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1975 RIDE QUALITY SYMPOSIUM

Held at
Williamsburg, Virginia
August 11-12, 1975

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • NOVEMBER 1975

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16. Abstract

The report consists of a compilation of papers presented at the 1975 Ride Quality Symposium held in Williamsburg, Virginia, August 11-12, 1975. The symposium, jointly sponsored by NASA and the United States Department of Transportation, was held to provide a forum for determining the current state of the art relative to the technology base of ride quality information applicable to current and proposed transportation systems. Emphasis focused on passenger reactions to ride environment and on implications of these reactions to the design and operation of air, land, and water transportation systems acceptable to the traveling public. Papers are grouped in the following five categories:

The Needs and Uses for Ride Quality Technology
Vehicle Environments and Dynamics
Investigative Approaches and Testing Procedures
Experimental Ride Quality Studies
Ride Quality Modeling and Criteria

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PREFACE

The 1975 Ride Quality Symposium, sponsored by NASA and the U.S. Department of Transportation, was held in Williamsburg, Virginia, August 11-12, 1975. In attendance at the symposium were representatives from 60 industry, university, and government organizations in the United States, Canada, and the United Kingdom. A predecessor symposium on Vehicle Ride Quality, reported in NASA TM X-2620, was held in July 1972.

The purpose of the symposium was to provide a forum for determining the current state of the art of ride quality technology applicable to current and proposed transportation systems. Emphasis was given to passenger reactions to ride environment and the implication of these reactions to the design, operation, and maintenance of air, land, and water transportation systems acceptable to the traveling public.

A five-man steering committee established policy with regard to the meeting format and subject material to be included. The steering committee was comprised of John J. Fearnsides and E. Donald Sussman of DOT and D. William Conner, David G. Stephens, and Raymond P. Whitten of NASA. Detailed planning and implementation of the symposium were carried out by members of the faculty and staff of the Department of Engineering Science and Systems of the University of Virginia under the direction of A. Robert Kuhlthau.

In addition to issuing a call for papers, four papers were invited to review specific areas deemed particularly important. These four papers reviewed ride quality technology needs of user organizations, measured vehicle ride environments, ride quality studies in the United Kingdom, and the International Standard ISO 2631 as it pertains to ride quality. The papers contained in this compilation have been edited only for clarity and format. Technical content and views expressed are the responsibility and opinions of the individual authors.

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Chairman: E. Donald Sussman
DOT Transportation Systems Center

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Chairman: David G. Stephens
NASA Langley Research Center

SESSION V - RIDE QUALITY MODELING AND CRITERIA

Chairman: Raymond P. Whitten
NASA Headquarters

INTRODUCTORY REMARKS

D. William Conner
NASA Langley Research Center

DEFINITION OF RIDE QUALITY

Vehicles expose man to ride environments which can interfere with comfort, working efficiency and, in some instances, health or safety. The present compilation of papers concerns only the comfort aspects of the ride environment and does not address either reduced proficiency or health aspects, both of which are associated with environmental inputs of much greater magnitude. Ride quality in the past has been defined either to embrace a number of environmental factors or to be restricted to only motion and vibration. Definition of ride quality as used in this symposium is therefore necessary to avoid possible misunderstanding and confusion. The definition which was adopted is:

> Impact on the passenger of all aspects of the carrier vehicle physical environment that affect his acceptance of the ride.

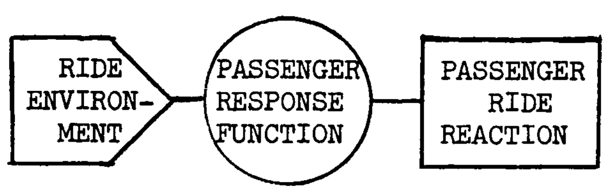
While motion and vibration are recognized as prime factors affecting ride environment, the definition allows inclusion of other factors (e.g., noise) which may be important as well.

PERSPECTIVES OF RIDE QUALITY

Ride quality, as affecting acceptance of the ride, can have different interpretations depending on the perspective. Appreciation of the different views can lead to a broader understanding of the significance of the various findings brought out in the ensuing compilation of papers. Three different views will be described.

Psychophysical View of Ride Quality

The psychophysical view of ride quality can be represented by the following three-element block diagram:



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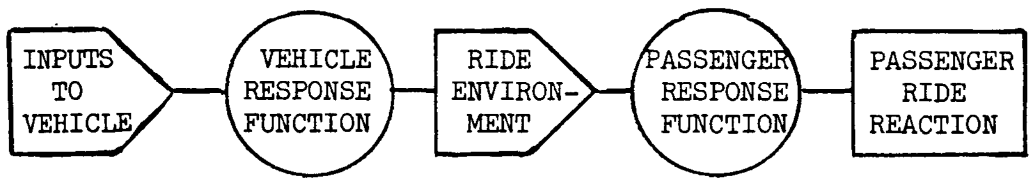
In this view of ride quality, the focus is on the passenger's comfort as affected by various ride environment inputs. Ride reaction depends on various psychological and physiological factors. The influences of the several factors are here labeled the passenger response function, which obviously will vary from person to person. To study passenger ride reaction, environmental inputs are first defined from field data of vehicles. Studies of subjective comfort reaction to these inputs, in component or combination form, are then carried out, generally under controlled conditions using simulators, to better understand and quantify the passenger response function. The resulting technology allows some prediction of passenger ride reaction to a given ride environment. In summary, the psychophysical view of ride quality concerns that which is desirable from comfort considerations.

Systems Engineering View of Ride Quality

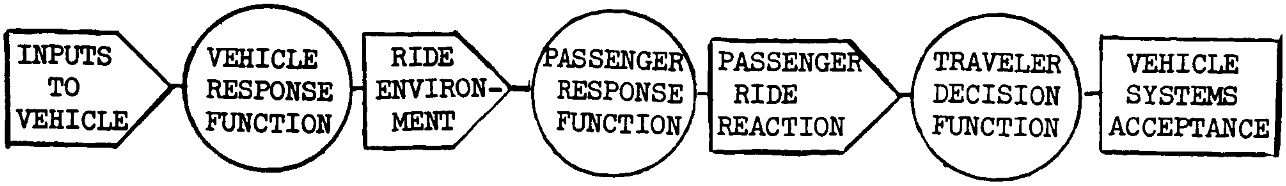
The systems engineering view of ride quality can be represented by the following five-element block diagram:



In this view of ride quality, the focus is on the passenger's comfort as affected by the perturbing inputs to the vehicle which create the ride environment, and on the characteristics of the vehicle in responding to these inputs. Some inputs (e.g., atmospheric turbulence for aircraft) must be accepted as they occur in nature, while other inputs (e.g., track roughness for trains) may be controlled by appropriate design and construction. Likewise the response characteristics of the vehicle to inputs can be tailored, at least to a degree, to meet specified objectives. Definition of the perturbing inputs to the vehicle and of the vehicle dynamic characteristics allows prediction of the ride environment which the passenger will experience. Effort is required to better understand and improve both the inputs to the vehicle and the vehicle dynamic characteristics in responding to these inputs. The resulting technology, coupled with the psychophysical studies technology described previously, will allow prediction of passenger ride reaction to a given vehicle system with its perturbing inputs. In summary, the systems engineering view of ride quality concerns that which is practical from engineering considerations.

Systems Marketing View of Ride Quality

The systems marketing view of ride quality can be represented by the following seven-element block diagram:



In this view of ride quality, the focus is on the decision of the traveler, as influenced by vehicle ride environment, to accept or reject use of the vehicle system for travel. Passenger ride reaction is just one of many varied inputs (e.g., trip cost, frequency of operation, availability of competing modes of travel, etc.) to the traveler decision-making process. The passenger's ride reaction may or may not be important to acceptance and use of a system. For example, if economic considerations demand that a system for a given market can only exist by increasing seat capacity in the vehicle, some increase in ride discomfort caused by using smaller seats spaced closer together may well be acceptable to the traveler if there are compensating advantages such as saving of time. A critical level of discomfort may be reached, however, in which a significant percentage of travelers rebel and refuse to use the system. Technology is needed, therefore, to allow determination, for a given market situation, of the relationship between the degree of discomfort in ride environment and the degree of traveler acceptance. As pointed out in the previous section, the ride environment is subject to improvement by appropriate upgrading of the vehicle and guideway (for ground vehicles). With technology sufficiently developed for all elements of this block diagram, cost/benefit trade-offs could be carried out of candidate changes to the vehicle system for improving ride quality. Marketing assessment of all-new vehicle systems could also be carried out during system design. In summary, the systems marketing view of ride quality concerns that which is justifiable from economic considerations.

The systems marketing view of ride quality is probably the most realistic approach for arriving at acceptable levels of ride comfort for a given situation. A comfortable ride for public transportation vehicles is presented not a guaranteed right of the traveler (although events may possibly lead to noncompetitive modes of public travel which have government imposed minimum standards for factors such as ride quality). No particular hazard to the physical well being is involved since the demarcation between comfort and discomfort occurs far below the levels of environmental inputs which pose a threat to health or safety. Design criteria may be developed which guarantee a comfortable ride but if the expense of reaching that level of comfort is economically prohibitive, use of the criteria may be academic. Greater benefits could result from ride quality technology which is sufficient to allow realistic assessment of cost benefits of ride improvement for any given vehicle/travel market situation.

ORGANIZATION OF COMPILATION

The papers in this compilation have been organized into five groups to address specific facets of ride quality technology. The first group of papers (papers nos. 1 to 3) was selected to illustrate the need for and use of ride quality technology and touches on all elements of the seven-element block diagram presented in the previous section. The papers report findings of an ongoing survey of users of ride quality technology, a frustrating experience in attempted use of present technology, and an example of the application of ride quality technology to evaluate traveler acceptance.

The second group of papers (paper nos. 4 to 10) is concerned with various facets of the vehicle ride environment and contributors to this environment and concentrates on the first three elements of the seven-element block diagram.

The third group of papers (paper nos. 11 to 13) concerns investigative approaches and testing procedures for psychophysiological studies on passenger reaction to given ride environments and concentrates on the middle three elements of the seven-element block diagram. The fourth group of papers (paper nos. 14 to 20) reports on the results of various psychophysiological studies on passenger reactions to ride environments.

The fifth group of papers reports activities underway in ride quality criteria (paper nos. 21 to 23) and modeling (paper nos. 24 to 26). These papers generally concern only the middle three elements of the seven-element block diagram. The last two elements of the block diagram were not considered in the criteria discussed by papers in this group but were included in the criteria employed in the ride quality application study reported in paper no. 3. Paper no. 21 critically examines the International Standard ISO 2631 with regard to its derivation and appropriateness as ride quality criteria. Paper no. 22 proposes absorbed power for expressing criteria as preferable to the plots of vibration intensity versus frequency employed in the International Standard ISO 2631. Paper no. 23, which was not presented in the symposium, was added to the compilation at the suggestion of its author to provide additional information not included in paper nos. 21 and 22 but pertinent to the derivation of International Standard ISO 2631 and to the meaningfulness of absorbed power for expressing criteria. Paper nos. 24 to 26 report the results of three experimental/analytical studies which employ differing approaches to ride comfort modeling of complex vibration environments.

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REVIEW OF RIDE QUALITY TECHNOLOGY NEEDS OF INDUSTRY AND USER GROUPS*

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SUMMARY

A broad survey of ride quality technology state-of-the-art and a review of user evaluation of this technology have been conducted. During the study so far, 17 users of ride quality technology in 10 organizations representing land, marine and air passenger transportation modes have been interviewed. Interim results and conclusions of this effort are reported in this paper.

INTRODUCTION

The quality of vehicle ride can be a significant factor in determining passenger acceptance and use of various modes of public transportation. Technology pertaining to the subjective aspects of ride quality is therefore needed to aid design and operation of vehicles and to achieve acceptance of existing and planned transport vehicle systems.

During the past few years significant efforts have been initiated to gain a better understanding of ride quality factors and to build a technology base adequate for supporting design of viable transport vehicle systems. Many of these ride quality technology programs (not including ride smoothing) have been conducted by research organizations rather than user organizations. Significant research has been accomplished to identify crew tolerance of acceleration in a military environment and has culminated in a portion of the military specification of Reference 1. Although this research is pertinent, it has resulted in identification of safety and proficiency levels rather than comfort levels as needed for evaluation of passenger response. This paper is confined to passenger ride response to commercial vehicles and its purpose is to present interim results of a critique of ride quality technology research activities from the viewpoint of user organizations.

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* This work is sponsored by the NASA Langley Research Center under Contract NAS1-13908.

Ride quality is important in the design of public transportation vehicles due to the influence of several factors. The primary factor that has been studied is the effect of vibrations on passenger response. The effect of vibrations has received the greatest amount of attention because it has an obvious influence on passenger comfort and is not as easily quantified as other factors such as temperature, humidity, etc.

It is easy to predict that the general increase in vehicular operational speed (except automobiles) which has occurred over the past several years presents a potential ride problem to the designer. This potential problem is common to different transportation modes.

In the case of aircraft, this potential problem results from the fact that vehicle (and passenger) vertical and lateral acceleration responses are approximately proportional to speed for a given turbulence environment. Short haul carriers tend to operate at lower altitudes where the probability of turbulence encounter is greater, and typical vehicles tend to operate at a comparatively low wing loading which in turn increases gust sensitivity. In addition, very large vehicles tend toward increased airframe flexibility and studies have shown that structural response to turbulence becomes significant in ride evaluation. Inputs such as these cannot be controlled directly but undesired vehicle response may be reduced by the application of automatic control concepts as demonstrated by the results presented in Reference 2.

In some cases, helicopters which were designed primarily for military missions have been used to provide taxi type service. Passengers are subjected to ride considerations for which they may not be prepared, such as noise, blade flicker or unaccustomed maneuvers.

Marine transportation systems face similar adverse environments including atmospheric turbulence as well as a varying sea state. As in aircraft, not much can be done to control the inputs short of avoiding the worst of them.

High speed ground transportation vehicles are also subject to the effects of turbulence and cross winds. The smoothness of rails, surfaces or guideways also directly affects the ride quality of such vehicles. Unlike atmospheric turbulence, this input can be controlled to a certain extent by original manufacturing requirements and by continuing maintenance.

Ride quality criteria in use for existing transportation modes primarily focus on vibration effects, although in most cases secondary attention is paid to other amenities such as seating, temperature, humidity, noise and decor. The user is sometimes faced with applying inadequate criteria or adapting criteria formulated for other vehicles to his purposes. He has encountered this situation because sufficient technology has not been developed or because existing data have not been transformed into a design format useful to him. This is the case for ride parameters such as exposure time, vehicle attitude, combined axis motion and multiple frequency effects.

The problem for today's user is the transformation of available ride quality knowledge or data where it is available into the proper format for his application. Subsequent sections will point out areas of technology weakness which impede the user in performing this task.

DEFINITIONS

Definitions of key words and phrases used throughout the paper are provided to establish a common basis for understanding and interpreting the results.

Ride Quality Technology and Criteria

Ride quality technology is defined as that body of knowledge which provides performance and cost data for the development of vehicle or system ride quality criteria. Ride quality criteria are defined as the performance standards for system design and development. The inter-relationship of technology and criteria implies that a lack or weakness of criteria is a result of an insufficient technology data base and there is a need for additional research.

Ride Quality/Passenger Acceptance

Ride quality means different things to different people. Traditionally, the term "ride quality" refers to the effects of vehicle motions such as acceleration response to inputs from equipment or maneuvers, or inputs from turbulence or guideway roughness. For the purpose of this paper ride quality is defined as shown in Figure 1. This definition is an extension of the traditional definition of ride quality in that the passenger's subjective response to the perceived vehicle motion is only one of many relevant factors.

The passenger's age, background, ride experience, motivation, physical and psychological condition also have a direct effect on his subjective rating of the ride experienced during his trip. For instance, at one time there was a monorail system serving one of the major airports from a remote parking facility. One could drive directly to the monorail station, leave his car to be parked by an attendant and board the monorail directly. The monorail system itself was rather jerky, noisy, and suffered from excessive roll conditions, but the alternative was to park in a crowded lot nearer the terminal and carry luggage a long distance. This monorail system probably would not have been able to achieve a quality of ride which would meet the criteria levied on rail systems today, but motivation dictated that this system was used and appreciated. Other examples of motivation dictating choice of transportation mode may be found in the "park and ride" rail or bus systems to be found in many large metropolitan areas. These examples do not indicate that ride quality is subordinate to motivation in the passenger's choice of transportation mode but simply indicate that there are trades that the passenger will make.

Vehicle response parameters such as motion, noise, and effects of other amenities such as seat geometry, temperature and odor, have been quantified to some degree but the varying effects of each or combinations of these parameters on passenger response have not been well defined.

Figure 1 is completed by the addition of a passenger acceptance transfer function. This term includes such things as passenger evaluation of cost, schedule convenience, mode prejudice and onboard services. The effect of these concepts on the passenger's choice of transportation mode is beyond the scope of this paper but is shown in the diagram to complete the perspective of passenger evaluations.

User of Ride Quality Technology and Criteria

Users of transportation vehicles and systems relate to ride quality technology and criteria in two distinctly different ways. Users may be governmental agencies responsible for procuring and/or operating a transportation system or a private company developing a vehicle which it hopes to sell to other companies or to the government. In this case the procurement organizations and company technology groups need an adequate technology base to develop criteria for a system specification or for an internal product development program. On the other hand, a user, such as a manufacturer, responding to a customer's requirements, is concerned with satisfying the specified criteria and has little need for the technology data from which the criteria were derived. This paper discusses both ride quality criteria and technology from the appropriate user point of view.

Personal interviews of typical users of ride quality criteria and technology were conducted to expedite the gathering of data for this paper. The type and number of users contacted to date are listed in Table 1.

VEHICLE RIDE QUALITY PROBLEMS - PRESENT AND FUTURE

Selected present and future vehicle ride quality problems are outlined in Table 2. Problems identified are those assumed to have greatest priority in terms of requirements for ride improvement for existing or future modes of transportation. Contents of this table are preliminary since much data bearing on this subject have not been received.

Vibration and noise environments account for a majority of user concerns with vehicle ride quality, both for existing and near-future transportation systems. Vibration sources presenting ride quality problems generally occur at the interface of the vehicle with the medium on or through which it is traveling. Noise sources exist in ground transportation systems at this same interface while a major contributor to noise level in water or airborne vehicles arises from the propulsion system.

There are sources of vibration and noise, however, that create unique ride quality problems for different transportation modes. Examples of these more unique problems are switch crossings, wheel squeal during turns (rail), transition from foilborne to hullborne status (marine), vibrations and noise associated with blade passage (helicopters) and effects of maneuvers (short haul aircraft).

This initial effort to anticipate ride quality problems associated with transportation vehicles of the near future did not identify many new areas of concern. Vibration and noise problems still appear to be primary and are aggravated due to higher speeds and more powerful propulsion systems. In the air transport mode, quiet short haul aircraft may have degraded ride due to low wing loading and runway roughness may be more of a problem as aircraft get larger and more flexible. These trends do not necessarily project a bleak picture for the future passenger, however, since application of vibration control and noise alleviation technology will likely solve the problems. These technologies will presumably be guided by more sophisticated and accepted ride quality criteria.

AN OVERVIEW OF THE EXISTING RIDE QUALITY DATA BASE

Persons responsible for specifying vehicle ride environment need an adequate data base to support this activity. The existing data base is given a cursory review here to establish a basis for discussion in the following section, the User View of Ride Quality Criteria. Elements of the ride environment addressed are temperature, humidity, airflow, barometric pressure, leg room, seat width, noise and vibration. An excellent starting point for persons interested in a more detailed review of the relevant literature is Reference 3, which develops initial environmental criteria for motion, noise, temperature, humidity and pressure.

This overview considers only those potential sources of criteria which are published in the general literature or which have been presented at technical meetings covering a specific area of ride quality data. This restriction excludes consideration of criteria based on passenger vehicle manufacturers' or passenger carriers' experience with consumer acceptance of their product, unless these criteria are in published form. Appropriate reference is made to ongoing ride quality related studies for which interim results have been presented.

Temperature, Humidity and Rate of Air Flow

These three components of the vehicle ride environment are commonly discussed jointly. It appears that there is general agreement among the handbooks regarding comfortable ranges although there may be minor differences. A comprehensive review of relevant data is found in References 4, 5 and 6.

Barometric Pressure

Primary concern in this area has been the rate of change of pressure that is acceptable to air travelers. References 6 and 7 present acceptable limits.

Leg Room and Seat Width

Space available to the seated commercial passenger is an important factor affecting assessment of vehicle ride quality, particularly if the trip is extended and if movement within the vehicle is restricted. Anthropometric data are available in standard design handbooks to establish these space requirements. In addition, Reference 6 proposes seat pitch and width to provide acceptable passenger comfort.

Noise

The data base from which ride quality noise criteria may be drawn is more fragmented than those for the elements of the ride environment discussed above for a number of reasons. Most of the literature relating subjective reactions of persons to noise levels deals with the problem of community reactions to noise sources such as road or rail traffic and airplane fly-overs. There is also disagreement on the most appropriate scale of noise measurement and the best means of measuring or evaluating the passenger noise environment. There are also problems in reaching a consensus on the level of subjective response that defines an unacceptable noise environment. References 3 and 8 contain relevant discussions of different approaches taken to define a noise exposure criterion.

Motion

There is a large amount of data available describing the human reaction to motion. References 3 and 9 provide results of literature searches that include most of the relevant reports. Reference 10 contains results of a survey of vibration research being conducted in Great Britain; final circulation of the survey was to 57 organizations, 27 of which reported ongoing research or research capabilities relating to human response to motion.

Most of the literature deals with human response to single frequency, single axis vibration (generally vertical or lateral). The most widely recognized criteria in this area are the ISO standards of Reference 11, which address human comfort response to vertical, lateral and longitudinal vibration in the frequency range above 1.0 Hertz. Other data sources available are contained in References 12 and 13. These two references are based on ground simulator and flight research experiments, respectively.

There is a lack of motion ride quality data for vibration frequencies below 1.0 Hertz and efforts are underway to fill this gap. An extension of the ISO standards to include the 0.1 to 1.0 Hertz range was proposed in 1974 as reported in Reference 14. An alternative frequency response weighting curve was proposed for ISO consideration by U.S. members of the ISO committee. An amendment or an appendix to the ISO standards to include human response to vibration frequencies below 1.0 Hertz is nearing completion although the added criteria may not be viewed as an extension of existing standards because of qualitative differences in human response to vibration frequencies above and below 1.0 Hertz. Further developments relating to the ISO standard will be presented during this conference.

Data describing passenger response to vertical and lateral vibration at frequencies below 1.0 Hertz were obtained in a research flight test program conducted at NASA FRC. These data are reported in Reference 13. Results of a study to link incidence of motion sickness with frequencies and acceleration of vertical motion are reported in Reference 15. Equal subjective intensity curves for the frequency region 0.25 to 4.0 Hertz (vertical vibration) are reported in Reference 16.

The data base relating human response to angular motions is very limited. Little data have been generated to investigate human subjective response to multiple frequency or multiple axis vibration. Some starts have been made to explore this general area as shown in References 13 and 17.

Combined Elements of the Ride Environment

Little research has been conducted to investigate effects of combinations of ride quality variables on passenger ride comfort. Research reported in Reference 18 indicates that combinations of heat, noise and vibration were judged more stressful than any component variable alone.

USER VIEW OF RIDE QUALITY TECHNOLOGY

The user's view of ride quality technology seems to be focused through the lens of the criteria he has available or can foresee developing from the existing data base. Consequently, discussions with ride quality technology users always center on the adequacy of ride quality criteria. In this section the user view of ride quality technology is discussed.

As previously mentioned, interviews were conducted with representatives of various facets of distinct public transportation modes to expedite collection of opinions and data. During this study it was determined that the user typically assigns a large weight to the effect of accelerations on passenger ride response compared to other influences. Due to this fact criteria relating to passenger acceptance of vehicle motion are emphasized in the following discussion.

Two basic types of ride quality criteria are in use today in the transportation field. First, there are specific criteria based on results of experiments performed with subjects placed in a pseudo-real passenger environment using moving base simulators. These criteria are usually expressed as limits on some expression of vehicle acceleration versus frequency as shown in Figure 2. Most experiments of the type generating motion response data have used a small number of subjects with professional or semi-professional backgrounds. Habitability variables are most often fixed and vibrational inputs including noise are varied to observe effects. Also, the vibrational inputs representing vehicle motion are often of a single frequency, single axis nature. Criteria derived from empirical studies of this type often do not agree in interpretations of acceptable limits of acceleration as revealed in references such as 6 and 19. An attempt has been made to resolve these differences as shown in Reference 11, but agreement on criteria specification among transportation modes is still not universal.

It should be pointed out that a passenger's ride response will probably be influenced by his expectations rather than an absolute basis. This means that an acceptable ride for a train where sway or lateral acceleration may be expected may not be an acceptable ride for an airplane. When different modes of transportation are considered, there may be variable requirements for acceptable levels of acceleration. This argues against the use of a single standard for all types of passenger vehicles. From another point of view, such a universal application of criteria could cause additional and unwarranted cost of design and manufacture if requirements leading to overdesign were established.

The second type of ride quality criteria is called in this paper the "As Good As" or AGA criteria. These criteria are usually more related to passenger response than to vehicle response although generally there is some attempt to characterize acceptability in terms of acceleration versus frequency. For instance, a potential customer may require that a new vehicle shall ride "as good as" vehicles with which he has had previous experience and confidence of good passenger acceptance. This method has occasionally been taken a step farther by requiring that the new vehicle exhibit accelerations "less than" those encountered with some previous vehicle.

The primary problem with the use of AGA ride quality criteria is that the vehicle manufacturer must first determine the ride quality of the vehicle being used as the goal and then devise a method to demonstrate compliance which meets the customer's agreement.

In some industries, criteria such as these have been the traditional means of stating desired ride quality and the method has worked well within a manufacturing company that has previous experience to rely upon. A major difficulty with this approach occurs when a new type or family of vehicles is to be developed.

Air Transportation Mode

Public air transportation may be divided into three basic categories; trunk lines, feeder lines and commuter lines. Of the three, the commuter and feeder lines encounter the more significant ride quality problems because they generally operate at lower altitudes where turbulence is more likely to be encountered and with small, light wing loading aircraft which are more responsive to turbulence than the large jets. In addition, their frequency of takeoff and landing and the accompanying degree of maneuvering motion is greater.

Air transportation is the clearest example of the use of AGA ride quality criteria. Typically, the airplane manufacturer relies heavily on past experience to determine what produces favorable passenger response to ride and designs to the dictates of this experience. During the preliminary design stage of an airplane, the vertical gust acceleration response sensitivity is evaluated in terms of its change in lift coefficient due to variation of angle of attack, C_{L_α}, or wing loading, lift per unit of wing area. A typical survey is shown in Figure 3 for comparison purposes. Here, vertical accelerations of several aircraft classes are characterized by their change in lift coefficient due to variation of angle of attack and compared to a baseline which is known to have good passenger ride response.

The situation is not as clear in the design of larger more flexible airframes where structural mode dynamics may have a significant role in passenger ride acceptance. Again the AGA criteria are used but a lack of definitive passenger subjective reaction models may lead to problems. The design goal of a recent large flexible airplane in the area of dynamic turbulence response was to be "as good as" a previous acceptable design. During the preliminary design stage it was known that aft body lateral acceleration response to turbulence was slightly greater than that exhibited by the baseline, but a review of passenger subjective response data and consideration of other factors resulted in a decision not to attempt a reduction. Subsequent service operations have revealed inadequate passenger response to aft body lateral ride in certain situations and an active control system has been designed for the airplane to alleviate this situation.

When the manufacturer begins the design of a new generation of aircraft not similar to previous designs, he is obliged to consider the ride quality situation in greater depth. For instance, during the conceptual design phase of the American Supersonic Transport, Boeing-Wichita conducted a broad range of studies to determine human reactions to vibrations ranging in frequency from 0.10 to 7.0 Hertz, as reported in Reference 17. These studies were undertaken because the slender, flexible fuselage of the design exhibited lower frequency larger amplitude response to turbulence than had previously been the case with conventional aircraft. This additional study was deemed necessary since passenger reaction to accelerations due to both turbulence and runway inputs was not clearly defined.

Contributory factors to the passenger ride response other than accelerations are listed in Table 3. The specific effect of each of these quantities as a modifier to ride response is not normally evaluated, but each factor has an effect on passenger comfort and apprehension, which in turn modifies the level of ride response.

Customer specifications or FAA egress regulations will normally determine basic seating factors as well as air conditioning, lighting and ventilation requirements. Noise and unusual odors are kept to a minimum and decor is specified by the customer but is designed to provide the passenger with an overall feeling of safety. Interior noise measurement techniques within the industry should be standardized and additional understanding of subjective reaction is necessary.

In the design of an aircraft, the cost of providing acceptable ride must be ranked in the overall economic equation and this rank will vary depending on the type service considered. Initial cost and return on investment are the two most important factors in the design of a commercial aircraft. A passenger must have a ride that will cause him to accept that airplane as a candidate for future flights but beyond that the benefit from increased cost to be devoted to comfort is difficult to ascertain. Normally if the ride is adequate in competition with similar services, costs associated with ride improvement will not be accepted by the airplane operator.

Helicopters present some unique facets of the same problems previously discussed. Noise, acceleration impulses due to blade passages and unaccustomed maneuvers are the primary adverse ride quality factors. Interior noise levels are generally required to be similar to existing conventional jet aircraft. Each noise source has its own characteristic frequency with engine noise being highest and least bothersome. Noise criteria are based on hearing loss, fatigue and on speech or communication requirements and are measured in several ways as shown in Reference 18. One serious deficiency in noise measurement is the inability to measure low frequency impulsive noise accurately using current techniques. The methods and units of noise measurement need to be standardized so that existing criteria can be evaluated.

In summary, the weak ride technology areas discovered relating to air transportation modes are:

- Passenger subjective reaction must be quantified and correlated with an easily measured vehicle parameter such as acceleration.
- Criteria need to be presented in terms that allow easy verification of compliance. This is a problem since the normal vehicle input is random but most criteria are based on single frequency inputs.
- Similarly, criteria need to take into account combined axis and multifrequency inputs.
- Noise measurement variables and techniques need to be standardized.
- Vehicle mission and type need to be recognized by criteria.
- Best criteria format needs to be established.

Rail Transportation Mode

The rail industry appears to divide naturally into three classes based on weight, size and number of cars per train. Light rail refers to streetcars and one or two car rapid transit trains operating at moderate speeds on elevated, grade level or subway type track. A middle ground is occupied by the regular subway trains such as used in New York that are larger, heavier, and operate in multicar trains. The third type is the heavier intercity type passenger train.

In rail transport vehicle procurements, both the specific criteria (usually accelerations) and the AGA criteria are used. For instance, the San Francisco Bay Area Rapid Transit (BART) system specifications incorporated specific criteria based on measured accelerations as shown in Figure 4. Another specific criterion is that for the State of the Art Car (SOAC) shown in Figure 5. On the other hand the AGA criteria used in the specification for new Chicago transit cars stated that ride quality should be equal to or better than that of certain serial number cars already in service. as determined by measuring vertical, lateral and longitudinal accelerations. Competitors for this contract had to determine how to measure the ride of the existing cars and then how to compare the ride of their proposed vehicle to show compliance. One complicating factor was that of track inputs. In order to keep inputs regulated, a track with known dynamic characteristics or a particular section of track must be specified. Power spectral density (PSD) must be specified and then, when compliance is to be demonstrated, a track with similar PSD must be used. If track dynamics were specified along with required accelerations, the manufacturer could analytically determine the adequacy of ride in his vehicle with respect to the criteria.

Here again the lack of quantified passenger subjective response is apparent. Either criteria are presented in terms of accelerations, or the ride is required to be as good as existing equipment known to have acceptable ride.

In the ride quality specifications for intercity railroad cars the National Railroad Passenger Corporation (AMTRAK) has taken the more sophisticated approach of specifying a particular track PSD and requiring that the resultant vehicle accelerations meet a certain rms level on one type car and, on another car, that measured acceleration PSD's of the new vehicle and an existing vehicle be analytically transformed to a perceived comfort level for comparison. A data base is being developed from actual measurements of track PSD, vehicle accelerations and passenger subjective reactions using experienced "raters".

The two main facets of ride quality in rail transportation are the vehicle dynamics and the rail dynamics. Rail construction specifications are always in terms of allowable static deflections per unit of distance traveled. This type criterion puts very little restraint on the resultant track dynamics at higher frequencies although the trend from jointed to welded rails has moved primary input frequencies away from those most objectionable to the passenger. The impact of track smoothness criteria on construction costs should be considered in selecting applicable criteria since the cost of building a dedicated rail system may be a large percentage of the total cost of the system.

As in aircraft, passenger amenities are specified separately from allowable acceleration with no attempt to show modifying influences. Noise measurements in dB(A) seem to be standard but the acceptable levels are open to question. A minimum level should also be specified in order to provide speech privacy.

In summary, the weak ride technology areas discovered relating to the rail transportation mode are:

- There is a proliferation of ride quality criteria.
- There is not much correlation of criteria with track dynamics.
- Track and car dynamic models are generally not adequate for extensive analysis.
- Cost impact of ride criteria needs to be carefully assessed.
- The data base must be expanded in track dynamics.
- Passenger subjective reaction must be quantified.
- It must be confirmed that criteria specified are applicable to the vehicle.

Marine Transportation Mode

One of the newest modes of marine commercial transportation is the submerged hydrofoil, hereafter referred to as the Jetfoil. The unique feature of this vehicle is that its lift is derived from submerged hydrofoil surfaces. This provides a ride impervious to sea state up to the capability of the system to keep the hull above wave crests. Ride quality criteria developed by the manufacturer for this system are similar in form to criteria used for aircraft and have been described in Reference 20.

The primary deficiency in ride quality technology for this transportation mode is for motions in the frequency range below 1 Hertz. Since this is the frequency range in which motion sickness is predominant, criteria in the range below 1 Hertz are of utmost interest in the design of marine vehicles. Information is lacking on the effects of motion and the effects of the duration of the motion. It is possible that different criteria might be required for passengers and crew due to the effects of duration in this low frequency range.

Another related deficiency is the effect of combined axis inputs on the passenger reaction to motions in this frequency range. As in other transportation modes investigated, habitability variables such as temperature, seating, etc., are specified but effects are not assessed to determine impact on ride. In the case of the Jetfoil, the goal was to provide passenger amenities "as good as" a current jet aircraft.

Another weak criteria area is in the specification of a sea model. Models similar to those used to define atmospheric turbulence have been developed to aid in marine vehicle analysis and synthesis, but work in this area is by no means complete or adequate. Once again the passenger subjective reaction needs to be quantified so that the manufacturer can predict passenger reaction to proposed marine vehicle ride. The manufacturer could then predict the percent of passengers that would be satisfied with ride in a particular customer's operating environment and more easily reach adequate contract agreements. This capability would also allow overdesign to be identified and reduced, thereby reducing cost.

In summary, the weak ride technology areas identified in the marine transportation mode are:

- Inadequate criteria in the frequency range below 1 Hertz.
- Inadequate definition of the effects of duration in this frequency range.
- Inadequate knowledge of multi-input axis effects.
- Lack of passenger subjective reaction quantification.
- Lack of adequate sea models.

Surface Transport Mode

In surface transport, as in rail transport, there is a proliferation of ride quality criteria as well as possible inappropriate application of these criteria. For instance, acceleration versus frequency criteria have been used to define acceptable ride for some recent rubber-tired automatic people-mover systems. There has also been some disagreement about correlation between these criteria and the passenger subjective reaction to the ride actually perceived. The need here is to provide the necessary subjective passenger reaction evaluation so that appropriate criteria may be determined and adjustments made if necessary.

Another facet of the ride criteria situation that is a candidate for close inspection is the required interior noise level. The ability to achieve required levels is affected by many factors. For instance, the fact that maintenance requirements may severely impact the noise level illustrates the need to consider the effects of all inputs. Maintenance requirements that dictate ease of cleaning and low susceptibility to vandalism can cause difficulty in achieving required noise levels. The conclusion then is that all factors affecting ride should be considered simultaneously, weights for each input established, and trade studies conducted to define costs.

The AGA criteria are also used in the surface transport mode. One such case is found in the TRANSBUS program sponsored by the UMTA where prototype transit buses were developed to a ride criteria goal of "as good as a 1973 Ford LTD". In order to apply this criterion, quantitative data had to be generated. This involved building a test track with simulated roadway anomalies and evaluting two automobiles of the type specified as well as an urban bus to serve as a baseline. Results are reported in Reference 21. Here again, as in other transportation modes, we find the AGA criteria being used with the result that these criteria must be quantified before they can be applied.

In some cases of commercial manufacture, this quantification step is bypassed by the use of subjective evaluations by experienced raters and management personnel. This approach has apparently worked well in the past in lieu of quantitative acceleration criteria.

The surface transportation modes face problems similar to those described for the rail transportation modes in the area of guideway surface criteria. Again the usual specification relates to static deflections and very little dynamic modeling information is available to the investigator so that he can realistically predict vehicle response to random inputs. Some work is being done in this area as shown in Reference 22 to try to quantify guideway surface dynamics and produce criteria other than the familiar acceleration criteria. The approach taken has been to generate a figure of merit based on a particular weighting of vehicle response variables. This approach has been investigated by the British Railways Board and is also being investigated at the University of Texas where an ISO weighted ride index has been developed that exhibits good agreement with passenger subjective reaction to automobile ride. Some results are presented in Reference 23.

In summary, weak ride technology areas identified in the surface transport mode are:

- Proliferation of criteria.
- Inappropriate application of criteria (criteria developed for one class of vehicle applied to a different class).
- Lack of correlation between acceleration criteria and passenger subjective reaction.
- Criteria weight (noise, etc.).
- Trade studies to identify undetermined criteria effects.
- Criteria cost impact (related to weight).
- Lack of ability to correlate acceleration response to random inputs with criteria based on single frequency inputs.

- Lack of adequate statistical definition of guideway surface.

- Methods of providing specification compliance.

DISCUSSION OF FINDINGS

The preliminary results of an effort to determine representative views of ride quality technology users in four distinct public transportation modes were presented in the previous section. A review of the findings reported discloses that there are many similarities among the needs presented. In fact, it appears that one list of user needs can be constructed that will suffice for all transportation modes. Such a list is presented in Table 4.

One of the first things necessary to satisfy user needs is standardization. This applies to both acceleration and noise technology. For instance, there is the question of applicability of acceleration criteria developed from technology based on the use of single frequency inputs in the evaluation of vehicle response to random inputs. The user wants to know how to reconcile any possible differences and how to evaluate realism effects such as passenger apprehension not present in moving base simulators. Also, information is limited on effects of motion below 1 Hertz. Another factor that generally lowers the user's evaluation of the available technology is the minimal knowledge of effects of combined axis inputs and multiple frequency inputs.

Standardization does not mean the application of one criterion to all vehicles. In fact, it is quite possible that criteria magnitudes should be adjusted for applicability to different modes and to different vehicles within each mode. Different criteria formats might be desirable. Such a format might be the figure of merit type discussed previously instead of the more familiar acceleration versus frequency format.

Agreement on standard units and methods of noise measurement is desired. Typical noise measurement locations, vehicle configuration and passenger loading should be defined.

The situation that allows a proliferation of criteria without sufficient guidance for application places an unacceptable burden on the contractor trying to demonstrate specification compliance. If compliance is to be demonstrated analytically, proper mathematical models of vehicle input such as a PSD of rail or guideway surface smoothness should be developed for use and standard methods of determining vehicle response should be agreed upon. In addition, standard methods of vehicle response measurement should be defined so that demonstration of specification compliance is adequate.

Second, passenger subjective reaction must be quantified and correlated with an easily measured vehicle response parameter, probably acceleration. This would allow the user to more precisely determine passenger ride response analytically. Benefits beyond preliminary assurance of specification compliance would include more intelligent marketing and the ability to eliminate some overdesign with subsequent lowering of manufacturing cost. In line with this quantification, the combined effects of varying other passenger comfort quantities such as noise, temperature, humidity, etc., should be determined.

Thirdly, the cost of applying ride quality criteria should be determined. Some vehicles within a transport mode may need more sophisticated criteria than others, depending on the job to be performed, but applying criteria without first determining the impact on system cost may penalize a particular transport mode by escalating initial cost. The percent of passengers satisfied with the ride versus the cost of providing the ride should be quantified so that the desired cost effectiveness can be determined. A plot typical of such a quantification is shown in Figure 6. Point A on the figure is representative of a ride that would satisfy only a small percent of passengers although the cost is lowest. Point B represents some optimum or desired trade between percent of passengers satisfied and cost of providing that satisfaction. Point C is included to demonstrate the cost of satisfying the last 5 or 10 percent of passengers can be quite high and it is probably true that not everyone can be satisfied no matter how much is spent.

CONCLUDING REMARKS

The interim results of this study show that ride quality technology users perceive technology weaknesses through the ride quality criteria that are subsequently developed. Technology weaknesses identified during this study were discussed in detail in the previous section and are concentrated in four areas.

Ride technology results need to be standardized so that standard criteria may be developed. In conjunction with this, units and methods of measurement should be standardized. Passenger subjective reaction to vehicle ride must be quantified so that the user can accurately predict the percent of passengers satisfied. Costs of applying technology to improve ride must be assessed so that the user can determine the level of ride he can afford. Finally, advanced techniques for specifying and evaluating guideway construction should be investigated.

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TABLE 1

RIDE QUALITY TECHNOLOGY USERS
PERSONAL CONTACTS

VEHICLE CLASSIFICATION	NUMBER OF ORGANIZATIONS	TYPE OF ORGANIZATION
MARINE	1	MANUFACTURER
BUS	2	MANUFACTURER GOVERNMENT AGENCY
HELICOPTER	1	MANUFACTURER
LIGHT RAIL	3	MANUFACTURER GOVERNMENT AGENCY
HEAVY RAIL	1	OPERATING AUTHORITY
LARGE AIRCRAFT	1	MANUFACTURER
MEDIUM AIRCRAFT	1	OPERATOR

TABLE 2

VEHICLE RIDE QUALITY PROBLEMS PRESENT AND FUTURE

VEHICLE CLASS	PRESENT PROBLEMS	FUTURE PROBLEMS
LIGHT RAIL	• RAIL/GUIDEWAY SMOOTHNESS AND INTERACTIONS WITH VEHICLE CARRIAGE SYSTEM – VERTICAL AND LATERAL MOTION • NOISE, INCLUDING WHEEL SQUEAL WITH TURNS • LONGITUDINAL ACCELERATION • ROLL LIMITS (MONORAIL)	• RAIL/GUIDEWAY SMOOTHNESS AND INTERACTIONS WITH VEHICLE CARRIAGE SYSTEM
HEAVY RAIL	• RAIL/GUIDEWAY SMOOTHNESS AND INTERACTIONS WITH VEHICLE CARRIAGE SYSTEM – VERTICAL AND LATERAL MOTION	• RAIL/GUIDEWAY SMOOTHNESS AND INTERACTIONS WITH VEHICLE CARRIAGE SYSTEM • OUT-THE-WINDOW VIEW DURING HIGH-SPEED TRAVEL • MOTION CAUSED BY HIGH-SPEED TRAVEL
BUS	• VERTICAL AND LATERAL MOTION • NOISE	
MARINE	• LOW FREQUENCY MOTION, SHORT AND EXTENDED DURATIONS OF EXPOSURE • PEAK ENCOUNTER • TRANSITION FROM FOILBORNE TO HULLBORNE	
HELICOPTER	• NOISE • VERTICAL AND LATERAL MOTION	
AIRPLANE – SHORT TO MEDIUM RANGE	• VERTICAL AND LATERAL MOTION • MANEUVERS • LOW FREQUENCY MOTION • NOISE	• MORE CRITICAL MANEUVERS (STOL) • TERMINAL CONFIGURED VEHICLE MANEUVERS
AIRPLANE – MEDIUM TO LONG RANGE	• VERTICAL AND LATERAL MOTION • LOW FREQUENCY MOTION • TAXI (EFFECTS ON CREW)	• HIGHER LEVELS OF MOTION OF FREE A/P (HIGHER SPEED AND MORE FLEXIBLE AIRFRAMES) • LONGER DURATION FLIGHTS AND LOWER WING LOADINGS (FUEL CONSERVATIVE TRANSPORTS) • TERMINAL CONFIGURED VEHICLE MANEUVERS

TABLE 3

CONTRIBUTING FACTORS TO PASSENGER RIDE RESPONSE

- BASIC SEAT ARRANGEMENT
- AISLE WIDTH
- SEAT WIDTH
- SEAT RECLINE
- SEAT SETBACK
- AIR CONDITIONING
- LIGHTING
- GENERAL NOISE
- VENTILATION
- ODORS
- DECOR
- UNEXPECTED EQUIPMENT NOISE

TABLE 4
FINDINGS

USER NEEDS	BENEFITS
STANDARDIZATION OF RIDE TECHNOLOGY • ACCELERATION – UNITS – EXTEND BELOW 1 Hz – SINGLE FREQUENCY VERSUS RANDOM – COMBINED AXIS EFFECTS – MULTIPLE FREQUENCY EFFECTS – FIGURE OF MERIT • NOISE – UNITS – MEASUREMENT TECHNIQUES – MEASUREMENT LOCATIONS – PASSENGER LOADING • ANALYTICAL REPRESENTATION OF INPUTS	• EASE OF COMMUNICATION BETWEEN CONTRACTING PARTIES • FIRM BASE FOR DEMONSTRATING SPECIFICATION COMPLIANCE • IMPROVED APPLICATION OF ANALYTICAL TECHNIQUES • INCREASED CONFIDENCE IN RESULTS
QUANTIFY PASSENGER SUBJECTIVE REACTION • CORRELATION WITH MEASURABLE RESPONSE PARAMETERS • DETERMINE COMBINED EFFECTS OF OTHER INPUTS	• ELIMINATION OF POSSIBLE OVERDESIGN • ASSURANCE OF CERTAIN PROBABILITY OF PASSENGER RESPONSE • INCREASED EASE OF MARKETING
DETERMINE COSTS OF IMPROVED RIDE • CRITERIA WEIGHT • IMPACT ON SYSTEM COST • TRADE STUDIES	• MORE INTELLIGENT APPLICATION OF CRITERIA • MINIMIZED SYSTEM COSTS
ADVANCED SPECIFICATION FOR GUIDEWAY CONSTRUCTION • DYNAMIC AS WELL AS STATIC	• CONTROLLED VEHICLE INPUT

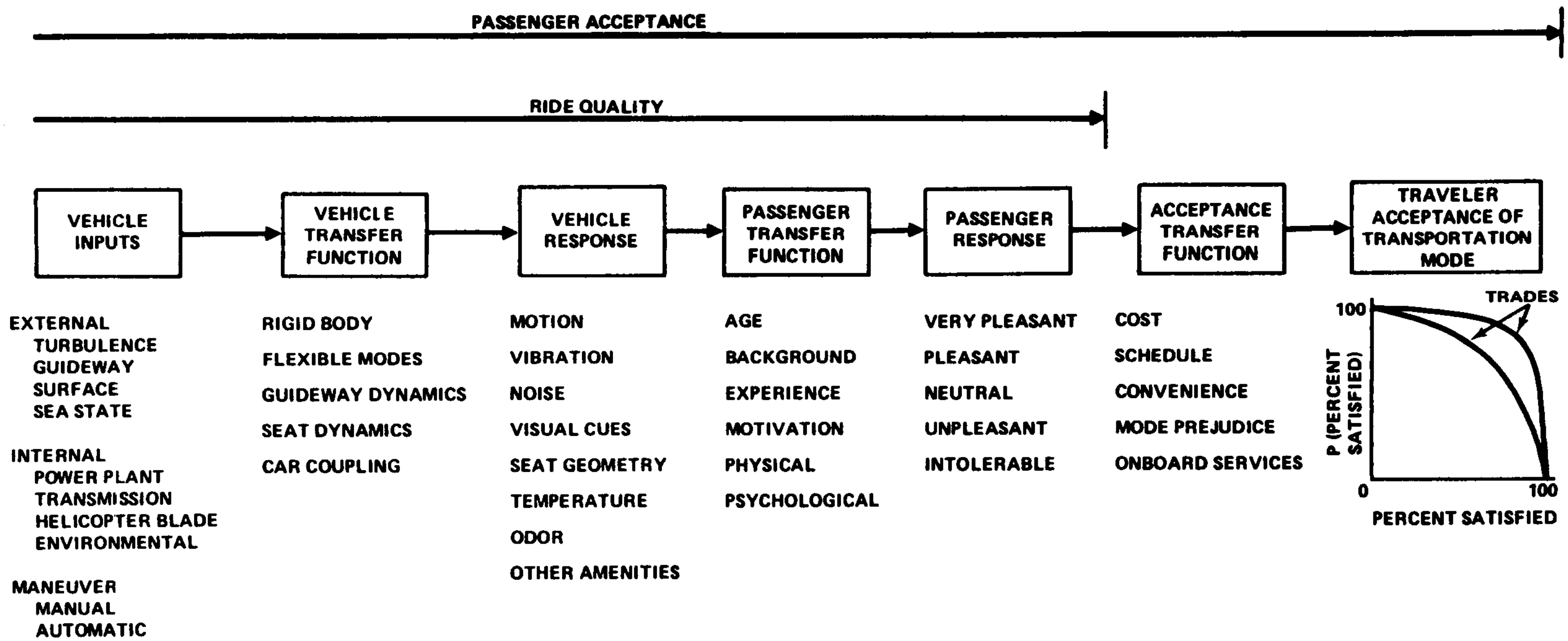

Figure 1.- Passenger acceptance diagram.

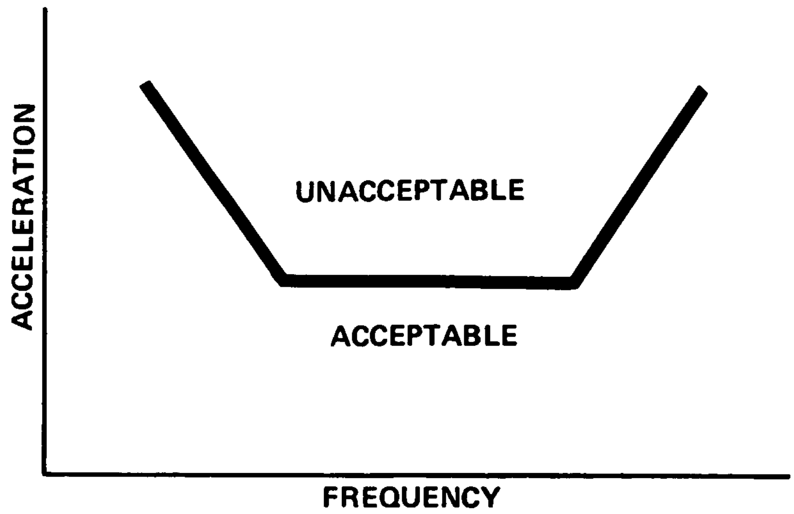

Figure 2.- Typical ride quality criteria format.

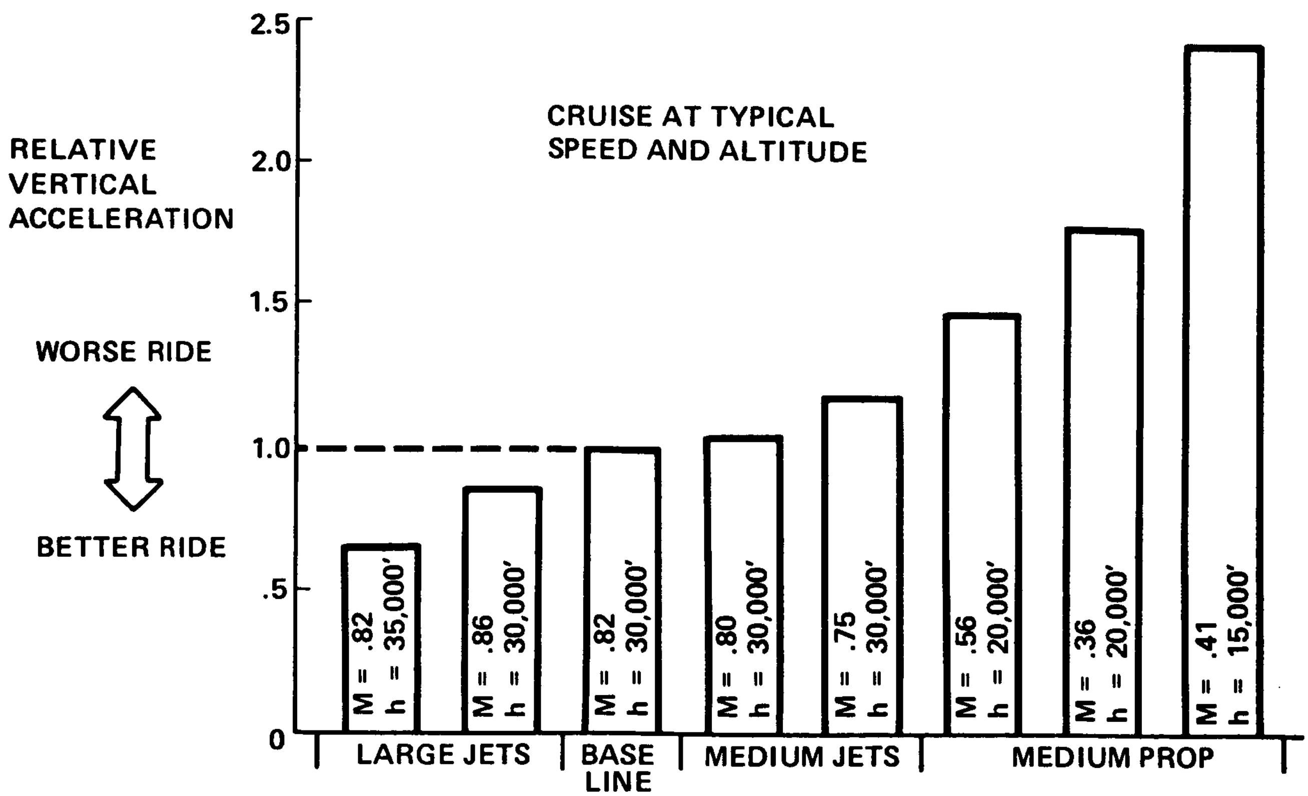

Figure 3.- Relative ride quality. M denotes Mach number and h denotes altitude. 1 ft = 0.3048 m.

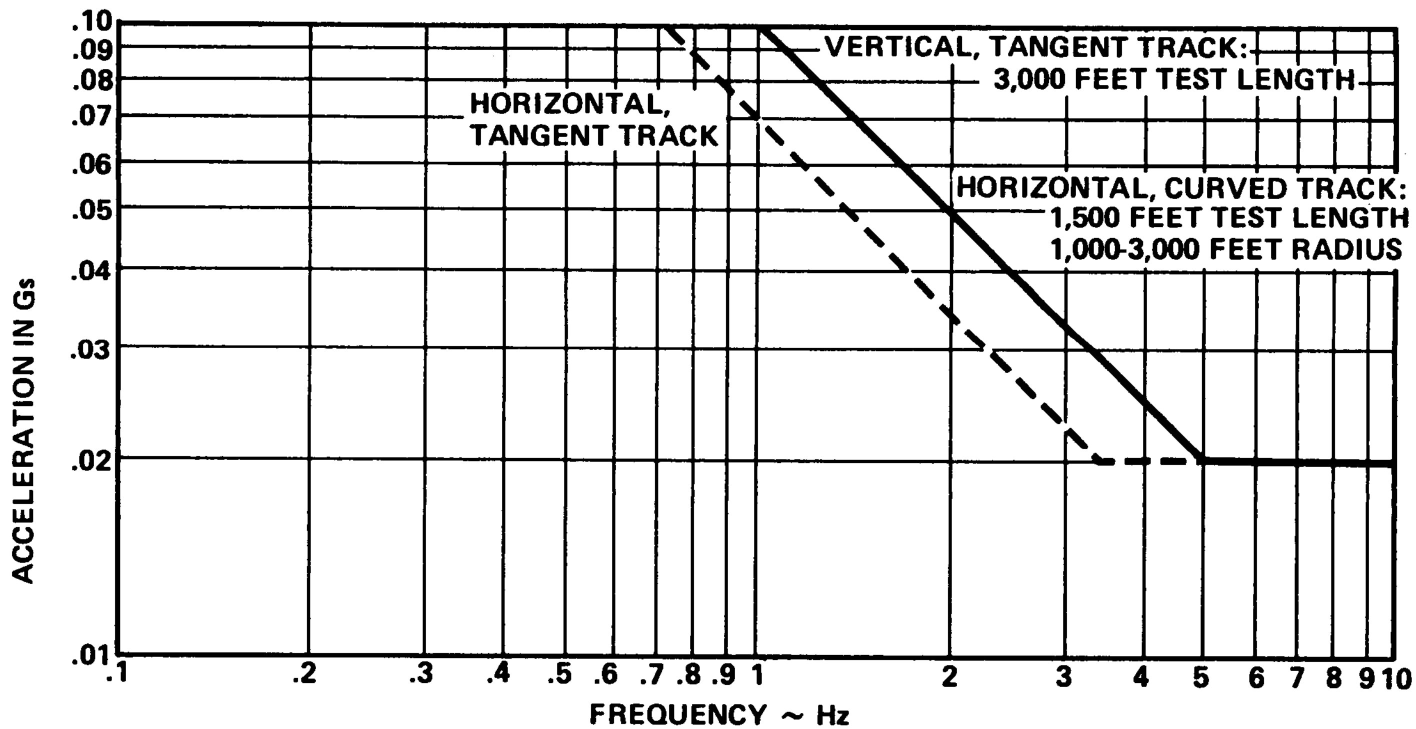

Figure 4.- BART ride quality goals. 1 ft = 0.3048 m.

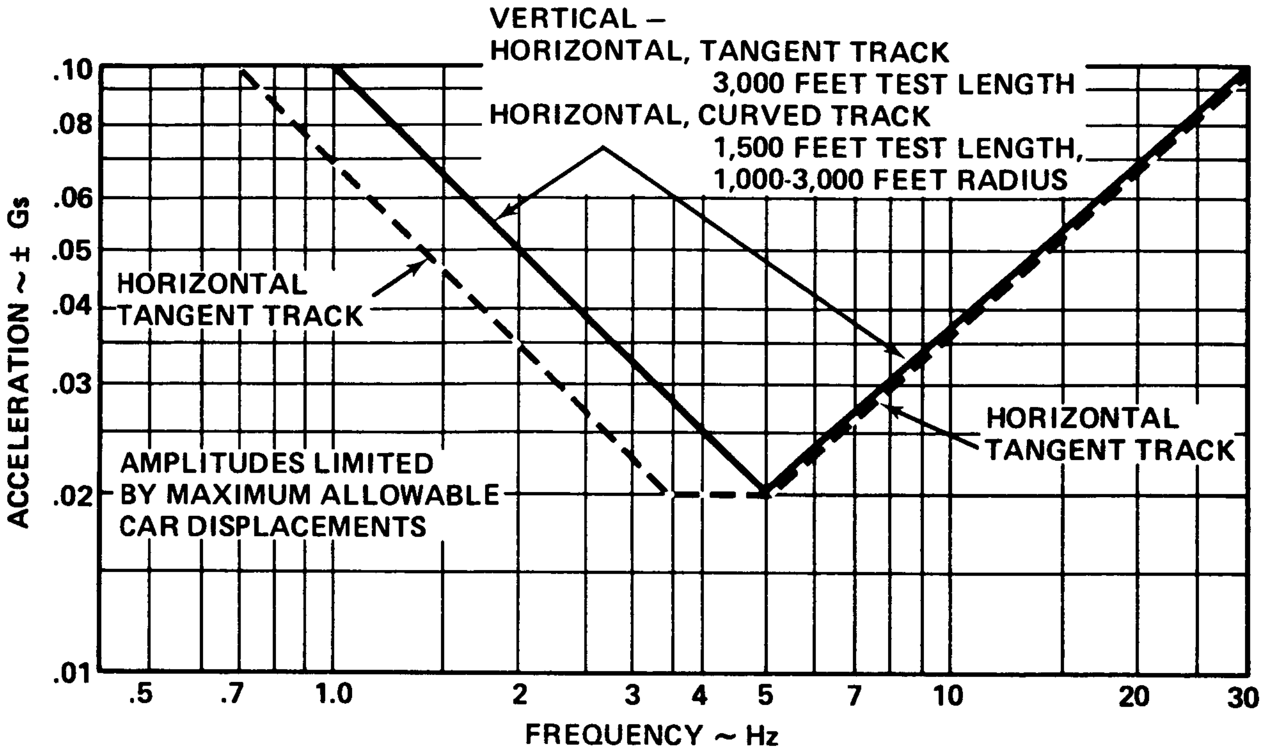

Figure 5.- SOAC ride quality goals. 1 ft = 0.3048 m.

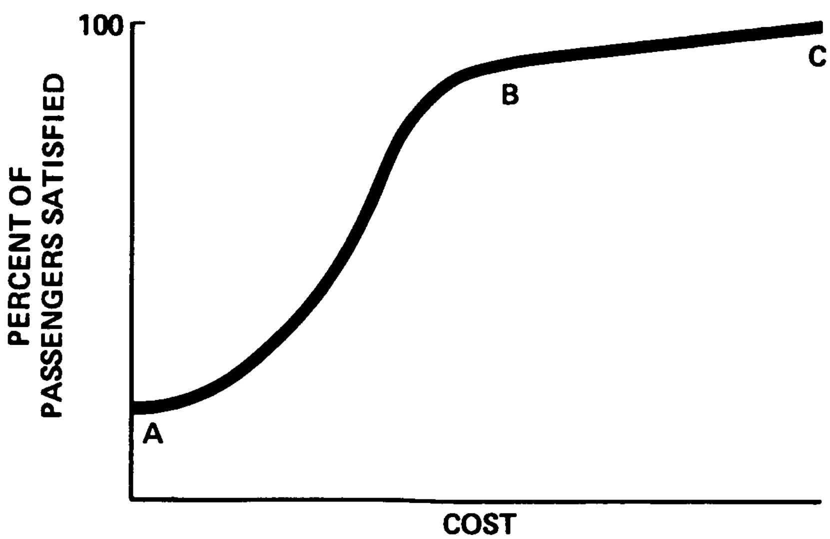

Figure 6.- Cost of satisfying passengers.

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RIDE QUALITY CRITERIA AND THE DESIGN PROCESS

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SUMMARY

Conceptual designs for advanced ground transportation systems often hinge on obtaining acceptable vehicle ride quality while attempting to keep the total guideway cost (initial and subsequent maintenance) as low as possible. Two ride quality standards used extensively in work sponsored by the U.S. Department of Transportation (DOT) are the DOT-Urban Tracked Air Cushion Vehicle (UTACV) standard and the International Standards Organization (ISO) reduced ride comfort criteria.

These standards are reviewed and some of the deficiencies, which become apparent when trying to apply them in practice, are noted. Through the use of a digital simulation, the impact of each of these standards on an example design process is examined. It is shown that meeting the ISO specification for the particular vehicle/guideway case investigated is easier than meeting the UTACV standard.

INTRODUCTION

One of the more difficult problems associated with the conceptual design of advanced transportation systems is achieving acceptable ride quality while simultaneously avoiding guideway structures which will be expensive to build and maintain. In analyzing new transportation concepts, especially those involving elevated guideway structures, the analyst must consider the vehicle/guideway combination as a system. The elevated guideway structure will deflect elastically under the moving vehicle(s) load, thus introducing a periodic disturbance input to the vehicle. In addition, those guideway inputs broadly classified as "roughness" also introduce undesirable vehicle motion. In the case of elevated guideway structures, it has been shown [1]* that roughness can be directly related to such guideway construction tolerances as pier survey error, pier settlement, camber and surface finish. It has also been demonstrated that guideway flexibility and construction tolerances can, in turn, be related to the ride quality performance of the vehicle. The most commonly employed ride quality standards in research sponsored by the U.S. Department of Transportation include the Urban Tracked Air Cushion Vehicle (UTACV) specification [2] and the International Standards Organization (ISO) criteria [3]. In previous work on conceptual vehicle/guideway design [1,4], some difficulties were encountered by the author in applying the aforementioned standards. In particular, when finite acceleration time histories were obtained from a vehicle/guideway computer simulation [1,4], a lack of preciseness in the accompanying instructional material for both the UTACV and ISO criteria seemed to permit a wide area of judgment to be exercised by the user. Thus, it seemed possible that two analysts working with the same data record could arrive at different conclusions regarding ride quality compliance. It was also discovered that an identical acceleration record could be in compliance with one standard

*Numbers in brackets indicate references.

and not the other. These quantifiable problems, in addition to some subjective observations concerning the UTACV and ISO standards, will be discussed in the following sections of the paper.

VEHICLE/GUIDEWAY COMPUTER SIMULATION

It is worthwhile to devote a brief section of the paper to discuss the origin of the acceleration time histories to be discussed. A digital computer simulation was developed for the purpose of studying conceptual vehicle/guideway systems with the ultimate objective of relating vehicle ride quality to guideway design parameters and construction tolerances [1]. Figure 1 is a schematic representation of the simulation and illustrates the significant program elements and input/output quantities. In particular, the simulation was used to study the performance of the conceptual air cushion vehicle/guideway configuration whose major system parameters are listed in Table 1. The vehicle properties are essentially self explanatory while the guideway parameters include the span fundamental bending frequency and the pier spacing (span length) ℓ. The construction tolerance parameters are also shown in table 1 and they represent, in general, maximum expected values in a statistical sense; a detailed discussion can be found in Reference 1. It should be emphasized that the vehicle acceleration records obtained from such a complete vehicle/guideway simulation should be typical of records obtained from actual vehicle test runs. The problems of processing this acceleration data in the UTACV and ISO formats and the areas open to interpretation (or misinterpretation) will be discussed in the following sections.

THE UTACV RIDE QUALITY STANDARD

In part, the UTACV ride quality standard [2] requires that the spectral composition of passenger cabin acceleration time histories, in a spectral density format, must not exceed the boundaries shown in Figure 2 over the frequency range 0.1 to 50 Hz. The only requirements on processing the acceleration data are that the time history shall be 30 seconds or longer and that the spectral density calculation be made with a frequency resolution of 1.0 Hz, and plotted at 1.0 Hz and each succeeding integral frequency in the passband. Considering the problems associated with estimating spectral densities from finite data records, the above stated UTACV guidelines are at best not sufficient. First, there is no information as to which estimation procedure (Blackman-Tukey, Fast Fourier Transform or direct band pass filtering) is to be used in a given situation. This is important as these methods can sometimes produce somewhat different results. Second, guidelines related to the original data collection and the required accuracy of the spectral density estimate are not given. In order to expand on this, it is necessary to introduce some important parameters. The length of the original data record T is given by:

$$T = Nh \tag{1}$$

where N is the number of data points and h is the sample spacing. The standard (statistical) error ε is a measure of the "goodness" of the spectral density estimate and is given by [5]

$$\varepsilon = \sqrt{1/B_e T} \tag{2}$$

or from eq. (1),

$$\varepsilon = \sqrt{1/B_e Nh} \tag{3}$$

where B_e is the frequency resolution bandwidth. A seemingly curious fact is that the finer the resolution, the larger the error; this is related to the problem of trading off frequency resolution for the confidence level in the final spectral density estimate. Note that the UTACV specification does implicitly set a maximum allowable standard error by fixing B_e = 1.0 Hz and T to at least 30 seconds; thus

$$\varepsilon_{UTACV} = \sqrt{\frac{1}{30}} = 0.183$$

An alternate way of expressing statistical error is to specify the statistical degrees of freedom (sdof) n where [5]

$$n = 2/\varepsilon^2 \tag{4}$$

For the UTACV specification,

$$n_{UTACV} = 60$$

For any specification employing spectral densities, the values of ε and/or n should be explicitly stated since their importance may not be apparent to the uninitiated user employing a packaged processing routine. For example, Figure 3 depicts two acceleration spectral density estimates computed from the same record.[1] The differences in the spectral density estimate for the case where ε = 0.22 (n=40) and the case where ε = 0.16 (n=80) are significant. Only the latter case satisfies the implicit UTACV accuracy requirement. Some insight into the meaning of standard error or, equivalently, sdof, can be gained from Figure 4 taken from Reference 6; it may be seen that 60 sdof corresponds to a 50% confidence level that the spectral density estimate will be within ±12% of the true value and a 90% confidence level that the estimate will be within ±30% of the true value. Similarly, 40 sdof corresponds to a confidence level of 90% that the estimate will be within ±37% of the true value. Some other interesting relationships [5,6] are as follows: The lowest frequency f_ℓ in the spectral density band of interest is related to B_e through:

$$f_\ell = B_e/2 \tag{5}$$

The cut-off (Nyquist) frequency f_c for the case of a vehicle traveling at velocity V is:

$$f_c = V/2h \tag{6}$$

and the frequency f_h below which no aliasing (frequency masking) is to occur is:

$$f_h = f_c/p, \; 3/2 \leq p \leq 2 \tag{7}$$

---

[1]The acceleration record is based on the vehicle/guideway parameters in Table 1 and a vehicle speed of 121 km/hr (75 mph).

The higher value of p gives the least amount of aliasing below f_h. Surprisingly, the UTACV specification gives no guidelines with respect to eliminating the potentially significant aliasing errors. Two interesting points with respect to the above formulas (5) through (7) should be noted: First, from eq. (5), substitution of the UTACV specified value of B_e = 1.0 Hz yields:

$$f_\ell = 0.5 \text{ Hz}$$

The UTACV specification however requires that f_ℓ = 0.1 Hz. and is therefore not self consistent. Secondly, equations (6) and (7) yield:

$$h = V/2p\ f_h.$$

Certainly if the upper frequency of interest for the UTACV specification is as stated, 50 Hz, then one would want no aliasing below 50 Hz; thus, setting p = 2 and f_h = 50 Hz.,

$$h = \frac{V}{200}$$

The time Δt between samples is then

$$\Delta t = h/V = 1/200 = .005 \text{ sec.}$$

For a 30 second record, 6000 data points are therefore required. Thus, the UTACV upper frequency of interest (50 Hz) can put a difficult and perhaps unnecessary burden on data collection requirements. In particular, for digital computer simulations, such requirements are costly. Moreover, most reasonably designed passenger compartments will be isolated from any significant 50 Hz disturbances and as 50 Hz is well beyond the critical frequency range for humans, there is no apparent reason why it should not be relaxed.

It seems clear that the lack of explicit guidelines within the UTACV standard, coupled with some inconsistencies and possible impractical data requirements, force the user to resort to "best engineering judgment". Under such conditions, it is not at all clear what it means to "meet" or "violate" the UTACV ride quality specification.

A more subjective observation about the UTACV specification is that its popularity with many users is based on the fact that guideway roughness disturbances are often presented in spectral density format: therefore, if $G(\omega)$ is the guideway spectral density input, and $H(j\omega)$ is the vehicle/suspension/ guideway transfer function, the acceleration spectral density for linear, stationary, ergodic systems is:

$$A(\omega) = |H(j\omega)|^2\ G(\omega).$$

As $A(\omega)$ can be compared directly with the UTACV limit, the UTACV standard is particularly convenient for the analyst. This approach is however, somewhat simplistic and tends to obscure some of the thorny problems associated with processing actual data. A few other personal objections are, first, that there is no way to judge the relative ride quality for differing acceleration spectral densities which, in part, are due to the lack of any time element associated with the UTACV limit; i.e., will the ride be comfortable for 15 minutes, 1 hour, 3 hours? Second and most important, it is also due in part to the lack of any statistical data on what percentage of the population could be reasonably expected to find the UTACV ride comfortable.

THE ISO SPECIFICATION

The ISO ride comfort specification [3] for vertical acceleration is illustrated in Figure 5. More correctly, the solid lines represent "fatigue reduced proficiency" limits as a function of time spent in the vibration environment. The boundaries limit rms acceleration as a function of frequency within the range 1.0-80. Hz. According to the ISO specification, the fatigue reduced proficiency boundary "specifies a limit beyond which exposure to vibration can be regarded as carrying a significant risk of impaired working efficiency in many kinds of tasks." The reduced comfort boundary is assumed to lie approximately at one-third (10 dB below) of the reduced proficiency boundary. For example, the one hour ISO reduced ride comfort boundary is illustrated by the dashed line in Figure 5. The TRW Corp. [7] has proposed an extension of the ISO specifications to below 1.0 Hz in order to limit acceleration at those frequencies where motion sickness is known to occur. When processing a finite data record, the user follows Paragraph 4.2 in the ISO Specification Document [3] and some problems arise almost immediately. For example, Par. 4.2.1 claims that the limits depicted in Figure 5 are valid for discrete frequency vibrations. Par. 4.2.2 states that when there are discrete multiple frequencies, i.e., "vibration present simultaneously at more than one discrete frequency in the range 1.0 to 80 Hz." the evaluation of ISO ride quality compliance is accomplished as follows: "The rms acceleration of each frequency component shall be evaluated separately with reference to the appropriate limit at that frequency". This instruction is disturbing for two reasons: Since no limit on the number of simultaneous multiple frequencies which can be processed is given, it implies (1) that there is no cumulative effect of multiple frequencies and (2) that any number of multiple frequency components, including those obtained from Fourier analysis could be evaluated according to Par. 4.2.2. Since acceleration output from the vehicle/guideway simulation discussed earlier was routinely processed by Fourier analysis, it was felt that the rms acceleration Fourier components at each frequency could be evaluated according to the ISO instructions for multiple frequency inputs. This procedure was initially adopted and applied to the nominal vehicle/guideway case defined in Table 1, with V = 242 km/hr (150 mph). The results are shown in Figure 6. It quickly became apparent, however, that for longer acceleration time histories of the same steady state vehicle response, the discrete frequency resolution became finer (more frequencies) and the rms value of each frequency component decreased, making it easier to "meet" the ISO specification. Obviously, the applicable paragraph for this situation is Par. 4.2.2 entitled "Broad-band Vibration"; this states that the rms acceleration in each 1/3 - octave band is to be evaluated separately with respect to the appropriate limit at the center frequency of that band. The result of this procedure is also illustrated in Figure 6. Nevertheless, Par. 4.2.2 on multiple frequencies remains vague for application purposes, and unsatisfying with regard to an intuitive feeling that a cumulative effect should exist. Other features lacking in the ISO specification include a limit on sustained acceleration and jerk (time derivative of acceleration) associated with vehicle operations such as starting and braking. As in the UTACV specification, statistical data on what segment of the population could be reasonably expected to find the ISO ride comfortable is not given.

EXAMPLE

Using the simulation discussed earlier in the paper, a design exercise was carried out to determine the effect of changing the guideway pier spacing on vehicle ride quality compliance. It is useful to know the range of acceptable span lengths (pier spacing) since a detailed design will generally involve an optimum economic pier spacing which involves a trade-off between smaller span cross sections (less span material) and shorter span lengths (more piers). A convenient non-dimensional span length parameter is the vehicle-to-guideway span length[2] ratio L. The nominal value for the case illustrated in Table 1 is $L = 0.3$. Two additional cases were run with $L = 0.24$ and $L = 0.4$. Conditions of interest included the situation where guideway flexibility is the only disturbance, and where both flexibility and nominal construction tolerances (see Table 1) are included. Figure 7 shows the acceleration response spectral density due to the flexibility input only for V = 242 km/hr (150 mph) and $L = 0.24$, $L = 0.3$, and $L = 0.4$. There is no major effect on UTACV ride quality compliance the significant difference being in the shift of the major peak. The frequency f_p associated with this peak is given by $f_p = V/\ell$. When the nominal construction misalignments are added, Figure 8 illustrates that there is a broader band response characteristic and that the UTACV ride quality standard is violated in all three cases. Moreover, the shorter pier spacing ($L = 0.4$) produced the "rougher" ride[3] despite the greater relative flexural rigidity associated with shorter span lengths. The reason for this is that the pier survey and pier settlement misalignments, assumed equivalent over the range $0.24 \leq L \leq .4$, produced a rougher guideway profile over the shorter span length. Comparing Figure 8 to Figure 6 however, it is clear that for the nominal case (including flexibility and construction misalignment) the one-hour ISO specification is easily met while the UTACV limit is exceeded. At this time there is no definite information on the relative stringency or leniency in the two standards. It would be costly, perhaps prohibitively so, to design new systems to meet ride quality standards which are overly stringent. On the other hand, people cannot be subjected to rides which are so rough, that the passenger feels in imminent danger. From a personal viewpoint, the major fault in the ISO and UTACV ride quality standards is the lack of statistical information which would provide system planners and designers with some information on what percentage of the population would refuse to ride a vehicle meeting a particular ride quality standard. Only then can the systems analyst determine the trade-off between increased cost to meet a more stringent standard and the additional revenues to be derived from increased ridership.

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[2]Vehicle length is conveniently taken as the length between fore and aft suspension mid-points; in the case of an automobile this would be the wheelbase.

[3]"Rougher" ride is arbitrarily defined in this paper by the higher rms acceleration level.

CONCLUSIONS

Designing advanced transportation systems to meet ride quality standards will place requirements on the vehicle suspension system, guideway construction tolerances and subsequent guideway maintenance. These requirements have a significant impact on initial capital costs and ongoing maintenance costs. It therefore seems imperative that reliable and meaningful ride quality criteria be developed. While the obvious operational shortcomings in the ISO and UTACV specification can be remedied, the lack of information on the percentage of ride quality acceptability by the population cannot be easily retrieved, thereby depriving the designer of important trade-off information. It is hoped that this paper has adequately emphasized the importance of meaningful ride quality criteria in terms of the potential design and cost impacts involved and that it has also illustrated the need for precise guidelines and language with regard to processing vehicle data in the required ride quality format.

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TABLE 1

EXAMPLE SYSTEM PROPERTIES

PASSENGER CABIN MASS = 22664 kg (1553 slugs)

PASSENGER CABIN PITCH MOMENT OF INERTIA = 1.7×10^5 kg-m^2 (1.5×10^6 lb-in-sec^2)

CUSHION MASSES(2) = 569 kg (39 slugs)

SUSPENSION BASE = 9.2 m (30 ft)

UNSPRUNG CABIN HEAVE FREQUENCY = 1.0 Hz

NOMINAL PIER SPACING = 30.5 m (100 ft)

NOMINAL SPAN FUNDAMENTAL FREQUENCY = 3.52 Hz

MAXIMUM PIER SURVEY ERROR = 1.27 cm (0.5 in.)

MAXIMUM CAMBER TOLERANCE ERROR = 20%

MAXIMUM ALLOWABLE PIER SETTLEMENT = 1.91 cm (0.75 in.)

CALIFORNIA PROFILE INDEX (SURFACE FINISH) = 2.5 cm/km (1.6 in./mi)

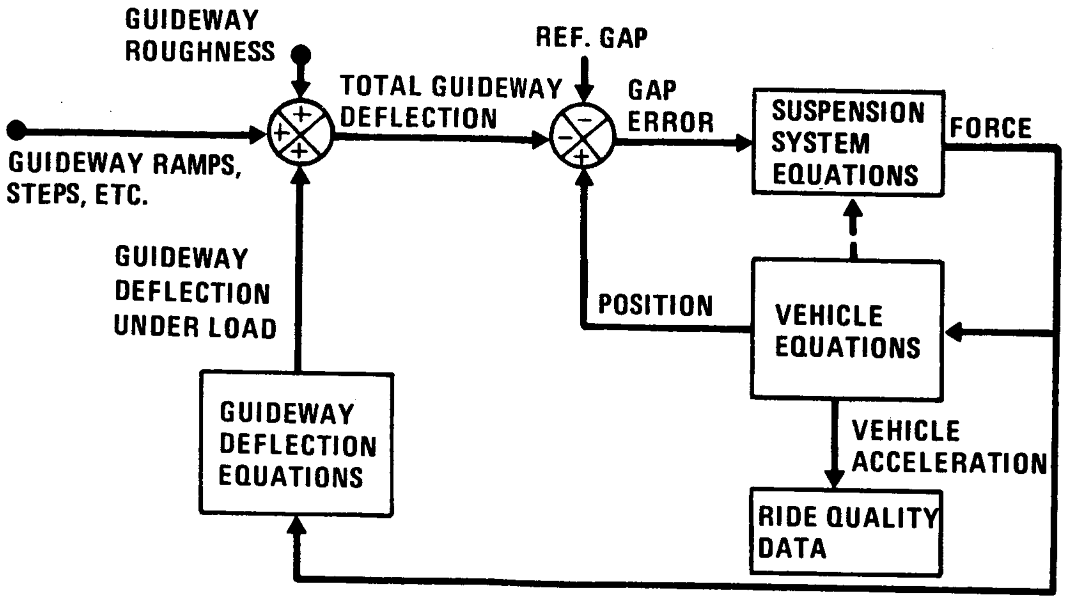

Figure 1 - General Simulation Flow Diagram

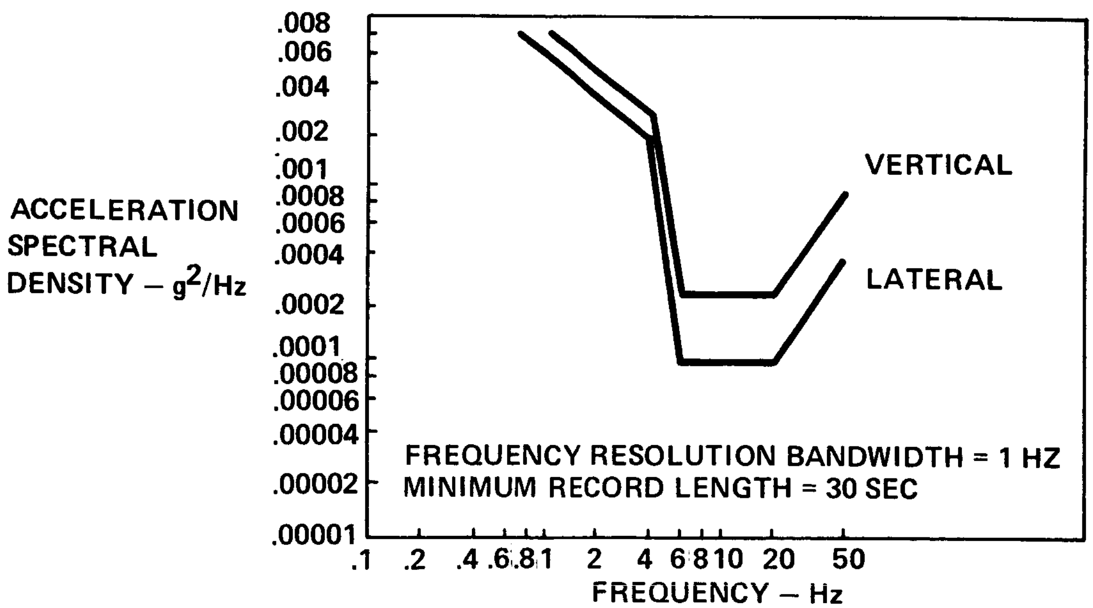

Figure 2 - The UTACV Ride Quality Standard

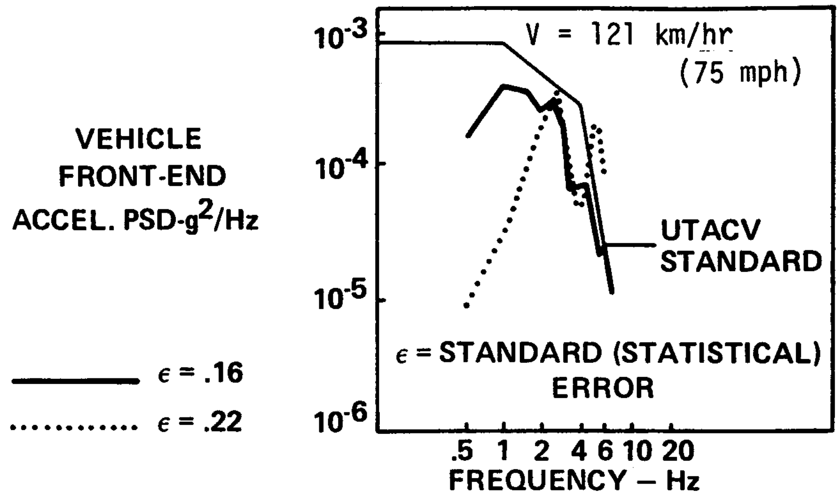

Figure 3 - Effect of Standard Error on PSD Estimate

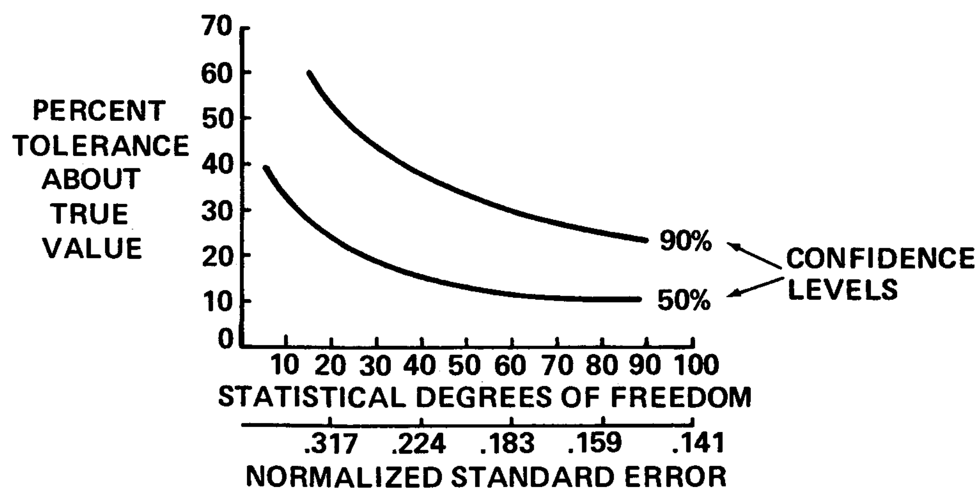

Figure 4 - Relationship Between Standard Error on Confidence in PSD Estimate

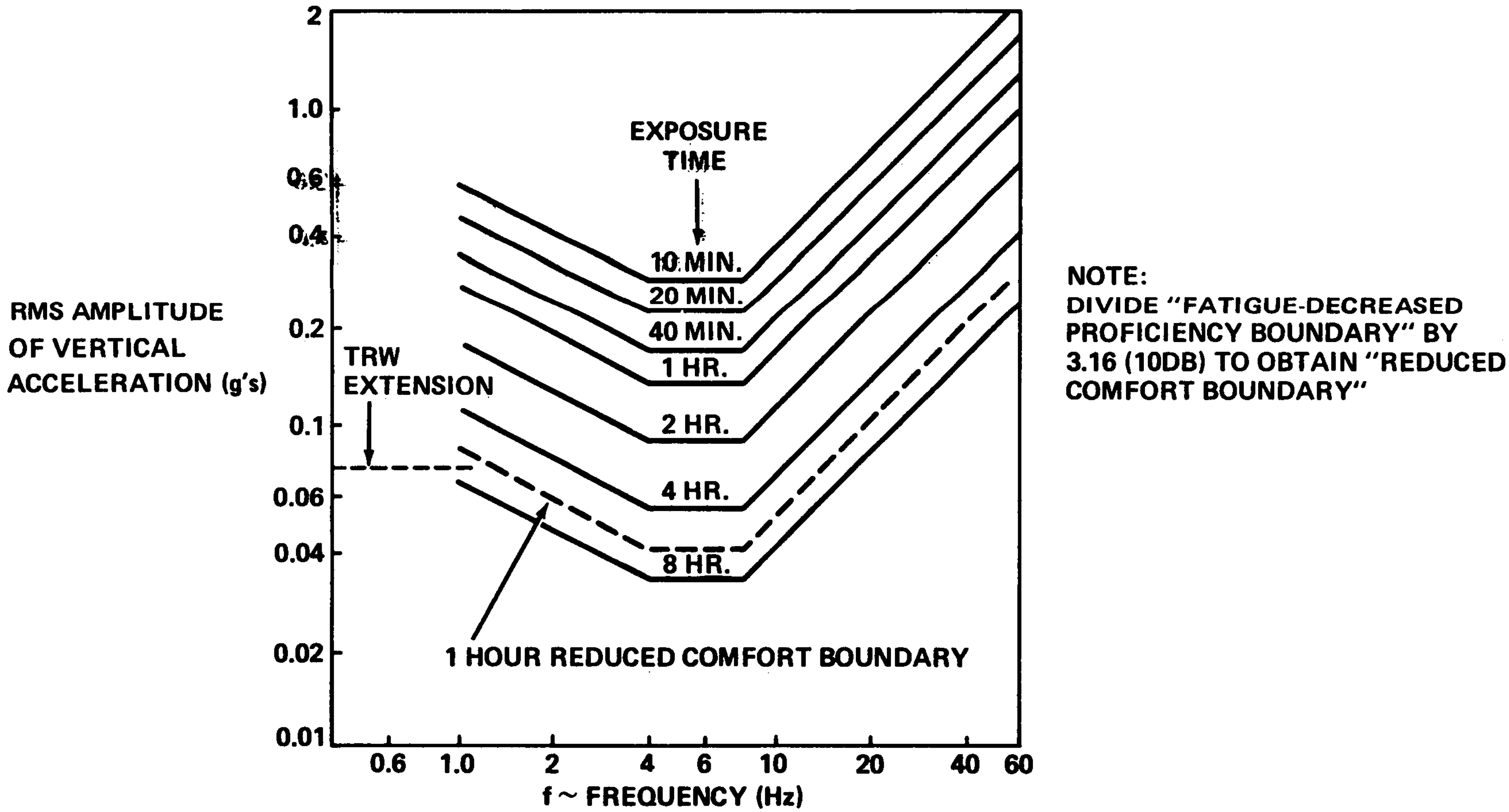

Figure 5 - The ISO Ride Quality Standard

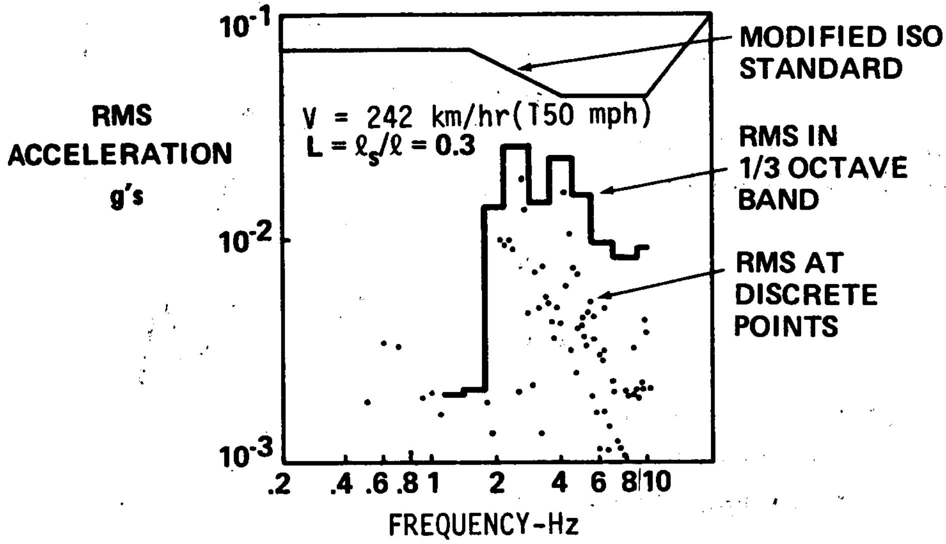

Figure 6 - Cabin Acceleration Spectrum in the ISO Format - Guideway Flexibility and Roughness Included

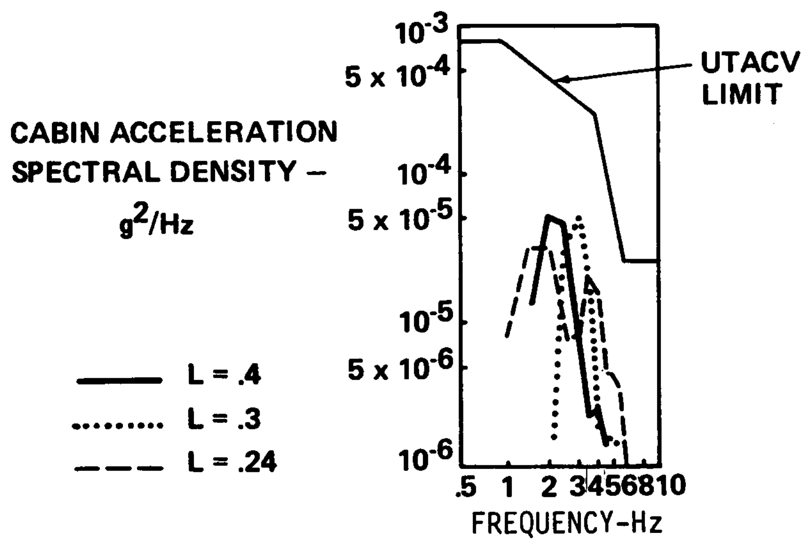

Figure 7 - Cabin Acceleration PSD Due to Guideway Flexibility Only: V = 242 km/hr (150 mph)

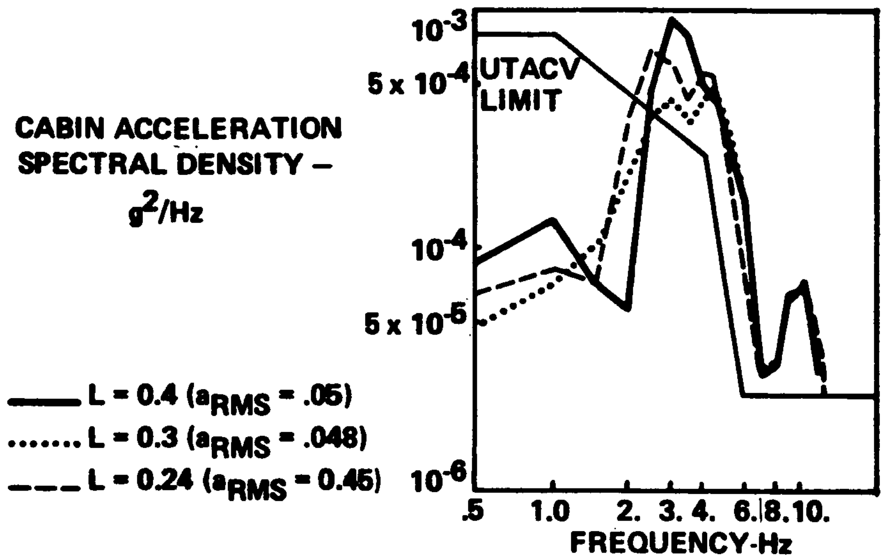

Figure 8 - Cabin Acceleration PSD Due to Guideway Flexibility and Roughness: V = 242 km/hr (150 mph)

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APPLICATION OF RIDE QUALITY TECHNOLOGY TO PREDICT RIDE SATISFACTION FOR COMMUTER-TYPE AIRCRAFT

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SUMMARY

A method has been developed to predict passenger satisfaction with the ride environment of a transportation vehicle. This method, a general approach, has been applied to a commuter-type aircraft for illustrative purposes. Here the effect of terrain, altitude and seat location were examined. The method predicts the variation in passengers satisfied for any set of flight conditions. In addition several non-commuter aircraft were analyzed for comparison and other uses of the model described. The method proposed has advantages for design, evaluation, and operating decisions.

INTRODUCTION

The purpose of this paper is to provide a method of assessing passenger satisfaction with the ride quality on transportation vehicles. The method is applicable to both existing systems as well as future ones, and can be used for evaluation, design and decision making. Basically it relates the environment in which the vehicle must be used and the performance characteristics of the vehicle to determine the probability of satisfying the passenger.

This analysis is based on previous work by the authors in assessing vehicle ride quality for the air mode. In refs. 1 and 2, a model of passenger comfort and satisfaction with a ride as a function of the motion of the vehicle was developed. This model coupled with standard techniques for analyzing a vehicle's motion allow us to examine such variables as: vehicle type, input forcing functions, operating characteristics, etc.

The method will be applied to commuter-type aircraft and variations in passenger satisfaction due to terrain, altitude, equipment and location in the vehicle described. Other uses of the technique are also suggested.

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SYMBOLS

a	acceleration
C	comfort rating
f	joint probability density function
L	length scale for turbulence spectrum
S	percent of passengers satisfied with ride
V_0	velocity of vehicle
v_g	turbulence gust velocity in transverse direction
w_g	turbulence gust velocity in vertical direction
μ	mean
ρ	correlation coefficient
σ	standard deviation for accelerations, rms for turbulence quantities
ϕ	power spectral density
ω	frequency
$\underline{\omega}$	angular velocity

Subscripts:

a_y	transverse rms acceleration
a_z	vertical rms acceleration
x	longitudinal direction
y	transverse direction
z	vertical direction

METHOD

Description

The method of analysis is shown in figure 1. A vehicle forcing function is converted into motion cues to the passenger using the appropriate transfer functions for the system being analyzed. Typical forcing functions are illustrated in table I along with the important properties of the transfer functions for several vehicles.

Table I

Vehicle	Primary Forcing Function	Characteristics of Transfer Function
Airplane	Atmospheric Turbulence	Aerodynamics, Mass Properties
Train	Rail Profile	Suspension System, Wheel Configuration, Mass Properties
Bus, Automobile	Road Surface	Suspension System, Wheel Configuration, Mass Properties
Ship	Sea Surface	Hydrodynamics, Mass Properties

In most cases the vehicle engines also contribute to the motion experiences (e.g. vibrations) however their amplitudes and frequencies compared with the primary forcing function shown above are usually negligible.

Vehicle functions generally depend on frequency; thus, both amplitude and frequency information of the input; or, the input power spectrum is necessary for the analysis. In addition the inputs can and usually are statistically varying quantities so that a probability density function for each of the inputs is necessary. In fact, as will be seen below, the method described allows for isolating components of the forcing function which contribute most to passenger dissatisfaction. In some cases this information may be used to find ways to improve the ride environment (e.g. treatment of roadways or active ride smoothing on the vehicle).

Vehicle motion can take the form of velocities, accelerations and rates of change of acceleration in each of six-degrees-of-freedom. Not all of them are appropriate for the ensuing analysis and only those needed in the subjective transfer function must be determined. In general the passenger's comfort will be functionally related to the motion parameters of angular velocity and linear acceleration and their derivatives

$$C = f(a_x, a_y, a_z, \underline{\omega}_x, \underline{\omega}_y, \underline{\omega}_z, \dot{a}_x, \dot{a}_y, \dot{a}_z, \dot{\underline{\omega}}_x, \dot{\underline{\omega}}_y, \dot{\underline{\omega}}_z) \qquad (1)$$

where C is the subjective comfort rating, a_x, a_y, a_z linear accelerations in the longitudinal, lateral and vertical directions respectively, and $\underline{\omega}_x, \underline{\omega}_y, \underline{\omega}_z$ angular velocities about the longitudinal, lateral and vertical axes respectively. The $\cdot$ denotes a time derivative, thus $\dot{a}_x$ is the longitudinal jerk, etc. The comfort model can be a simple function of rms motion variables through a more complex frequency-dependent psychophysical model (see e.g. ref. 2). In the "real world" other factors also contribute to the passenger's comfort (e.g. noise, temperature, etc.) however these will be neglected here. A more complete analysis should include them.

The mathematical procedure for arriving at the comfort rating is straightforward but somewhat tedious to perform. The joint probability density function for the motion variables, $f(a_x, a_y, a_z, \ldots)$ is integrated over motion space to arrive at a probability function for the passenger's comfort level. That is, the probability that the comfort rating is less than or equal to some value C' is given by

$$P(C \leq C') = \int_{\dot{\underline{\omega}}_z=0}^{C'-\bar{C}_{\underline{\omega}_z}} \int \cdots \int_{a_x=0}^{C'-\bar{C}_{a_x}} f(a_x, a_y, a_z, \ldots, \dot{\underline{\omega}}_z)\, da_x\, da_y\, da_z \ldots d\dot{\underline{\omega}}_z \qquad (2)$$

where $\bar{C}_i$ is the value of the associated motion variable given by the comfort equation, each one being eliminated as the integration progresses. Since the motion can vary with location in the vehicle, the above analysis must be repeated at each station of interest.

The last step in the analysis relates the derived comfort rating to a value judgement. This value judgement is taken to be passenger satisfaction with the ride which is related to comfort rating. The percentage of passengers satisfied, S, is a simple function

$$S = f(C). \qquad (3)$$

Thus for any comfort rating the value judgement transfer function transforms C to S by the above equation. The actual decision process is much more complex, being dependent on other variables as well as competing modes. These have been neglected in this analysis, assuming that if a passenger were dissatisfied a sufficient number of times he would seek an alternate means of reaching his destination.

The remainder of this paper will apply this method to a particular vehicle type--commuter aircraft--however it is important to note that the method is by no means restricted to this mode. At the present time this is the only mode for which data were available.

Application to Commuter-Type Aircraft

Input Forcing Function

For aircraft the input forcing function is atmospheric turbulence, which can be characterized by velocity power spectra in all six-degrees-of-freedom, longitudinal, lateral, vertical, pitch, roll, and yaw. However, since previous work (refs. 1 and 2) has shown that the comfort models require only vertical and lateral linear accelerations, only these components of the turbulence field will be considered. The amplitude probability as well as the frequency content are functions of terrain, altitude, and weather. Typical examples are shown in figure 2, where the variation in vertical, σ_{w_g}, and lateral, σ_{v_g}, rms gust intensity is seen versus altitude for mountain terrain (refs. 3 and 4). Similar curves are available for water and flat terrain. The power spectra for these are given by a Dryden model

$$\phi(\omega) = \sigma^2 \frac{L}{\pi V_0} \frac{1 + 3\left(\frac{L\omega}{V_0}\right)^2}{\left[1 + \left(\frac{L\omega}{V_0}\right)^2\right]^2} \tag{4}$$

where V_0 is the aircraft velocity, ω, the frequency, σ the rms gust intensity and L, the length scale which is a function of altitude (ref. 4). A typical power spectrum is shown in figure 3.

Vehicle Transfer Function

Aircraft transfer functions are a function of aerodynamics and mass properties and can be found in many references (see e.g. ref. 5). Here we assume a rigid body model (no structural bending) and neglect gyroscopic effects. The particular vehicle first considered is the deHavilland Twin Otter aircraft, which was selected because of the abundance of data available concerning its aerodynamic characteristics. It is regrettable that functions for the aircraft suitable for potential use in the commuter market are not readily available.

Motion Spectra of Vehicle

The outputs of interest for the comfort model to be used below are the rms accelerations in the vertical and lateral directions. These can be obtained by integrating their power spectral densities over frequency space which are given by

$$\phi_{a_z}(\omega) = \left|\frac{a_z}{w_g}\right|^2 \phi(\omega) \tag{5a}$$

$$\phi_{a_y}(\omega) = \left|\frac{a_y}{v_g}\right|^2 \phi(\omega) \tag{5b}$$

where $\phi_{a_z}(\omega), \phi_{a_y}(\omega)$ are the power spectral densities for vertical and lateral accelerations, and $\left|\frac{a_z}{w_g}\right|$ and $\left|\frac{a_y}{v_g}\right|$ are the transfer functions for these accelerations relating them to the turbulence field. For the Twin Otter, the rms acceleration cumulative probability distribution is shown in figure 4 for a typical case. Typical spectra for these accelerations are given in figure 5.* As can be seen in figure 6 the acceleration in the vertical direction closely approximates a normal (Gaussian) distribution. The same behavior can be seen for actual flight data in reference 6. Transverse acceleration behaves similarly. This allows us to write the probability density functions for each separately and for both combined using a normal distribution. From flight data (refs. 2 and 6) the cross correlation between vertical and lateral accelerations is 0.8 thus the joint probability distribution function is given by

$$f(a_y, a_z) = \frac{1}{2\pi\sigma_{a_y}\sigma_{a_z}} \cdot \frac{1}{\sqrt{1-\rho^2}} \exp\left\{-\frac{1}{2(1-\rho^2)}\left[\left(\frac{a_y-\mu_{a_y}}{\sigma_{a_y}}\right)^2 - \frac{2\rho(a_y-\mu_{a_y})(a_z-\mu_{a_z})}{\sigma_{a_y}\sigma_{a_z}} + \left(\frac{a_z-\mu_{a_z}}{\sigma_{a_z}}\right)^2\right]\right\} \tag{6}$$

where μ_{a_y}, μ_{a_z} are the mean rms accelerations, $\sigma_{a_y}, \sigma_{a_z}$ are the standard deviations of rms accelerations, and ρ is the correlation coefficient between accelerations. The values for the μ's and σ's for different terrain, altitude and vehicle location can be found by computing values of the motion variables for $P(a) = .5$ and .84 respectively.

Subjective Transfer Function

A subjective comfort model has been developed (ref. 2) based on extensive field data taken on commercial airlines (refs. 6, 7, 8) and in-flight simulator (ref. 9) experiments. This model relates the subjective comfort response to rms vertical and transverse accelerations in g's as

$$C = 2 + 11.9a_z + 7.6a_y \text{ when } a_z \geq 1.6a_y, \tag{7a}$$

and

$$C = 2 + a_z + 25a_y \text{ when } a_z < 1.6a_y, \tag{7b}$$

*The contribution to the rms acceleration for the vertical direction can be divided into two frequency regimes-below and above 1 rad/sec. The region above 1 rad/sec contributes 88 percent of the total power and is thus more important in determining comfort.

where C is restricted to values 2 through 5, with the following descriptors:

$$
\begin{aligned}
C &= 2 \quad \text{comfortable} \\
&= 3 \quad \text{neutral} \\
&= 4 \quad \text{uncomfortable} \\
&= 5 \quad \text{very uncomfortable.}
\end{aligned}
$$

For motions in which vertical acceleration dominates (i.e. $a_z \geq 1.6a_y$), subjective judgments lean more heavily toward the vertical stimulus, however transverse acceleration is more important otherwise. For pure motion in either direction these models predict twice the sensitivity to the transverse direction compared with the vertical direction.

Comfort Determination

Using equations 7 we compute the comfort rating corresponding to any given vertical and transverse accelerations. However the accelerations are described by the joint probability distribution function (given in equation 6). Thus the probability of exceeding a given comfort level C' is obtained from equation 6 using equations 7 to describe the integration space as

$$
\begin{aligned}
P(C \geq C') = &\int_0^{a_{y_0}} \int_{1.6a_y}^{(C'-2-7.6a_y)/11.9} f(a_y,a_z)\,da_y\,da_z \\
&+ \int_0^{a_{z_0}} \int_{a_z/1.6}^{(C'-2-a_z)/25} f(a_y,a_z)\,da_z\,da_y .
\end{aligned} \tag{8}
$$

For the Twin Otter aircraft the first, μ, and second, σ, moments describing the probability distribution f are given in table II, for the center of gravity of the aircraft. Similar data have been generated for other positions within the craft.

Table II

	Terrain					
	Mountain		Water		Flat	
Altitude	152 m	3,048 m	152 m	3,048 m	152 m	3,048 m
μ_{a_z} (g's)	.055	.035	.019	.012	.048	.031
μ_{a_y} (g's)	.015	.0094	.0051	.0032	.013	.0082
σ_{a_z} (g's)	.024	.015	.0052	.0033	.019	.012
σ_{a_y} (g's)	.0066	.0041	.0014	.00088	.0051	.0032

Equation 8 is numerically integrated to determine the cumulative probability distribution for each case of interest (i.e. terrain, altitude, seat location). A typical result is shown in figure 7 which illustrates the variation due to terrain for a fixed altitude and location within the aircraft. Thus for this case there is a 90% probability that the subjective comfort rating will be less than 2.3 for flight over water, less than 2.75 for flight over flat terrain, and less than 2.9 for flight over mountain terrain. Stated alternatively the probability that the comfort rating of the ride will be less than or equal to 2.5 is 100%, 54%, and 42% for flight over water, flat and mountainous terrain respectively. Similar comparisons can be made for any set of conditions or to compare different aircraft for a single set of conditions.

Value Transfer Function

The calculated comfort judgments must now be related to a more value-oriented variable. We choose as this quantity the percentage of passengers satisfied with the ride, that is, the fraction of passengers who would willingly take another flight at least without hesitation. This quantity has been determined in previous work (ref. 1) to be related to the subjective comfort rating as shown in figure 8. As can be seen, from a statistical point of view, there are approximately 7% of the passengers who will not be satisfied with the ride environment even when the ride is rated comfortable by most of the passengers. This is seen more clearly when examining distributions of passenger responses (see e.g. ref. 2).

This transfer function, figure 8, has been applied to data on subjective comfort responses, to obtain the probability of satisfying a given percentage of the passengers. Typical graphs are given in figures 9 and 10 for the aircraft center of gravity and an extreme aft seat location as a function of terrain and altitude. As an example of using these graphs, they indicate that at the center of gravity there is a 45% probability of satisfying at least 85% of the passengers flying at 152 m over mountain terrain, while there is an 84% probability of satisfying the same number of passengers flying at 3,048 m over mountain terrain. Similarly over the same terrain at 152 and 3,048 m respectively there is a 36% and 78% probability of satisfying 85% or more of the passengers at an aft seat location. This illustrates that a) the aft seat locations are less comfortable than those near the center of gravity, and b) flying at higher altitudes increases the probability of satisfying passengers. Thus the more conservative approach would be to design to the low altitude, aft seat location results.

Comparison to Other Aircraft

Several other aircraft have been analyzed using the method described. Transfer functions were obtained from references 10 and 11. The aircraft are the Breguet 941, Douglas DC-8, Cessna 182, and an externally blown flap (EBF) aircraft still in the design stage. These aircraft have the following characteristics:

Aircraft	Weight (kg)	Approximate No. of Passengers
DC-8	91,000	200
Cessna 182	1,360	4
Breguet 941	20,000	45
EBF	122,000	270

Figure 11 illustrates the variation in percent satisfied by aircraft type for cruise at 3,048 m altitude over mountain terrain. As is seen the DC-8 is the best aircraft and the EBF the worst.

Applications of the Method

The method described can be used to assess the satisfaction of passengers with the ride environment of a given vehicle. In addition it can be used to perform sensitivity analyses of the effects of vehicle variables, through variations in the vehicle transfer function and of input variables through the forcing function. These can be used to determine maximum design payoffs in the case of the vehicle or operating conditions and surface requirements (for roadways/rail) in the case of the forcing function.

Another application would be to incorporate an optimization routine and use the method inversely to determine the optimum design under engineering constraints for a desired satisfaction level.

Lastly, the method can be applied to validation studies of models of comfort and/or satisfaction by testing over a wide range of conditions with a limited set of field data. This would be accomplished by inserting the appropriate transfer function to replace those described above.

Conclusions

A method has been developed to predict passenger satisfaction with the ride environment of a transportation vehicle. This method, a general approach, has been applied to a commuter-type aircraft for illustrative purposes. The effects of terrain, altitude and seat location were examined. The method predicts the variation in passengers satisfied for any set of flight conditions. Several non-commuter aircraft were also analyzed for comparison and other uses of the model described. The method proposed has advantages for design, evaluation, and operating decisions.

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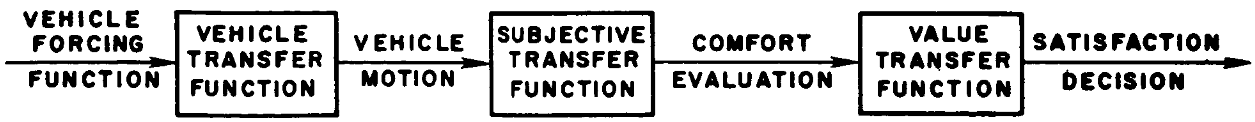

Figure 1.- Schematic for determining passenger satisfaction with ride quality.

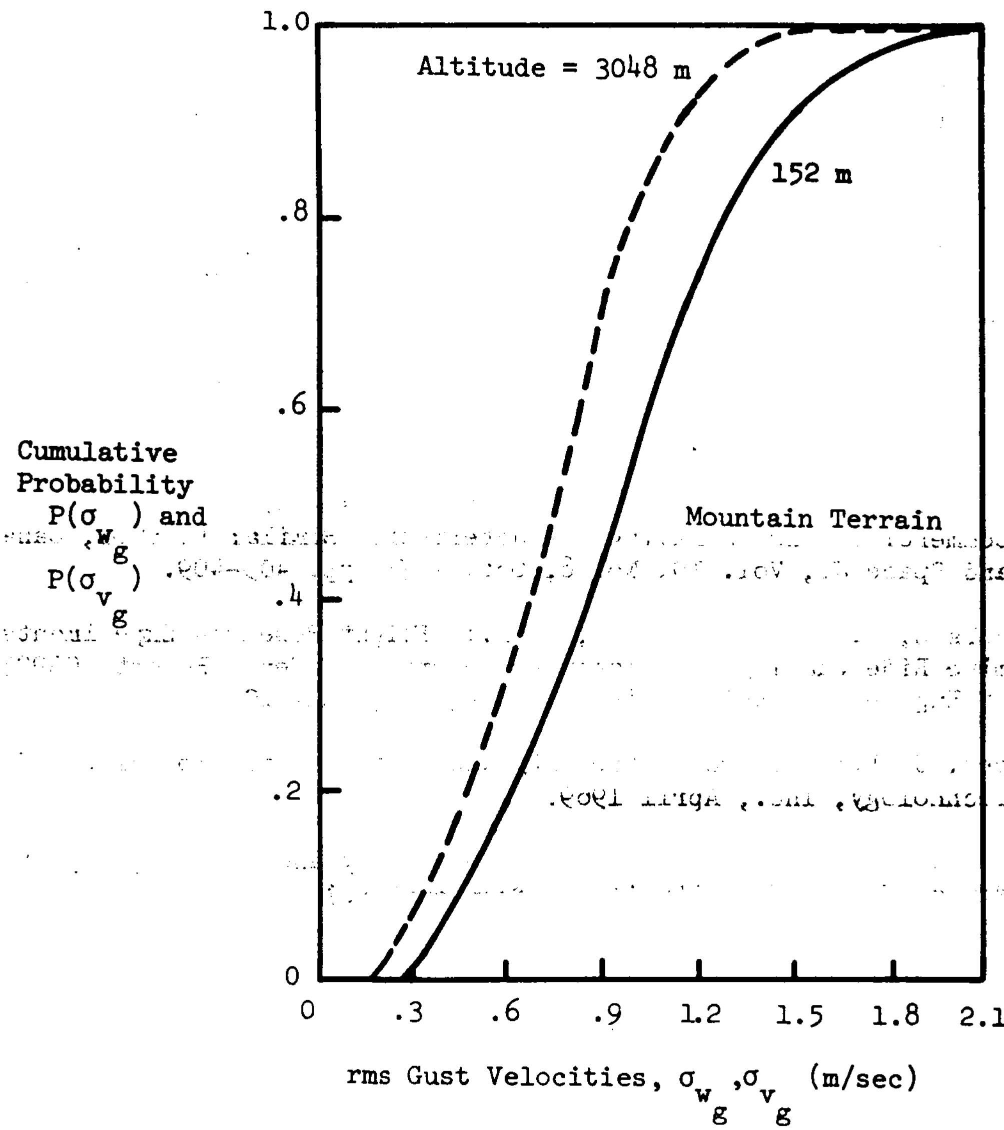

Figure 2.- Turbulence probability distribution.

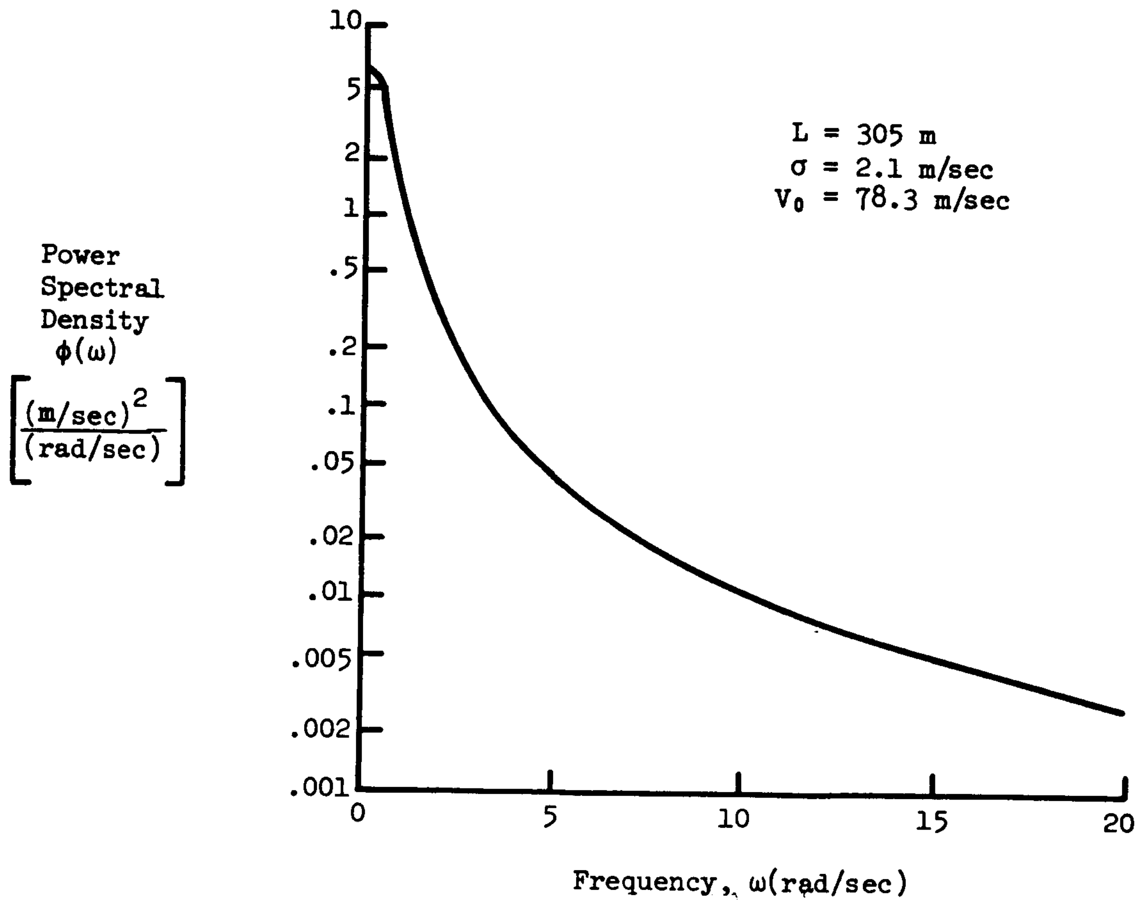

Figure 3.- Turbulence power spectrum.

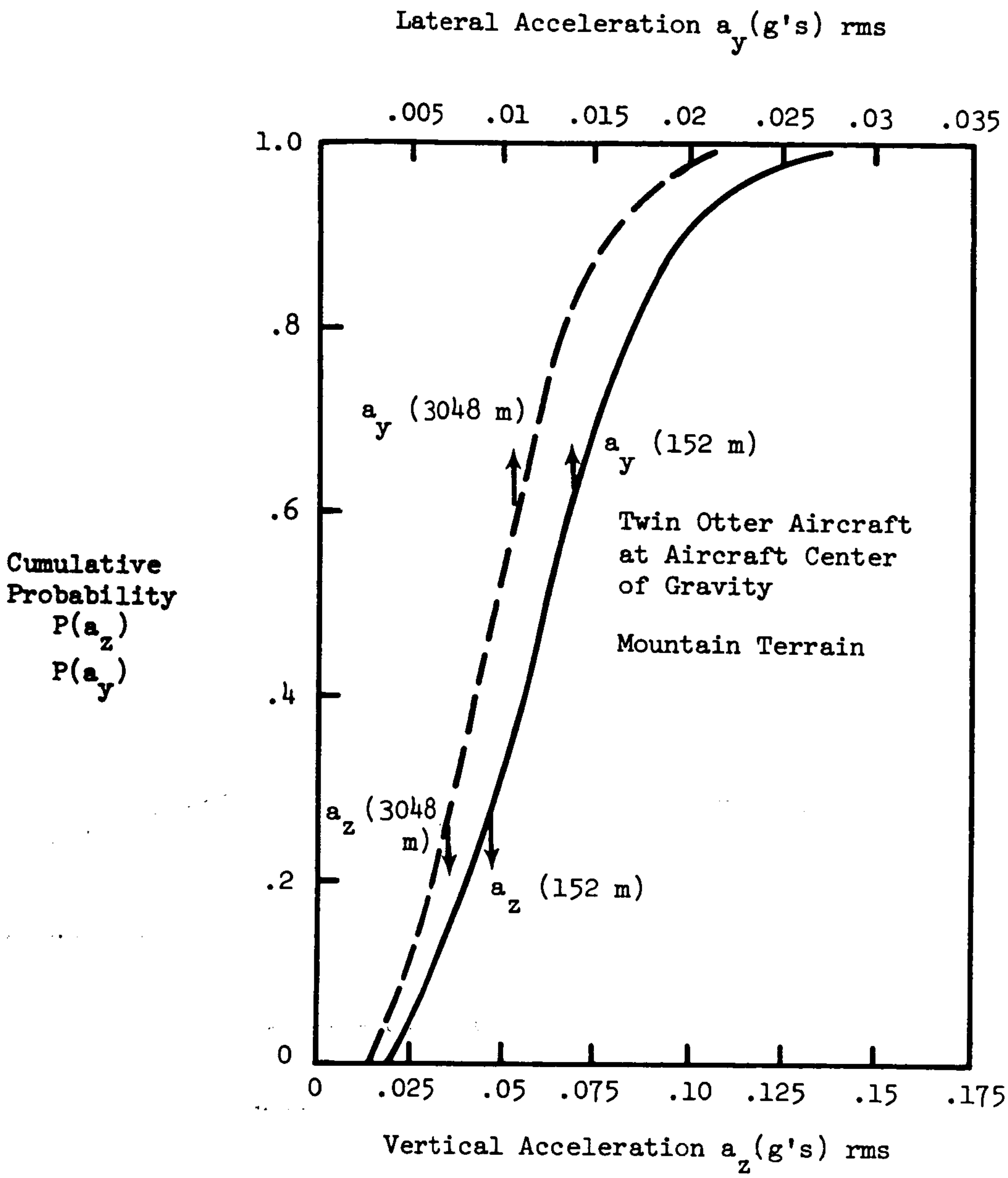

Figure 4.- Acceleration probability distribution.

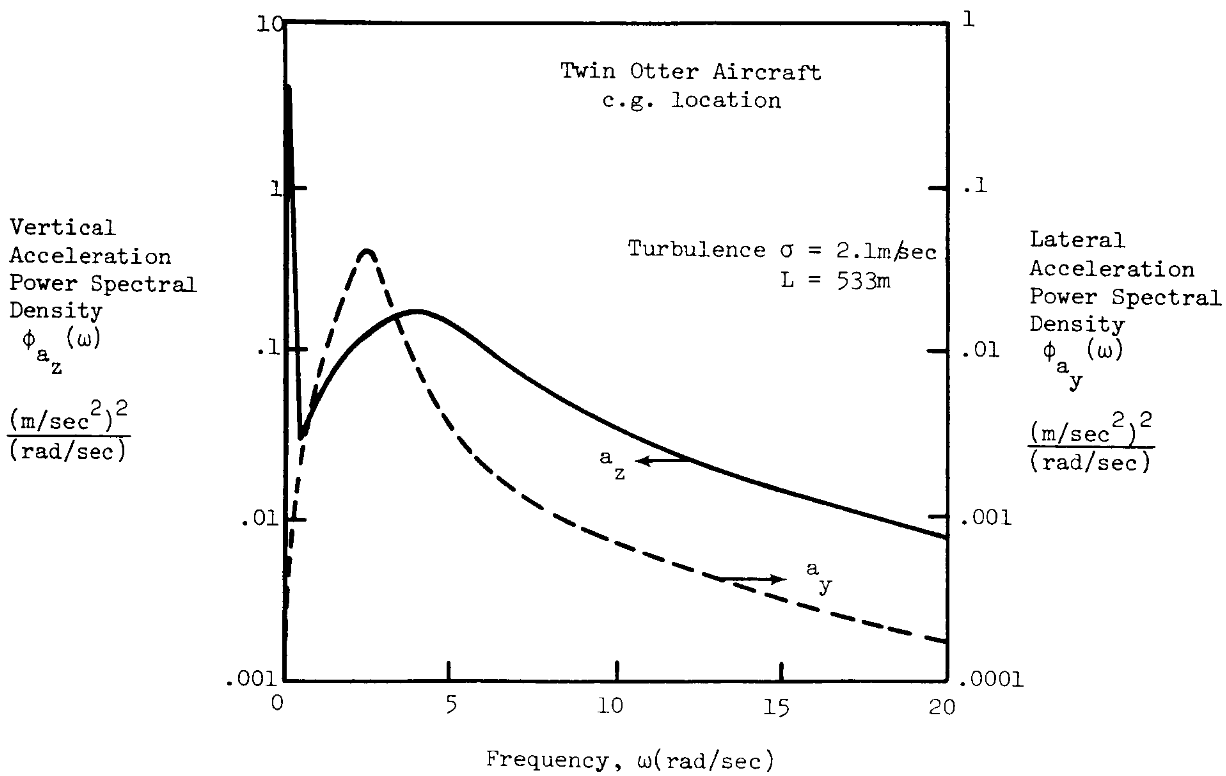

Figure 5.- Acceleration power spectrum.

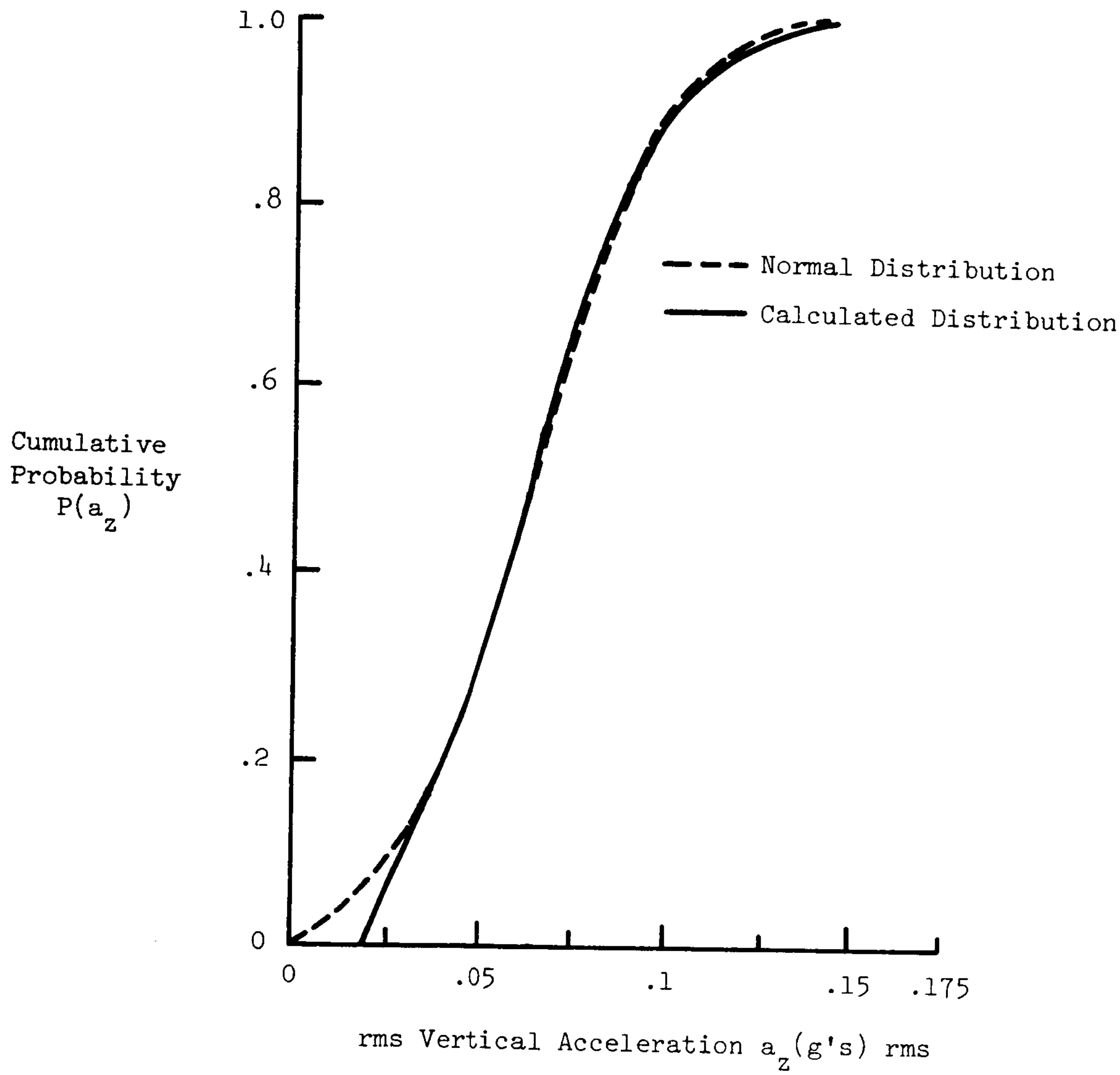

Figure 6.- Comparison of normal acceleration distribution with Gaussian distribution.

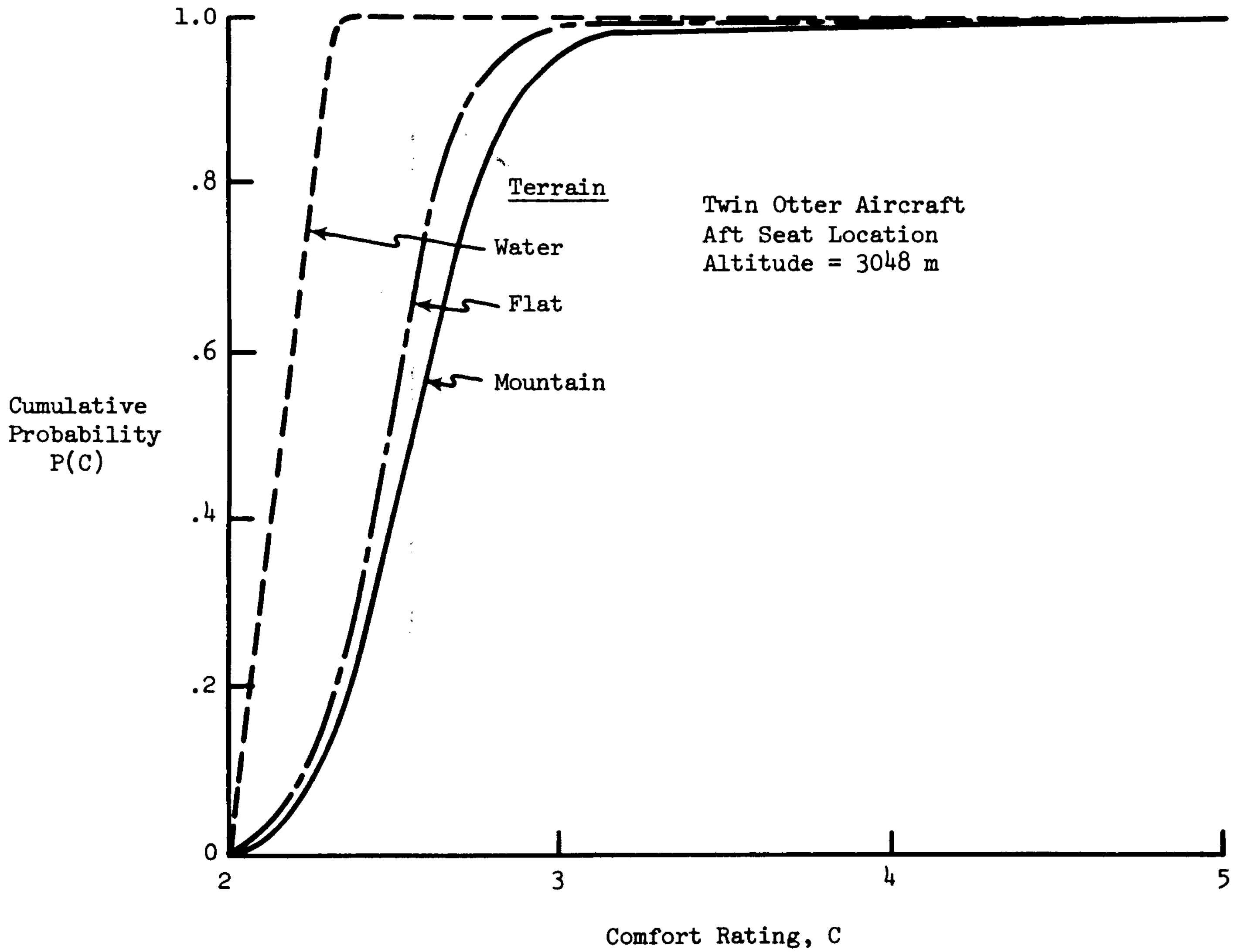

Figure 7.- Subjective rating probability distribution.

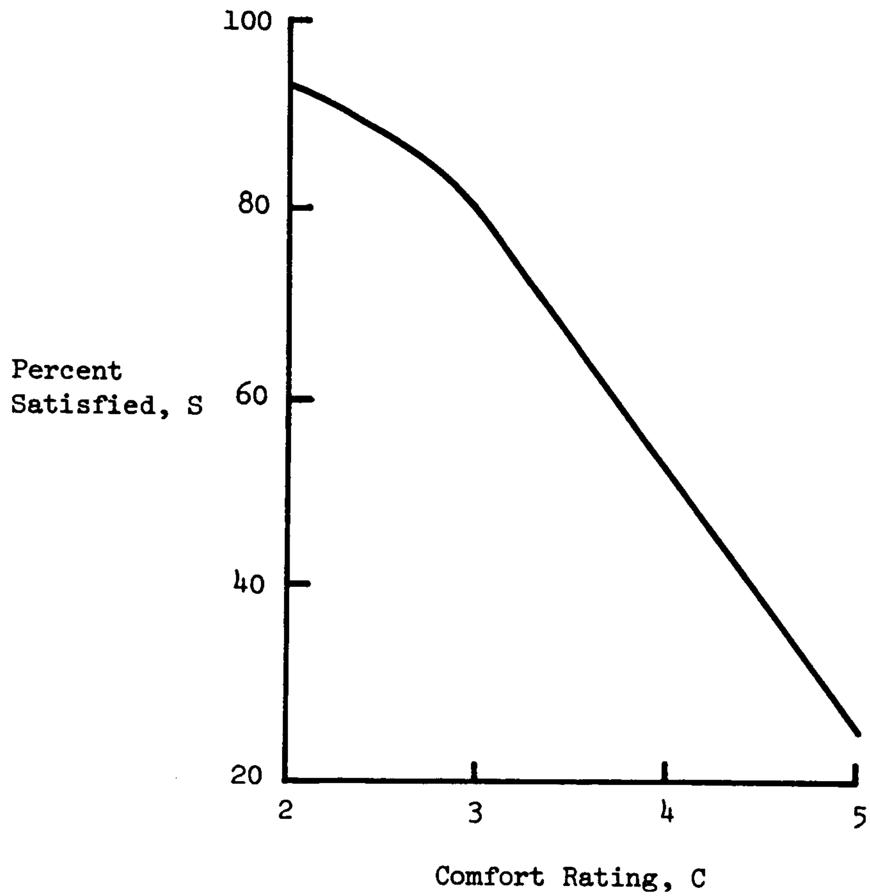

Figure 8.- Value transfer function.

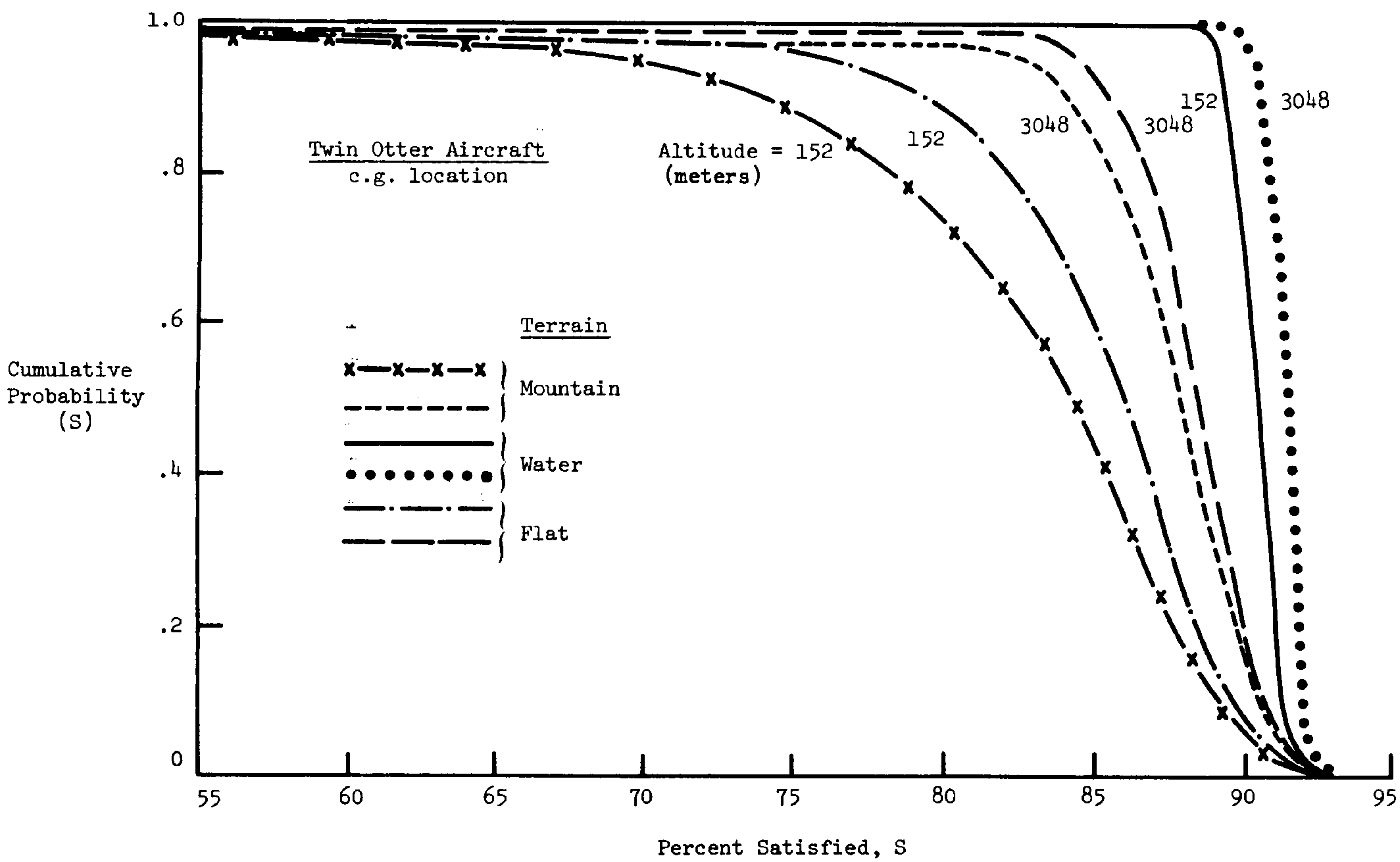

Figure 9.- Satisfaction probability distribution - Twin Otter, center of gravity location.

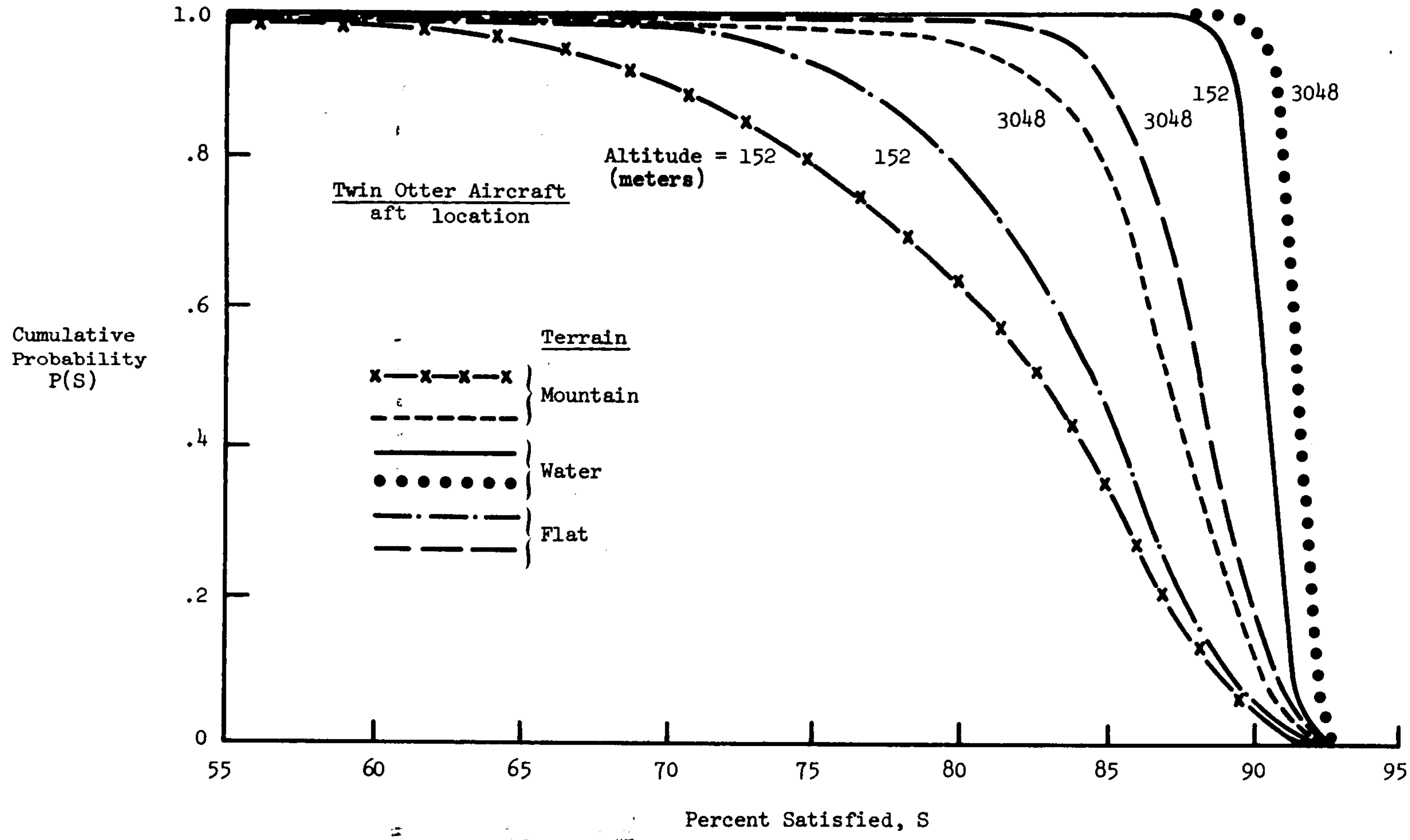

Figure 10.- Satisfaction probability distribution - Twin Otter, aft seat location.

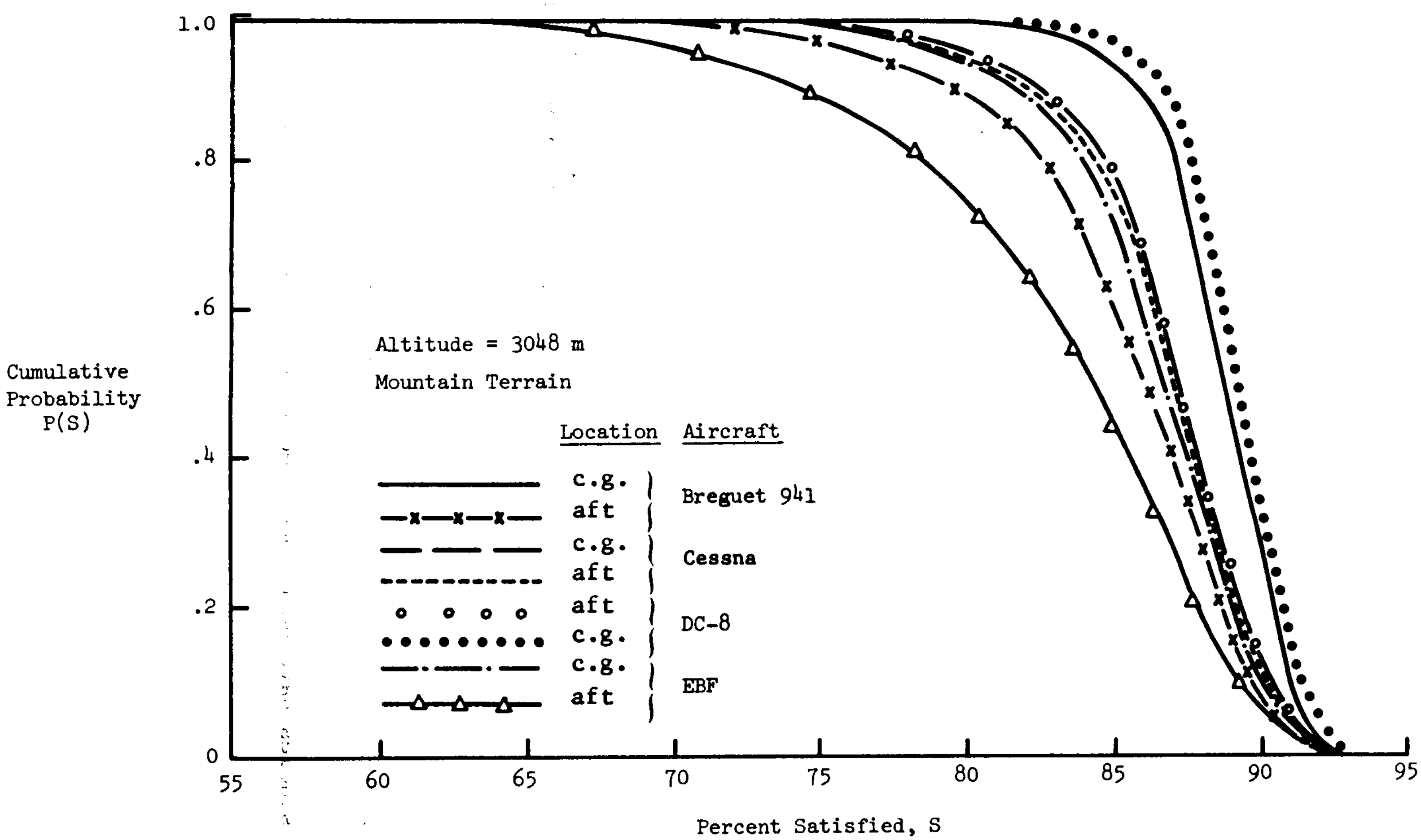

Figure 11.- Satisfaction probability distribution by aircraft type.

N76-16758

REVIEW OF MEASURED VIBRATION AND NOISE ENVIRONMENTS EXPERIENCED BY PASSENGERS IN AIRCRAFT AND IN GROUND TRANSPORTATION SYSTEMS

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SUMMARY

Measured vibration and interior noise data are presented for a number of air and surface vehicles. Consideration is given to the importance of direction effects; of vehicle operations such as take-off, cruise, and landing; and of measurement location on the level and frequency of the measurements. Various physical measurement units or descriptors are used to quantify and compare the data. Results suggest the range of vibration and noise associated with a particular mode of transportation and illustrate the comparative levels in terms of each of the descriptors. Collectively, the results form a data base which may be useful in assessing the ride of existing or future systems relative to vehicles in current operation.

INTRODUCTION

The vibration and interior noise environments of current and future vehicles are important to the ride quality and passenger acceptance of the transportation system. To fully evaluate the influence of vibration and noise on ride quality and passenger acceptance, the dynamic characteristics of the vehicle environment as well as the response of passengers to these stimuli must be well understood. Furthermore, such an understanding of the environment and its effects is essential to the development of ride-quality and passenger-acceptance criteria and the development of ride-improvement technology.

Numerous studies have been conducted in which the environment and/or the passenger response have been examined (refs. 1 to 6). However, very few studies have been conducted in which both the environment and the passenger response have been simultaneously measured over a wide range of environmental conditions. As a consequence, a comprehensive understanding of the effects of vibration on comfort does not exist. In particular, methods for assessing the combined effects of vibration level, duration, frequency, noise, and seat dynamics of the type encountered in transportation systems are not well understood. This lack of understanding has hindered the development and acceptance of descriptors for characterizing the environment of vehicles and the subsequent development of a comprehensive data base for current vehicle systems.

Measured vibration and interior noise data are presented herein for a variety of operational vehicles. The purpose of this presentation is to illustrate some of the important considerations and factors in quantifying the environment and also to provide comparative data for a variety of air and surface vehicles in terms of several physical descriptors.

VIBRATION AND NOISE MEASUREMENTS

Research Programs

The data presented in the following sections were collected in conjunction with research programs being conducted at the Langley Research Center in the areas of ride quality and aircraft interior noise. Although these programs are closely related, the ride-quality program (ref. 7) has emphasized the vibration environment of air and surface transportation systems and the influence of vibrations on passenger acceptance. The interior noise program is a relatively new program at Langley and includes both objective and subjective studies of the noise levels within vehicles as well as interior noise prediction and noise control. As mentioned, these programs have many common aspects and future ride-quality studies at Langley Research Center will stress combined vibration and noise environments.

Measurements have been obtained on a wide variety of vehicles in the course of these programs. These measurements have been used for purposes such as: vehicle absolute and/or comparative ride assessment; identification of vibration and/or noise sources and paths; identification of external sources of vibration and noise (rail track inputs, for example); evaluation of vibration or noise control fixes; inputs for laboratory studies; and development of criteria. As a result of these studies, a relatively large data base exists which can be used in assessing the ride quality of existing or future transportation systems relative to vehicles in current operation.

Measurement Methods

Vibration measurements are obtained by using the specially developed portable, battery-operated, instrumentation system shown in figure 1 and described in reference 8. The system consists of one or more acceleration packages, each containing three linear servoaccelerometers to measure vibration in the vertical, lateral, and fore-and-aft directions. The accelerometer data are recorded on a multichannel FM recorder and later digitized for frequency and amplitude analyses using a time series analysis program (ref. 9). The quasi-steady values of acceleration are removed from the recorded signals by passing the data through a high-pass filter which excludes values below 0.1 Hz.

In examining the vibration environment of a vehicle, the acceleration time history for a particular event, the amplitude of the vibration, and the frequency characteristics are of importance. In addition to providing important information for assessing comfort, the acceleration time history and the frequency analyses are often useful in diagnosing the source of the vibration input. For example, the acceleration time history may be used to identify a rough area in the runway whereas the frequency content may provide information on the wavelength of the input or the characteristic response frequencies of the vehicle.

Sound pressure measurements are usually obtained by recording the output of a microphone and a type 1 (precision scientific) sound level meter. The recorded data are subsequently digitized and a time series analysis program is used to obtain both numerical and graphical outputs in terms of octave-band, 1/3-octave-band, and narrow-band analyses.

RESULTS AND DISCUSSION

Vibration data obtained in the Langley Research Center ride-quality programs are presented for both aircraft and surface vehicles. Selected data are used to illustrate the characteristics of recorded vibration data for a variety of conditions. This is followed by comparative data for several vehicles presented in terms of various physical descriptors to illustrate the character of the descriptors as well as to provide a data base for future use.

Interior noise data include the comparative levels and spectra for several vehicles along with selected data samples to illustrate the unique noise characteristics of certain aircraft being studied. In all cases, the vibration and noise data presented in this paper were obtained from rides described by the test engineer as a normal or average operating condition. Furthermore, the rides of the CTOL aircraft are believed to be quite comfortable.

Vibration

Measurement considerations.- A great many variables must be considered in measuring the vibratory ride environment of a vehicle, and there are a comparable number of options available for describing the measured results. Certain of these considerations are listed in table I and graphically presented in figure 2 to illustrate the characteristic effects of direction of vibration, range of vibration level, operating condition, and mode of transportation. The level as a function of frequency of the vibration stimuli is presented by means of a power spectral density (PSD) plot. The data were recorded on the floor of the vehicle near the center of gravity and the PSD results were obtained from selected samples of the ride having a sample duration of approximately 2 minutes. The aircraft was a CTOL aircraft having three fuselage-mounted jet engines. Figure 2(a) presents typical vertical and lateral PSD functions during cruise operation. The levels of the selected

PSD's (2-minute sample) represent the maximum values observed during a normal flight of the aircraft. The general vibratory response of the aircraft is seen to be similar in both the vertical and lateral directions, with the highest levels of vibration occurring in the vertical direction. The vibratory energy is concentrated at frequencies less than 4 Hz. The range of vibration levels encountered during a typical flight of this aircraft is shown in figure 2(b) for the vertical direction. The frequency characteristics are similar except at the low end of the frequency range. In the smooth case, a relatively larger portion of the energy occurs at frequencies below 1 Hz. For the PSD's shown, the rms values of acceleration differ by a factor of about 4 and are discussed in more detail in subsequent sections. Figure 2(c) illustrates the difference in frequency response which results from differences in vehicle operation. As shown, the landing produces higher levels of vibration as well as frequency characteristics which are quite different from those for cruise. The high frequency response during landing is attributed to landing-gear—vehicle interactions. The response of the aircraft on the ground is not unlike that of many surface vehicles. As can be seen in figure 2(d), there is a significant difference between the response of the CTOL aircraft in cruise and that of an automobile; however, there are similarities between the aircraft during landing and the automobile. The automobile has considerable energy between 10 and 20 Hz due to wheel hop and response of the structure. The energy at approximately 1 Hz results from the fundamental suspension tuning and is typical of most surface vehicles. More detailed information for air and surface vehicle vibration level is presented in the next section.

Comparative vibration data.- In an effort to provide a comparative data base for future use as well as to provide insight into some of the vibration units, measured data are presented for a variety of vehicles and physical descriptors. Among the suggested units for describing the vibration associ-ation with a particular vehicle, the following descriptors are of interest and were selected for this study:

g_p — the maximum amplitude of vibratory acceleration associated with a selected time history

g_{rms} — the overall root-mean-square value of acceleration for a selected frequency band (0.1 to 30 Hz or 1/3 octave for this study)

$g_{\ell 10}$ — the level of vibratory acceleration that is exceeded 10 percent of the time

g_w — the root-mean-square value of the acceleration resulting from an acceleration signal that is weighted or filtered to better reflect human response to vibration

The values of these descriptors may in some cases vary depending upon the time duration of the measurement sample. As previously noted, all data were obtained from samples having a duration of approximately 2 minutes.

The levels presented represent the range of maximum values recorded during several normal operations. The weighted values g_w were obtained by filtering the data as recommended by the International Standards Organization (ISO) to reflect recommended equal comfort contours (ref. 10).

Comparative data obtained on a number of vehicles during cruise are presented in figures 3 and 4 in terms of the various descriptors. The vehicles are ranked according to the maximum level of vertical acceleration. The range of g_p observed in examining numerous 2-minute data samples for each of the vehicles is presented in figure 3(a). A comparison of the various vehicles suggests that the maximum values of g_p cover a range of about 3 to 1 ($0.5g > g_p > 0.15g$) in the vertical direction. In general, the vertical levels are higher than the lateral levels and the ground vehicles have higher acceleration than the aircraft. A similar trend is noted in terms of g_{rms} (fig. 3(b)). Again, the maximum values of g_{rms} cover a range of about 3 to 1 in the vertical direction. In terms of $g_{\ell 10}$ (fig. 3(c)), the vehicle ranking, with the exception of the helicopter, is identical to that obtained with g_{rms}. The relatively high values of $g_{\ell 10}$ associated with the helicopter are due to discrete frequency vibration observed at the blade passage frequency.

The vehicle vibration data are presented in figure 3(d) and figure 4 in terms of descriptors which reflect both the amplitude and the frequency of the vibration. In figure 3(d), for example, the acceleration is weighted according to the ISO equal comfort contours (ref. 10). Data are presented for the vertical direction only. It is noted that the values of g_w are lower than the values of g_{rms} (unweighted) in figure 3(b) as would be expected; however, the vehicle ranking remains approximately the same. These findings are further amplified in figure 4 in which 1/3-octave-band data are presented for the surface vehicles and aircraft and are compared with the ISO 4-hour reduced comfort boundaries. The 1/3-octave amplitude-frequency distribution provides a clear picture of the vibratory frequency which is useful in determining the source of vibration.

In considering the various descriptors, the single units such as g_p, g_{rms}, $g_{\ell 10}$, and g_w all appear to provide a simple, relatively consistent or similar description of the ride and may be adequate for assessing ride quality in many applications. The selection of a preferred descriptor will depend upon the specific application as well as upon the development of more information on subjective response to vibration. For example, g_p may be preferred for examining aircraft landing vibration whereas g_w may be preferred for examining longer term cruise conditions. For examining the source of vibration, the narrow-band analyses such as PSD or the 1/3-octave analyses are useful. Although the data presented in figures 3 and 4 do not represent a large sample for certain vehicles, collectively the data are believed to be consistent and to represent a relatively large data base in comparison to previously published data on vehicle vibration. The data may be used for a comparative assessment of the ride quality of a particular vehicle of interest relative to the vehicles presented herein or in specifying design criteria for future systems in terms of currently acceptable vehicles.

In an effort to develop a statistically larger data base, measurements have been taken on two different CTOL aircraft during a total of 13 flights including taxi, climb to altitude, cruise, and landing. These data are presented in reference 11 and are summarized in figure 5. The vibration behavior of the two aircraft are very similar. As would be expected, the best ride occurs during cruise. Furthermore, the vibration levels in the vertical direction are seen to exceed the lateral levels by a factor of about 5 during cruise and of somewhat less than 5 during ground operations. As previously indicated, figure 5 represents a relatively large data base obtained from vehicles which are believed to be good riding, acceptable transportation systems.

Seat/passenger response.- The physical data presented in the previous sections have been obtained on the floor of the vehicle. In order to have a better understanding of how the measurements taken at the floor of the vehicle compare with the levels actually experienced by the passenger, simulator studies have been conducted (ref. 12) to determine the transmissibility of various seats. Tourist-class and first-class aircraft seats and bus seats were examined with seated passengers for single-axis sinusoidal inputs in the vertical and lateral directions. The acceleration measured at the seat/passenger interface is shown in figure 6 in terms of the amplitude response ratio (ratio of seat acceleration to floor acceleration) for a range of sinusoidal input frequencies. As noted, the resonant frequency in the vertical direction is in the range of 4 to 7 Hz with a maximum amplification of about 1.4. For lateral inputs, an amplification of about 1.5 is observed in the frequency range of 2 to 3 Hz. By coincidence, the area of greatest human sensitivity, according to the ISO standards, also occurs in these regions, as shown in the figure. The importance of considering seat transmissibility in the development of ride-quality criteria is currently under study in a simulator program wherein subjective ride-quality measurements are being compared with both seat and floor measurements.

In concluding this section on vibration, it is again noted that a data base does exist for a variety of vehicles in terms of several descriptors. However, the "best" descriptor (if such exists) as well as ways to compare the vibrations occurring in different directions will require extensive subjective testing in the laboratory and in the field.

Interior Noise

Interior noise spectra are presented in figure 7 for several aircraft and an automobile. As in the case of vibration, the vehicle noise spectra are dependent upon many factors such as vehicle type and operating condition; however, the selected spectra are believed to be representative in terms of relative amplitude and frequency for the particular class of vehicle. As shown, the interior noises of the aircraft are higher than those of the automobile and the noise levels of the STOL, helicopter, and general aviation vehicles are generally considered to be uncomfortable by most observers. These three vehicles have, in addition to the high levels, relatively low frequency characteristics which make noise control difficult.

The sources and detailed characteristics of the interior noise for the CTOL, general aviation, STOL, and helicopter are quite different. The boundary layer is an important noise source in the case of CTOL, whereas, it is relatively small in the other vehicles. The main sources in general aviation vehicles are the propeller and reciprocating engine, whereas the helicopter has, in addition to the rotor, a number of discrete inputs associated with gear clash in the transmissions. The STOL (powered-lift) levels are estimated to be high because of the impingement of the engine exhaust on the listing surfaces and the inboard location of the engines. These and other details are shown in figures 8 to 11.

The CTOL spectra are shown in figure 8 for three locations in a jet transport having fuselage-mounted engines. The highest levels are recorded at the aft cabin location in the proximity of the engines. At the pilot location, the noise is higher in frequency and is attributed to the boundary layer. Measured levels for a single-engine, light aircraft (ref. 13) are shown in figure 9 for several values of rpm and indicated airspeed (IAS). Note that an increase in rpm results in an increase in the dB(A) level but a decrease in the overall sound pressure level (OASPL). This results from the shift in frequency (crossover) and the frequency weighting in the dB(A) unit. As shown in figure 10, the STOL levels (ref. 14) are highly dependent on the operating condition. The externally-blown-flap (EBF) configuration has high levels during powered lift but lower levels during cruise, where powered lift is not required. If powered lift is utilized during cruise of the upper-surface-blowing (USB) configuration, the levels would be relatively higher than those of the EBF configuration as shown. The helicopter data of figure 11 were obtained on the Langley Research Center Civil Helicopter Research Aircraft (ref. 15) during hover and with an untreated cabin. These data show that the main noise source occurs at approximately 1370 Hz which corresponds to first-stage planetary gear clash in the main gear box. The peak amplitude at 1370 Hz is at least 10 dB above all other peaks in the spectrum, which indicates that for this flight condition the other sources of interior noise do not significantly contribute to the overall noise level.

Two other frequencies are emphasized in the figure. The tail rotor-blade passage frequency occurs at approximately 53 Hz; main bevel and tail take-off gear clash occurs at approximately 2700 Hz. The acceleration PSD also has peak amplitudes in the spectrum at 1370 Hz and 2700 Hz, which suggests that some relationship exists between noise and structural vibration at these frequencies.

For comparative purposes, the A-weighted interior noise levels for the aircraft are presented in figure 12 along with levels for bus, rail, and auto vehicles and the OSHA 8-hour limit of 90 dB(A). The data shown were obtained from references 6 and 13 to 21. Again, these data emphasize the fact that aircraft levels are considerably higher than those of the surface vehicles. Furthermore, the fact that several of the aircraft exceed the OSHA 8-hour limit suggests that better noise control is needed. The interior noise program currently underway at Langley will emphasize the noise reduction of STOL, helicopter, and general aviation vehicles as well as the establishment of acceptable levels (criteria) of interior noise for the safety and comfort of crew and passengers. Safety considerations will include speech intelligibility and auditory effects, whereas the comfort studies will emphasize passenger acceptability and speech interference.

CONCLUDING REMARKS

Measured vibration and interior noise data are presented for a number of air and suface vehicles. In comparing air and surface vehicle environments, the vibration levels are relatively high in the ground vehicles and the noise levels are relatively high in the aircraft. For a particular vehicle, large variations in level are observed throughout the operating envelope of the system due to external effects (turbulence, for example) as well as the effects of vehicle operation and measurement location. The aircraft vibration and noise data base appears to be larger than that of the surface vehicles. However, when taken collectively the measurements form a data base which may be used in assessing the ride of existing or future systems relative to vehicles in current operation.

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TABLE I.- DESCRIPTION OF RIDE ENVIRONMENT

MEASUREMENT CONSIDERATION	EXAMPLE
STIMULI	LEVEL, FREQUENCY, TIME
DIRECTION	VERTICAL, LATERAL, COMBINED
RANGE	SMOOTH, ROUGH
OPERATION	TAKE-OFF, CRUISE, LANDING
MODE	AIR, GROUND
DESCRIPTOR	PSD, 1/3-OCTAVE, g, dB
LOCATION	FLOOR, SEAT, FORE/AFT

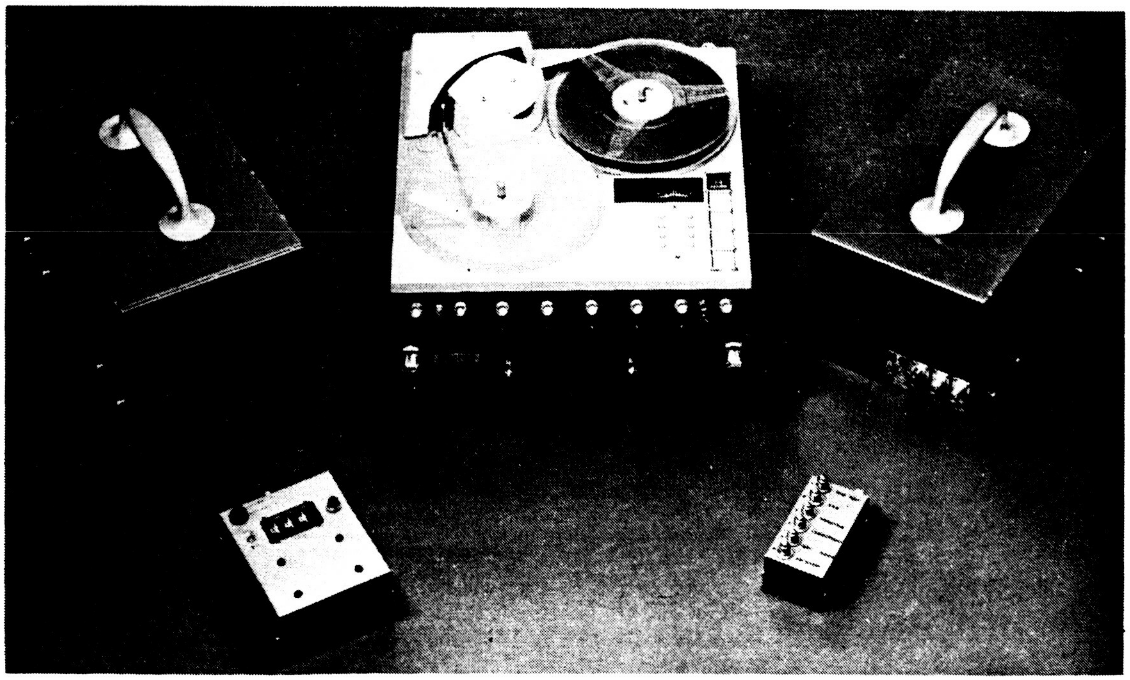
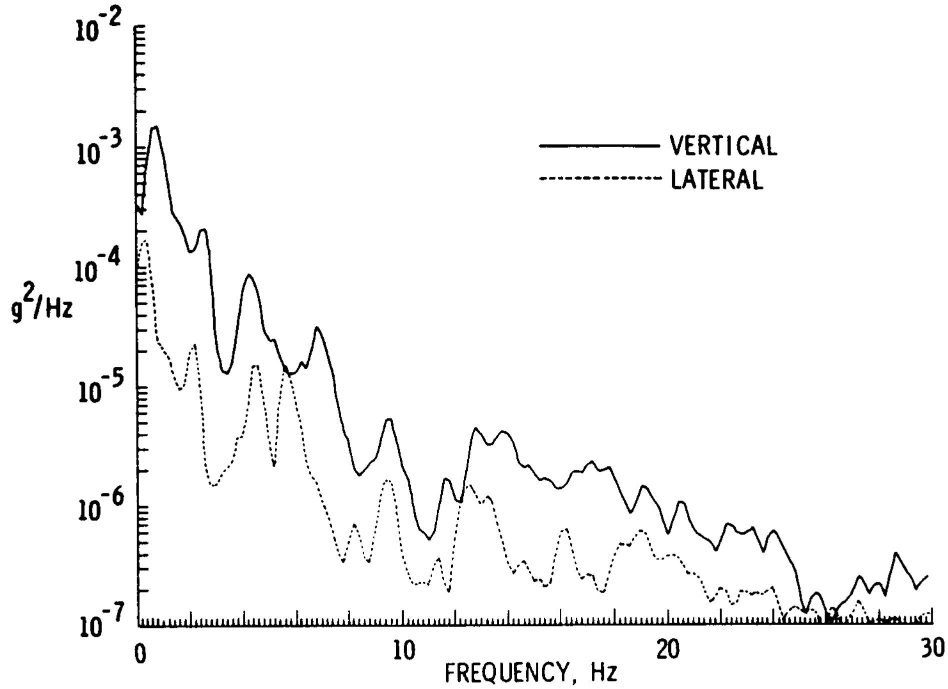
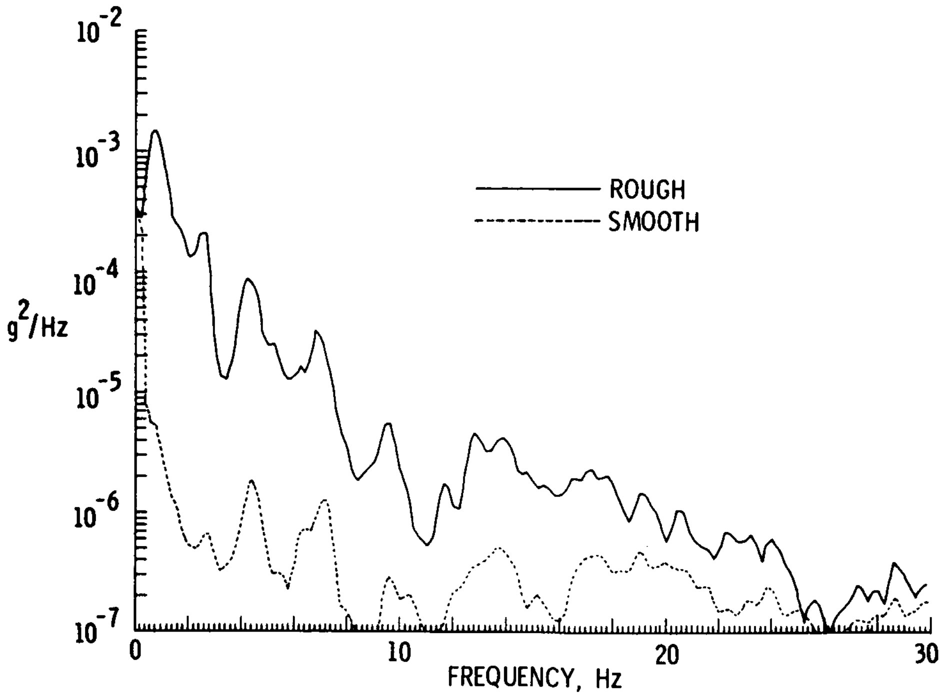


Figure 1.- Vibration measuring and recording system.

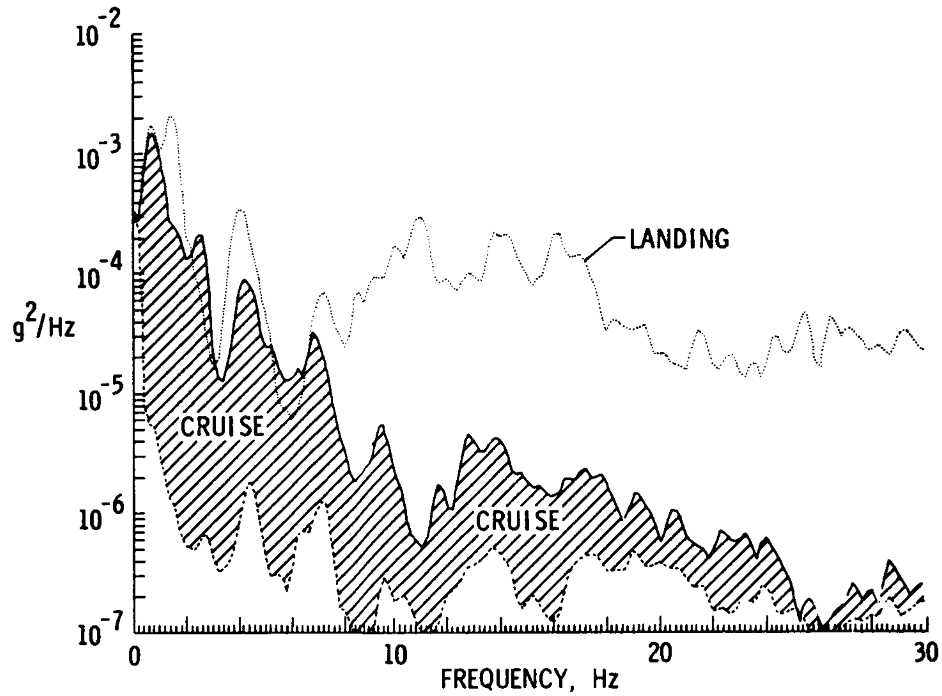


(a) Direction effects for CTOL aircraft in cruise.

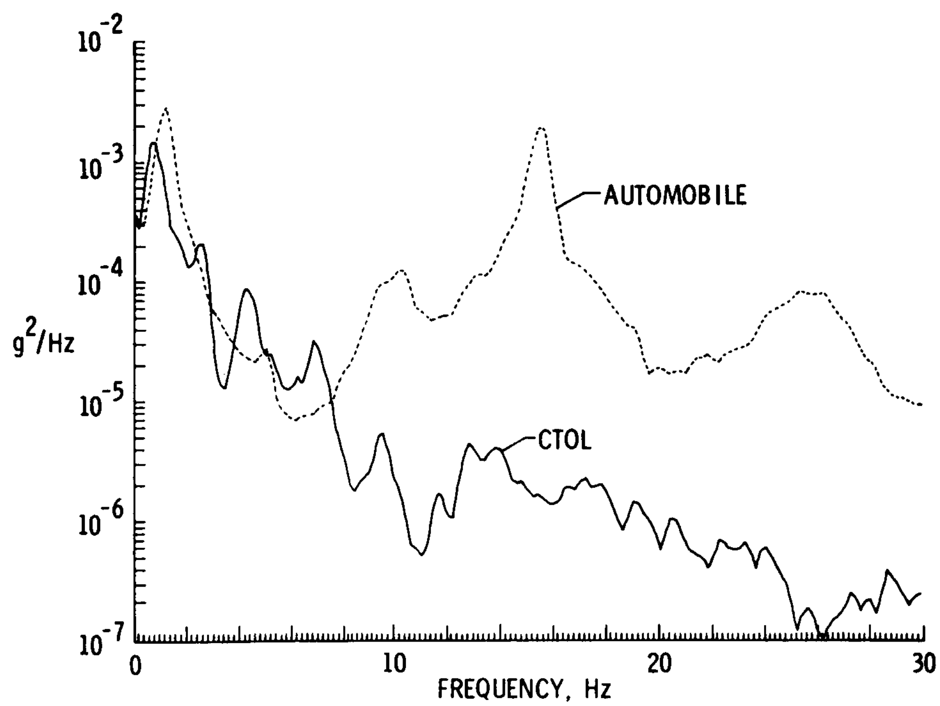


(b) Range of vibration in vertical direction for CTOL aircraft in cruise.

Figure 2.- Considerations for measuring and describing vibratory ride environments.

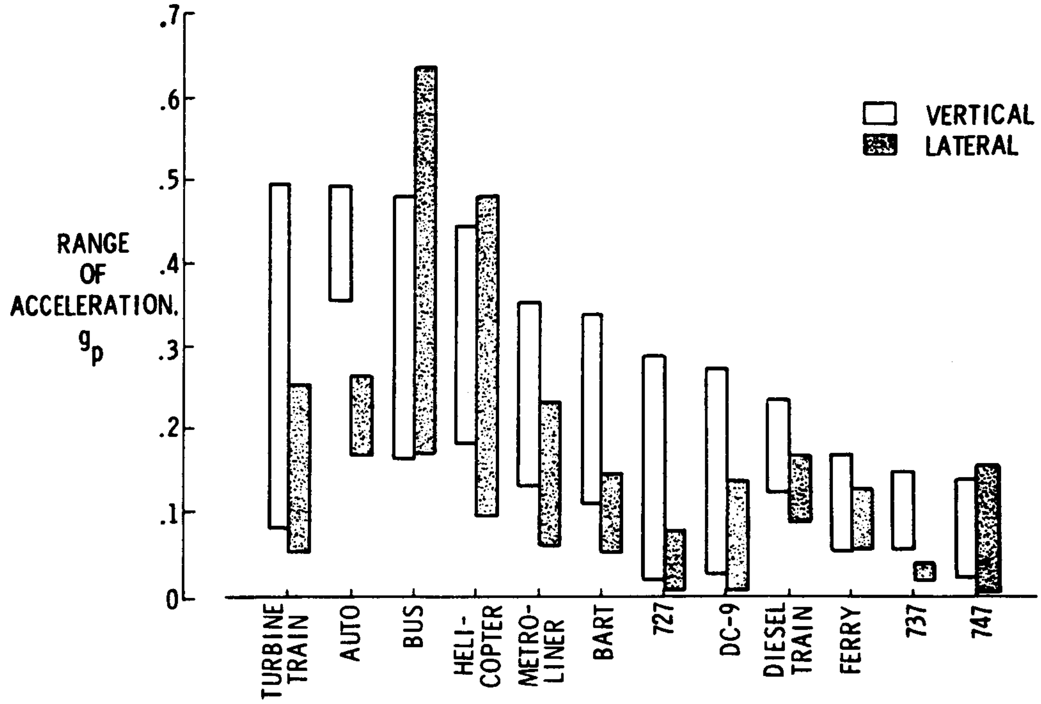


(c) Effects of vehicle operating condition on vertical vibration for CTOL aircraft.

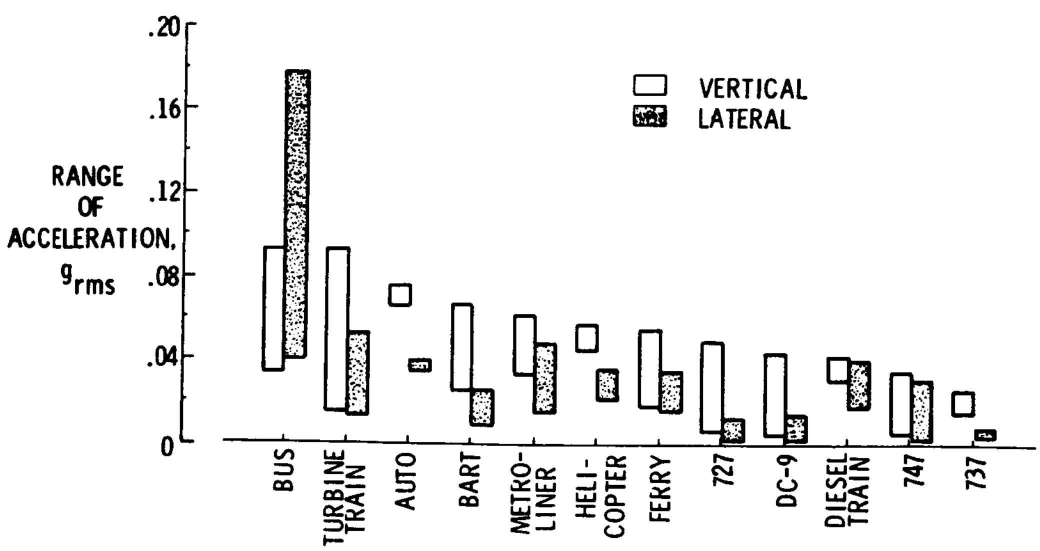


(d) Effects of mode of transportation on vertical vibration in cruise.

Figure 2.- Concluded.

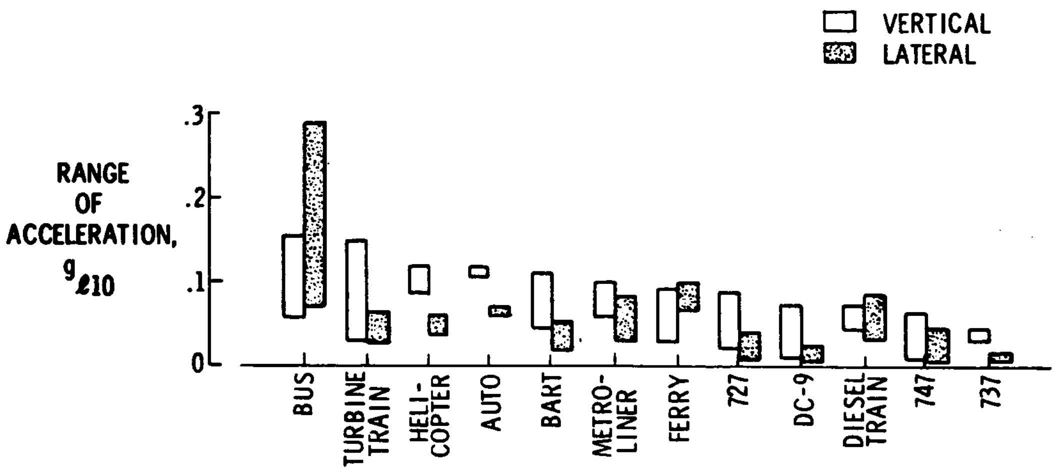


(a) Peak acceleration range.

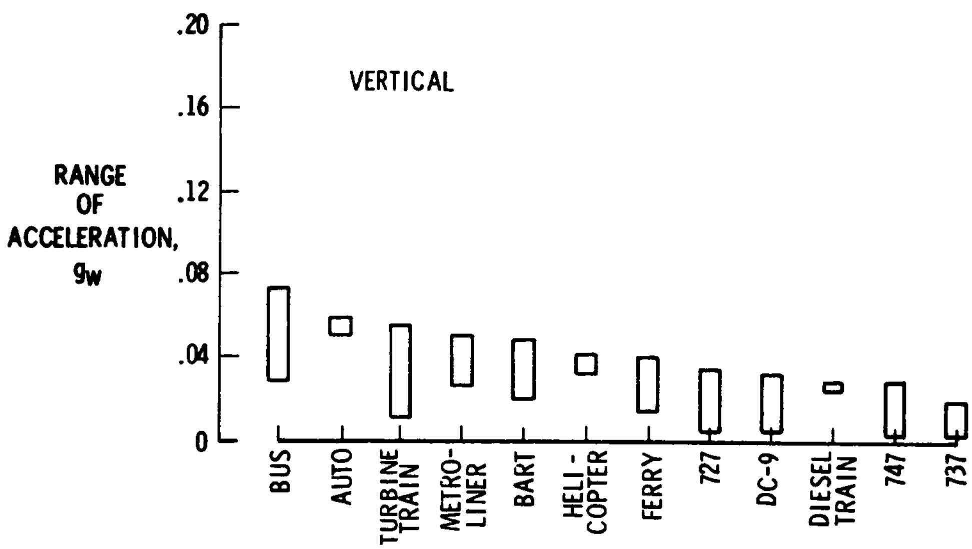


(b) RMS acceleration range.

Figure 3.- Vibration levels recorded during cruise.

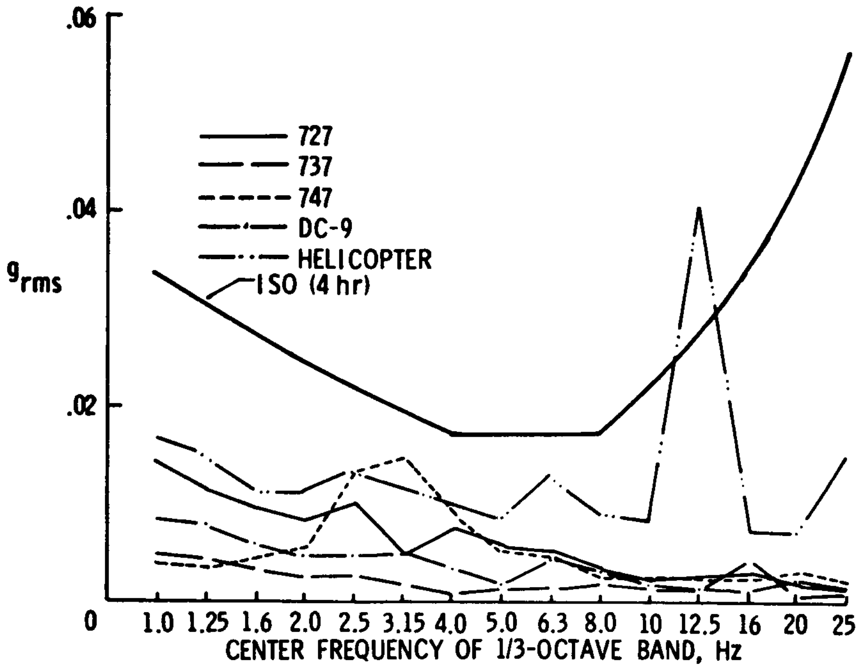


(c) Level exceeded 10 percent of time.

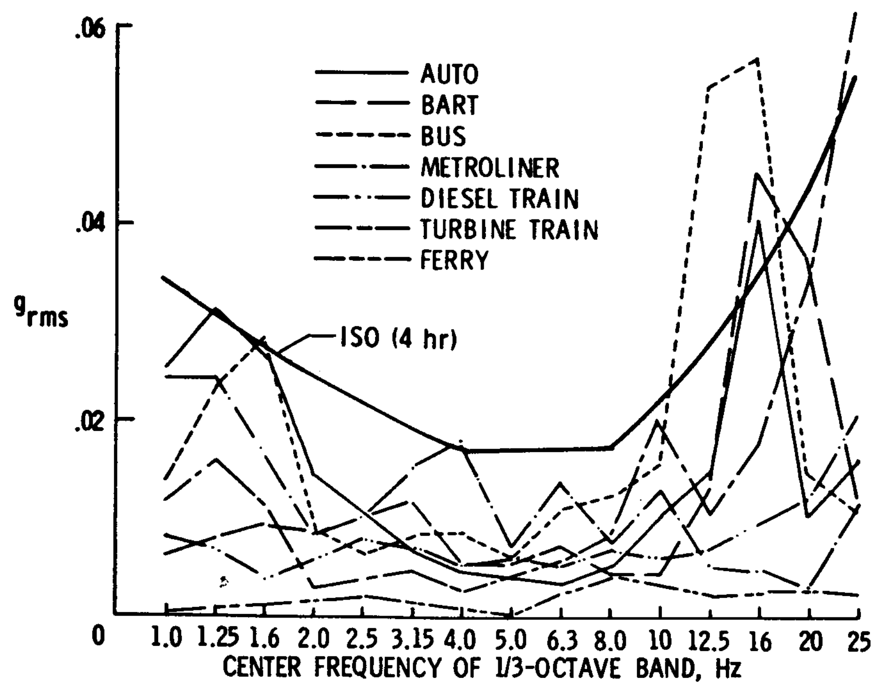


(d) ISO weighted RMS acceleration.

Figure 3.- Concluded.

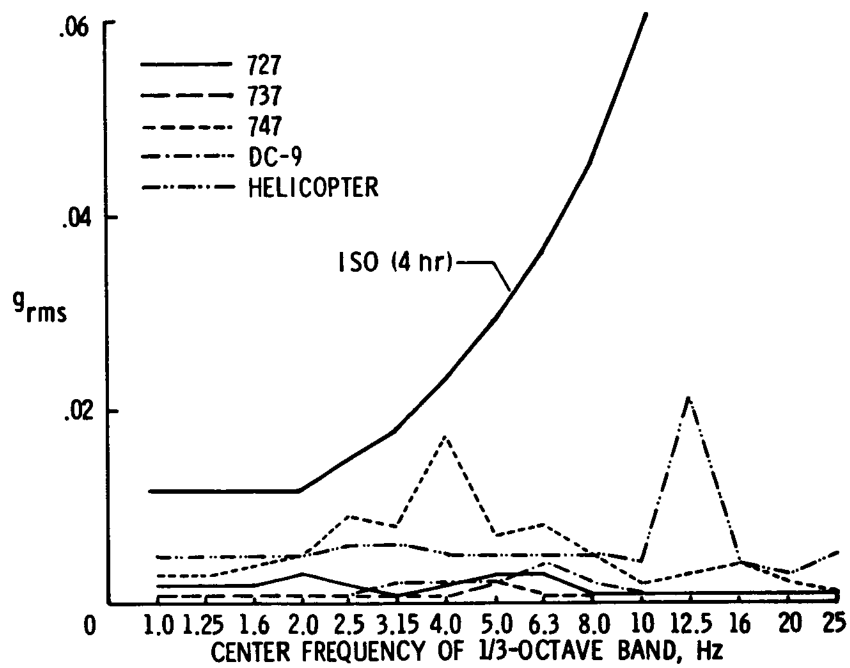


(a) Vertical response for aircraft.

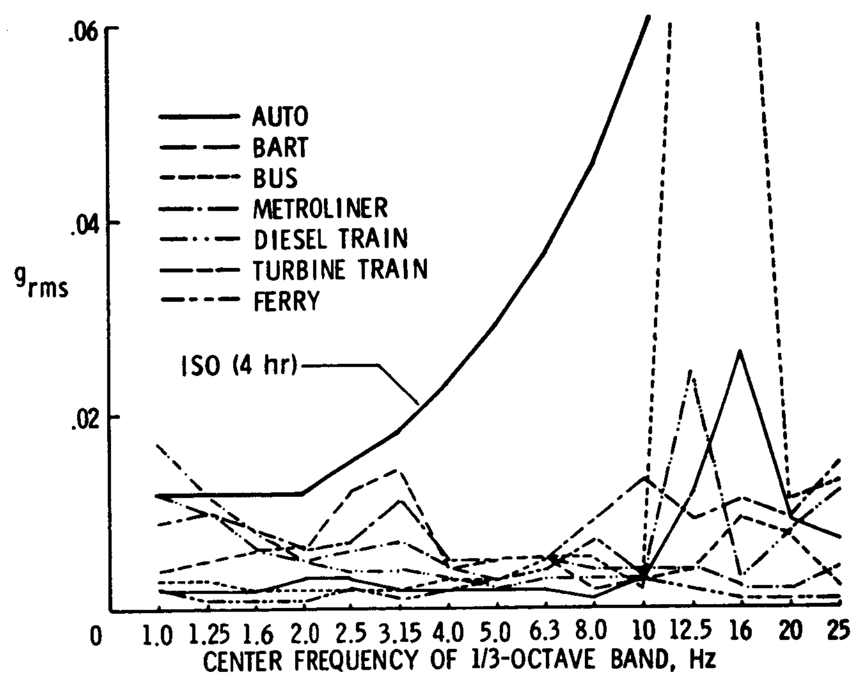


(b) Vertical response for surface vehicles.

Figure 4.- One-third-octave-band spectra recorded during cruise.



(c) Lateral response for aircraft.



(d) Lateral response for surface vehicles.

Figure 4.- Concluded.

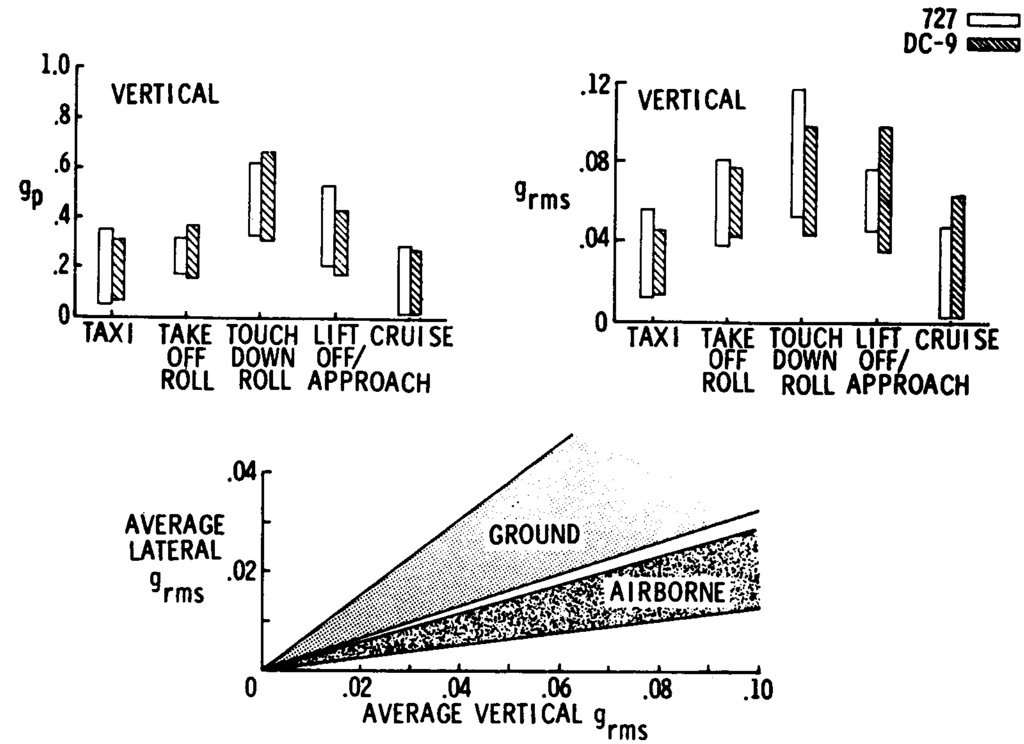


Figure 5.- CTOL vibration data base.

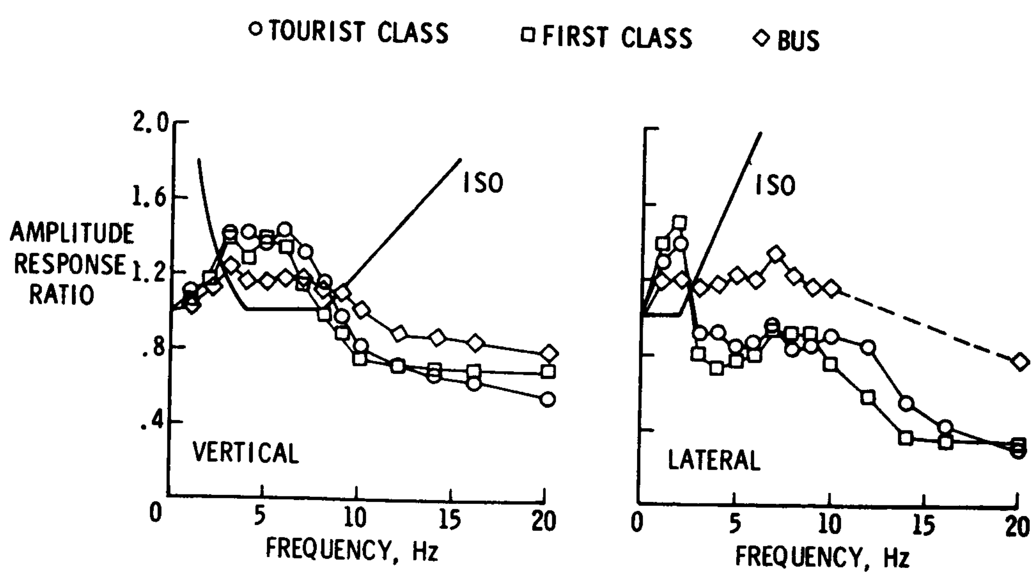


Figure 6.- Seat transmissibility.

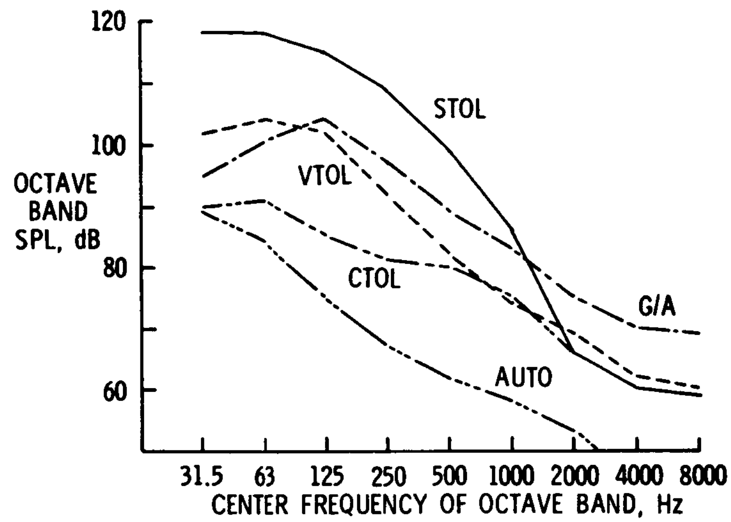


Figure 7.- Interior noise spectra for selected vehicles during cruise.

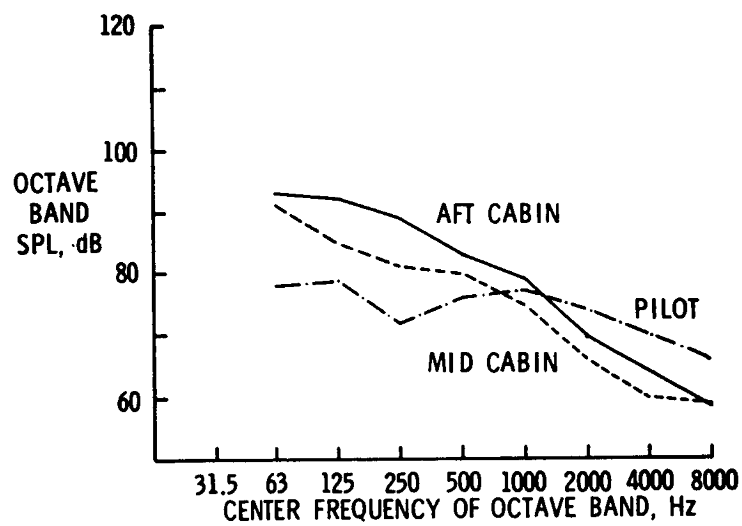


Figure 8.- Effects of measurement location on recorded CTOL noise during cruise.

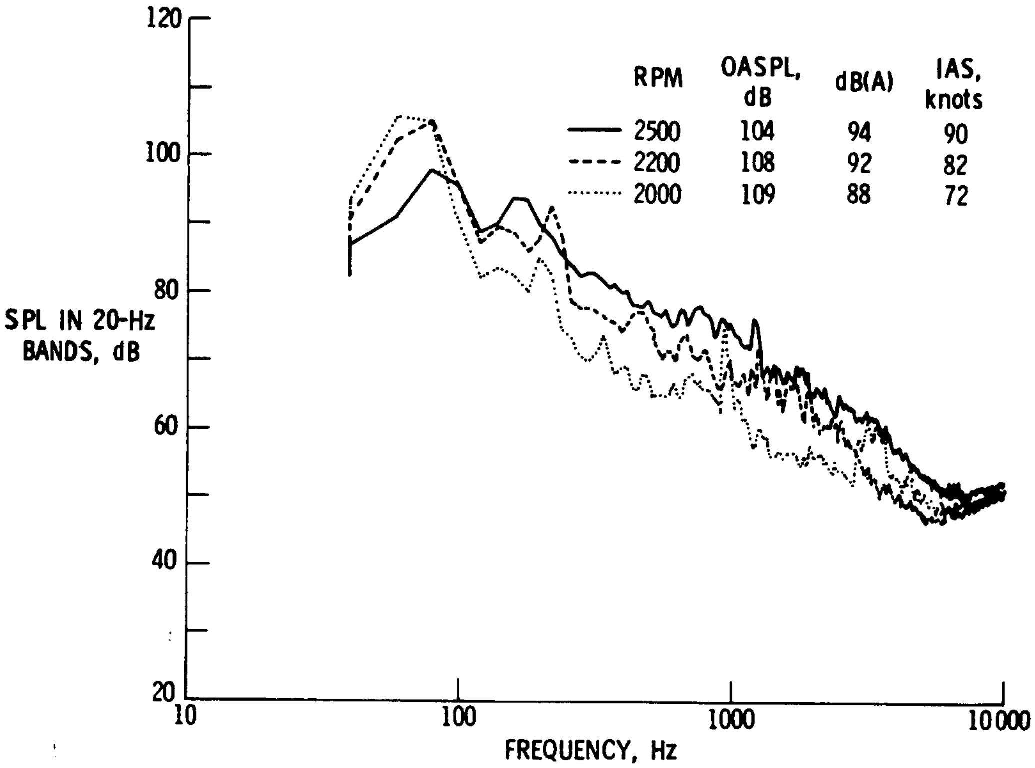


Figure 9.- General-aviation interior noise characteristics for single-engine aircraft cruising at 305 meters.

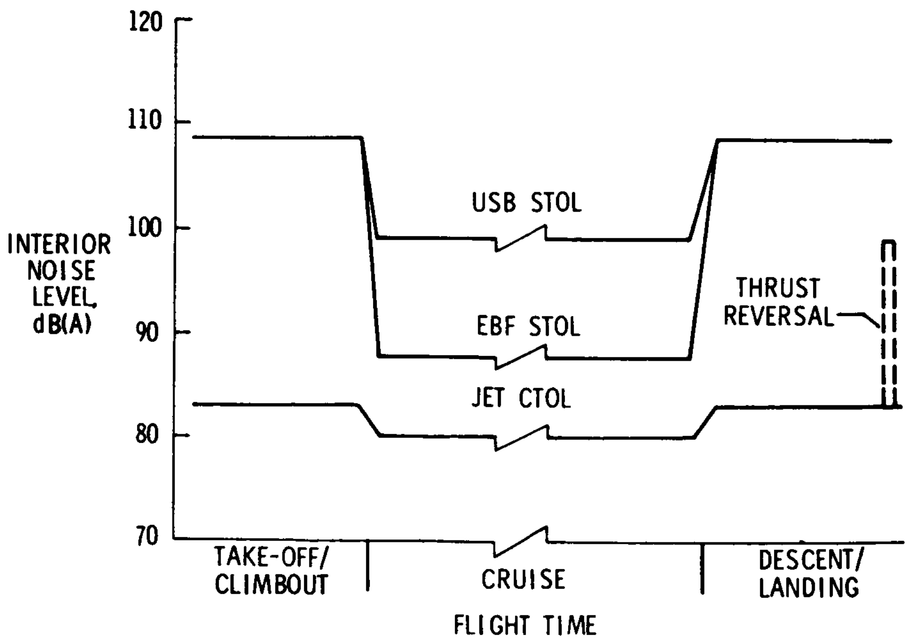


Figure 10.- STOL interior noise characteristics.

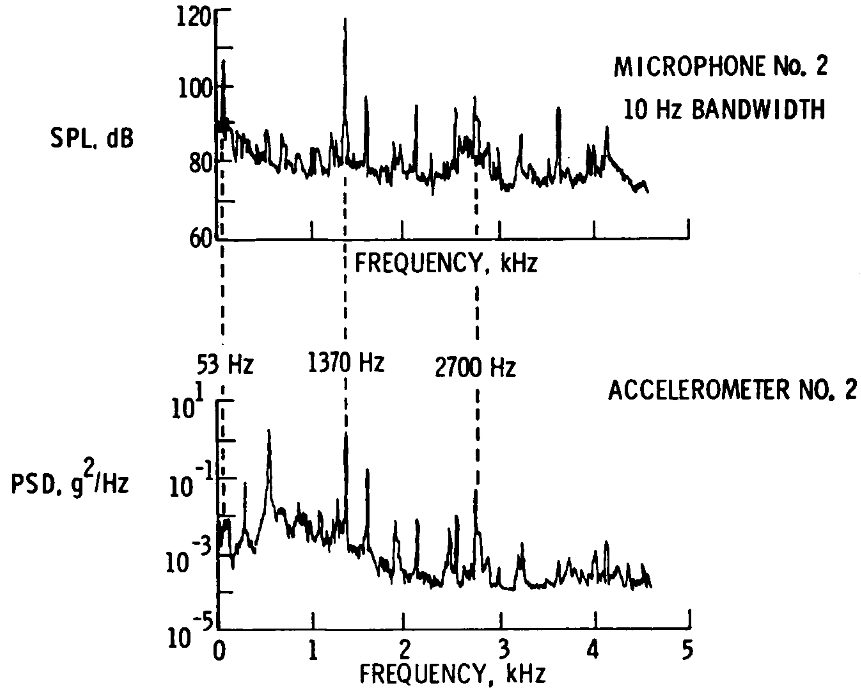


Figure 11.- Interior noise and acceleration for untreated helicopter during hover out of ground effect.

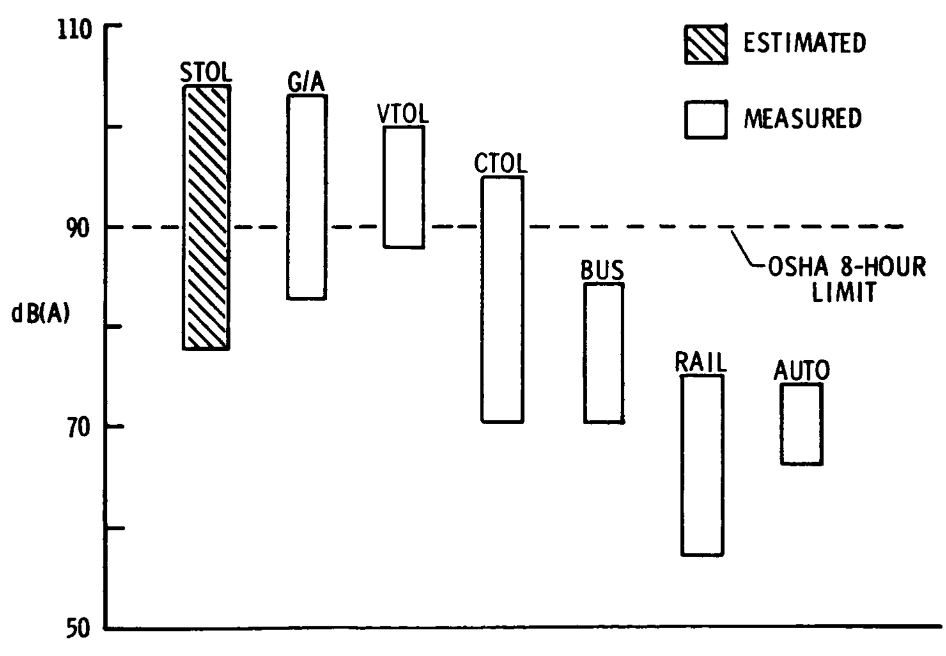


Figure 12.- Comparative interior noise levels during cruise.

N76-16759

NONMOTION FACTORS WHICH CAN AFFECT RIDE QUALITY

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SUMMARY

Data pertaining to nonmotion factors affecting ride quality of transport aircraft have been obtained as part of NASA in-house and sponsored research studies carried out onboard commuter-airline and research aircraft. From these data, quantitative effects on passenger discomfort of seat width, seat legroom, change in cabin pressure, and cabin noise are presented. Visual cue effects are also discussed.

INTRODUCTION

Ride quality can be defined as the impact on the passenger of all aspects of the carrier-vehicle physical environment that affect his acceptance of the ride. Within this definition, environmental factors other than motion and vibration would be included. These other factors are the subject of this paper. Surveys of travelers using a given mode of transportation are useful in identifying the importance of various environmental factors. Abridged results from one British survey (ref. 1) are shown in figure 1. In this survey, travelers were asked to rank in preference 18 suggested improvements for railway coaches. The 18 improvements covered a variety of items ranging from Extra entertainment to More luggage space. In the cumulative order of preference, four of the top five suggested improvements concerned ride quality: Less vibration, More space, Less noise, and Better seats. These results are typical of survey findings for other modes of transportation (refs. 2 and 3), where various ride-quality factors rank high in importance from the traveler's viewpoint.

Some new information pertaining to nonmotion factors affecting ride quality of aircraft has been obtained as part of NASA in-house and sponsored research studies carried out onboard airline and research aircraft. No attempt was made in these studies to determine systematically the effects of varying specific nonmotion factors on ride quality. Data which will be presented were only incidentally obtained and therefore are of limited scope. A brief overview will first be given of the airline traveler surveys from which much of the data originated to provide a proper background for subsequent discussion of individual factors.

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AIRLINE TRAVELER SURVEYS

Small transport aircraft used by feeder lines and commuter lines can provide much valuable ride-quality information because a significant percentage of travelers often rate their ride as marginal at best. Lack of good ride quality is associated with a lack (by economic constraints) of features and characteristics common to the much larger and heavier jet transports. Four such small commuter aircraft, shown in figure 2, were the subject of recent ride quality studies (ref. 4) carried out by the University of Virginia under NASA grant. The aircraft include the 19-passenger De Havilland of Canada DHC-6 twin-engine turboprop, commonly known as the Twin Otter; the 26-passenger Aerospatiale Nord 262 twin-engine turboprop; the 13-passenger Beech 99 twin-engine turboprop; and the 26-passenger Sikorsky S-61 helicopter. Each of the aircraft was used by a different airline; hence, operating conditions differed somewhat between aircraft. All operations took place, however, in the north-eastern part of the United States.

Table I provides an overall summary of traveler survey information. More than 800 travelers participated. Trip time ranged between 20 and 60 minutes for the three fixed-wing aircraft and between 7 and 10 minutes for the heli-copter. Near the end of the trip the passengers rated the overall trip ride on a 7-point (undefined) descriptor scale ranging from Very uncomfortable to Very comfortable. The ride was given some form of uncomfortable (Somewhat uncomfortable, Uncomfortable, or Very uncomfortable) rating by slightly more than 30 percent of the passengers riding in each of the three fixed-wing air-craft and by 12 percent of the passengers riding in the helicopter. The relatively low percentage of helicopter passengers who expressed discomfort may have resulted from the relatively brief time of the trip. On the basis of general observations of passenger reactions to riding helicopters, an increase in trip time to 30 or 40 minutes could well result in a greater percentage of passengers who would be uncomfortable. In addition to obtaining the passengers' assessment of the overall trip ride quality, the investigators obtained passengers' opinions concerning their satisfaction with various motion and nonmotion factors believed to affect ride comfort. Some of the nonmotion factors identified as significant are discussed in the following section. A complete discussion of all factors is given in reference 5.

EFFECTS OF NONMOTION FACTORS

Seat Width

In small aircraft, constraints on interior volume and the economic need to accommodate as many passengers as safety will allow limit the width of seats to a value considerably less than that normally used in larger aircraft and in other public transport vehicles. Figure 3 presents in bar graph form the percentage of passengers expressing dissatisfaction with the width of the seats for each of the four aircraft of the survey. Values range from 39 percent to

67 percent. Also shown in each bar graph is the percentage of passengers that indicated strong feelings about seat-width discomfort. Strong feelings were expressed by less than one-fourth of those giving a discomfort rating to seat width.

Seat width differed between types of aircraft. In the lower part of figure 3 is presented the variation of percent passengers dissatisfied as a function of measured seat width. The data indicate a linear relationship, which, if extrapolated to zero dissatisfaction, indicates that seats ideally should be about 60 cm wide. Although seat shape and firmness differed between aircraft, questionnaire answers revealed that the great majority of passengers considered shape and firmness characteristics of these seats to be satisfactory.

Legroom and Workspace

In addition to width, there are other factors in seating which can affect ride comfort to some extent. In tourist-class sections of conventional-size jet transports, each seat row generally consists of a group of three seats on each side of the aisle. Although seat width may be ample, a passenger sitting in the center seat, when both adjacent seats are occupied, has little freedom of movement, particularly if a task, such as eating a meal, is being performed. Also, both reduced floor width for window-seat passengers adjacent to an incurving fuselage shell and insufficient seat pitch (fore-and-aft spacing) will tend to limit passenger leg movement, which in turn can adversely affect ride comfort, particularly for long trips.

Figure 4 presents in bar graph form the percentage of passengers expressing dissatisfaction with the amount of legroom. Values range from 63 to 73 percent for the fixed-wing aircraft and 28 percent for the helicopter. Of those expressing discomfort, up to 40 percent indicated strong feelings. In the lower part of figure 4 is presented the variation of percent passengers dissatisfied as a function of legroom measured from the front edge of the seat. A nonlinear relationship is indicated, with rapid reduction in dissatisfaction as the legroom is increased beyond 24 cm. Extrapolation of the results indicates that a legroom of about 28 cm should be satisfactory to practically all passengers.

In the survey, passengers were also queried about adequacy of workspace (which may be inferred to include both side-to-side and fore-and-aft space in front of the passenger). Responses were very similar to those for legroom, with values for percent uncomfortable ranging from 66 to 81 percent for the fixed-wing aircraft and 43 percent for the helicopter.

Change In Cabin Pressure

The rate of change of cabin pressure which occurs in aircraft depends on how the aircraft is operated and whether or not the cabin is pressurized. Smaller transports oftentimes are not pressurized, and, except for the Nord

262, this is true for the four aircraft of the survey. Terminal-area maneuvers used to minimize time and costs for short-haul operations generally involve rates of climb and descent which change cabin altitude (pressure) at a far greater rate than recommended. Such was the case for the aircraft surveyed, as shown in figure 5. At the top of the figure is presented in bar graph form the percent passengers dissatisfied for each aircraft. Values range from 26 to 60 percent.

Measured rates of change of altitude or pressure are not available for the four aircraft surveyed but are available from another study utilizing the U. S. Air Force Total In-Flight Simulator (TIFS) research aircraft. In this study, also reported in the present compilation (ref. 6), the effect of various flight maneuvers on ride quality was determined. During the course of the study, written comments incidentally offered by the test subjects for certain maneuvers oftentimes indicated discomfort due to change in cabin pressure. Data for 423 test-subject—maneuver situations are available and have arbitrarily been divided into 5 groups with each group covering a specific, nonoverlapping range of rate of change in cabin altitude. Within each group the percent of passengers offering comments of dissatisfaction was determined and the results are presented in the lower part of figure 5 as a function of rate of change of cabin altitude. At rates from zero to 150 meters per minute, no dissatisfaction was expressed. Dissatisfaction was first evidenced by a small percentage of passengers in the range of rates between 250 and 350 meters per minute and then increased almost linearly to more than 50 percent dissatisfied when the rate of change of cabin altitude was between 850 and 1100 meters per minute. Although a direct relation between these data and the airline survey data cannot be established, the trends shown in figure 5 certainly indicate that all four aircraft must have engaged in rather rapid rates of change of altitude during some portion of their journeys.

Cabin Noise

Passenger surveys indicate cabin noise to be a common source of discomfort for various air, surface, and marine forms of public transportation. Even in large jet transports the noise level can be quite low near the front of the aircraft but can be uncomfortably high near the rear. The four commuter aircraft of the airline survey were no exceptions, as can be seen by the bar graphs presented in figure 6 for percent passengers expressing discomfort from cabin noise. Discomfort levels ranged between 60 and 70 percent for all four aircraft, with 10 to 25 percent feeling strongly. The discomfort results from a noise environment which varies during the trip as the aircraft climbs, cruises, and descends to landing. No attempt has yet been made to equate cabin noise dissatisfaction from this survey with measured noise environment.

Limited information is available, however, concerning effects of cabin noise level on passenger comfort rating. A brief in-house flight study was recently carried out on a Boeing 737 airplane at Langley Research Center. In the study 13 passenger subjects rated their comfort associated with noise and vibration during 1-minute segments of straight and level flight. A range of vibration and noise levels was obtained by varying aircraft thrust, forward speed, and position of the landing gear and drag brakes. Figure 7 presents the variation of average rating of comfort with cabin noise level for a constant (vertical) vibration condition of 0.047g. Comfort rating was indicated by the test subjects on a 5-point scale, with 1 as Comfortable and 5 as Uncomfortable. The data showed a reasonably consistent trend, with average comfort rating increasing from a value of 1 to 4 as noise level in the cabin increased by about 15 dB(A).

The interrelated effects of noise and vibration, however, were not clearly established during the study. This was particularly true when noise level was maintained constant and vibration level was allowed to vary. Much more research is required to establish quantitatively the contributions of noise to passenger discomfort in combined environment situations. Such information is needed since likely candidate concepts of advanced transports (large civil helicopters, powered-lift jet aircraft, etc.) may well have significantly high levels of interior noise in combination with other worrisome environmental inputs. In a paper presented earlier in this compilation (ref. 7), Stephens describes the magnitude of the problem and suggests areas for future research.

Visual Cue Effects

Another factor which can affect passenger ride comfort is the presence or absence of visual cues from outside the vehicle. Most vehicles are equipped with windows for various reasons, some of which are psychological (e.g., to minimize claustrophobia). Although quantitative information regarding visual cue effects on ride comfort is lacking, several observations made during recent ride-quality investigations are worth mentioning. For random-motion ride environments, presence of a window adjacent to the seat appears to have a slightly favorable effect on comfort as observed in preliminary checkout studies of a helicopter to be used for ride-quality research (ref. 8). If an aircraft carries out tight turns at a relatively low altitude, however, an unfavorable effect can result because of the passenger's natural instinct to turn the head simultaneously to look out the window at the rapidly changing visual scene. The resulting change in force vector on the vestibular organs produces a discomfort sensation which can be significant if head motion is rapid. Another unfavorable visual cue situation results from flickering light due to interruption of sunlight by the rotor blades. This situation occurred during the helicopter checkout studies cited above. Only a small percentage (<10 percent) of passengers are generally affected, but the effects can be quite severe. Light flicker at appropriate frequencies can even lead to seizures by persons prone to epilepsy. Fortunately, light flicker can be minimized by darkening light-reflecting surfaces and by tinting the windows.

CONCLUDING REMARKS

For various modes of travel by air, surface, and water, passenger surveys have identified nonmotion factors as important contributors to ride discomfort.

Ride-quality information has been obtained for such factors in passenger studies onboard four types of commuter aircraft. Considerable discomfort was specifically identified for seat width, legroom, and workspace and was quantitatively related in terms of percent passengers dissatisfied as a function of pertinent dimensions. A significant percentage of passengers were dissatisfied with excessive rate of change of cabin altitude (pressure). In a separate study using a research aircraft, percent of passengers dissatisfied was quantitatively related to rate of change of cabin altitude. In the commuter aircraft, a majority of passengers were also dissatisfied with the cabin noise levels. Preliminary exploration in a jet transport indicated that although passenger comfort rating could generally be related to noise level, the combined effects on comfort of noise, vibration, and other factors are complex, and much research is required to better understand and quantify contributions of individual factors to overall passenger discomfort in combined-environment situations. Visual cue effects by passengers sitting adjacent to windows were indicated to affect ride comfort unfavorably for two situations which could occur in transport aircraft.

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TABLE I.- TRAVELER SURVEY SUMMARY BY AIRCRAFT TYPE

AIRCRAFT	DHC-6	N-262	B-99	S-61
TRAVELERS SURVEYED (Total Number)	200	156	133	339
AVERAGE TRIP TIME (Minutes)	20-25	35	25-60	7-10
UNCOMFORTABLE TRIP RIDE (Percent Passengers)	33	31	31	12

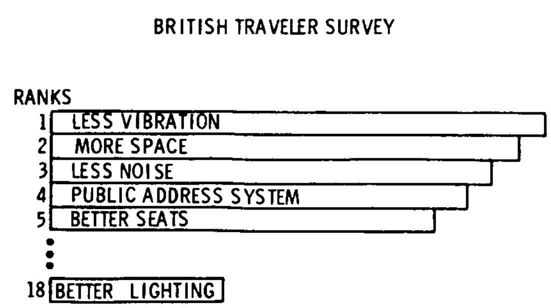

Figure 1.- Preferred improvements for railway coaches from British traveler survey.

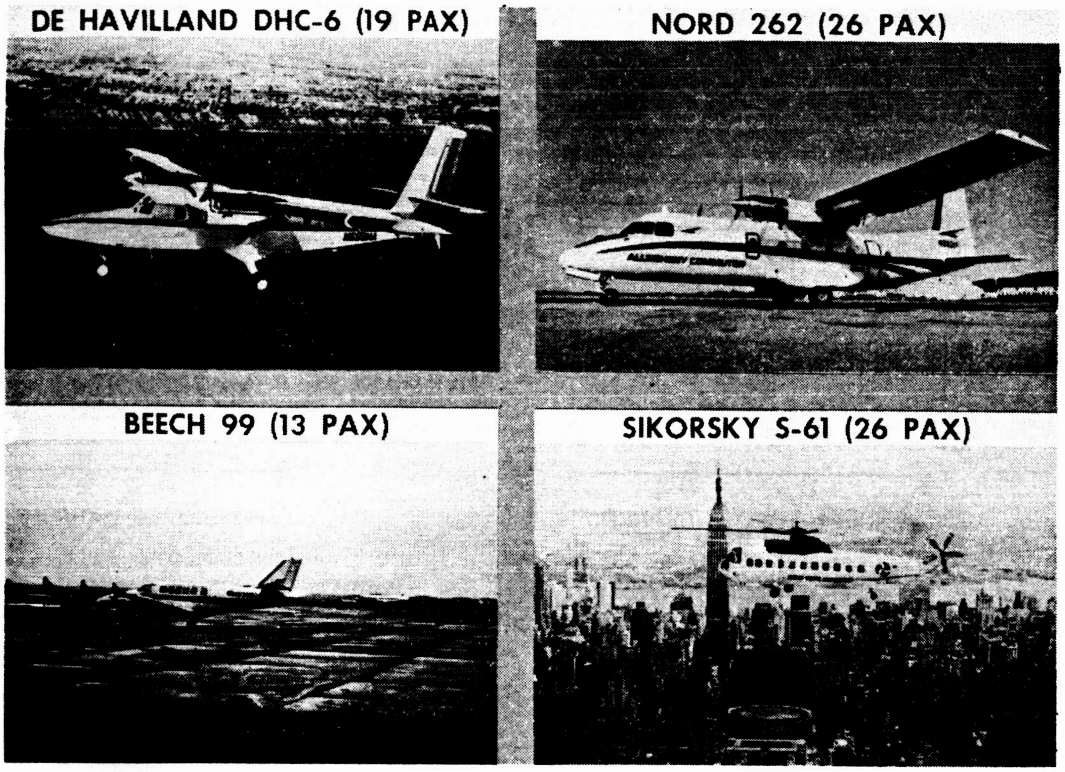

Figure 2.- Aircraft of commuter airline traveler survey.

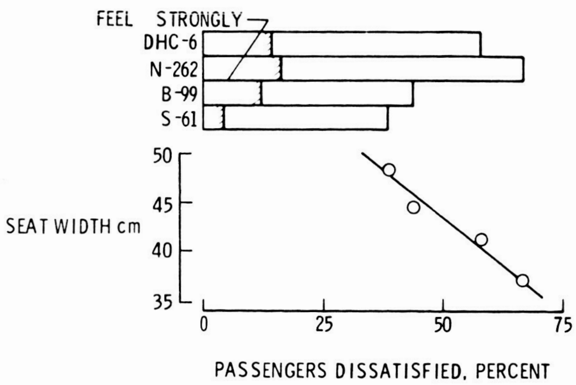

Figure 3.- Discomfort from seat width from airline traveler survey.

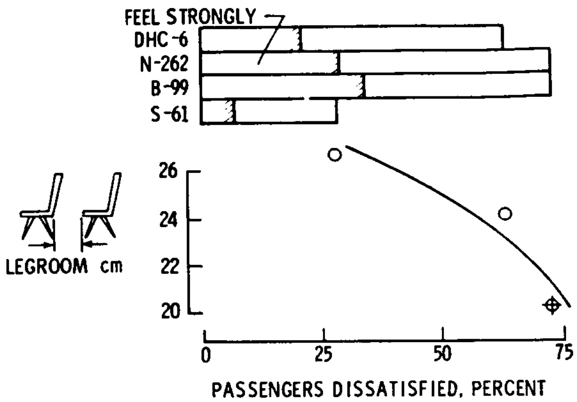

Figure 4.- Discomfort from seat legroom from airline traveler survey.

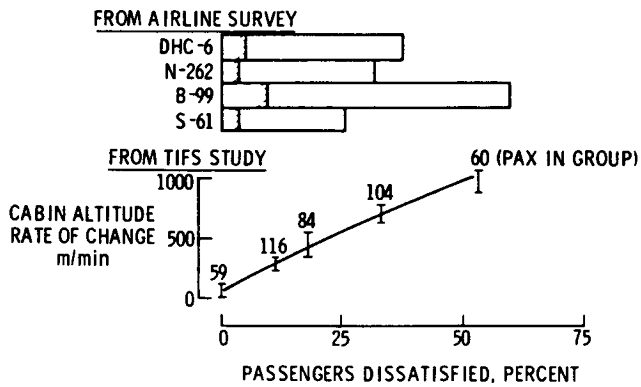

Figure 5.- Discomfort from change in cabin altitude (pressure).

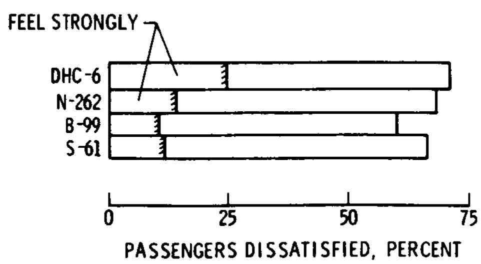

Figure 6.- Discomfort from cabin interior noise from airline traveler survey.

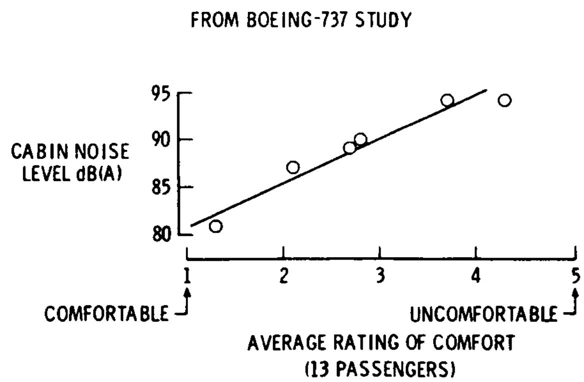

Figure 7.- Effect of noise level on passenger comfort rating.

N76-16760

VEHICLE FOR CIVIL HELICOPTER RIDE QUALITY RESEARCH

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SUMMARY

A research aircraft for investigating the factors involved in civil helicopter operations has been developed for NASA Langley Research Center. The aircraft is a reconfigured 17000 kg (36000 lb) military transport helicopter. The basic aircraft has been reconfigured with advanced acoustic treatment, air-conditioning, and a 16-seat airline cabin.

During the spring of 1975, the aircraft was flight tested to measure interior environment characteristics - noise and vibration - and was flown on 60 subjective flight missions with over 600 different subjects. Data flights established noise levels somewhat higher than expected, with a pure tone at 1400 Hz and vertical vibration levels between 0.07g and 0.17g.

The noise and vibration levels were documented during subjective flight evaluations as being the primary source of discomfort. The aircraft will be utilized to document in detail the impact of various noise and vibration levels on passenger comfort during typical short-haul missions.

6

INTRODUCTION

Civil helicopter exploitation has taken a tremendous upsurge in recent years; the onset of tremendous growth in offshore oil operations and the identification of numerous new applications for the helicopter have been contributing factors in a nearly 10 percent per year growth in sales. If this growth is to continue and, particularly, if any inroads are to be made into the short-haul passenger market, then substantial improvement must be made in the vehicles. It was with this idea that the NASA Langley Research Center embarked on a program to upgrade civil helicopter technology. One of the primary areas of concern in the civil helicopter effort is the evaluation of ride quality aspects of short-haul helicopter operations. As part of this effort, a vehicle has been developed for research studies of a broad range of civil helicopter problems including noise, vibration, and other factors affecting ride quality. (See ref. 1.)

The vehicle to be used as a test bed for civil helicopter studies is a reconfigured CH-53A military transport helicopter. The vehicle has been acoustically treated and configured with passenger seats and air-conditioning to simulate an airline interior. While the formal flight studies with the

CH-53A have not been initiated, the aircraft has been involved in both a subjective flight evaluation involving several hundred subjects and in a number of interior noise and vibration data flights. The interior noise related results of the latt~~ are presented in reference 2.

The present paper discusses the results of the subjective flight evaluation with the Civil Helicopter Research Aircraft and how the characteristics of the aircraft impact ride quality testing.

DESCRIPTION OF AIRCRAFT

Airframe and Systems

The Civil Helicopter Research Aircraft is a reconfigured CH-53A military transport helicopter (fig. 1). The basic characteristics of the aircraft, as reconfigured, are presented in table I. The aircraft was modified from its baseline configuration by the addition of uprated engines which produce nearly 3 MW (4000 shp) each as opposed to about 2.1 MW (2800 shp) each for the original engines. Uprated transmissions to accept the higher power engines were also incorporated. The present control system, rotors, and avionics are unchanged from the basic CH-53A.

Interior

The interior of the Civil Helicopter Research Aircraft (figs. 2 to 5) consists of four basic areas - the cockpit, a vestibule, a passenger cabin, and a rear cabin compartment.

Cockpit.- The cockpit is a basic CH-53A design with some modifications to accommodate the changes made to the electrical system for the heater, cabin air-conditioning, and lighting systems. There is direct access between the pilot's compartment and the vestibule. A jump seat is provided between, and slightly to the rear, of the pilot and copilot. A night flying curtain separates the cockpit and the vestibule area.

Vestibule.- The vestibule is located to the rear of the cockpit from fuselage station 162 to station 222. The walls are covered with nonacoustically treated decorative panels compatible in color and general design to that of the cabin. Located in the vestibule is a passenger air stair entrance door on the right side of the aircraft, an attendant's seat forward of the door, and a galley and coat locker located opposite the entrance door. The vestibule is shown in figure 2.

Main Cabin.- The main compartment (figs. 3 and 4) is a 4.06-m (13.3-ft) long passenger compartment located to the rear of the vestibule between fuselage stations 222 and 382. The passenger compartment contains eight airline quality double seats (seating for a total of 16 passengers) mounted on tracks with a continuously adjustable seat pitch from 76 to 94 cm

(30 to 37 in.) in 2.54-cm (1-in.) increments. The two individual sections of each double seat are separated by an armrest and have individually adjustable backrests. The minimum aisle width, between seat armrests, is 41 cm (16 in.), and the individual seat sections are approximately 43 cm (16.9 in.) between armrests.

The cabin acoustic treatment is comprised of fiberglass batting, skin damping material and a laminate of polyurethane foam, leaded vinyl, and polyurethane foam. The acoustic treatment is capable of achieving a transmission loss of approximately 40 dB in the preferred speech interference level, PSIL (arithmetic average of the 500-, 1000-, and 2000-Hz center frequency octaves). The cabin interior trim is a molded plastic shell attached to the aircraft structure through rubber isolators.

The floor is raised on either side of the aisle by approximately 6.9 cm (2.7 in.) in order to provide better ground level visibility for the passengers. The seat tracks are mounted on the floor and structurally attached to the aircraft floor frames. The entire floor, including the center aisle, is furnished with carpet padding and high pile carpet.

The forward and rear bulkheads are structurally isolated from the airframe by isolators. The bulkheads are acoustically treated and are covered on the passenger side by a cork covering. In the center of each bulkhead is an acoustically sealed door with a break-open feature and a foot operated floor latch to hold it in the open position.

The cabin has both indirect lighting in the valances located over the seats and direct lighting located down the center of the aisle ceiling. The lighting intensity is controlled in the vestibule and has two intensity positions. No individual lights are provided for the passengers. Emergency exit, no smoking, and fasten seat belt signs are also provided in the cabin.

Cabin equipment consists of fire extinguishers, first aid kits, fire axes, and a telephone intercom system capable of communicating with the crewmembers. There are six speakers spaced throughout the cabin through which can be played 8-track tapes or instructions from a microphone located in the cockpit and accessible to the vestibule.

There are four real windows, two on each side of the aircraft, and twelve simulated windows located in the cabin. The real windows are located at the first and third seat rows. Program economics prevented real windows at each seat location. The window size is approximately 38 cm by 38 cm (15 in. by 15 in.). The real windows are of double pane construction, with the inner pane attached to the acoustic treatment, lightly tinted, and provided with an opaque shade.

The cabin contains air distribution ducts for heated and cooled air. The air inlets are from floor ducts located at the bottom of the sidewalls and downward facing valance ducts. The air return duct is in the upper portion of the valance, between the valance and ceiling, and provides a circuitous distribution flow field down the sidewalls, out from the bottom of the walls, up the center aisle, and into the return valance ducts. The normal aircraft heating system provides heat for all compartments. The freon air-conditioning system is located in the compartment aft of the passenger compartment. The air-conditioner is designed to provide a total cooling capacity of approximately 17.58 kW (60,000 Btu/hr) while operating in an ambient temperature as high as 44.5° C and 50-percent relative humidity.

Individually adjustable gaspers for recirculated air are provided for each passenger.

Aft Compartment.- The compartment aft of the passenger compartment (fig. 5) contains the air-conditioner and duct distribution system as well as the cabin lighting power supply. This compartment is partially treated with military type fiberglass blankets placed on the walls and ceiling. The aft compartment contains three windows, each of which is an emergency exit type. The aft compartment will house flight instrumentation systems and an engineer's station for the NASA flight research program.

FLIGHT PROGRAM DESCRIPTION

The flight program is actually independent efforts to define the subjective and objective characteristics of the aircraft. The first was a flight effort with limited instrumentation to define the vibration and noise levels within the cabin. The second was an extensive subjective flight evaluation.

Noise and Vibration Flights

The measurement of noise and vibration levels in the CH-53A was carried out by NASA and Sikorsky engineers. The measurements were accomplished in part during Sikorsky check flights and during scheduled NASA test flights. During the check flights, vibration levels at the blade passage frequency were mapped over the cabin floor area during hover and cruise flight. Likewise, the interior noise levels were mapped during both hover and cruise flight. The NASA test flights included a range of flight conditions - hover, climb, cruise, and descent. During the test flights, fixed microphone and accelerometer locations were utilized. Test flights were flown both before and after the interior was installed. An extensive program to measure environmental conditions, such as noise and vibration, is planned in the near future.

Passenger Evaluation Flights

The passenger evaluation flight program was considerably more extensive than the noise and vibration flight program. The program encompassed a broad geographic spectrum from Boston to Los Angeles, as shown in figure 6. The typical flight mission (fig. 7) entailed a 304.8- to 457.2-m/min (1000- to

1500-ft/min) ascent to cruise altitude (although conditions occasionally required much higher rates of climb), cruise at altitude with an approximate airspeed of 130 to 140 knots, in-flight shutdown of one engine, and descent and landing.

A total of 60 flights were flown during this evaluation.

RESULTS AND DISCUSSION

The results of separate data-measuring flights and subjective evaluation flights of the NASA Civil Helicopter Research Aircraft are discussed in the following sections.

Noise and Vibration Flights

While measurements were taken in a variety of flight conditions, only the cruise data will be discussed.

Figure 8 presents the vertical and lateral vibration levels at the floor for each seat location during a 130-knot cruise; not all locations were measured directly as some were interpolated from the closest available points. The variation in levels is, of course, a function of the mode shapes of the airframe. The levels shown are at a frequency of approximately 18 Hz, or the blade passage frequency of the rotor.

The variation in lateral vibration levels is between ±0.12g and ±0.17g. The range of vertical vibration levels is between ±0.07g to ±0.17g. The corresponding spectra for the vertical and lateral vibrations are shown in figures 9 and 10. The data correspond to vibration levels in the aft cabin, starboard seat locations. The data present the spectrum up to 30 Hz for the 130-knot cruise condition for the vertical and lateral directions. The predominant frequency in both directions is the blade passage frequency of the main rotor, which is 18.3 Hz.

The measured vertical vibration levels (less than 0.1g) in the forward end of the cabin should be acceptable from a passenger acceptance standpoint; however, the lateral levels (greater than 0.1g) are in a more questionable area for passenger comfort and require further study.

A map of the measured interior PSIL (preferred speech interference level) noise levels at each seat location during the 130-knot cruise flight is presented in figure 11. The levels vary from 74 dB PSIL in the forward cabin to 82 dB PSIL in the aft cabin. These levels correspond to levels in the older jet transport aircraft (727, etc.) in the mid- to aft cabin; however, these levels do not adequately reflect a pure tone at 1400 Hz caused by the first stage planetary gear clash in the main transmission. This gear clash frequency, while not in the hearing damage range, is annoying because of its pure tone nature at a level above other noise in the cabin. Further discussion of the interior noise can be found in reference 2.

Subjective Flight Evaluation

The following section presents a discussion summarizing over 60 flights and over 600 subjective reactions to the ride qualities of the aircraft.

During each subjective flight of 15- to 20-minute duration, the subject was requested to complete a questionnaire (table II). A summary of the occupational backgrounds of the test subjects is presented in tables III and IV. The subject sample was generally representative of the helicopter industry and related fields including government (foreign and domestic). The flight experience background of the subjects is as follows: 22 percent had never flown in a helicopter; 29 percent had flown less than 10 times; and 49 percent had flown over 10 times. The average rating on a scale from 1 to 9, where 1 represents very comfortable, was 2.5.

Table V presents a summary of the five top environmental conditions that caused discomfort to the passengers/subjects. High frequency noise was the most frequent problem area, causing discomfort to 64 percent of the subjects. Vibration was the next greatest complaint, with 46 percent experiencing discomfort. Cabin pressure, low frequency noise, and workspace complete the list. It should be noted that the cabin pressure problem was related to rapid climbs and descents which did not occur on every flight. Had the rapid climbs and descents occurred on every flight, the rapid changes in cabin pressure may have been a more widespread problem. Table VI presents the general results of passive problems with the aircraft; that is, problems with the fixed location or fixed facilities within the cabin. The primary complaints were a function of the window locations and size.

In general, according to the subject survey data, the subjects felt the aircraft was competitive with fixed-wing aircraft in overall comfort and were willing and, in the majority of cases, eager to take another flight. The negative aspects most frequently brought out were the high frequency noise, vibration, and the window locations.

Looking now in somewhat more detail at the data, table VII presents the overall rating matrix of each seat location. The number of ratings at each comfort level is shown against seat location. It can be seen that the two-seat rows with windows had lower ratings than the rows without windows. Likewise, the ratings in the rear of the cabin with the higher noise levels and vibration levels are the highest ratings. There is no general trend indicated when either the noise or the vibration levels are compared with the average rating at each seat location; however, there is (as shown in fig. 12) a correlation between the average rating and the noise level for the two rows of seats without windows. Comparing the two rows of seats with windows does not show the effect of the increased noise level. It appears that the lack of windows increases the sensitivity to noise annoyance.

One additional problem area that arose during the testing that may be significant is blade flicker (stroboscopic effect of sunlight through rotor). The problem was not widespread but deserves further attention.

GENERAL DISCUSSION

The Civil Helicopter Research Aircraft, as a tool for ride quality testing, presents a challenging opportunity to investigate a wide variety of conditions. The environment is generally acceptable for short duration missions, although certain aspects have been shown to be marginally acceptable, including the vibration levels and interior high frequency noise levels.

As an instrument for ride quality testing, it would be desirable to have certain conditions where the vehicle would be totally acceptable to the average subject; however, this does not appear possible with this aircraft due to the main transmission noise level being objectionable in most all flight conditions. The vibration level can be varied considerably and can probably be made acceptable at certain airspeeds, although a complete documentation through all conditions and configurations (cg, gross weight, airspeed) has not been conducted to date. An additional area that still requires further definition is the impact of much lower vibration levels at the lower harmonics of rotational speed of the main rotor. From the data, it is obvious that the blade passage frequency of 18 Hz (6 times the rotor speed) dominates all other frequencies by at least an order of magnitude; however, the lower harmonics (1 and 2 times the rotor speed) may be unacceptable because they are nearer the comfort zone frequencies of the body.

The most important area that can be investigated with this aircraft is that of the long-range effects of vibration and noise levels on flights of up to two hours. For flights of this nature that could simulate short-haul missions, the aircraft can carry up to 16 subjects. The aircraft has sufficient variability in vibration level to investigate the reaction of subjects to prolonged exposure to several levels of vibration.

Variables such as seating direction, seat pitch, attitude, and airspeed will all be investigated with the vehicle. Terminal-area maneuvers, blade flicker, and breadboard treatments to reduce reverberation in the cabin will also be investigated.

CONCLUSIONS

A modified version of the CH-53A military transport helicopter has been flown in an extensive program to obtain in-flight subjective evaluation of the general characteristics of large helicopter airliners. The vehicle has also been flight tested by NASA and Sikorsky engineers to obtain preliminary noise and vibration data on the aircraft. This paper has presented a summary of the results of these two flight test efforts and the following conclusions are drawn.

The most serious drawback of the Civil Helicopter Research Aircraft as a ride quality research vehicle is the high frequency noise transmitted from the main transmission. This problem reduces the probability of establishing a totally acceptable baseline condition. The capability to systematically increase the cabin noise levels does exist, however.

Vibration at rotor blade passage frequency and the lower harmonics of rotor speed is somewhat higher than desirable, but it is felt that these levels can be brought to acceptable levels by proper choice of flight conditions and configurations.

Blade flicker, window size and location, and seat pitch have been identified as items requiring further investigation.

The Civil Helicopter Research Aircraft presents an opportunity to investigate not only the many aspects of large helicopter environments that affect passenger comfort, but also to investigate techniques for noise reduction and vibration reduction and to establish the effects of prolonged flight and the exposure to maneuvers that may be required in future terminal-area operations.

REFERENCES

1. Daniell, R. F.; and Warren, R. E.: Short Haul Transportation - The Helicopter's Time is Now. [Preprint] 750598, Soc. Automot. Eng., May 1975.

2. Howlett, James T.; and Clevenson, Sherman A.: A Study of Helicopter Interior Noise Reduction. NASA TM X-72655, 1975.

TABLE I.- CIVIL HELICOPTER RESEARCH AIRCRAFT CHARACTERISTICS

	SI	U.S. Customary
Mission gross weight	16586 kg	36573 lb
Empty weight	11575 kg	25525 lb
Alternate gross weight	19047 kg	42000 lb
High speed cruise	304 km/hr	164 knots
Normal speed cruise	278 km/hr	150 knots
Range	448 km	242 n. mi.
Length	17.2 m	56.46 ft
Height	5.07 m	16.63 ft
Width (blades folded)	4.72 m	15.50 ft
Main rotor diameter	21.9 m	72 ft

TABLE II.- QUESTIONS FOR FLIGHT EVALUATION SURVEY

1. What is your primary occupation or professional title?
2. What organization, industry, or special service do you represent?
3. Please specify your seat location.
4. How many times have you traveled by helicopter?

   This is my first time   1-5   6-10   More than 10

5. Please indicate your overall reaction to this demonstration flight:

1	2	3	4	5	6	7	8	9
Very Comfortable								Very Uncomfortable

6. Check the box which indicates your feelings about each of the following items on this demonstration flight:

	Comfortable	Some Discomfort	Uncomfortable
Pressure (on ears)			
High Frequency Noise			
Low Frequency Noise			
Odors			
Temperature			
Ventilation			
Workspace			
General Vibration			
Sudden Jolts			
Acceleration			
Up and Down Motion (bouncing)			
Backward and Forward Motion			
Sudden Descents			
Turning			

TABLE II.- Concluded.

7. Include your reactio to each of the following statements:

	Yes	No	Comment
The seat has enough leg room			
The window size is satisfactory			
The firmness of the seat is satisfactory			
The window height is satisfactory			
The seat is wide enough			
The window location is satisfactory			
The shape of the seat is satisfactory			
The window location had very little effect on my comfort			
The seat can be adjusted to satisfaction			

8. How does this demonstration flight compare to your experience in a fixed-wing aircraft?

Much better   Better   Equal   Worse   Much Worse

9. After experiencing this demonstration flight, I would: (check only one)

Be eager to take another flight

Take another flight without any hesitation

Take another flight, but with some hesitation

Prefer not to take another flight

Not take another flight

10. Comments.

TABLE III.- OCCUPATION OF FLIGHT EVALUATION SUBJECT SAMPLE

Management	150
Technical	68
Politics	47
Business	23
Pilot	66
Aircraft Ground Support	12
Housewife	7
Miscellaneous	227
No Answer	5
	605

TABLE IV.- EMPLOYING ORGANIZATIONS OF FLIGHT EVALUATION SUBJECT SAMPLE

Oil Industry	38
Helicopter Airline	67
FAA	47
Army	2
Navy	7
Air Force	1
NASA	64
Foreign Military	27
Other Government (Local, State, Federal)	105
Transportation Industry	42
Helicopter Manufacturer	53
Miscellaneous	141
No Answer	11
	605

TABLE V.- PRIMARY ENVIRONMENTAL FACTORS* CAUSING DISCOMFORT TO SUBJECTS

	Comfortable	Some Discomfort	Uncomfortable
High Frequency Noise	36%	49%	15%
General Vibration	54%	42%	4%
Cabin Pressure (On Ears)	64%	31%	4%
Low Frequency Noise	78%	21%	1%
Workspace	84%	15%	1%

* Eleven other factors were noted as causing some discomfort by 11% or less of the subjects.

TABLE VI.- PRIMARY CONFIGURATION FACTORS* THAT ELICITED NEGATIVE COMMENTS

	Yes	No
Window location had little effect on comfort	72%	28%
Window size is satisfactory	77%	23%
Seat is wide enough	77%	23%
Window location is satisfactory	84%	16%
Window height is satisfactory	86%	14%

* Other factors elicited 6% and less negative comments.

TABLE VII.- SEAT LOCATION VERSUS OVERALL RATING

	Seat	Comfortable 1	2	3	4	5	6	Uncomfortable 7	8	9	Average	Totals
Window Row	1	16	10	10	3	3	2	2	0	0	2.59	46
	2	15	8	8	2	1	1	0	1	0	2.28	36
	3	14	12	12	3	2	0	0	0	0	2.23	43
	4	15	9	15	4	2	1	1	0	0	2.49	47
	5	10	3	8	8	3	0	0	0	0	2.72	32
	6	10	5	6	2	0	3	1	0	0	2.63	27
	7	8	1	10	2	1	1	1	0	0	2.75	24
	8	9	4	10	5	0	0	0	0	0	2.39	28
Window Row	9	15	14	12	3	4	0	0	0	0	2.31	48
	10	18	7	12	1	0	2	0	0	0	2.10	40
	11	17	7	9	6	1	0	0	0	0	2.17	40
	12	14	11	13	7	2	0	1	0	0	2.50	48
	13	4	5	4	6	0	0	1	0	0	2.85	20
	14	6	5	6	2	3	2	0	0	0	2.87	24
	15	4	2	7	0	2	1	0	0	0	2.81	16
	16	3	2	6	2	3	0	3	0	0	3.63	19
							Grand Average				2.50	

Note: Averages were obtained by weighing scores by the number of their overall reaction. A rating of 1 received a weight of 1, a rating of 2 received a weight of 2, etc.

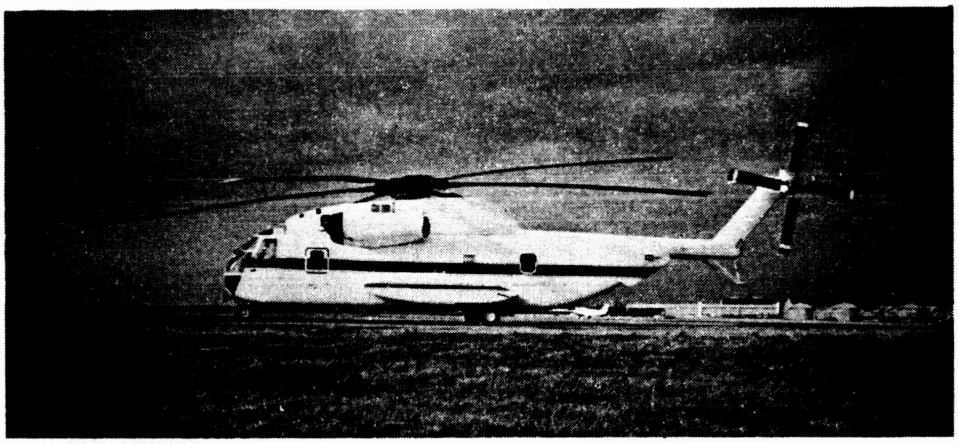

Figure 1.- Civil Helicopter Research Aircraft.

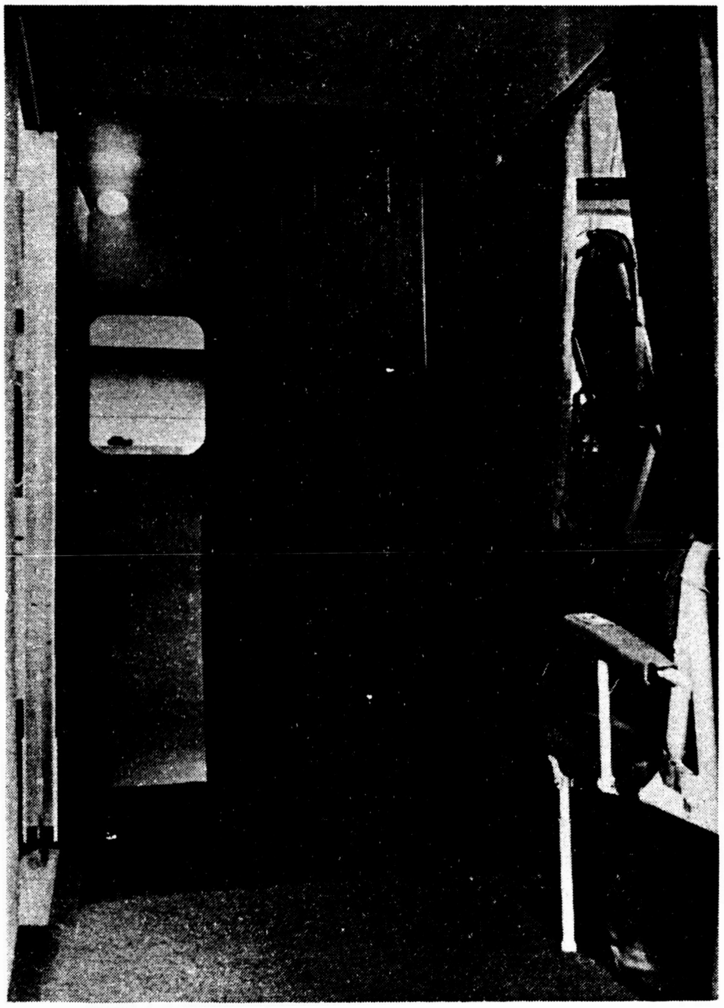

Figure 2.- Civil Helicopter Research Aircraft. Vestibule.

Figure 3.- Civil Helicopter Research Aircraft.
Main cabin (looking forward).

Figure 4.- Civil Helicopter Research Aircraft.
Main cabin (looking aft).

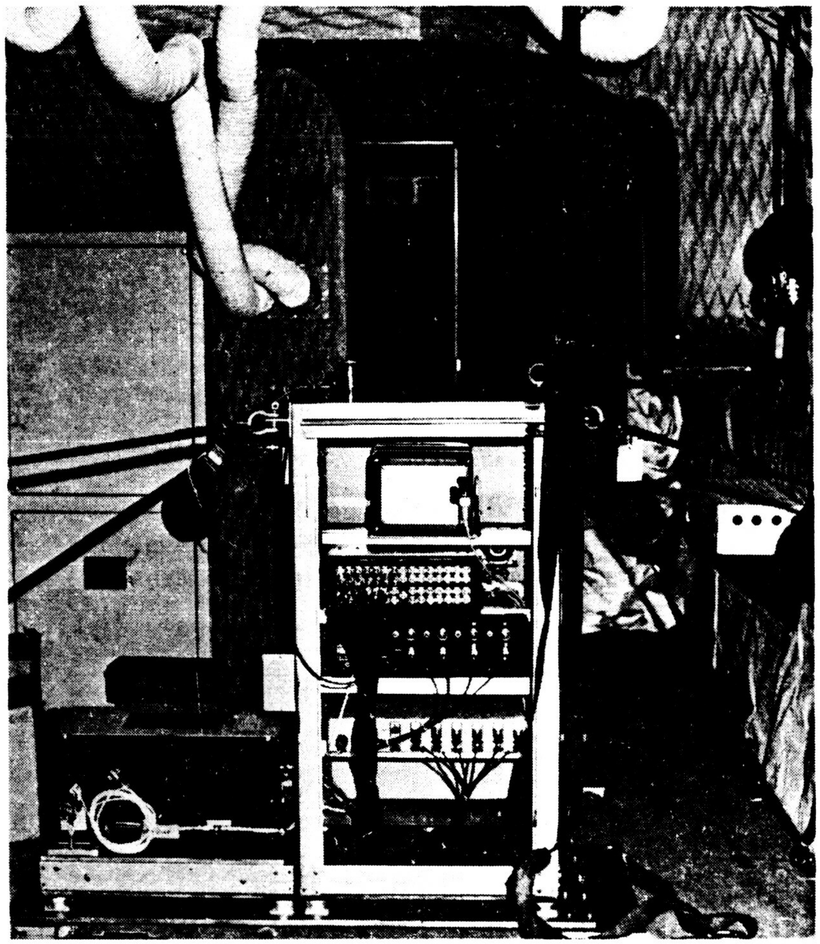

Figure 5.- Civil Helicopter Research Aircraft.
Aft compartment (looking forward).

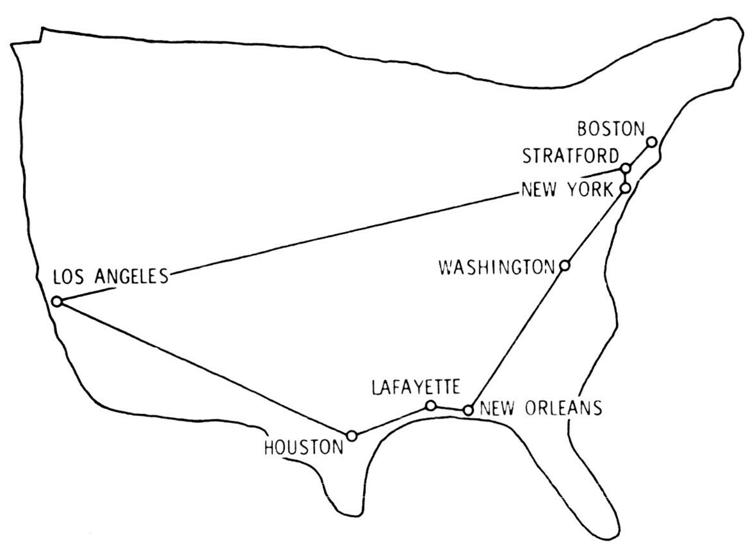

Figure 6.- Locations for Civil Helicopter Research Aircraft
subjective flight evaluation.

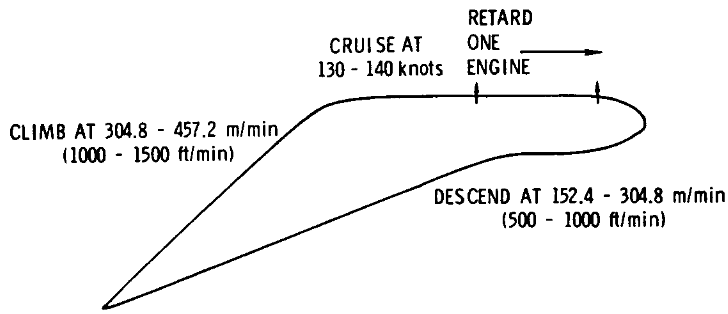

Figure 7.- Typical flight evaluation mission.

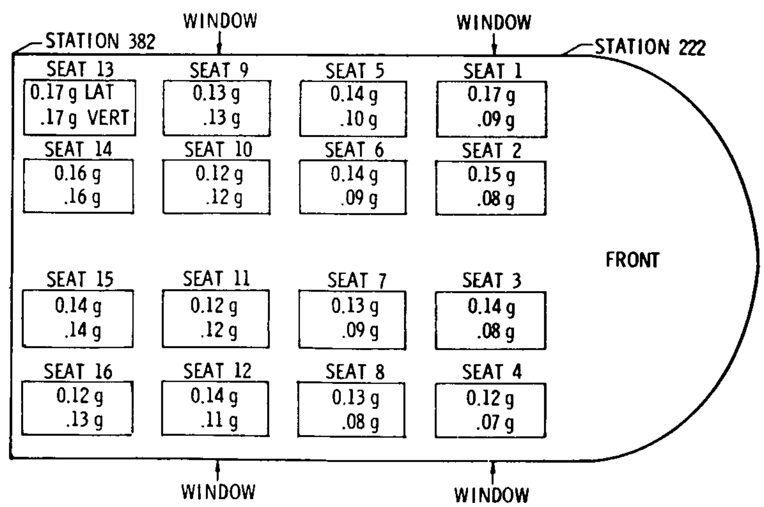

Figure 8.- Aircraft vibration environment versus seat locations.

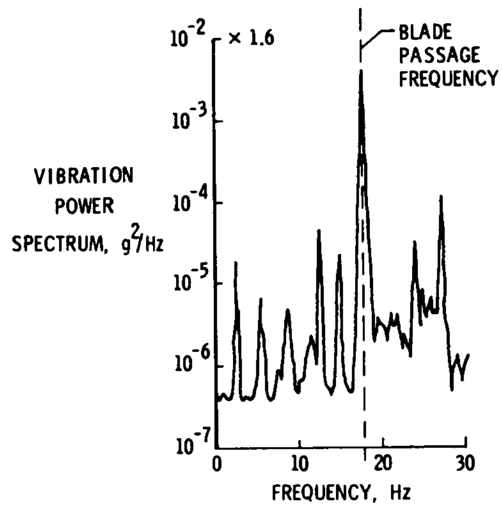

Figure 9.- Vertical vibration power spectrum.

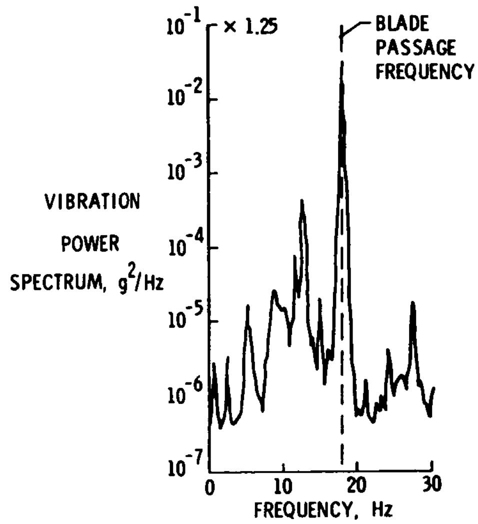

Figure 10.- Lateral vibration power spectrum.

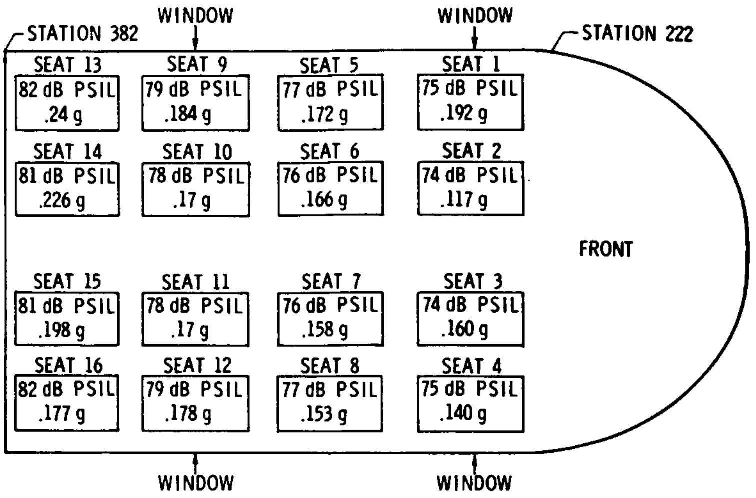

Figure 11.- Aircraft PSIL noise environment versus seat locations.

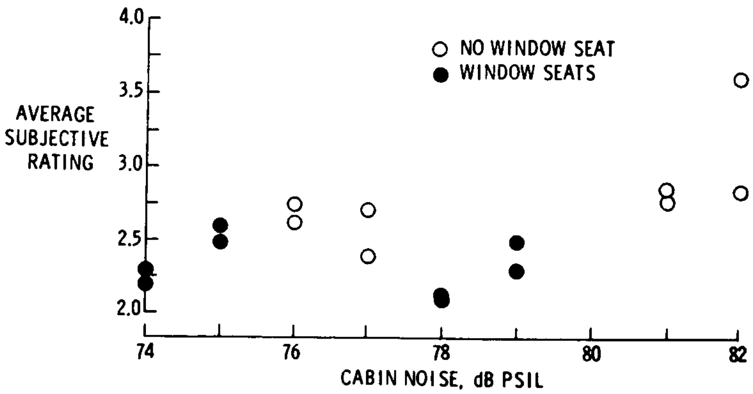

Figure 12.- Average subjective rating versus PSIL noise levels.

N76-16761

COMPUTER ANALYSIS OF RAILCAR VIBRATIONS

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SUMMARY

Computer models and techniques for calculating railcar vibrations are discussed along with criteria for vehicle ride optimization. The effect on vibration of carbody structural dynamics, suspension system parameters, vehicle geometry, and wheel and rail excitation are presented. Ride quality vibration data collected on the State-Of-the-Art Car (SOAC) and Standard Light Rail Vehicle (SLRV) is compared to computer predictions. The results show that computer analysis of the vehicle can be performed for relatively low cost in short periods of time. The analysis permits optimization of the design as it progresses and minimizes the possibility of excessive vibration on production vehicles.

INTRODUCTION

Historically, the analysis of railcar vibrations has been limited in scope presumably because of the complexity and time-consuming nature of the calculations required to solve systems with multiple degrees of freedom. Large numbers of simultaneous equations are necessary to accurately and completely describe the dynamic response of a complete vehicle. Detailed carbody dynamic analysis, for example, can best be made by using a finite-element model which may have as many as 3000 degrees of freedom and perhaps an equal number of structural elements.

At the time the bulk of the intraurban and intercity vehicles in use today were analyzed and constructed, the solution of a complex problem of this type was not practical. The high-speed electronic computer and associated programs that have been developed over the past decade now permit a total system approach and provide solutions to railcar ride quality problems on a timely basis; thus analysis is permitted to guide a vehicle design. The technique presented herein involves two steps: a separate solution of the carbody dynamics as a free-free structure and an integrated total system analysis considering the carbody modal solutions along with truck and track dynamics. This approach is recommended since vehicle response occurs over a wide frequency range and results from excitation of rigid body as well as carbody flexible modes. This can best be illustrated in figure 1, a typical railcar ride quality vibration goal, the 3-hr endurance ISO standard, and the predominant modes of response.

The railcars discussed in this paper fall into the main categories of rapid transit and light rail, the latter being the new SLRV now being built by the Boeing Vertol Company for Boston and San Francisco.

It is the intent of this paper to illustrate how computer analyses can be applied to the prediction and optimization of railcar ride quality.

WHEEL/RAIL EXCITATIONS

Dynamic forces which excite the carbody vertically through the primary and secondary suspension systems result from track misalignment and wheel eccentricity. On jointed rail, typically 11.89 m (39 ft) length, the predominant rail-induced excitation frequencies occur at rail joint and twice rail joint frequencies because of geometrical stagger. Wheel excitations occur at the fundamental rotational frequency of the wheel/axle assembly and are significantly higher in frequency than the periodic rail inputs. Since rapid transit and light railcars usually operate below 112 km/hr (70 mph), it is the rail inputs which excite the low frequency (1 Hz to 1.5 Hz) rigid-body suspension modes and wheel eccentricity which predominantly excites the flexible bending modes of the carbody structure. For typical railcars these structural resonances occur above 7 Hz. It should be noted that excitation of the flexible modes also occurs even with concentric wheels when rail joints are impacted. This response is, however, much less severe than at the critical speeds where the wheel rotational frequency coincides with a carbody flexible natural frequency. The variation with vehicle speed of the wheel and rail joint excitation frequencies is shown in figure 2.

CARBODY STRUCTURAL ANALYSIS

One of the first and most important steps in computer modeling of the vehicle is accurately determining the flexible modes of vibration of the carbody structure. This assessment of the dynamic characteristics of the carbody structure is necessary to avoid undesirable vibration under actual operating conditions. Historically, calculations of the natural frequencies and mode shapes of the carbody were performed by representing the structure as a uniform beam or a series of beam elements. This is a very misleading approach since railcar structures are far from being uniform beams because of their many cutouts for doors and windows. Even attempts to represent such a complex structure by a series of beam elements with shear and bending stiffness properties is unlikely to yield correct results, especially when determining higher order bending modes which contribute significantly to vibration at higher vehicle speeds. Effects of local structure such as floor beams, side sills, and attachments of heavy components demand representation of three-dimensional effects, such as section breathing, bulging, or lateral parallelogramming. Traditionally, carbody structural analysis only involved bang tests to determine the fundamental mode with the carbody shell mounted on a simulated suspension system.

A NASTRAN, finite-element, structural representation of the carbody provides a method for accounting for actual details of the structure, including effects of cutouts for doors and windows. This mathematical model of the three-dimensional gridwork of node points, structural elements, coordinates, and mass data that represents the distribution of mass and stiffness in the actual vehicle is used to form mass and stiffness matrices from which natural modes and frequencies are computed. The SLRV carbody NASTRAN structural dynamic model of the SLRV is shown in figure 3.

Dynamic analysis of the SLRV carbody structure was performed to

(1) Detune carbody structure to insure minimal flexible carbody vibration. Those primary sources of excitation in the operating speed range which were avoided by detuning the carbody structure are shown in figure 4.

(2) Optimize any structural changes required to achieve placement of the carbody vertical-bending natural modes above 13-Hz objective with minimum weight penalty. This insures that these modes will not cause amplification of vibration in the important frequency range of 4 to 7 Hz where human sensitivity is greatest.

Since the carbody is suspended on soft secondary springs, the structural natural frequencies can be considered decoupled from the rigid-body suspension frequencies. This allows the carbody to be analyzed as a free-free structure.

Early in the design of the SLRV, calculations using this finite-element model indicated that the first vertical-bending frequency, the primary source of the carbody flexible vibration, was only 8 Hz with the structural members sized on static load considerations. This meant that the frequency of the first harmonic of wheel rotation, a major source of excitation, would coincide with this natural frequency in the operating speed range and would result in high vibration throughout the car. Examination of the modal deflection data from the finite-element analysis indicated that the two large cutouts required for the center doors contributed most to this problem. Several structural modifications were evaluated on the computer.

The lightest and simplest design which met the 13-Hz frequency objective involved the designing of a truss at the rear of the longitudinal equipment enclosure compartments mounted on both sides of the car underframe. This arrangement provided two longitudinal beams approximately 0.6096 m (24 in) deep running almost from the forward bolster to the articulation bolster. The analytical results were available rapidly and the necessary structural design changes incorporated to ensure that the required natural frequency goal for optimum ride quality was achieved.

RIDE QUALITY VIBRATION MODELS

Once the carbody natural frequencies, mode shapes, and modal masses are determined, a ride quality computer model similar to that shown in figure 5 can be used to optimize suspension system parameters and predict vehicle vibration levels at any desired carbody location. This model was developed for the prediction of vibrations which affect ride quality and for the prediction of primary and secondary suspension dynamic loads. It considers the dynamics of the entire vehicle (truck and carbody) which permits a total systems analysis. Historically, truck and carbody parameters evolved independently, and consideration of the dynamic characteristics of the total vehicle was neglected. To accurately represent the important modes of vibration, appropriate car and truck geometry along with the following degrees of freedom should be included as a minimum:

(1) Carbody: Vertical, pitch, and roll

(2) Flexibly mounted body component: Vertical, pitch

(3) Truck: Vertical, roll, independent side frame pitch

(4) At least two carbody flexible modes.

As main line vehicle speeds increase to 241 km/hr (150 mph) wheel excitation frequencies will occur in the frequency range above 15 Hz. This implies that higher order bending modes will be significant contributors to carbody vibration and must be considered in the analysis. The carbody flexible modes can be described to the model from either finite-element analyses such as NASTRAN or shake test data. The flexibly mounted body components are included to analyze the effect on ride quality of massive sprung components such as the 1587.6 kg (3500 lb) motor alternator on the SOAC or energy storing flywheels on the Advanced Concept Train (ACT). Independent truck side frame pitch, coupled through a torsional spring, is necessary to model trucks which equalize by mechanical pivots or truck frame flexibility.

Elastomer springs and dampers should be made nonlinear by specifying an appropriate table hookup for each element. This is important when analyzing suspension configurations employing elements which can be deflected through large amplitudes or are made highly nonlinear after small initial linear deflections. A common example of this type of suspension arrangement is shown in figure 6. In addition to the features described above, the capability to excite the vehicle at each wheel/rail interface with phased displacement inputs is required. These inputs should be sinusoidal excitations to determine the vehicle acceleration transfer functions and track dynamic profiles to simulate actual running conditions.

The equations of motion for the figure 5 model described above were derived using LaGranges' method and have been programmed at Boeing Vertol on an IBM Continuous System Modeling Program (CSMP) and on a Xerox Sigma 9 machine using the SL1 language.

RESULTS OF PARAMETRIC STUDIES

To illustrate the importance of accurately simulating not only the amplitude and phase of rail excitation, but also the vehicle geometry and suspension characteristics, selected ride quality computer predictions of carbody vertical acceleration over bolster are presented in figures 7 to 11. These results are for a 22.86 m, 31752 kg (75 ft, 70000 lb) rapid transit car having a first vertical bending frequency of 5.88 Hz, concentric wheels, traveling 128.72 km/hr (80 mph) on 11.89 m (39 ft) jointed rail. The computer model used was similar to that shown in figure 5. The intent is to illustrate, for example, the integrated approach to evaluate the effect of staggered rail joints. This requires consideration of carbody and truck modes. For each two seconds of analytical data shown, the dynamic track profile was phased to all eight wheels based on vehicle speed, truck wheelbase, and truck spacing.

Figure 7 compares over bolster vertical acceleration levels calculated by exciting the model with measured right and left dynamic track profile excitation to acceleration levels determined where the left rail profile is assumed similar to the right rail profile. From these data several significant conclusions can be determined. Although the left rail profile looks significantly different than the right, the wavelength content (11.89 m (39 ft)) and overall amplitude levels are similar. The large spike at 1.75 seconds occurs because of an anomaly in the right rail characterized by a dip between rail ends. The mixed frequency of the waveform is composed of 3-Hz and 6-Hz vibration corresponding to rail joint and twice rail joint excitation frequencies at 128.72 km/hr (80 mph). (See figure 2.)

The effect of the 5.9 m (19.5 ft) stagger between rail joints can be seen in figure 8. A comparison is made between acceleration levels calculated using two identical profiles; one having the right rail mathematically displaced 5.9 m (19.5 ft) to shift the right to left input phasing. These data show that the predicted amplitude with the rail excitation not staggered is in error by as much as 100% and does not have twice rail joint frequency content.

Figure 9 presents a comparison of vibration levels calculated using a measured track dynamic profile to predictions using an "idealized" jointed rail profile. Both the amplitude and frequency content of the waveform are similar except for the previously discussed spike at 1.75 seconds. The importance of considering rail input phasing is well illustrated in figure 10 where the effect of staggered versus no stagger idealized rail is shown. In this case the amplitude and frequency content is significantly in error where input phasing is neglected.

An example of the effect of truck geometry can be seen in figure 11. Comparison is made between a truck having an 208 cm (82 in) wheelbase and an identical vehicle modeled with the wheelbase assumed equal to zero. Over bolster acceleration levels for the zero wheelbase case are approximately twice those calculated for the 208 cm (82 in) wheelbase. This attentuation in response occurs because the resultant input amplitude to the secondary suspension springs is reduced as each wheelset encounters a rail joint.

COMPUTER ANALYSIS OF SOAC RIDE VIBRATIONS

Background

The SOAC was developed to demonstrate the state-of-the-art and was assembled from available carbody structure and truck components using analyses common to the industry at that time. The car has demonstrated good riding qualities in testing at Pueblo over several combinations of jointed and welded rail and received favorable comment in public service at NYCTA, MBTA, CTS, CTA, and SEPTA. The ride demonstrates the advantage of an application of modern state-of-the-art in rubber chevron primary suspension and airspring secondary suspension. Analyses of the SOAC ride, however, indicate further improvements are possible by tuning the carbody and trucks for compatibility.

Ride quality, carbody shake tests, and wheel concentricity tests were performed on the State-Of-the-Art Car (SOAC) at the High Speed Ground Test Center at Pueblo, Colorado. These ride quality tests conducted on the welded fail sections of the 128.72 km/hr (80 mph) UMTA test oval indicated that there were noticeable vertical car floor vibrations near 72.4 km/hr and 128.72 km/hr (45 mph and 80 mph). Test data at empty car weight show that near 128.72 km/hr (80 mph) the floor vibrations are predominantly 15 Hz and that near 72.4 km/hr (45 mph) the carbody vibrations are predominantly in the 7.8 to 8.2 Hz frequency range.

Shake testing of the SOAC indicated that there are two vertical carbody flexible modes of interest occurring at frequencies below 20 Hz, the first vertical bending made at 8.1 Hz, which is a characteristic of the primary structure of the car, and a 15.2 Hz higher order mode involving vertical bending of the underfloor lateral motor alternator support beams and the primary side sill structure in the area of the rear door cutouts.

The 1587.6 kg (3500 lb) motor alternator, located at mid car, is flexibly mounted on elastomers giving an uncoupled vertical frequency of 15 Hz. This counting causes the motor alternator to act as a highly damped dynamic absorber, attenuating response from the second bending mode at 15.2 Hz. This was confirmed by mechanically "locking out" the elastomer mounts during the shake test. Figure 12 compares frequency response curves with the motor alternator flexibly and rigidly mounted to the underfloor structure.

Near 72.4 km/hr (45 mph) a resonant condition exists where the wheel rotational frequency coincides with the first vertical bending frequency at 8.1 Hz. Acceleration data shows that vertical motion at the forward end of the car is out of phase with vertical motion at the mid car location. This is expected since these two positions are located on opposite sides of the node of the first mode. Near 128.72 km/hr (80 mph) the wheel excitation frequency coincides with the 15 Hz higher order body bending mode resulting in carbody vibration throughout the car.

Data at speeds slightly higher and lower than 128.72 km/hr (80 mph) and 72.4 km/hr (45 mph) show that vibration levels are reduced, and this is expected since the wheel excitation frequency is then separated from the carbody bending frequencies. From these data, it could be concluded that wheel excitations resulting from wheel eccentricity provide significant harmonic inputs in the frequency range of the two carbody flexible modes. Figure 13 shows the frequency spectrum for the SOAC vehicle.

Wheel concentricity tests indicated that wheel out-of-round was typically 0.018 cm (0.007 in) with a maximum of 0.025 cm (0.010 in).

Test Results and Analytical Predictions

SOAC ride quality data collected at 72.4 km/hr (45 mph) and 128.72 km/hr (80 mph) is shown in figure 14 and compared with analytical predictions. Using the ride quality computer model shown in figure 5, calculations were made at 128.72 km/hr (80 mph) using welded rail dynamic track profile excitation superimposed with 0.025 cm (0.010 in) wheel out-of-round excitation at the wheel rotational frequency. The predicted vertical acceleration levels agree well with the test data, both amplitude and frequency content. If the carbody second flexible mode had not been included in the model, only the low-frequency response at the rigid-body frequencies would have been predicted. These data clearly show the importance of analytically representing the carbody dynamics including flexible and rigid-body modes on the suspension system, truck dynamics, and rail plus wheel excitations.

It should be noted that the State-Of-the-Art Car was developed from an existing carbody structure and truck components and was not optimized by extensive computer analysis.

COMPUTER ANALYSIS OF SLRV RIDE VIBRATIONS

Background

The SLRV is a 32659 kg (72000 lb), three truck, articulated car having a maximum speed of 96 km/hr (60 mph). Vertical ride quality data was collected on this vehicle at the Boeing Vertol jointed rail test track in the 32.2 km/hr (20 mph) to 64.4 km/hr (40 mph) speed range. Previous NASTRAN dynamic analysis of the carbody structure indicated that the first vertical bending frequency at 13 Hz was well above the wheel rotational frequency throughout the operating speed range; thus minimum flexible response is insured.

Test Results and Analytical Predictions

SLRV ride quality data, mean vertical acceleration at station 55 on the car centerline as a function of carspeed, is shown in figure 15 and compared to analytical predictions. Calculations were made at 8 km/hr (5 mph) increments from 32.2 km/hr (20 mph) to 64.4 km/hr (40 mph) using the analytical model shown in figure 16 and jointed rail dynamic track profile excitation. This model includes the three rigid-body vertical/pitch modes associated with an articulated car. Predicted vertical acceleration levels agree well with the test data over the speed range investigated and showed that vibrations at the rigid-body suspension frequencies dominated the response and were maximum near 56.3 km/hr (35 mph). At this speed the rail excitation at joint frequency is close to the car out-of-phase rigid-body pitch frequency on the secondary suspension; the amplitude of response being limited by the orifice damping provided by the airsprings.

Figure 17 compares ride quality vibration predictions on welded rail to measurements collected during tests conducted at Boston. These data are presented against the SLRV ride quality goal and show the low vibration levels throughout the entire operating speed range. This results predominately from the low-frequency secondary suspension and the detuned carbody structure.

CONCLUDING REMARKS

The computer technology which has been developed over the years primarily for aerospace applications provides the ability to solve many railcar ride quality problems that only a decade ago were treated with oversimplified analyses. It is now a reality that these computations can be performed accurately, for relatively low cost, in short periods of time prior to the detail design of the carbody structure and suspension components. Experience in applying these new analytical tools is still a prerequisite for success but the computer models described in this paper permit analysis to impact a design, reduce costs, and lower the possibility of problems on the production vehicle.

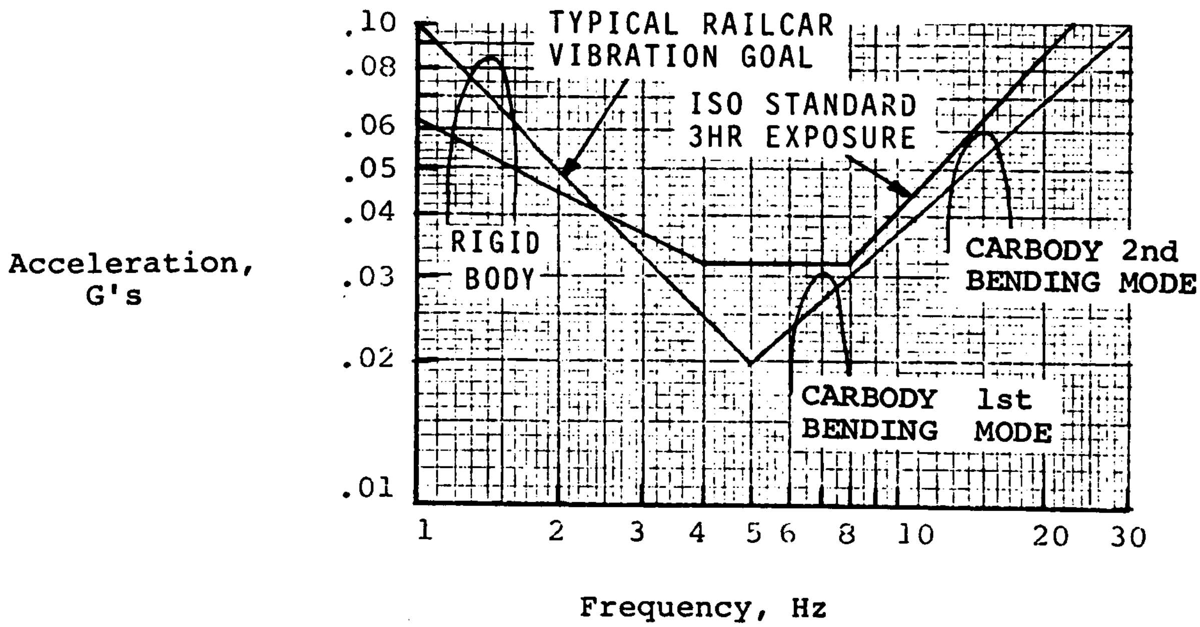

Figure 1.- Typical ride quality vibration goal and modal responses.

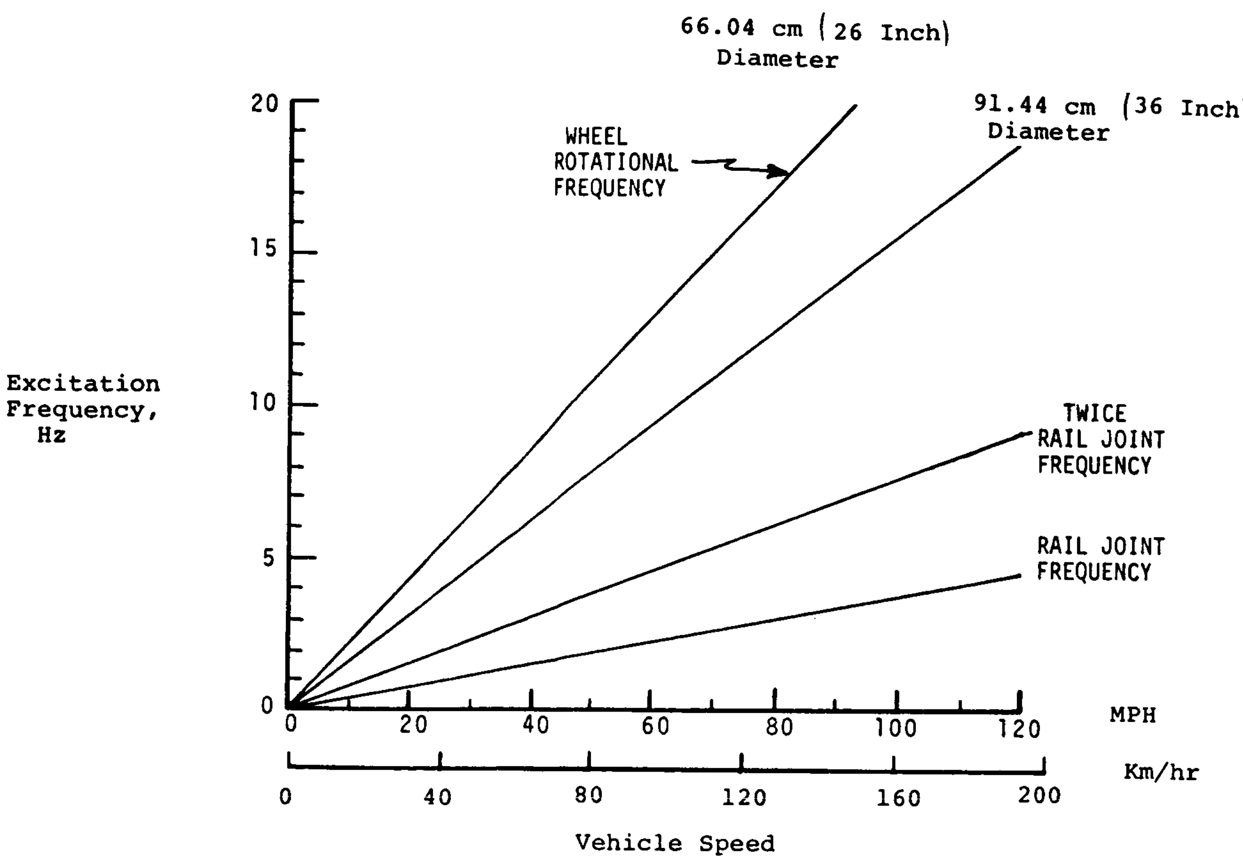

Figure 2.- Wheel/rail excitation spectrum.

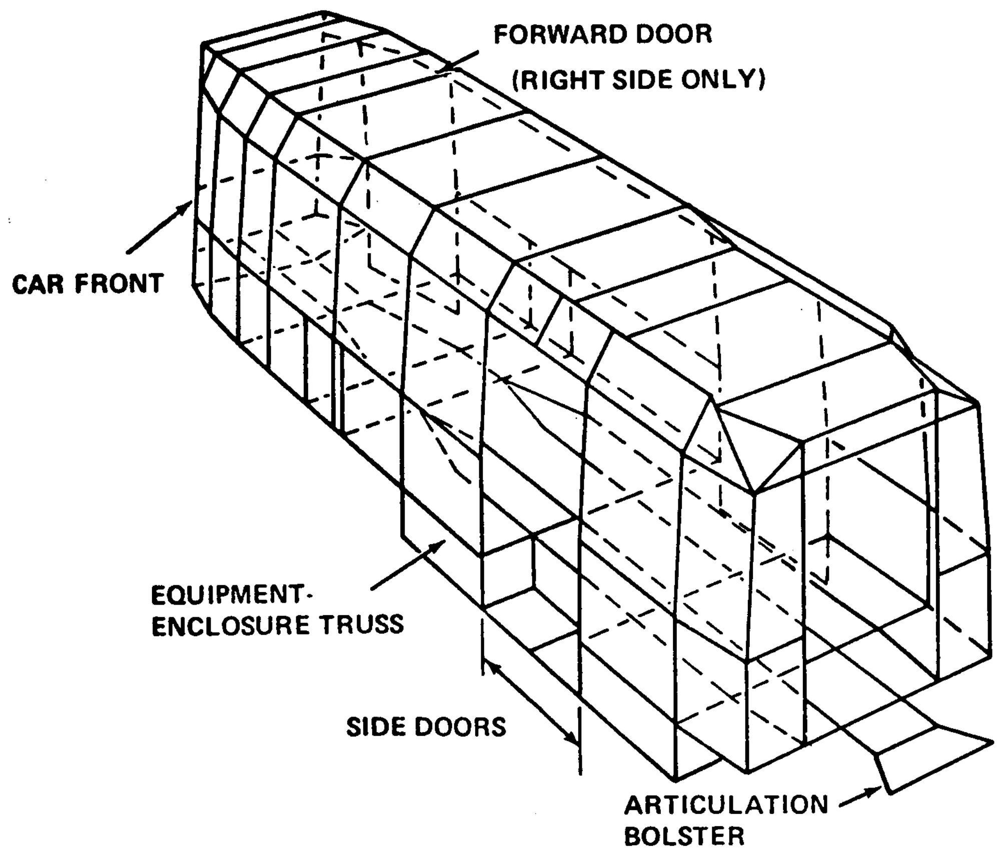

Figure 3.- NASTRAN idealization of SLRV car body.

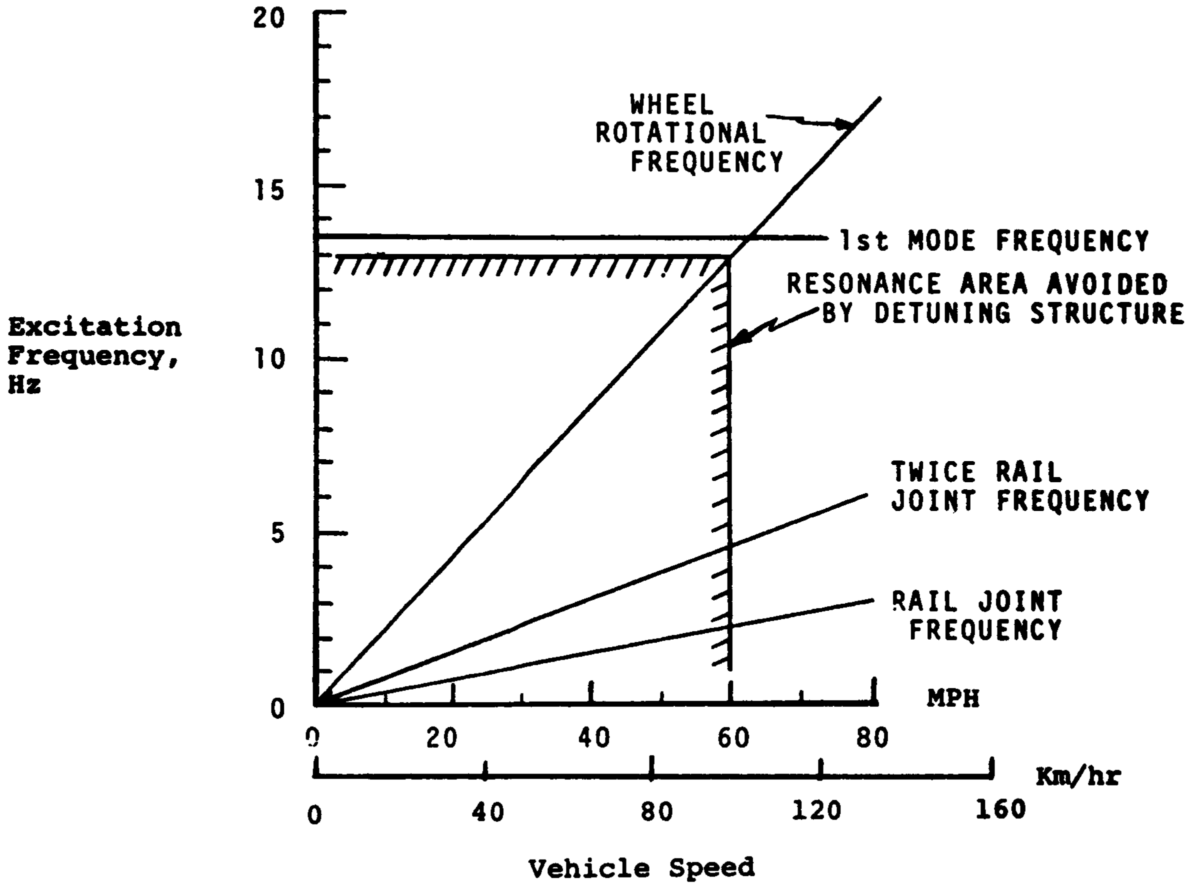

Figure 4.- SLRV wheel/rail excitation spectrum.

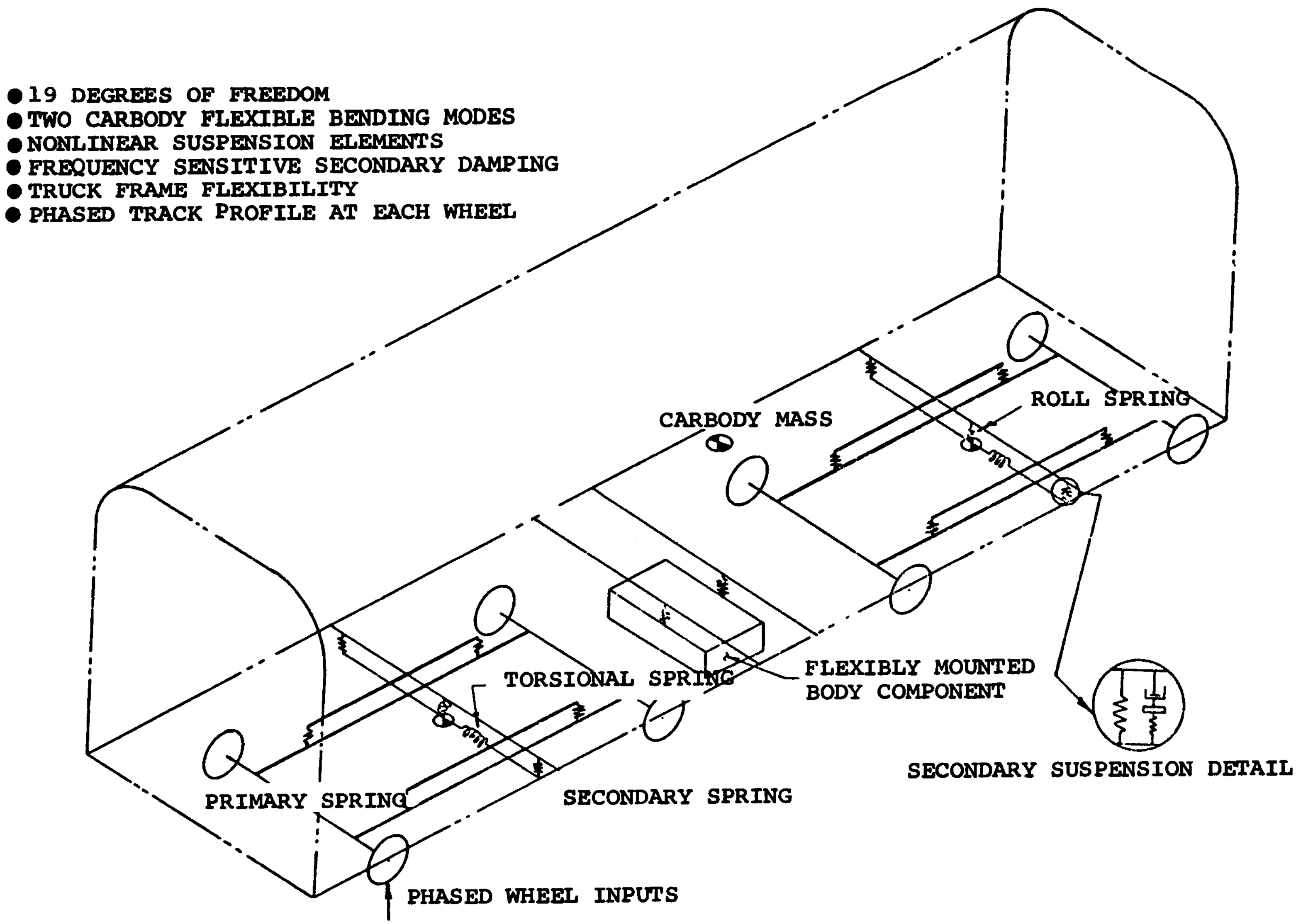

Figure 5.- Railcar ride quality vibration model.

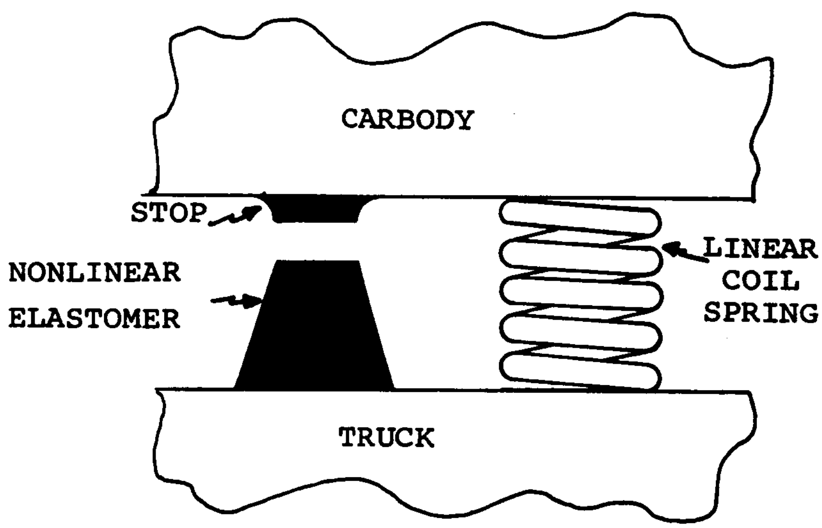

Figure 6.- Typical nonlinear secondary suspension system.

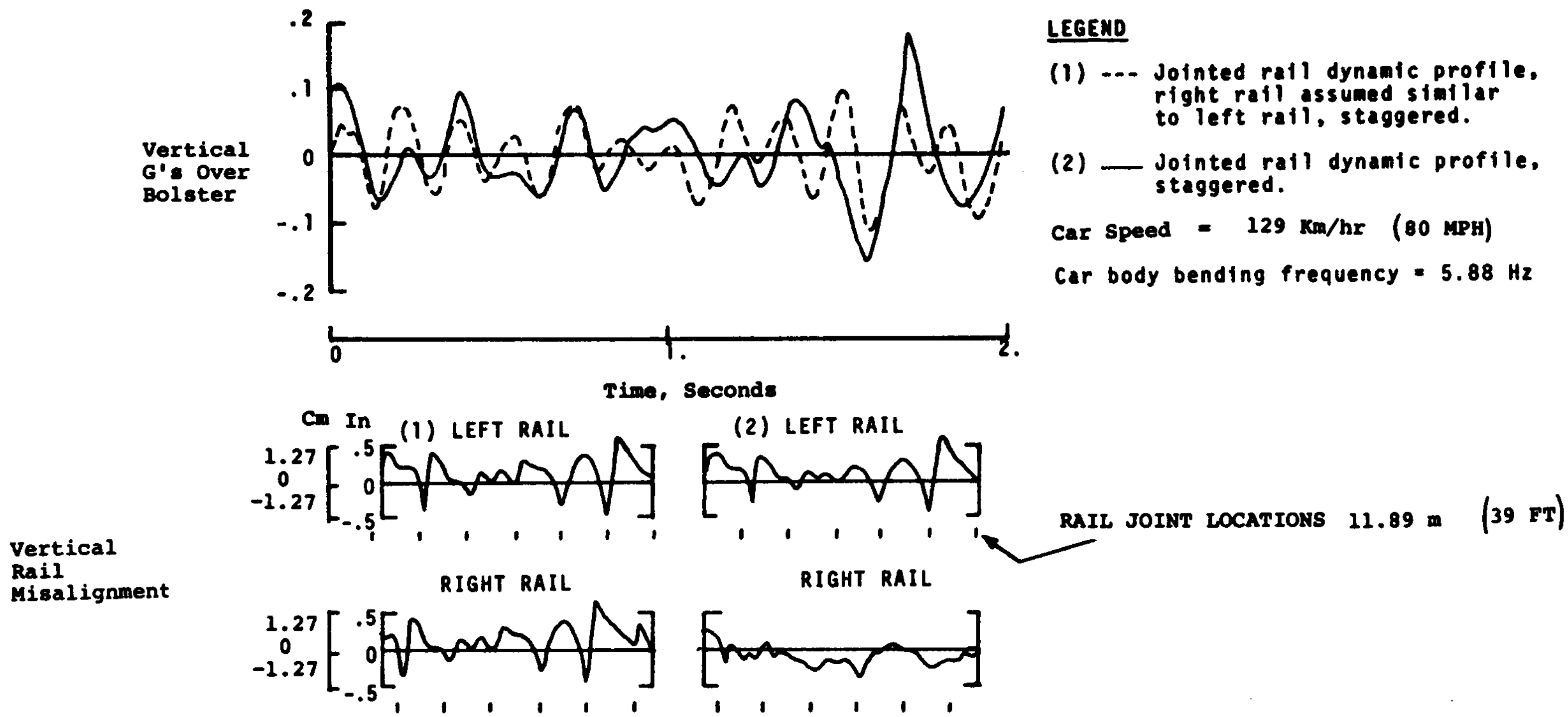

Figure 7.- Predicted car body vertical acceleration over bolster. Measured profiles.

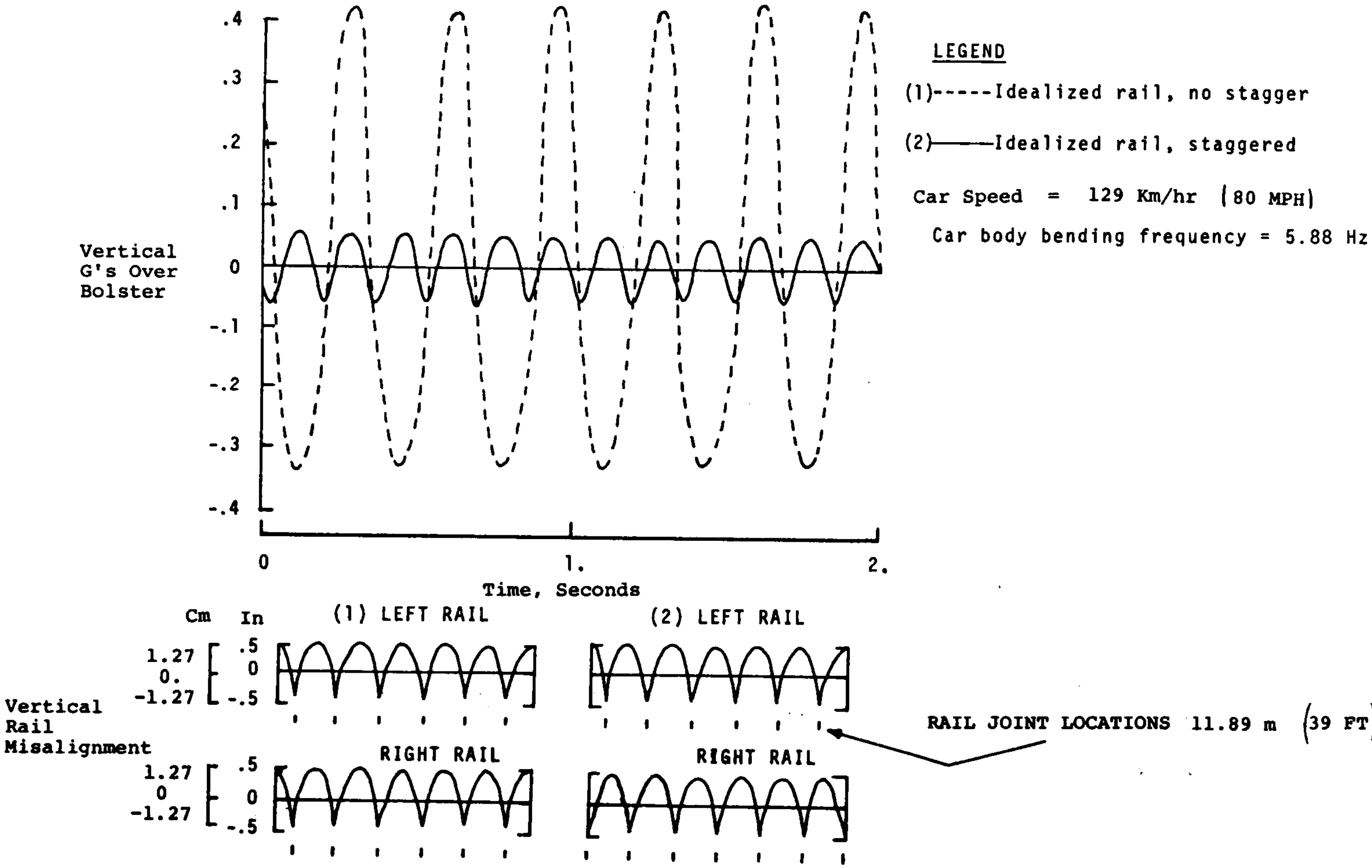

Figure 8.- Predicted car body vertical acceleration over bolster. Stagger.

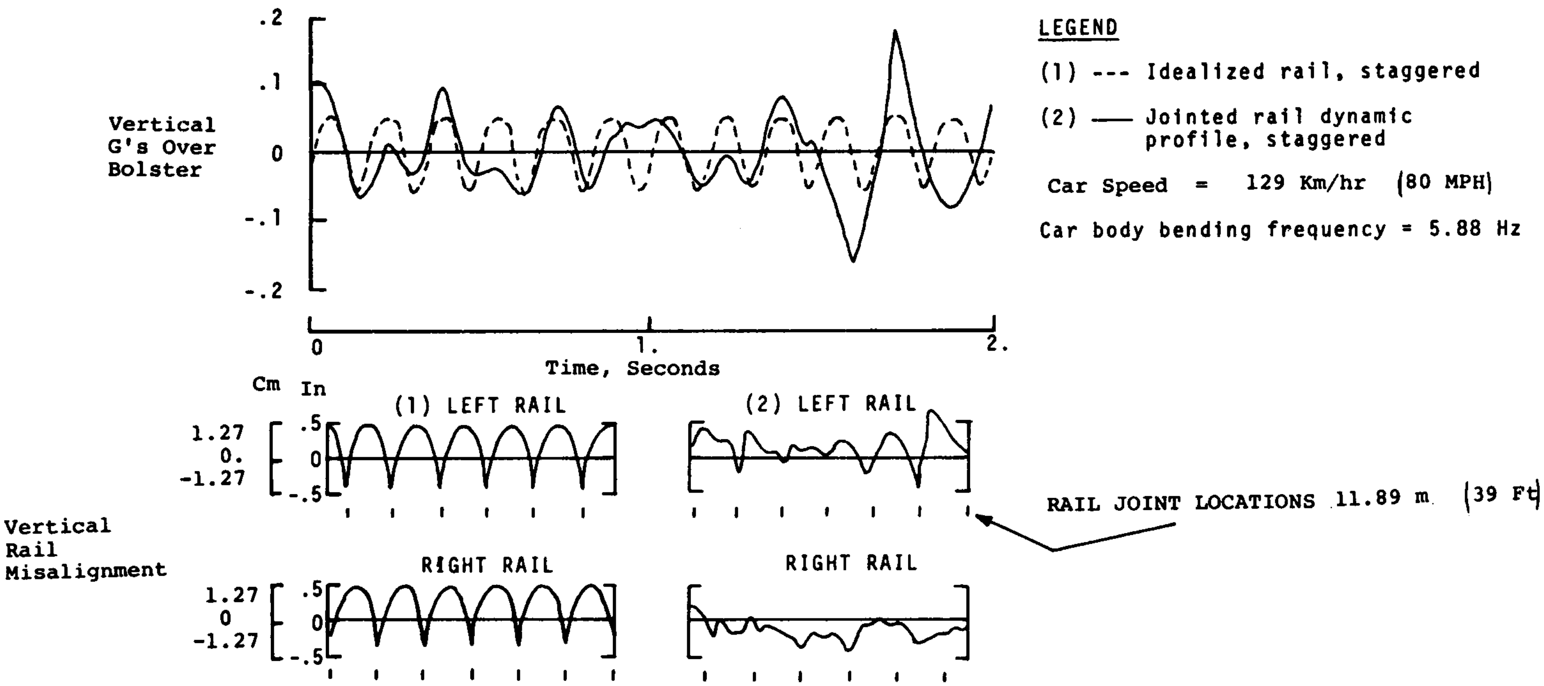

Figure 9.- Predicted car body vertical acceleration over bolster.
Measured values compared with those of ideal profile.

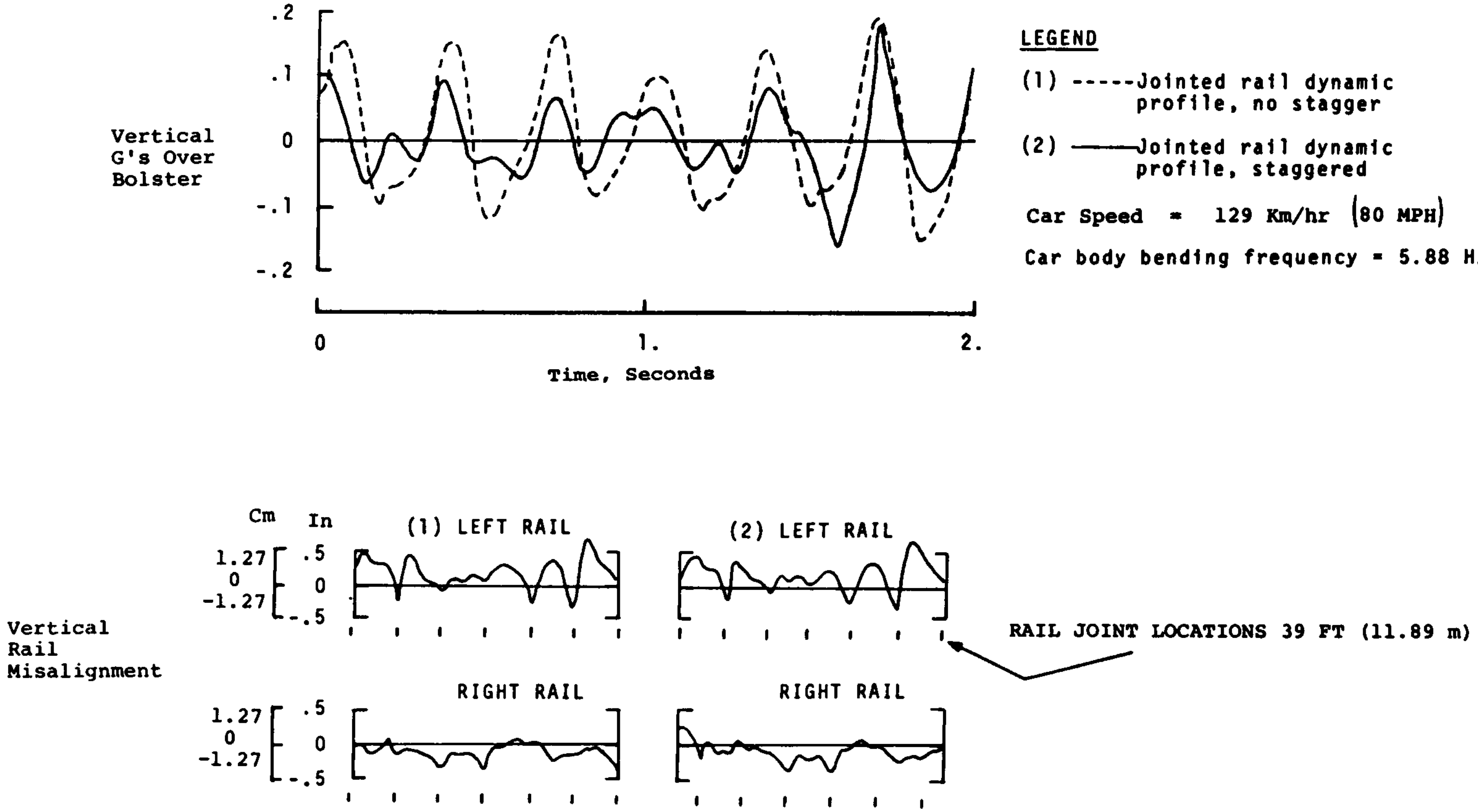

Figure 10.- Predicted car body vertical acceleration over bolster. Stagger.

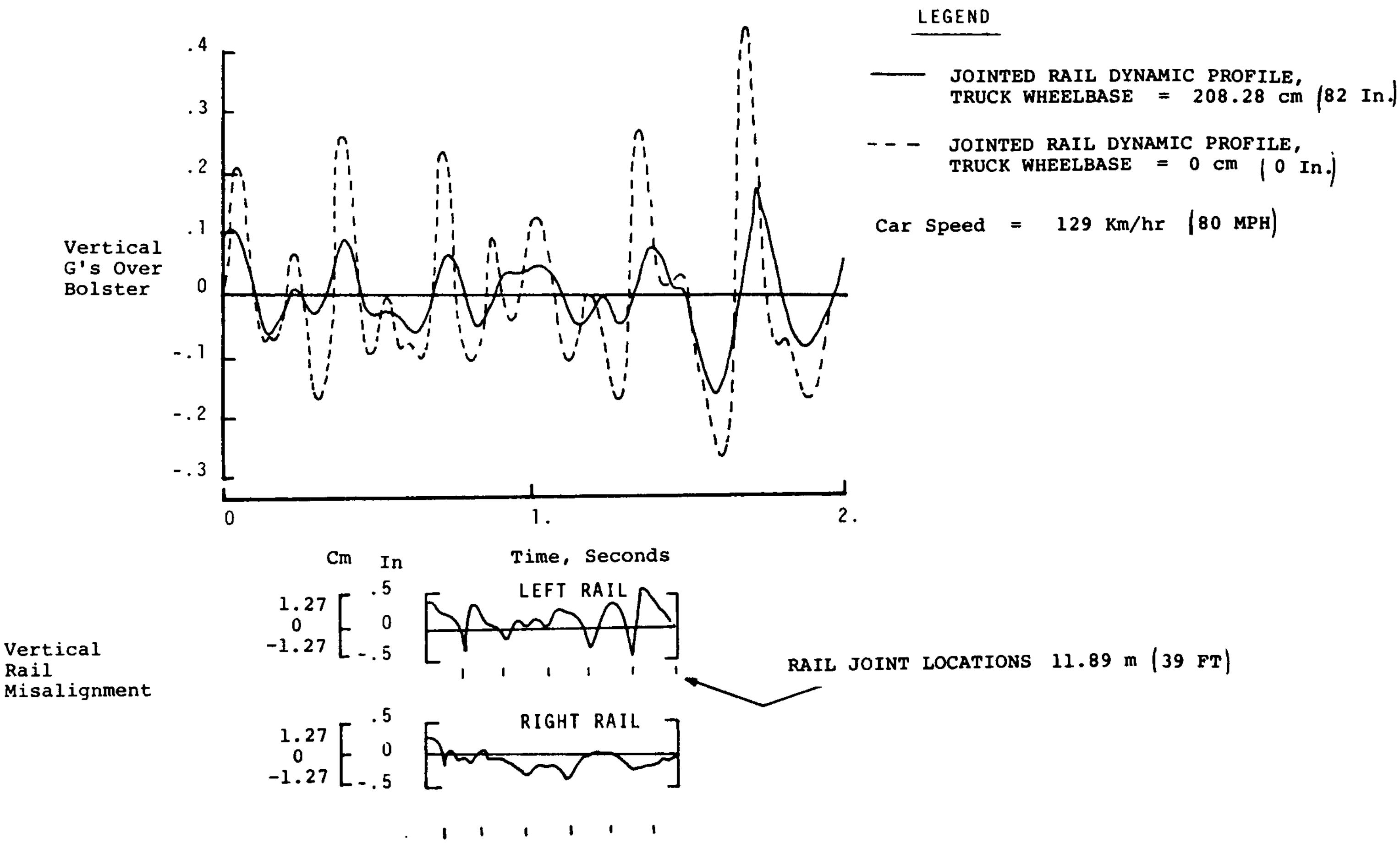

Figure 11.- Effect of truck wheelbase on car body vertical acceleration over bolster.

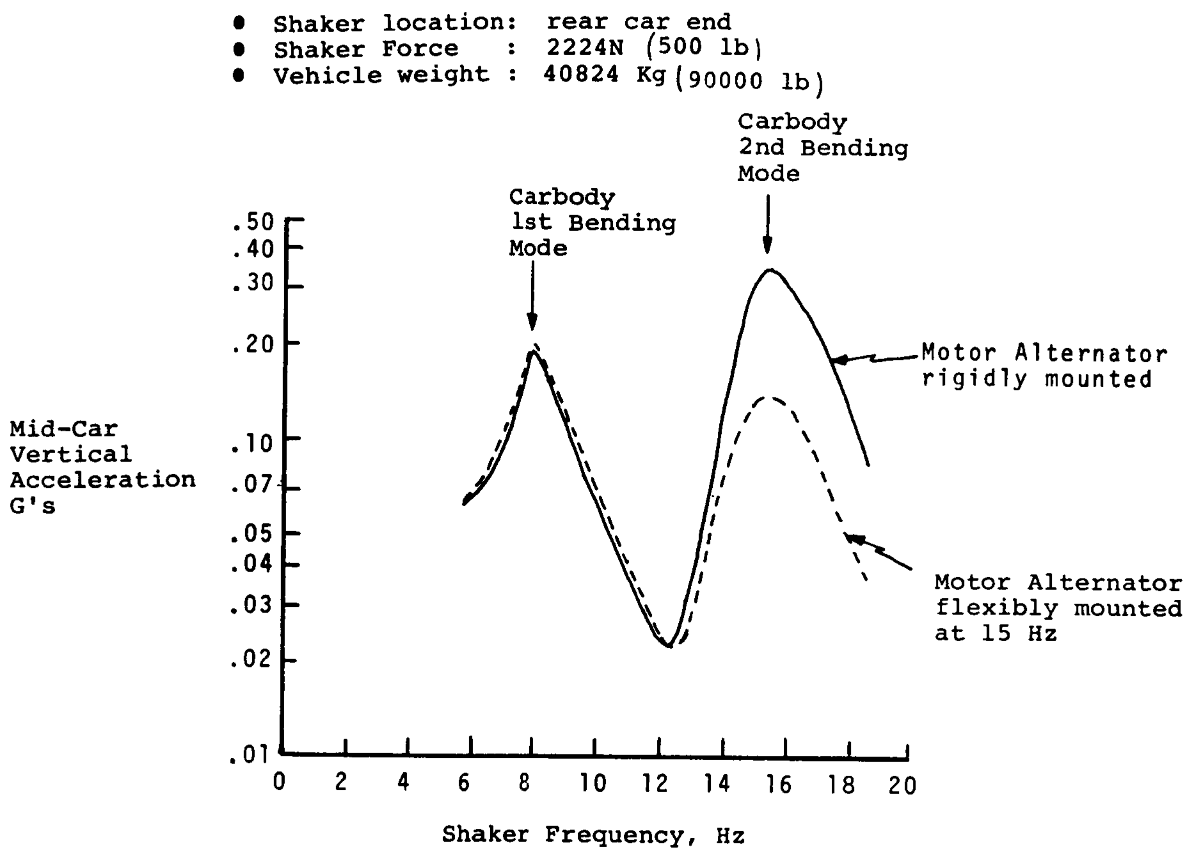

Figure 12.- SOAC shake test frequency response curves.

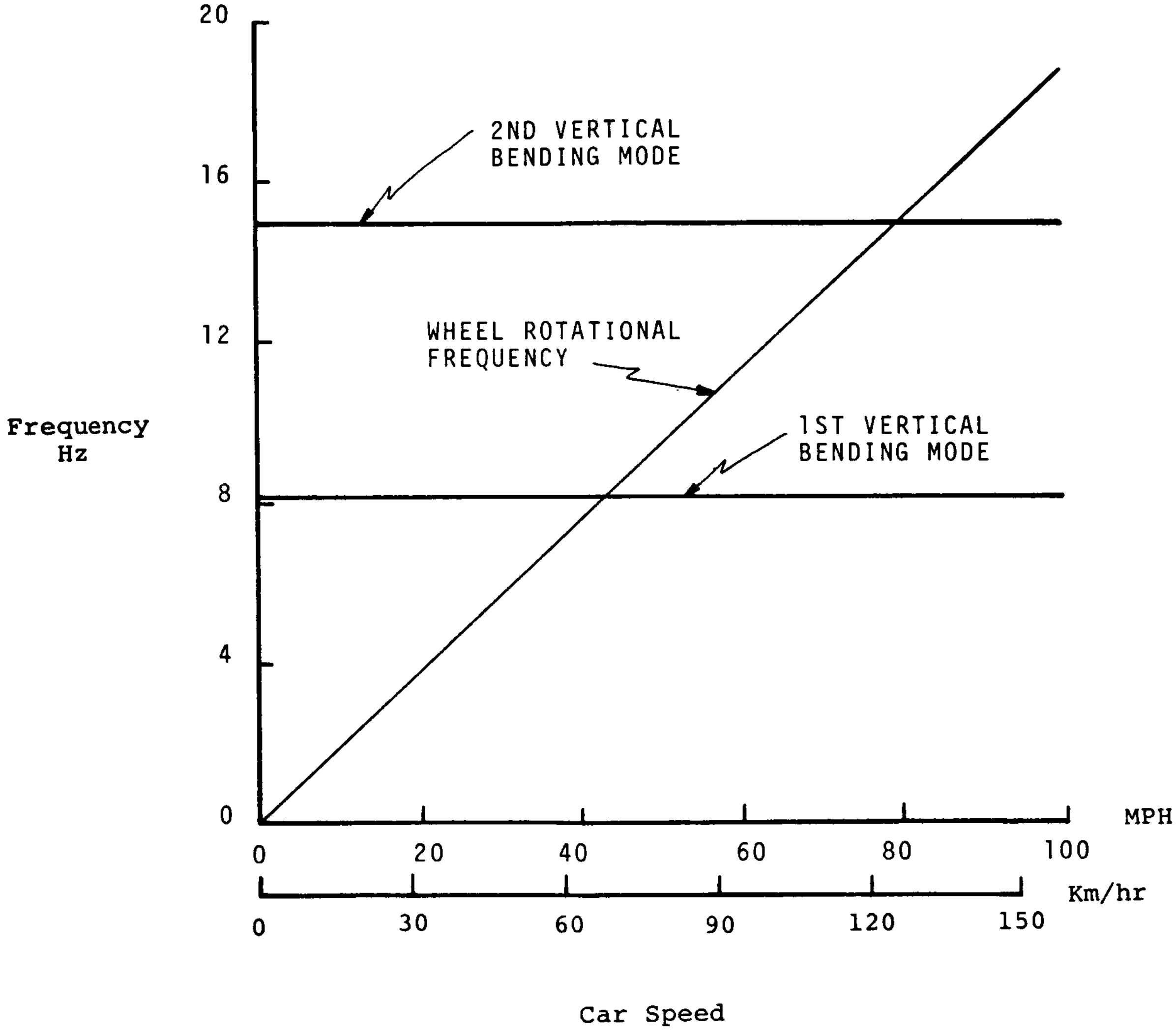

Figure 13.- SOAC frequency spectrum.

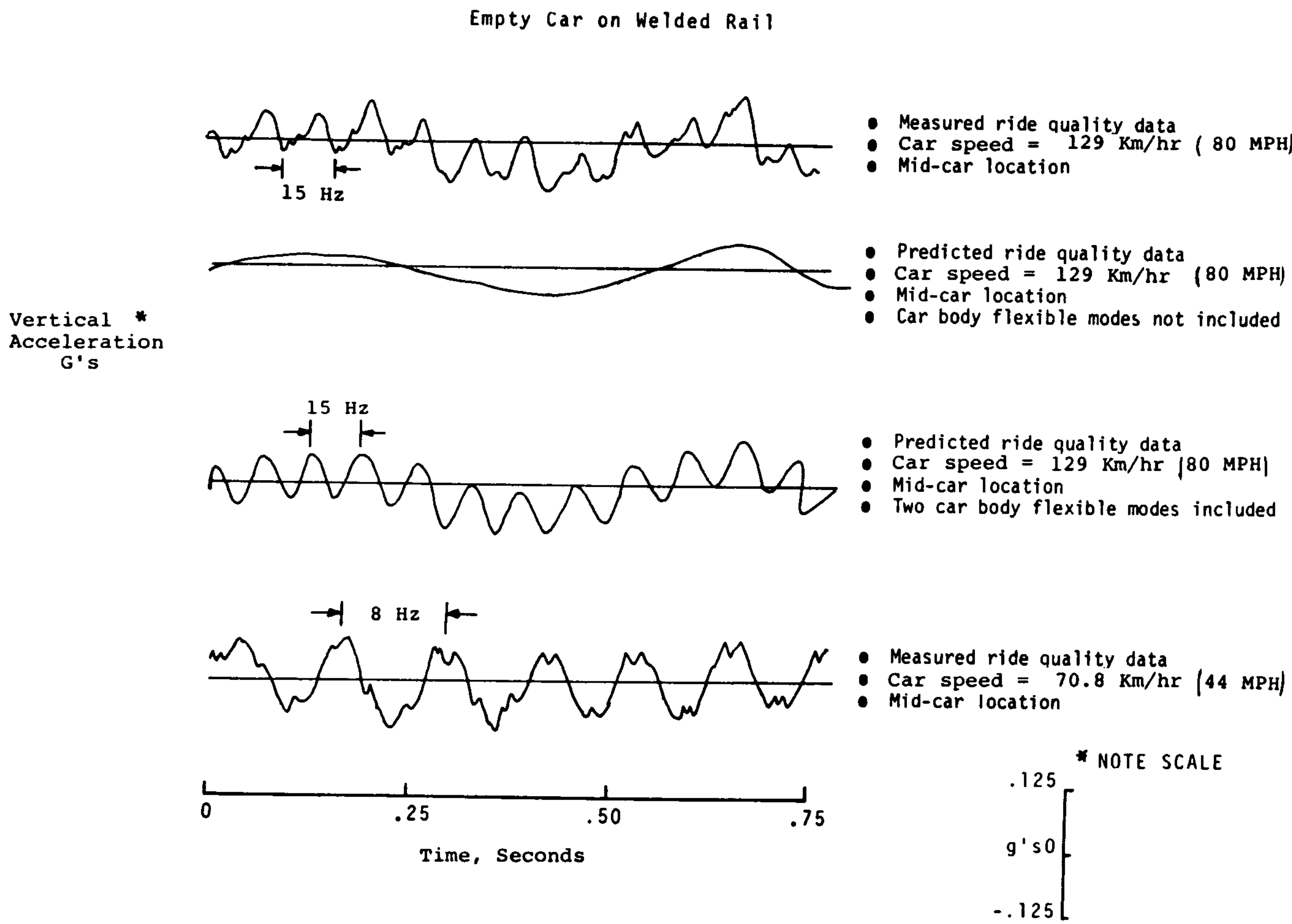

Figure 14.- SOAC ride quality vibration data.
Test and analytical predictions.

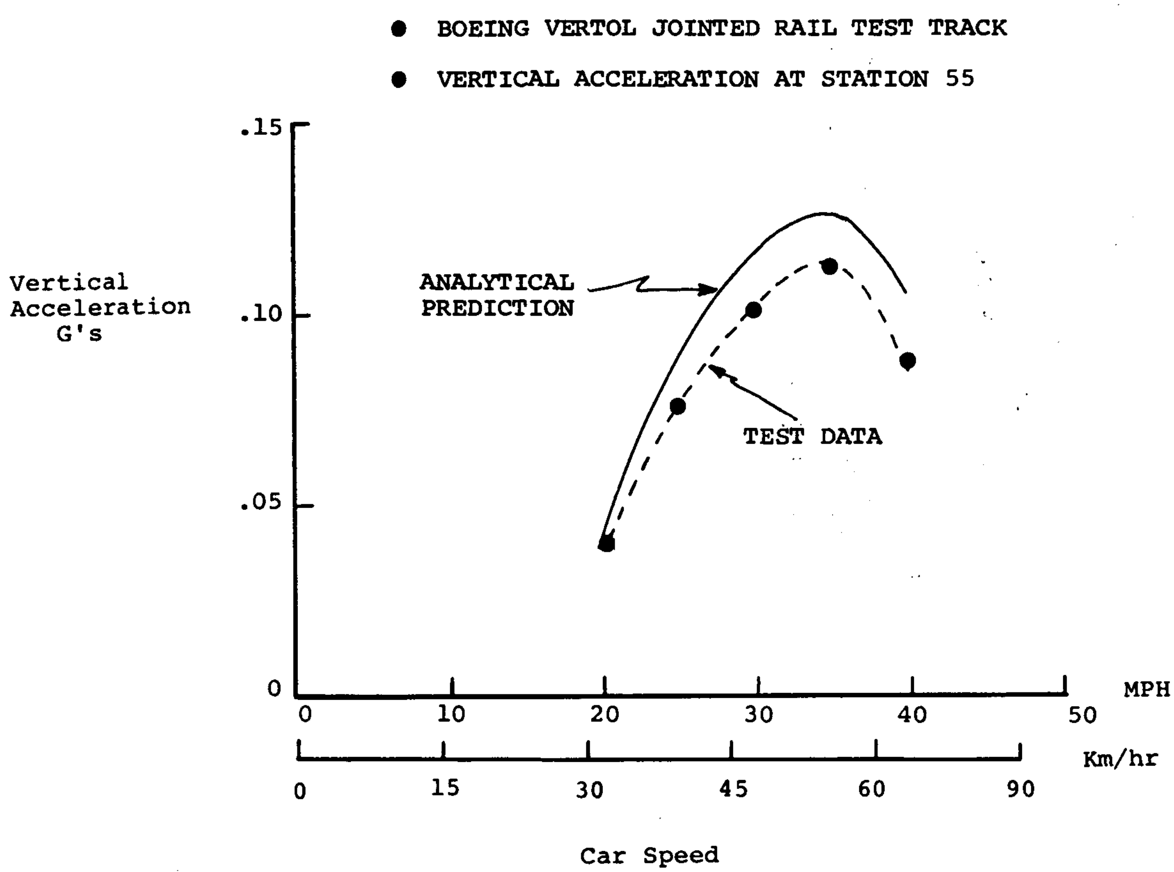

Figure 15.- SLRV ride quality vibration data.
Test and analytical predictions.

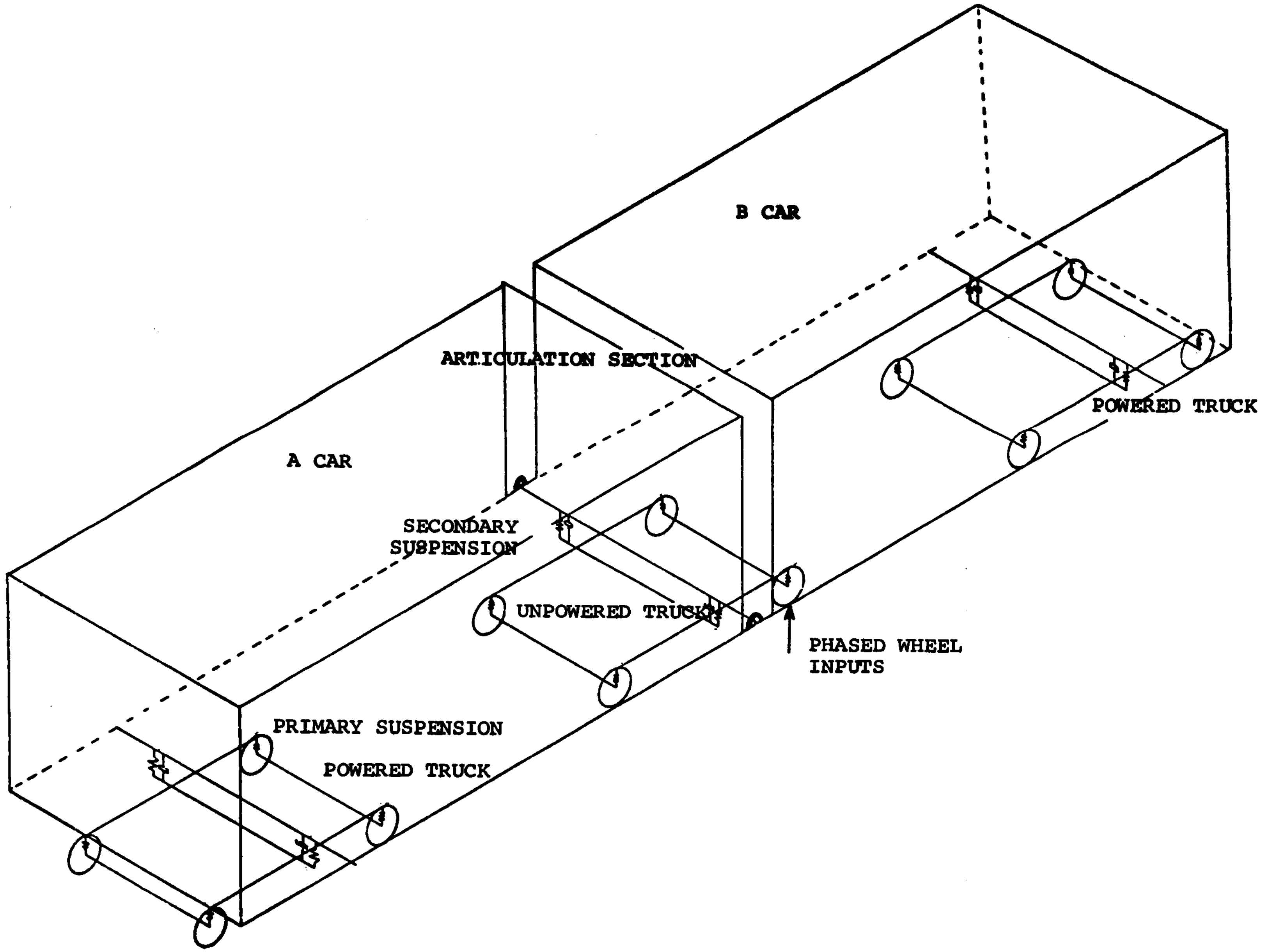

Figure 16.- SLRV ride quality vibration model.

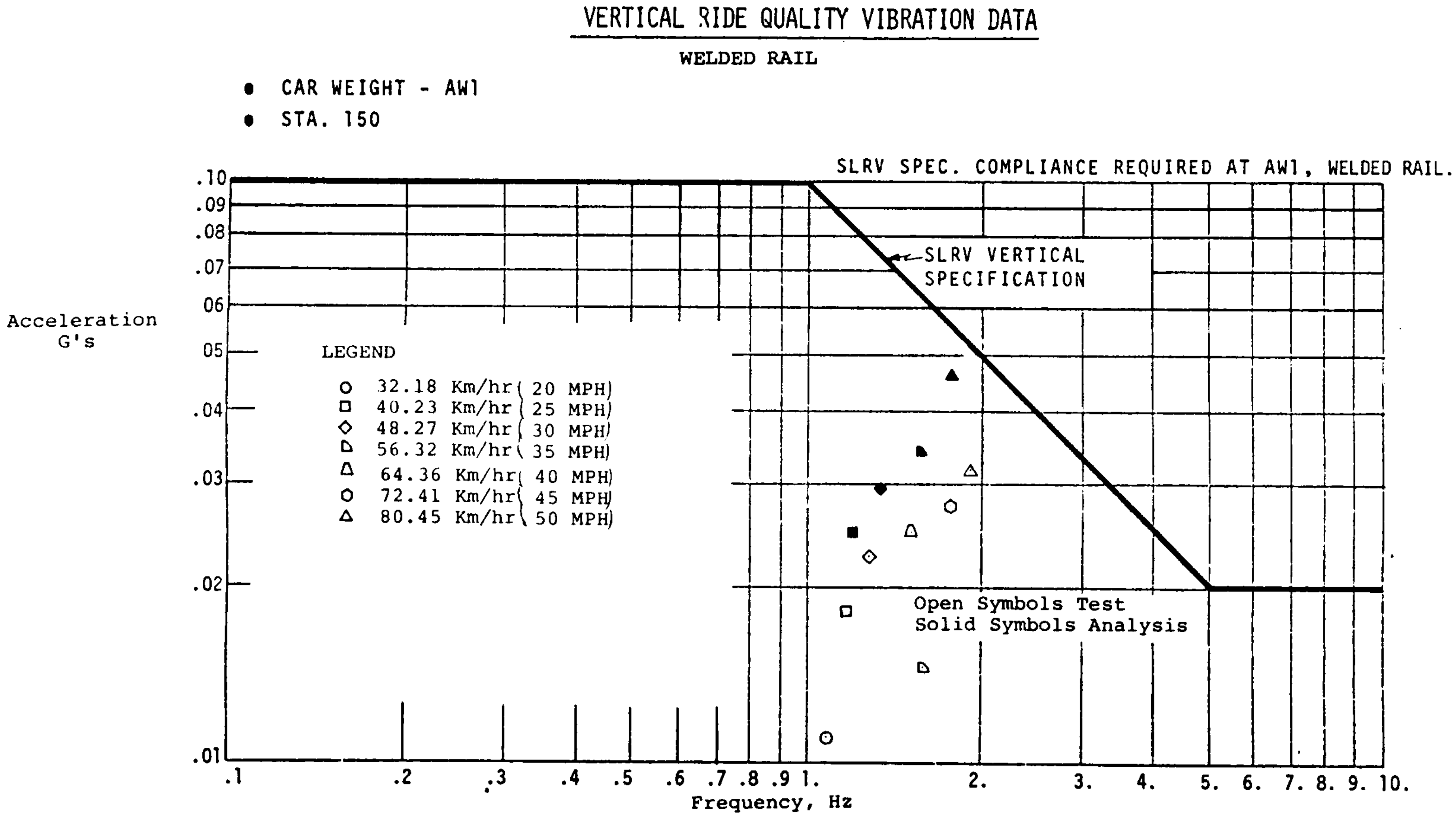

Figure 17.- SLRV ride quality vibration data.
Test and analysis.

N76-16762

INTERCITY RAIL-PASSENGER CAR RIDE QUALITY TEST PROGRAM

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SUMMARY

The Federal Railroad Administration's research and development program relating to intercity rail-passenger ride quality focuses on developing ride quality design criteria and specifications. This paper discusses the FRA ride quality test program and some of the techniques being used to analyze and evaluate the design criteria of the program.

PART I--FRA PROGRAMS

The Federal Railroad Administration's research and development program relating to intercity rail-passenger equipment ride quality focuses on developing ride quality design criteria and specifications. This paper will be limited to the ride quality test program and to some of the techniques for analysis and evaluation of design criteria. Only the baseline programs will be discussed in detail.

PASSENGER SYSTEMS R&D PROGRAMS

A portion of the FRA's Office of Passenger Systems R&D programs is oriented toward intercity rail systems. A subprogram under the Suspension, Support, and Guidance Program is the Intercity Rail-Passenger Car Ride Quality Test Program. Two other major programs that include ride quality considerations include Candidate Train Evaluations and the Improved Passenger Train.

Under the Suspension, Support, and Guidance Program is the Intercity Passenger Car Truck Test Program. Current efforts under this program include design of experimental tests, acquisition and analysis of test data, and development of analytical tools to describe ride quality in a form that is useful for design criteria and specifications. Still another program is the Improved Passenger Car Truck which is an effort to improve ride quality on the AMTRAK Metroliners.

TRUCK TEST PROGRAM

One objective of this 12-month effort is to establish a baseline of coordinated truck and ride performance data. In obtaining baseline data, tests are conducted on six current, relatively new or prototype intercity rail-passenger car trucks that are capable of speeds at least 160 km/hr (100 mph). In coordinating the tests, the same test sequence is used; that is, speeds, track locations, instrumentation, and test cars.

Another objective is to contribute to the establishment of ride quality design criteria and standards. By implication, this program will not necessarily answer all the unknowns. We believe, however, that a compilation of car and truck acceleration, motion, and displacement data of various truck configurations will provide insights which have not heretofore been available. This program is not straightforward, and there are still a number of pitfalls which will have to be skirted, such as the variation in track conditions between each test.

The activities of this program are to place, sequentially, up to six different truck sets (both foreign and domestic) under the same rail-passenger car and test them under the same conditions. The test car and one of the two trucks will be instrumented with accelerometers, displacement transducers, strain gages, pressure transducers, a video camera, and a sound level meter. An adjoining car will collect the physical characteristics of the track. After the data is reduced by the techniques to be discussed in this presentation and by other techniques that may be developed between now and the end of the test program, the effort will be directed toward developing analytical tools and methods for establishing ride quality criteria.

IMPROVED PASSENGER CAR TRUCK

The Improved Passenger Car Truck program is an effort to assist AMTRAK by improving the ride quality for current high-speed equipment. This objective is accomplished by a new high-speed truck that has completed its test program under a Metroliner car. Another program aimed at achieving an incremental improvement in ride quality on the Metroliner is an upgrade of existing trucks. The secondary coil springs will be replaced by an air bag suspension system, and various damping devices will be changed. The Metroliner is a Federal Railroad Administration demonstration program that, since 1968, has been directed at evaluating traveler response to improved intercity railroad passenger service.

TRAIN EVALUATIONS

The Passenger Train Evaluation Program represents still another area in which ride quality is a part of the effort. The FRA intends to prepare a specification, in coordination with

AMTRAK, for a new train system that will be called an Improved Passenger Train (IPT). The specification will outline a major subsystem program to determine technological requirements and deficiencies. Parallel with this effort, there are six non-U.S. built prototype or new-in-production train systems that will be evaluated to establish the capabilities of currently available new equipment, including ride quality. Many of the present techniques for collecting and reducing ride quality information, which are common to other ride quality test programs, will be used during these train evaluations. This information will add to the data bank of ride quality information.

RIDE QUALITY CRITERIA AND SPECIFICATIONS

Figure 1 illustrates, in general, the steps we will complete to arrive at our ride quality objectives which, in turn, will assist AMTRAK to define its specification requirements for ride quality. The left side of the figure illustrates that we are observing current equipment tests, such as the FRA truck improvement tests, and the tests on new intercity rail-passenger equipment that AMTRAK is currently conducting. We also are reviewing specifications, such as those for the Metroliner prepared in 1965. Along with this, we plan to review the specific designs that these specifications have produced.

Next, we are developing and improving the ride quality data acquisition techniques and methods of collection, reduction, presentation, and analysis. The literature review is a necessary part of this phase of the effort. The human factors portion is shown separately from our effort, because the Research and Technology Office of the Secretary of Transportation has a multimodal program which we are following and which will certainly have an impact on our ride quality programs.

At the present time we are involved with the design of experimental tests, data acquisition and reduction, and analysis and development of analytical tools and methods (next three activities in Figure 1). We believe that methods for specifying ride quality will evolve, at which time a hypothesis can be tested and validated. Once the methods are validated, the ride quality criteria and model specifications will be prepared, and AMTRAK will have the option either to use the information directly or to modify it for various intercity rail-passenger equipment. If the new rail equipment encompasses an R&D program, it may become an FRA development program that would be coordinated with AMTRAK. If the equipment is a straightforward state-of-the-art train system or cars, it would be an AMTRAK procurement. In the first case we would conduct the ride quality verification; in the second case we would obtain the data from AMTRAK or assist them in verifying ride quality performance.

We have briefly covered the program aspects of the intercity rail-passenger ride quality efforts FRA is undertaking. Now we will shift to a problem statement, and then summarize the current ride quality specification requirements and discuss the multiplicity of what has been required or requested.

PROBLEM STATEMENT

First, we do not have an efficient way of describing how much ride quality should be built into the equipment for the cost or how much flexibility or rigidity in an application should the equipment have designed into it. Second, it is difficult to define ride quality and what is "better." Ride quality literature is replete with conflicting views and results and different approaches to this complex subject. Thus, we need a better way to define ride quality or establish criteria on what kind of ride quality we want. With regard to the last part of the second element, What is better? possibly the designer will eventually determine quantitatively a zone of indifference for the particular application and thereby determine the most cost-effective ride quality elements that should be included in a particular new design. This will bring in the trade-off elements of human factors, ride motion, and cost.

SPECIFICATION DESIGN VALUES

Table I summarizes the ride quality requirements taken from current intercity passenger equipment specifications. Included in the table are the ride quality specification requirements of the Prototype Tracked Air Cushion Vehicle--the PTACV. The PTACV is a 240 km/hr (150 mph) FRA developmental air-cushion vehicle that is being tested at the DOT Transportation Test Center in Pueblo, Colorado.

The values in this table point out the different ways that ride quality acceleration information has been expressed in specifications. If this small sample is representative, the tendency has been to shift from time-domain requirements to frequency-domain requirements, or to specify both. The specification for the AMTRAK Bi-Level Car expresses ride quality in the frequency domain, but not to some absolute levels. Note the comparison with another rail car--the Hi-Level car. This comparative test is an effort to circumvent the track as a variable. Also note that different test equipment locations are cited. Standardization of location should be achieved in order that more comparative information would become available.

In summary, if we can better define or describe ride quality or ride comfort in terms of what is desired, then possibly we can provide a more cost-effective and pleasant ride for the intercity rail passenger.

TABLE I. SPECIFICATION RIDE QUALITY ACCELERATION DESIGN VALUES (PARTIAL REQUIREMENTS)

Equipment	Speed km/hr (mph)	Data Domain	G's Vertical	G's Lateral	G's Longitudinal	Test Equipment Location	Track Condition Reference
Metroliner (1965)	256 (160)	Time	0.03 Frequently 0.05 Occasionally	0.02 0.04	0.02 0.04	Floor over truck and mid-point	Geometric limits
AMCARS (1974)	152 (95)	Time	0.050 (99% of Time) ← 20 Sec. Data →	0.060 (99%)	0.010 (95%)	Over truck and midpoint	Referenced test run and PSD
Bi-Level car (1974)	24-176 (15-110)	Frequency	Bi-Level Car PSD comparison to Hi-Level car (run in same test)			Over truck and midpoint at seat height	-----
LRC (Canadian) (1974)	152 (95)	Time	0.070 (99% of time)	0.057 (99% of time)	-----	Over truck	Class 5 or 6
Prototype Tracked Air Cushion (PTACV)	240 (150)	Time and Frequency	← 10 Sec. Data → 0.094 (95% of time) PSD specified, 0.1 to 50 Hz ≤ 30 Sec. Data →	0.078 (95%)	0.078 (95%)	Passenger compartment	Guideway conditions specified

PART II--ENSCO RESEARCH IN SUPPORT OF FRA PROGRAMS

The purpose of developing the capability to measure and analyz vehicle acceleration data is to provide a means of evaluating the ride quality of rail vehicles and to provide information for the establishment of meaningful guidelines for vehicle designs. While ENSCO is not involved in basic research of perceived ride quality, we are interested in the results of basic research in this area and how this information might be applied to the development of ride quality specifications for rail vehicles.

DATA COLLECTION

A portable data collection system, known as the Portable Ride Quality (PRQ) package (Figure 2), has been developed for the Federa Railroad Administration by ENSCO. This system consists of a magnetic tape recorder, a conditioning and coding unit, and an accelerometer package. The accelerometer package contains six accelerometer three linear and three angular. Table II provides details of this package.

TABLE II

CHARACTERISTICS OF ACCELEROMETERS

Measurement	Full-Scale Capability	Natural Frequency
Vertical	±1 G	60 Hz
Longitudinal	±1 G	60 Hz
Lateral	±1 G	60 Hz
Yaw	±1 Rad/Sec	30 Hz
Pitch	±1 Rad/Sec	30 Hz
Roll	±5 Rad/Sec	30 Hz

The conditioning and coding unit converts the current output of each accelerometer to a proportional signal voltage suitable for recording. The unit provides metering for signal monitoring and calibration. It also contains batteries and associated charging and regulator circuits, which provide power to the system during portable operations.

The magnetic tape recorder accommodates eight channels of data. Six channels are used for recording accelerometer signals; one channel is used for voice annotation; and one channel is used for a multiplex recording of two external data signals, an internally generated digital annotation and a reference signal for wow and flutter compensation. The total weight of the system is approximately 41 kg (90 pounds).

A signal reconstruction unit is used in the playback mode of operation. The unit conditions all signals to a level comparable with data processing, and provides wow and flutter compensation. A block diagram of the PRQ system is shown in Figure 3.

Many problems arise in specifying the conditions under which ride quality data is to be collected. These problems involve:

- Speed or speeds of the vehicle
- Duration of recorded signals
- Track conditions
- Position of the accelerometer package in the vehicle

Answers to these problems will depend on the purpose of ride quality experiments and the analytical procedures applied to the data.

DATA REDUCTION

The recorded analog signals are converted to digital form for data reduction. The digitizing process involves anti-alias filtering of the data and conversion of the filtered analog data into 12-bit digital words. The conversion rate or sampling frequency in this process occurs 256 times per second. The digitized data in this form is compatible with a number of data reduction techniques, including both frequency domain and time domain. Methods used to reduce the data include:

- Histograms
- Standard deviations
- Cumulative distribution functions
- Density functions
- RMS time plots
- Power spectral density (PSD)
- One-third octave band filtering
- DC bias versus time

The block diagram for time-domain data reduction is shown in Figure 4. A digital high-pass filter is used to remove any DC bias in the accelerometer signals. The rationale for choosing these methods is that:

(1) Histograms provide information on the distribution of the acceleration levels, including peak acceleration. Estimates of the distribution function and the density function can easily be produced from the histograms. The data presented by these functions represent more usable forms for some applications.

(2) Standard deviations can be easily generated from the histograms and serve to define the signal in the time domain.

(3) RMS time plots provide a short-term average of the accelerations. Special events (i.e., large accelerations of significant deviations) can be quickly determined.

For the frequency-domain process, two methods of reduction are applied to the data. A block diagram for frequency-domain processing is shown in Figure 5. The first is a narrow-band type of processing that presents results in PSD form. Typical processing bandwidths are 0.1 Hz to 0.25 Hz. A typical PSD for ride quality data collected on passenger trains is shown in Figures 6 and 7.

The second method of frequency-domain processing is with one-third octave band filters. In this type of processing, the bandwidth increases with frequency. One-third octave band filtering is the appropriate method for applying the International Organization for Standardization Standard (ISO) 2631 for ride quality.

COMFORT CRITERIA

The one-third octave band frequency technique and the ISO Standard provide a method of applying a signal number to the measured vibration environment. The RMS G levels corresponding to the one-third octave bands between 1 Hz and 80 Hz are determined. Using the reduced comfort criterion from the ISO Standard, these values are converted to exposure limits. Exposure limits are measured in hours. An exposure limit of 5 hours means that a passenger experiences "reduced comfort" after being exposed to the vibration environment for a period of 5 hours. The minimum exposure time for the entire frequency range is taken as a single description of the ride. Results of this type are shown in Figure 8.

In correlating the results of ride quality test programs, we have found that much of the vehicle vibration data is presented in PSD form. Comparison of PSD's is difficult. One useful tool for comparison of ride quality data is to translate the ISO ride quality standard into an equivalent PSD form. The assumption in making this conversion is that the power in each of the one-third octave band filters is evenly distributed.

The resultant form of the ISO standard is shown in Figure 9 for the head-to-foot (vertical) direction and in Figure 10 for the side-to-side (lateral) direction. Also shown in these figures are PSD's for the lateral and vertical accelerometer data collected on a Metroliner passenger car. For the vertical direction, an exposure time of 4 hours at a frequency of 5 Hz is obtained. For the lateral direction, an exposure time of 8 hours at a frequency of 2 Hz is obtained. These results are similar to those for the one-third octave band filtering method.

While the PSD form of representing ride quality data provides a convenient method of condensing long time records of ride quality data, it tends to mask special events. By special events, we mean periods of high acceleration levels and significant duration. Time-domain processing can be used to account for special events. For rail vehicles, special events are usually related to "bad spots" in the track. One method of addressing this problem is to determine the percentage of time that the magnitude of the acceleration signal exceeds a given value or, conversely, the magnitude of the signal which brackets some fixed percentage of the data. The difficulty with this technique is that the duration of the peak acceleration levels is not taken into account (i.e., are the peak accelerations isolated spikes, or do they occur during a single period of high acceleration). Obviously, the results of this type of processing will depend on the bandwidth of the processing system. By calculating the RMS value of the accelerometer signals, the effect of duration can be seen, but again the bandwidth of the signal is an important factor. From experience, a bandwidth of 1 to 15 Hz appears appropriate for rail vehicles.

Table III shows a comparison of two Metroliner vehicles using both the ISO ride quality standard and time-domain processing. This data was collected between Baltimore and Wilmington. For the northbound run, the test zone was between milepost 35 and 40; for the southbound run, the test zone was between milepost 81 and 83. The accelerometer package was located on the floor in the center of the car. The Metroliner 850 vehicle was equipped with an improved truck design, while the 855 was a standard vehicle. The performance of the 850 vehicle is superior for all comparisons.

TRACK GEOMETRY

A means of describing the track conditions is required to correlate the results of ride quality experiments. Since both the vehicle vibration environment and the sensitivity of the passenger to vibration can be described in PSD format, it is convenient to use this format for describing track geometry. The PSD allows a three-way comparison of: the input to the vehicle, the output of the vehicle, and the sensitivity levels of the passenger. Track geometry data is usually collected with a distance-based data collection system, with the reduced data presented in the format

TABLE III.- METROLINER RIDE QUALITY TEST RESULTS

Type of Measurement	Vehicle	Northbound		Southbound	
		Vertical	Lateral	Vertical	Lateral
Standard Deviations**	850 855	0.031 G 0.045 G	0.024 G 0.026 G	0.027 G 0.040 G	0.024 G 0.030 G
99% Level**	850 855	0.098 G 0.037 G	0.068 G 0.083 G	0.088 G 0.111 G	0.073 G 0.085 G
95% Level**	850 855	0.064 G 0.090 G	0.050 G 0.060 G	0.055 G 0.080 G	0.051 G 0.060 G
ISO* Reduced Comfort Criterion	850 855	4.0 hr (5 Hz) 2.5 hr (5 Hz)	13.8 hr (2 Hz) 9.2 hr (2 Hz)	4.95 hr (5 Hz) 4.36 hr (5 Hz)	15.9 hr (2 Hz) 8.48 hr (1.3 Hz)

Vehicle 850 = Metroliner with improved truck design

Vehicle 855 = Standard Metroliner

*** 128 seconds of data**

**** 60 Hz bandwidth**

Speed 170 km/hr (106 mph)

in centimeters squared per spatial frequency versus spatial frequency. The format can be converted to the form of cm^2/Hz versus frequency by assuming a constant vehicle speed. To convert this form to the desired form requires that each point of the curve be multiplied by a factor of $(2\pi F)^4$ and that the acceleration level in cm^2/sec^4 be converted to G's. This operation in the frequency domain is equivalent to double differentiation of the track profile data. The process is shown in Figure 11.

Track geometry parameters of interest include mean profile, mean alignment, and crosslevel. Typical PSD curves for mean profile, mean alignment, and a 128-km/hr (80-mph) speed are shown in Figures 12 and 13.

The ISO curves for reduced comfort have been added to these figures. Note the peaking of the PSD levels at frequencies of 3 Hz, 6 Hz, 9 Hz, and 12 Hz. At 128 km/hr (80 mph), the 3-Hz frequency corresponds to a wavelength of 11.88 meters (39 feet) in the track (the basic length for bolted rail). The remaining frequencies represent harmonics of the encounter frequency. The interpretation of these curves is that the input from the track must be attenuated by the suspension system of the vehicle to lie below the appropriate exposure time curve. For ride quality tests performed on different sections of track, the relationship between the track input and the ride output (vehicle acceleration data) can be used to normalize the results of the test programs.

CONCLUSION

A number of data-processing techniques have been presented for reducing and analyzing vehicle acceleration data. The common format of PSD representation of track geometry, human tolerance, and vehicle response will be used to investigate and compare the "ride quality" of a number of vehicle designs. In addition, a number of time-domain techniques have been developed to investigate vehicle ride quality.

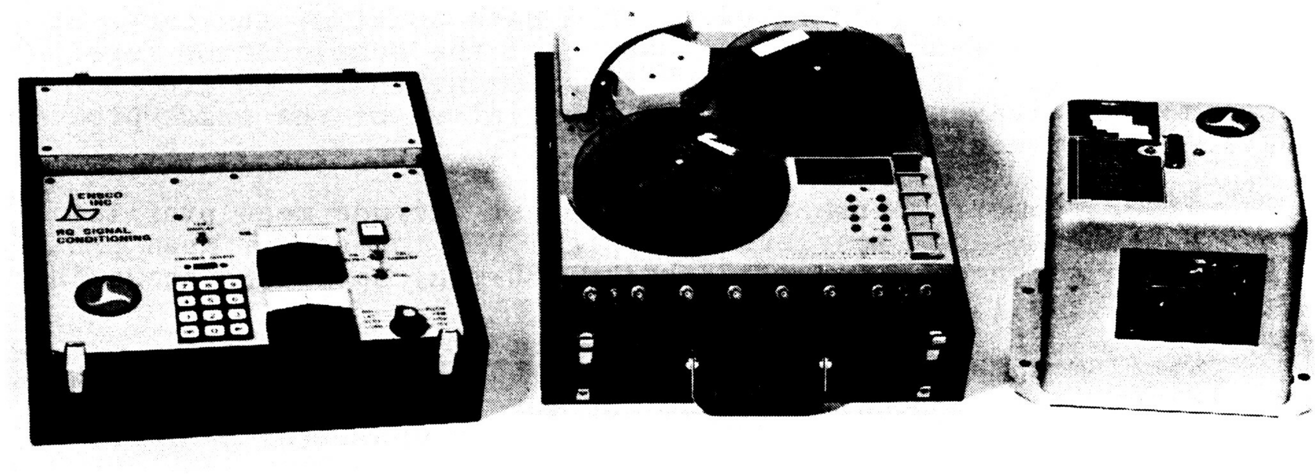


Figure 1.- Ride quality criteria and specifications.

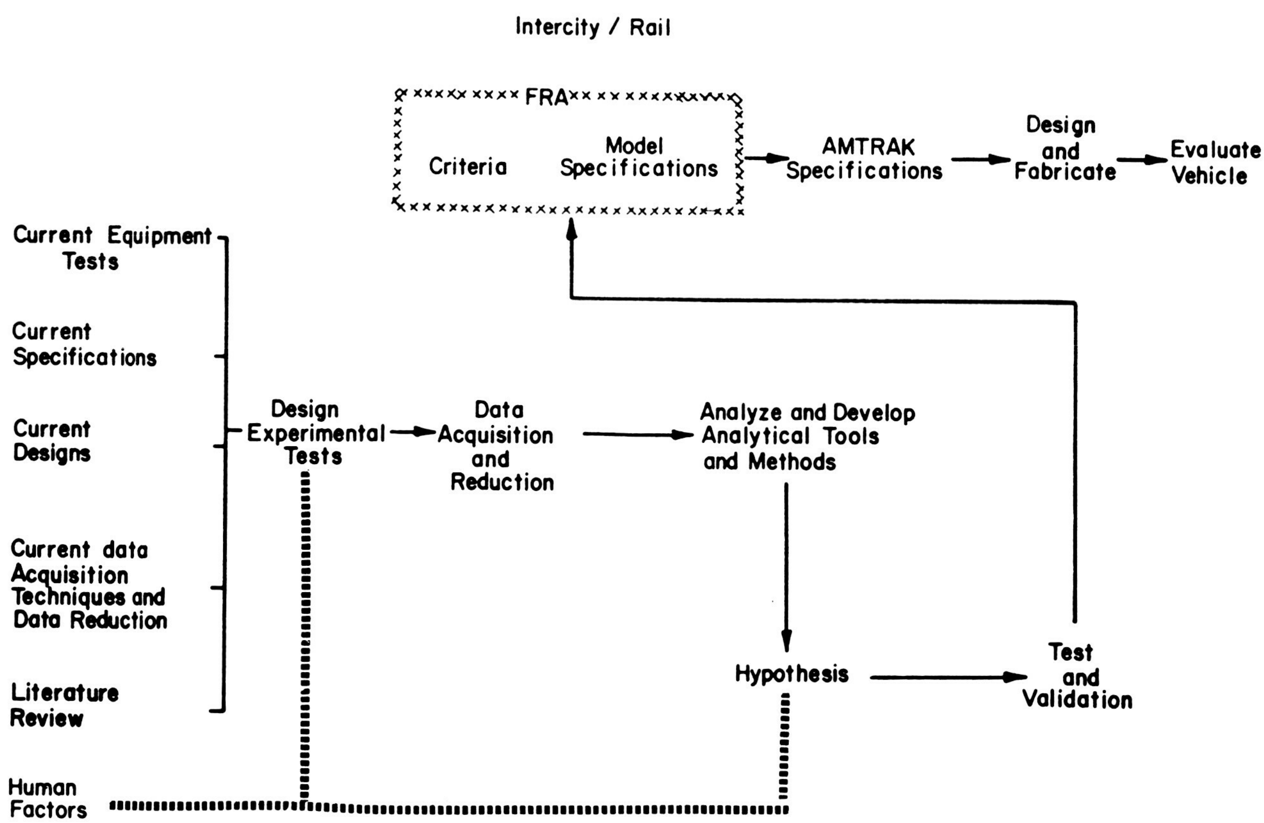


Figure 2.- Portable ride quality system.

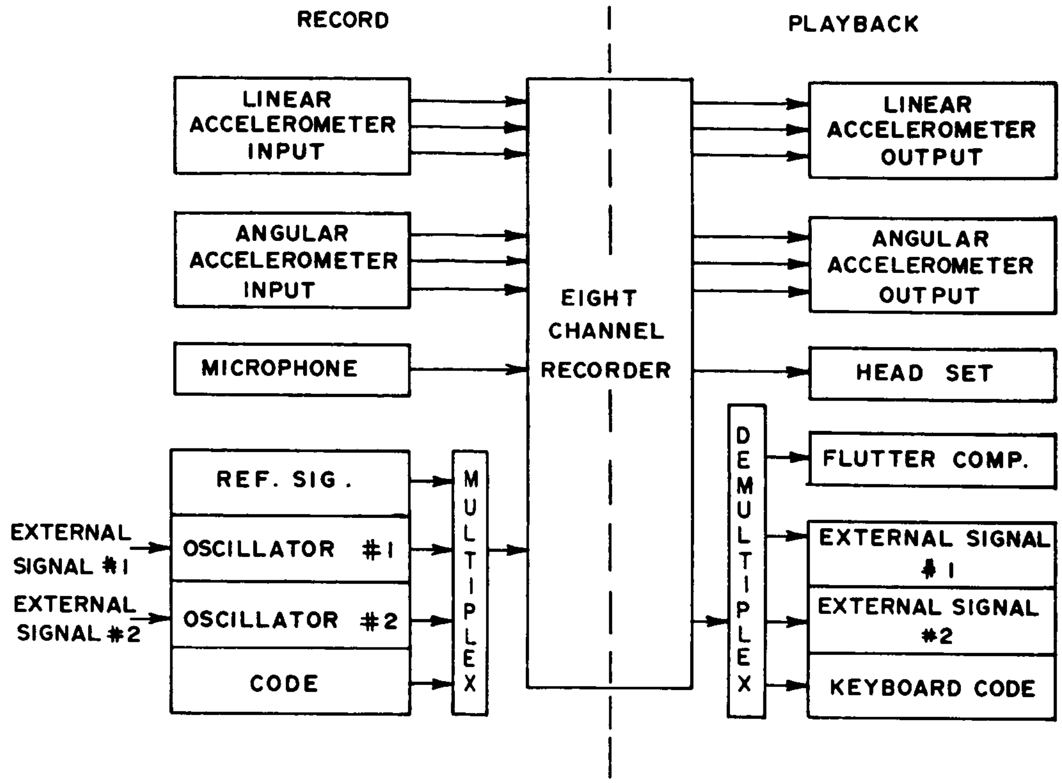
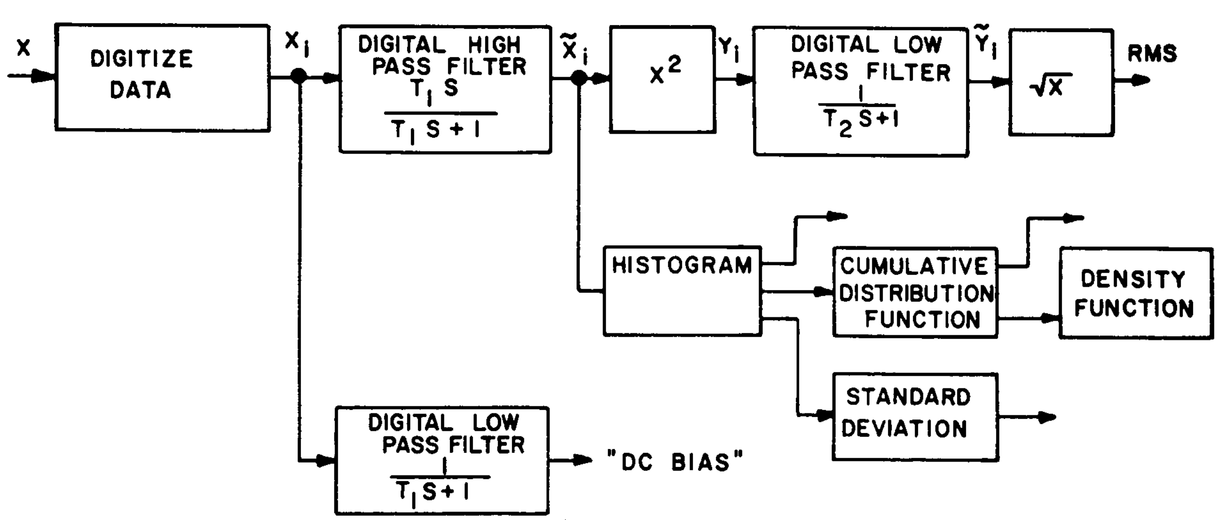


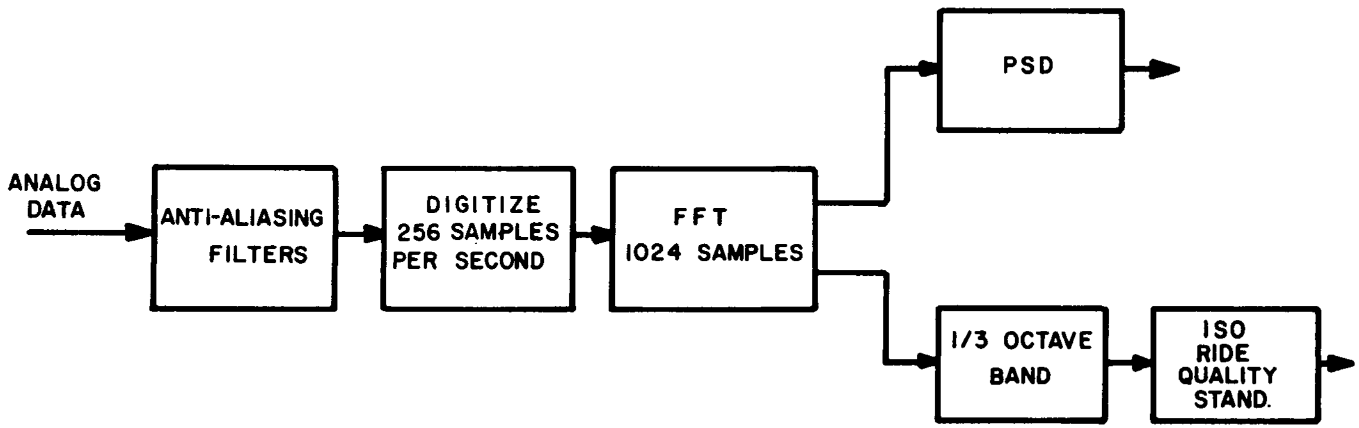
Figure 3.- Block diagram of portable ride quality system.



S = LAPLACE OPERATOR

$T_1 = T_2 = 0.33$ SECONDS

Figure 4.- Block diagram of time-domain processing.



FFT = FAST FOURIER TRANSFORM

Figure 5.- Block diagram of frequency-domain processing.

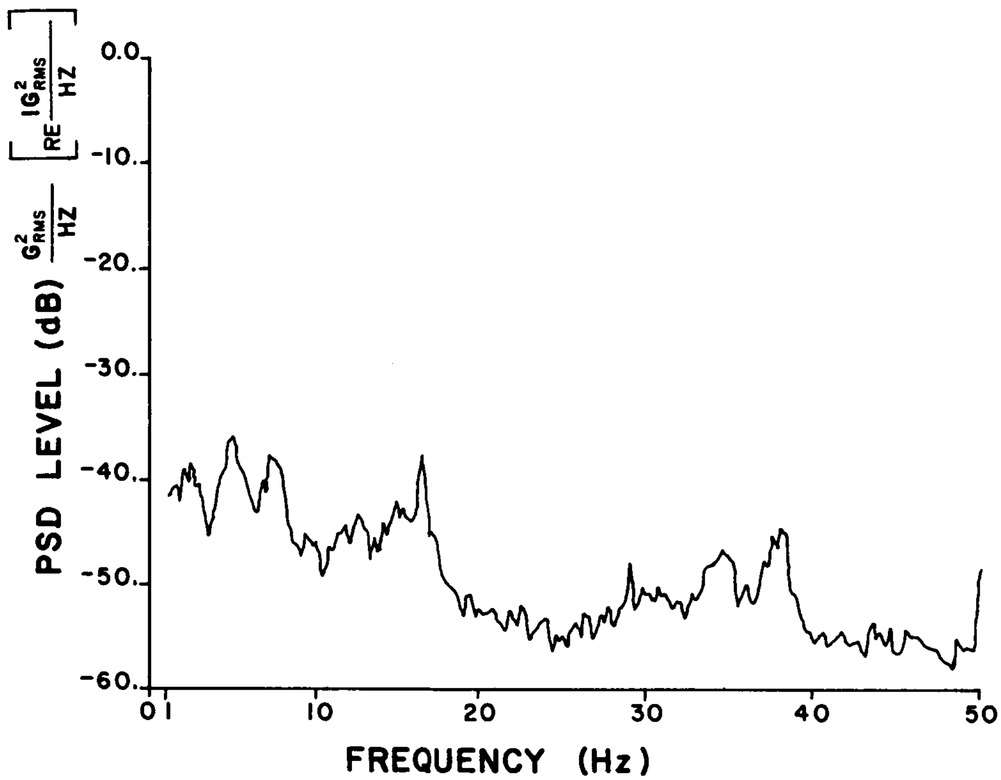


Figure 6.- PSD for vertical acceleration.

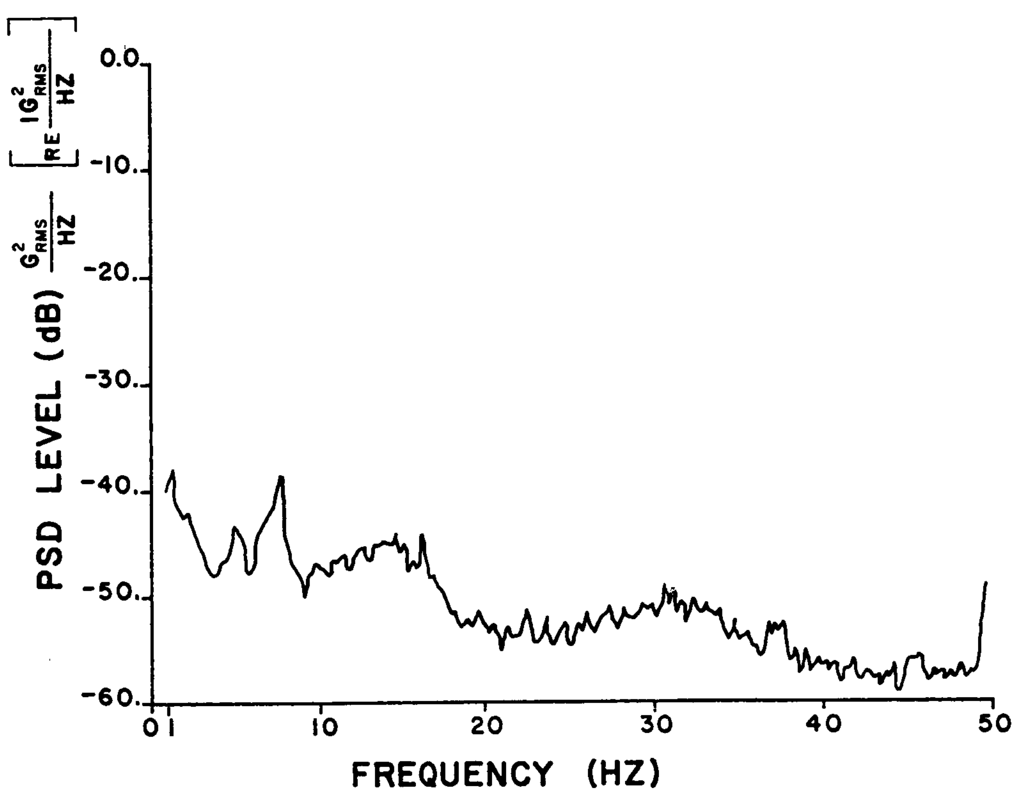


Figure 7.- PSD for lateral acceleration.

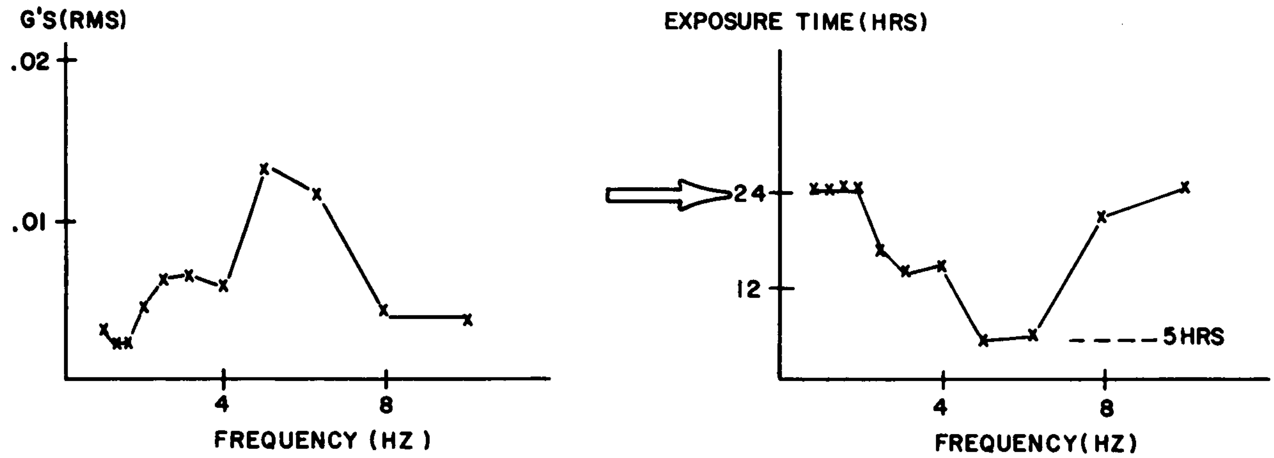


Figure 8.- ISO ride quality standard for random vibration.

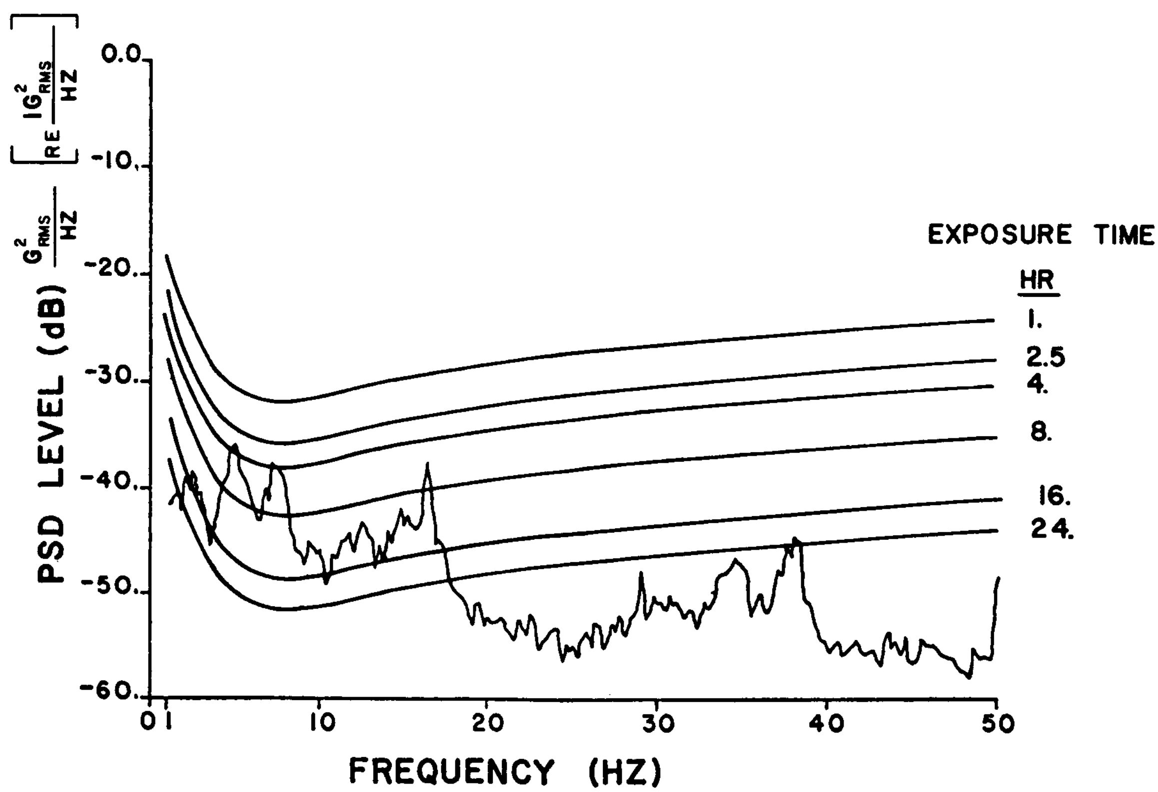


Figure 9.- PSD curve for vertical acceleration with ISO Standard overlay.

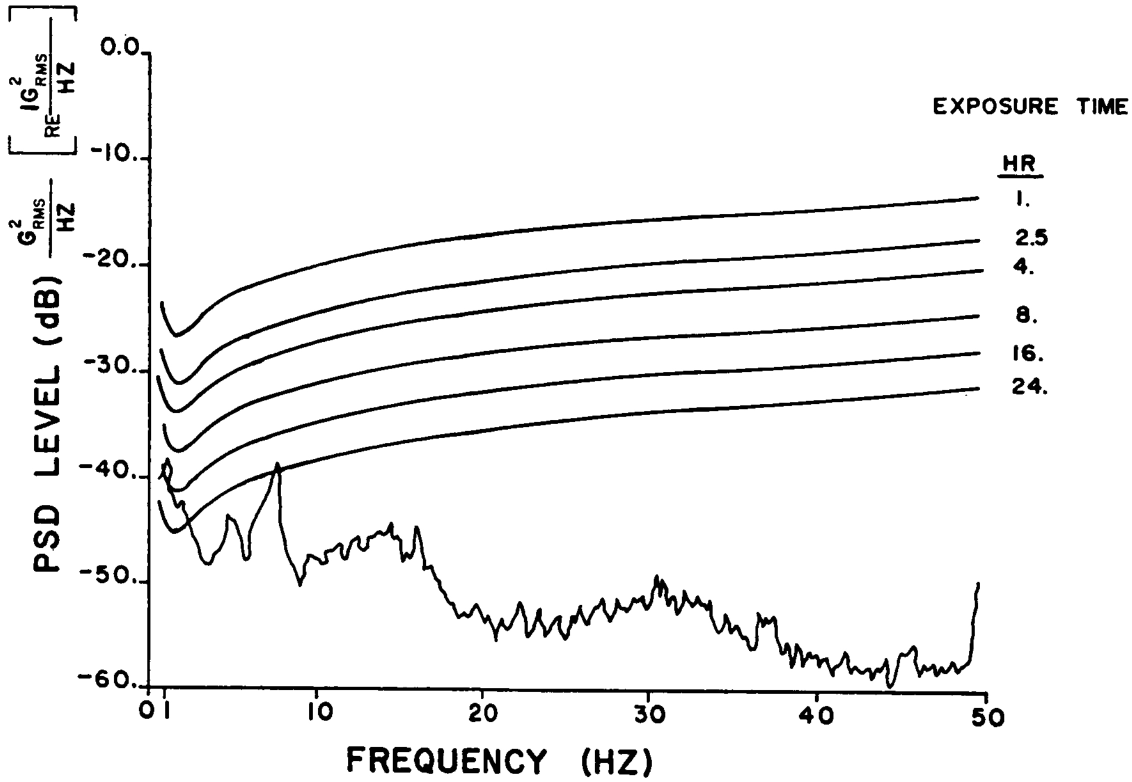


Figure 10.- PSD curve for lateral acceleration with ISO Standard overlay.

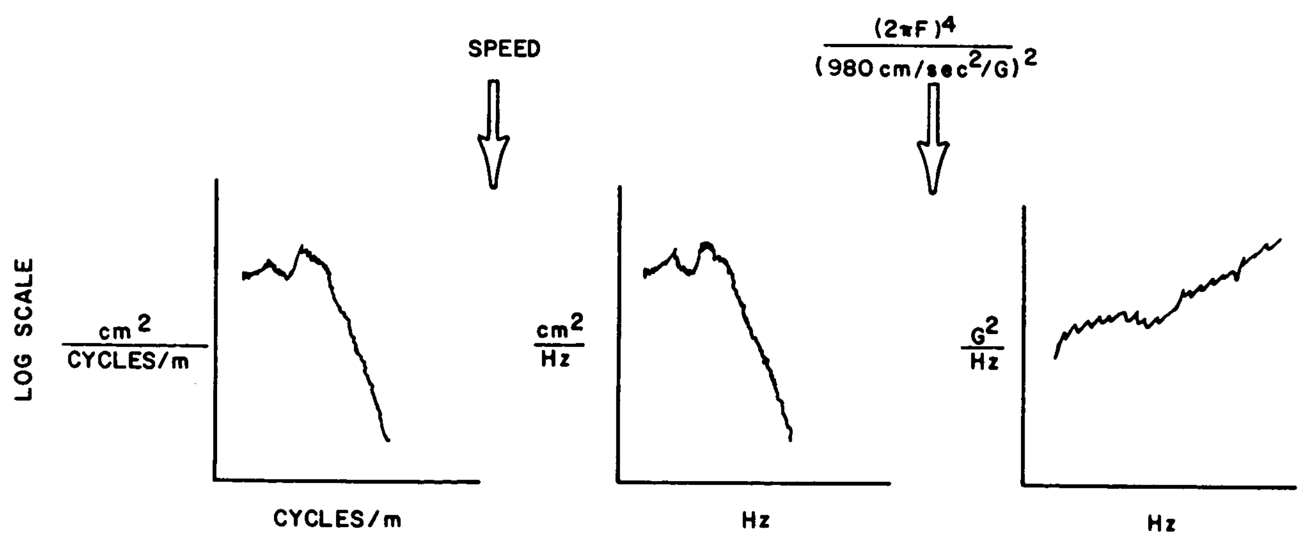


Figure 11.- Conversion of track geometry data from distance-based format to PSD format.

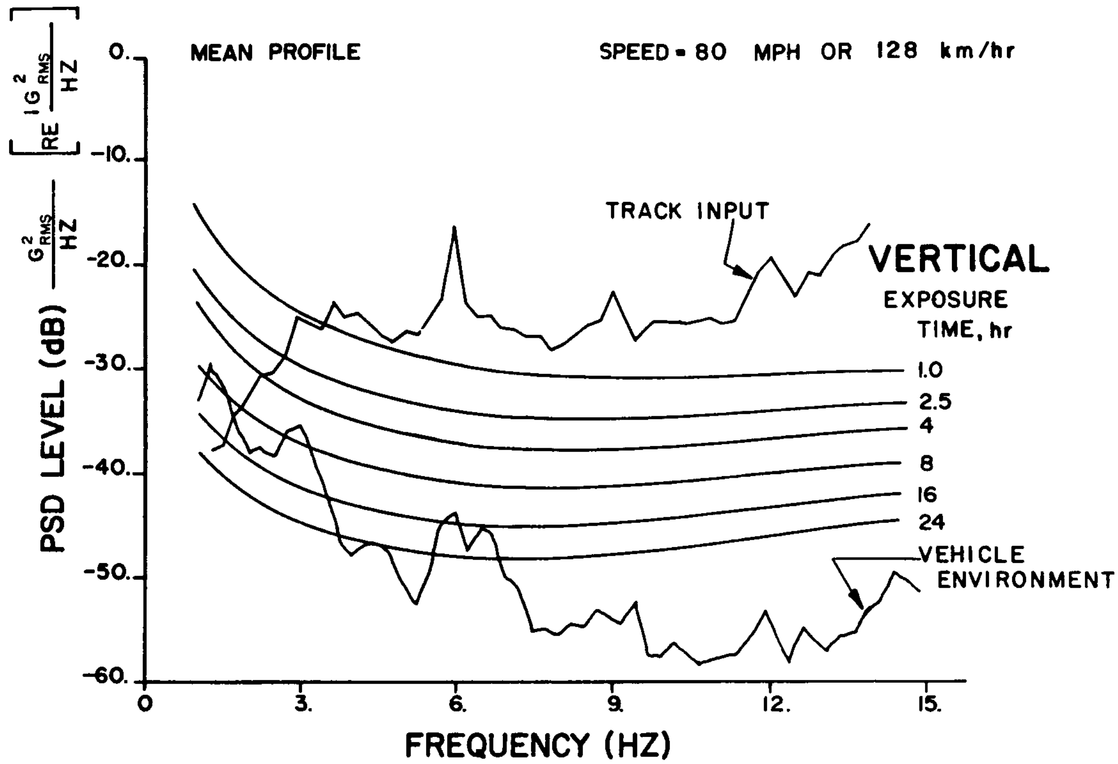


Figure 12.- PSD curve for mean profile with ISO Standard overlay.

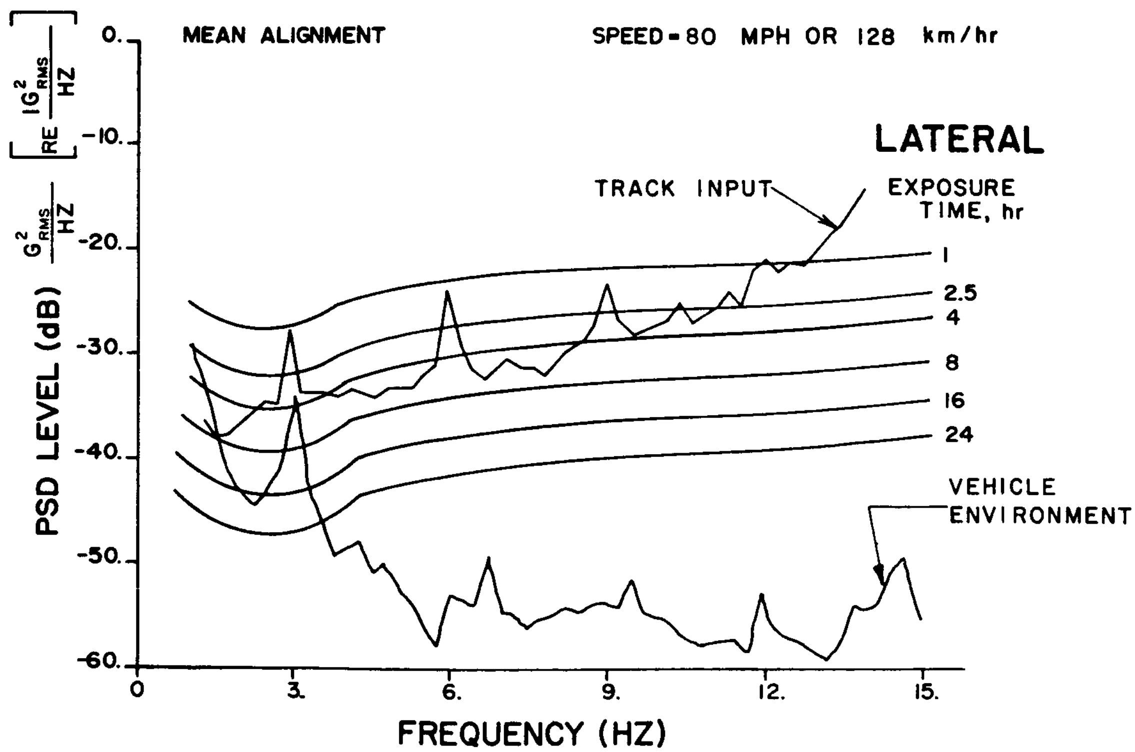


Figure 13.- PSD curve for mean alignment with ISO Standard overlay.

N76-16763

THE PITCH-HEAVE DYNAMICS OF TRANSPORTATION VEHICLES

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SUMMARY

The analysis and design of suspensions for vehicles of finite length using pitch-heave models is presented. Dynamic models for the finite length vehicle include the spatial distribution of the guideway input disturbance over the vehicle length, as well as both pitch and heave degrees-of-freedom. Analytical results relate the vehicle front and rear accelerations to the pitch and heave natural frequencies, which are functions of vehicle suspension geometry and mass distribution. The effects of vehicle asymmetry and suspension contact area are evaluated. Design guidelines are presented for the modification of vehicle and suspension parameters to meet alternative ride quality criteria.

I. INTRODUCTION

A fundamental problem in the development of high performance transportation vehicles is insuring adequate passenger ride quality when operating over economically feasible guideway surfaces and structures. While passenger comfort has long been an important factor in vehicle design, recent developments in high speed ground transportation vehicle technology have identified ride quality as a primary constraint to successful implementation of systems capable of speeds of 200 to 500 km/hr.

Prediction of passenger ride quality is based on three components: quantification of guideway irregularities and aerodynamic conditions that act as disturbances to the vehicle; analysis of the dynamic response of the vehicle body, suspensions, and other subsystems to external disturbances; and finally comparison of the resultant vibrational environment to be experienced by the passengers to suitable measures of ride quality.

At present significant deficiencies exist in all three components of ride quality analysis when applied to the development of new transportation systems. No measure of passenger ride quality has yet gained wide acceptance due to the differences that exist between typical applications in transportation and the controlled experiments conducted to date [1]*. Ride quality is a subjective evaluation of a random, multidirectional, vibrational environment that may vary among individuals in a population and may be altered by exposure times and environmental factors other than motion. Currently available data does not systematically represent the above variables, either quantifying subjective human evaluations of one-dimensional, single frequency steady sinusoidal motion [2] or limited samples of vibrational environments on specific vehicles [3]. Existing ride quality measures or specifications are useful primarily as a qualitative indication of the nature of human sensitivity to vibration but have limited utility as absolute standards for system design.

*Numbers in brackets refer to the list of references.

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Guideway irregularity and aerodynamic disturbances are represented either stochastically, as empirically derived spectra of surface roughness and atmospheric turbulence, or deterministically, as measured contours of specific guideways and computed coupled reactions between vehicles and flexible guideways or between passing vehicles. However, a lack of relevance between the spectral representations of guideway smoothness and meaningful guideway construction specifications has led to recent re-examinations of measures of guideway irregularities [4].

The dynamic response of vehicles to disturbances has been analyzed by two distinct methods. First, simulation models of widely ranging complexity [5,6, 7] have been formulated to represent vehicle response including multidimensional body motions, body flexibility, suspension displacements, and component vibrations. (Even more complicated models have been developed to include guideway flexibility.) In theory such simulations can be made as accurate as desired through the addition of elements to the model; however, the usefulness of these detailed simulations is limited by several factors:

(a) The analysis of vehicle response involves solution or simulation of equations of very high order, with a large number of design parameters. Such complex systems are not amenable to efficient computational and optimization techniques, and can frustrate the use of intuitive design procedures.
(b) Because of the large number of parameters and complex system structure, it may be difficult to generalize simulation results to other vehicles.
(c) It is difficult to validate the model experimentally at a level of detail commensurate with that of the analysis.

The second approach has been to use simple conceptual models of vehicles and suspensions as bases for intuitive design procedures and for closed-form or iterative optimization techniques. One-dimensional vehicle models, supported by either specific or generalized suspensions, subjected to a variety of disturbance, have been studied extensively, yielding mathematical descriptions of optimal suspensions for given sets of disturbances, performance criteria, and design constraints [8,9], or parametric descriptions of specific suspensions that approach the mathematical optima [5,10]. While valuable insights may be obtained by their use, one-dimensional models also have limitations:

(a) The one-dimensional model does not accurately represent the vibration environment experienced by the passenger, and is thus inadequate as a component of the prediction of ride quality.
(b) Suspensions designed to be optimal using the one-dimensional analysis may be suboptimal when the dynamics of the complete vehicle are considered. In turn, specification of non-optimal configurations may lead to costly overspecification of system parameters such as guideway smoothness and stiffness.

The dynamic characteristics of well-designed vehicle systems are such that the analysis and design of the complete vehicle/suspension system can be segmented into several uncoupled subsystems. The dynamics of a complete vehicle/suspension system traveling at constant forward speed can be categorized as

(a) Motion of the vehicle body: Of primary interest are the body heave and pitch in the vertical plane, sway and yaw in the horizontal plane, and roll. The cumulative effect of these motions is the motion perceived by the passengers within the vehicle, thus determining ride quality. These motions will be of low frequency (less than 15 Hz) in well-designed vehicles.

(b) Bending modes of the vehicle body: Due to body flexibility, an infinite number of natural modes can be excited, many at high frequencies. Analysis of these modes can be quite complex for bodies with discontinuous structures, such as transit cars with numerous doors. These vibrations may affect ride quality, passenger compartment noise, and the structural integrity of the body.

(c) Primary suspension motions: The motion of the cushion (or wheel) is primarily perpendicular to the guideway plane. If the cushions are separated from the vehicle by secondary suspensions, the dominant cushion mass oscillation will be of high frequency (greater than 25 Hz), due to the high gap stiffness and low cushion mass (less than 0.1 of the vehicle mass).

(d) Inter-suspension coupling: The motions of powered suspensions may be coupled, via the feeding system in air cushion vehicles or via the magnetic fields of magnetic cushions.

The one-dimensional heave mode suspension model is appropriate for predicting cushion stability and displacements subject to random disturbances. However, representing the vehicle motion as primarily one-dimensional neglects two important effects:

(a) The vehicle is capable of pitch and bending as well as heave. Numerous studies of high speed vehicles traveling over irregular and flexible guideways have shown that the pitch mode has a strong influence on passenger ride quality [11,12,7].

(b) The finite lengths of the cushions produce filtering of guideway irregularities having wavelengths shorter than the length of the pad. The heave mode vehicle model assumes all other wavelengths to be much longer than the vehicle length, which is not true for multicushioned vehicles.

The finite length vehicle model developed here includes both pitch and heave motion of a rigid body vehicle, as well as the effects of guideway irregularities seen along the vehicle length.

The objective of this paper is to help bridge the gap between the well-understood one-dimensional analysis and complex full vehicle models by presenting an analysis of finite length vehicles with pitch and heave* degrees of freedom. The analysis relates vehicle front and rear accelerations to the pitch and heave natural frequencies, which are functions of vehicle suspension geometry, suspension dynamic characteristics, and vehicle mass distribution. The vibrational environment used to determine ride quality is evaluated for vehicles subjected to stochastic and deterministic guideway disturbances. Design guidelines are presented for the modification of vehicle and suspension parameters to meet alternative comfort criteria.

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*The analysis in this paper applies to motion in the vertical plane (heave and pitch) and the lateral plane (sway and yaw). The discussion refers only to heave and pitch, but the extension to the lateral case is always implied. The analysis assumes that body motions in the vertical and lateral planes are uncoupled.

II. PREVIOUS RESEARCH

An extensive literature has developed on the dynamics of vehicle suspensions. Physical models of both wheeled and tracked levitated suspensions have been synthesized and in most cases experimentally verified [13 to 16]. Suspensi characteristics have been commonly quantified in terms of primary and secondary suspension stiffness and damping, sprung and unsprung mass, and suspension/guideway contact area [17,10]. For levitated and some wheeled vehicles, suspension motion is primarily one-dimensional, so that the suspension force transmitted to the vehicle body is a function of the relative displacements and velocities between the suspension attachment point on the vehicle and the adjacent guideway location. (If the suspension mass is large, its acceleration must also be included in the model.)

Research conducted in parallel to the physical modeling studies exploited the commonality between the various suspension types by developing generalized suspensions that were mathematically optimal for specified sets of input disturbances and performance measures. Inputs from surface roughness, wind gust loading, and guideway flexibility have been considered, with performance indices composed of weighted RMS body accelerations* and relative body-guideway displacements [8,9]. Transfer functions of the optimal suspensions are obtained as a result. These functions indicate the limits of performance of suspensions under various conditions as well as optimal parametric values for suspension stiffnesses, damping, and mass.

Virtually all suspension designs resulting from both optimal suspension formulations and parametric studies of the physical models have been based on one-dimensional vehicle models. Both vehicle body accelerations and suspension excursions are inadequately represented by the one-dimensional analysis, as shown in this paper; at the same time both variables are of critical importance in vehicle/suspension/guideway design.

Coupled pitch and heave vehicle motions have been studied principally via simulation; however, some analytical results have been developed for special cases. As described above, the detailed complete vehicle simulations that have been performed have seen limited use because of their complexity. In-depth studies of automobile and transit vehicle ride quality [12,7], vehicle/flexible guideway dynamic interactions [11], and rail vehicle stability [16] have shown the importance of including the pitch mode in vehicle dynamic analyses. Pitch-heave, finite length vehicle models were used in flexible guideway studies to improve the ride quality prediction and to account for the distribution of the traveling load of the vehicle along the guideway. Body hunting in rail vehicles results from coupling between yaw and sway body modes, plus suspension (truck) motions; most attention in this area has been focused on elimination of instabilities below top operating speed, so that the effects of yaw on rail vehicle ride quality have not been fully explored.

Since the input seen by the front suspension propagates back along the vehicle to succeeding suspensions, the suspension inputs are correlated via pure time delays. The time delay formulation can appear in transfer function

*Many suspension parametric studies have used comfort criteria to develop standards or weighting functions [3,18] rather than RMS acceleration as measures of ride quality.

[19,20] or state variable [21] forms. Several studies have neglected the correlation between these guideway inputs, with significant degradation in predictive accuracy [7,21]. Hedrick, et al. [21] has shown that the effects of correlated inputs can be included in the Lyaponov's equation method for computing RMS vibration levels. However, no systematic study has previously determined the properties of pitch-heave, finite length vehicles.

III. GOVERNING EQUATIONS

Model Description and Basic Assumptions

The dynamic model for the finite length vehicle model differs fundamentally from the simple heave model in that the input disturbances are distributed spatially over the vehicle length. In this paper only the guideway inputs are considered. In general aerodynamic loading will be spatially distributed also, but very little information is available on the correlation between front and rear loading, and a quantified description of this input is likely to be dependent on the specific body shape.

The pitch and heave natural frequencies and damping ratios (i.e. the roots of the system transfer functions) are determined by the vehicle mass geometry and the suspension stiffness and damping. The spatial distribution of the guideway input affects the phase relationship between the motion of each point in the vehicle and the ground motion below it (i.e., all spatial terms appear in transfer function numerators). The phasing of the inputs and the vehicle motion depends on vehicle geometry and forward speed and the wavelengths of the guideway irregularities.

The basic model as shown in Figure 1 consists of a rigid body vehicle of mass M and inertia I about a center of gravity of arbitrary location. The coordinate system employed in the analysis is detailed in the figure. The governing equations are based on the following assumptions:

(a) All cushion dynamics are neglected; this is valid when the cushion mass natural frequency is much higher than the pitch-heave frequencies.
(b) The vehicle height is small compared to its length, and pitch angles are small so that each suspension is essentially in heave motion.
(c) The guideway displacement profile can be represented by a static description of either random or deterministic irregularities. This is the case for a guideway characterized by statistical roughness and for supported guideways not subject to dynamic excursions under transient vehicle loading.

Suspension Inputs

Each suspension exerts a force on the vehicle as a function G(s) of the relative suspension displacement,

$$\Delta\overline{F}_n = -G(s)\ (\Delta\overline{y}_{2,n} - \Delta\overline{y}_{o,n}) \tag{1}$$

In general G(s) represents the cumulative effects of the primary and secondary suspensions, and unsprung mass. For the analysis of the vehicle pitch-heave motions, only the secondary suspension is assumed to be important. Since the dynamics of the cushion are neglected, Equation (1) is equivalent to assuming that the pad follows the guideway. This assumption is verified by detailed simulation in Section V.

Each secondary suspension is defined to have stiffness and damping equal to the total vehicle heave secondary suspension divided by the number of suspensions N, attenuated by the finite pad length function ψ_v,

$$G(s) \approx \frac{1}{N}(b_b s + k_b) \cdot \psi_v \tag{2}$$

The guideway input to the nth suspension is correlated with the input to the first suspension, and is expressed using equivalent time delay operators as a function of the distance, $x_{1,n}$, from the center of the first cushion to the center of the nth cushion,

$$\Delta\overline{y}_{o,n} = e^{-T_n s}\Delta\overline{y}_{o,1} \equiv e^{-(x_{1,n}/V)s}\Delta\overline{y}_o \tag{3}$$

This time delay formulation preserves the proper phase relationships between successive pads in a manner consistent with the finite pad length analysis described below.

Vehicle Pitch-Heave Motion

The heave motion of the center of gravity $\Delta\overline{y}_2$ is found by summing the suspension forces $\Delta\overline{F}_n$ *

$$Ms^2\Delta\overline{y}_2 = -G(s)\sum_n(\Delta\overline{y}_{2,n} - e^{-T_n s}\Delta\overline{y}_o) \tag{4}$$

From geometry $\Delta\overline{y}_{2,n} = \overline{y}_2 - \theta \cdot x_{2,n}$, so that

$$(Ms^2 + b_b s + k_b)\,\Delta\overline{y}_2 = \frac{1}{N}(b_b s + k_b) \cdot \psi_v \cdot$$

$$[-N\eta\, L_v\, \overline{\theta} + \sum_n (e^{-T_n s}\Delta\overline{y}_o)] \tag{5}$$

where η defines the location of the c.g.

The pitch motion θ is found by summing the moments about the center of gravity $\Delta F_n \cdot x_{2,n}$, where $x_{2,n}$ is the distance from the pad attachment point (pad center) to the body center of gravity.

$$Is^2\overline{\theta} = \sum_n G(s)x_{2,n}[(\Delta\overline{y}_{2,n} - e^{-T_n s}\Delta\overline{y}_o)] \tag{6}$$

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*All summations in this paper are from n = 1 to n = N. Vehicles with missing midbody suspensions, such as wheeled vehicles, are treated by modifying the summation; unless otherwise stated the results in this paper generalize to both vehicle classes.

or

$$(Is^2 + \frac{\gamma^2}{N} b_b s + \frac{\gamma^2}{N} k_b)\ \overline{\theta} = \frac{1}{N} (b_b s + k) \cdot \psi_v \cdot [-N\eta L_v \Delta\overline{y}_2 - \sum_n x_{2,n} (e^{-T_n s} \Delta\overline{y}_o)] \tag{7}$$

where $\frac{\gamma^2}{N}$ is the mean squared value of the distance $x_{2,n}$.

Equations (5) and (7) are dynamically coupled through their cross terms in $\Delta\overline{y}_2$ and $\overline{\theta}$. Transfer functions relating accelerations in the vehicle to the guideway input have fourth-order characteristic equations. For inputs that have wavelengths much longer than the vehicle length, these transfer functions are equivalent to those of the one-dimensional heave model.

The vertical accelerations along the vehicle above the nth suspension are given by

$$s^2\Delta\overline{y}_{2,n} = s^2\Delta\overline{y}_2 - x_{2,n} s^2\overline{\theta} \tag{8}$$

Finite Pad Length

Finite pad length analysis is based on a description of the guideway profile, either random or deterministic, as a sum of sinusoidal profiles of the form

$$y_o(x) = y_o \left(\sin \frac{(Vt + x_o)}{\lambda/2\pi}\right) \tag{9}$$

where x is the coordinate along the guideway, and the guideway profile is assumed to be uniform across the pad area. By direct integration the effects of changes in exit gap in air cushions, in active volume in air cushions and magnetic suspensions, and in deformation in the contact patch of flexible wheels can be determined. The distribution of the guideway input over the finite contact area causes a frequency dependent filtering of wavelengths λ shorter than the pad length L_p, given by the filtering function,

$$\psi_v = \frac{\lambda}{\pi L_p} \sin\left(\frac{\pi L_p}{\lambda}\right) = \frac{2V}{L_p\omega} \sin\left(\frac{L_p\omega}{2V}\right) \tag{10}$$

for the active volume and contact patch cases [17]. (A similar expression exists for the exit gap case.)

IV. PROPERTIES OF PITCH-HEAVE MODELS

Symmetric Vehicles

For arbitrary location of the center of gravity Equations (5) and (7) are coupled. They become decoupled, and hence considerably simplified, for symmetric vehicles ($\eta = 0$). The heave mode natural frequency ω_h and damping ratio ξ_h for a vehicle of mass M with suspensions as previously described are identical to those of the one-dimensional model:

$$\omega_h = \sqrt{\frac{k_b}{M}} \qquad \xi_h = \frac{b_b}{2\sqrt{k_b M}} \tag{11) (12}$$

The pitch mode natural frequency ω_p and damping ratio ξ_p are

$$\omega_p = \sqrt{\frac{\gamma^2 k_b}{NI}} \qquad \xi_p = \frac{b_b \frac{\gamma^2}{N}}{2\sqrt{\frac{\gamma^2 k_b}{N} I}} \tag{13) (14}$$

The inertia I of the vehicle can be expressed as

$$I = r^2 I_o = r^2 \left(\frac{1}{12} M L_v^{\ 2}\right) \tag{15}$$

where I_o is the inertia of a uniform bar of length L_v. If the vehicle has its mass concentrated near its center, $r < 1$; $r > 1$ for vehicles with mass concentrated at the body ends. Then,

$$\frac{\omega_p}{\omega_h} = \frac{\xi_p}{\xi_h} = \frac{1}{r}\sqrt{\frac{12\gamma^2}{NL_v^{\ 2}}} \tag{16}$$

As the number of pads N goes to infinity, $\frac{\gamma^2}{N} \to \frac{L_v^{\ 2}}{12}$. For a vehicle with only four pads, $\frac{12\gamma^2}{NL_v^{\ 2}} = 0.985$; for symmetric vehicles with four or more suspensions along the length the pitch frequency and damping are given by

$$\frac{\omega_p}{\omega_h} = \frac{\xi_p}{\xi_h} \approx \frac{1}{r} \tag{17}$$

The relative magnitudes of the pitch and heave natural frequencies are important in determining the responses of the front and rear of the vehicle. If the pitch and heave modes are in phase, then the heave and pitch motion seen at the front of the vehicle will add, while they will tend to cancel each other in the rear. The reverse is true if the heave and pitch modes are 180° out of phase. At the center of the vehicle the distance $x_{2,n}$ equals zero, so that the acceleration is determined by the heave motion alone.

The relative phase of the heave and pitch motions is determined by examination of Equations (5) and (7) for symmetric vehicles, rewritten here in transfer function form,

$$\frac{\Delta\bar{y}_2}{\Delta\bar{y}_o} = \frac{\frac{1}{N}(b_b s + k_b)\ \psi_v \sum_n (e^{-T_n s})}{(Ms^2 + b_b s + k_b)} \tag{18}$$

$$\frac{\bar{\theta}}{\Delta\bar{y}_o} = -\frac{\frac{1}{N}(b_b s + k_b\ \psi_v \sum_n (x_{2,n} e^{-T_n s})}{(Is^2 + \frac{\gamma^2}{N} b_b s + \frac{\gamma^2}{N} k_b)} \tag{19}$$

The $(b_b s + k_b)$ and ψ_v terms in (18) and (19) will always contribute identical phase angles at all frequencies. The phase angles due to the quadratic denominators will be in the range $(0, -\pi)$, passing through $-\frac{\pi}{2}$ at their respective natural frequencies.

The phase difference between the delay terms in (18) and (19) can be shown to be constant for all frequencies $s = j\omega$ for any symmetric vehicle,

$$\angle \Sigma \ (e^{-jT_n\omega}) - \angle \Sigma \ (x_{2,n}\ e^{-jT_n\omega}) = +\frac{\pi}{2} \tag{20}$$

The relative phase difference between heave and pitch is then a function only of the natural frequencies and damping ratios. For the limiting $(N \geq 4)$ case, the behavior as a function of normalized frequency ω/ω_h can be shown parametrically by the inertia factor r^2. The phase difference is shown in Figure 2 and the amplitude responses of the front, rear, and c.g. body positions to sinusoidal guideway inputs are shown in Figure 3. For comparison the heave model response is also plotted.

For $r^2 > 1$, the pitch natural frequency is lower than the heave frequency. Between the two frequencies the phase difference between heave and pitch is close to zero, resulting in larger front accelerations, and lower rear ones, than that experienced by the c.g. The converse is true for $r^2 < 1$. At $r^2 = 1$, the phase difference is constant (at $-\pi/2$) and the front and rear experience identical accelerations. Depending on r^2, ξ_h and ω/ω_h the end point accelerations may be greater or less than those predicted by the heave model, as shown in Figure 3.

The acceleration at the c.g. is always less than, or equal to, that predicted by the heave model, and unaffected by the value of the pitch natural frequency. The magnitude of the heave damping ξ_h does not affect the qualitative behavior discussed above for underdamped ($\xi_h < 1$) suspensions; the principal effect of increased ξ_h is the reduction in the sharpness of the resonant peaks at ω_h and ω_p.

Asymmetric Vehicles: The Effect of Variable c.g. Location

The mass distribution within the vehicle body is a design function subject to engineering constraints. Moving the c.g. along the body away from the geometric center (of a symmetric vehicle) has two principal effects:

(a) By coupling the heave and pitch motions, the characteristic polynomial is modified as follows,

$$0 = D(s) = MIs^4 + (b_b I + \frac{b_b M\gamma^2}{N})s^3 + (k_b M\frac{\gamma^2}{N} + k_b I + b_b{}^2(\frac{\gamma^2}{N} - \eta^2 L_v{}^2))s^2 +$$

$$2b_b\, k_b\, (\frac{\gamma^2}{N} - \eta^2 L_v{}^2)s + k_b{}^2\, (\frac{\gamma^2}{N} - \eta^2 L_v{}^2) \tag{21}$$

where η is the fraction of vehicle length the c.g. is located aft of the geometric center. Since the c.g. location η appears in (21) only as a squared term, the characteristic polynomial, and hence the pitch and heave natural frequencies, are changed equally for positive and negative η. The effect of η is to cause the roots of (21) to spread for constant M, b_b, k_b, N and I. Whichever pole pair is of higher frequency, pitch, or heave becomes larger in frequency and damping ratio. Causing the higher natural frequency to increase into the 4 to 30 Hz human sensitivity range in many cases will deteriorate ride quality.

(b) The distances between the c.g. and the vehicle front and rear will, of course, change. Whichever distance increases will make that vehicle location more sensitive to pitch. This effect can be used as a design tool to "balance" the ride quality in the rear and front (desirable since ride quality is usually judged by the worst condition in the passenger compartment).

Effects of Pad and Vehicle Length

All previous suspension design studies have concluded that pad length L_p should be as long as possible to obtain the greatest attenuation of the guideway irregularities. Pad length has been principally constrained by geometric considerations such as minimum guideway curvature radii [22].

Finite vehicle length effects in the pitch-heave analysis cause the pad length effect to be much less important. The acceleration of the c.g. of a symmetric vehicle with suspensions uniformly distributed along the body, Equation (18), is independent of the pad length. The magnitude of the time delay term times the finite pad function is independent of pad length (using $L_p = L_v/N$):

$$\left|\frac{1}{N} \cdot \psi_v \cdot \sum_n e^{-T_n s}\right| = \frac{2V}{L_v\omega} \sin\left(\frac{L_v\omega}{2V}\right) \tag{22}$$

Thus for the heave mode the entire vehicle length filters the guideway input. The expressions for the pitch mode and for the coupled pitch-heave modes of asymmetric vehicles are much more complex; studies of their limiting behavior show that for typical high speed vehicles (see Table 1), ride quality is unaffected by changes in pad length below L_p = 9.15 m (30 feet). The guideway geometry often constrains pads to this length or less. Therefore for vehicles with suspensions distributed along the vehicle body, the individual pad length is not important. However, pad length effects will still be beneficial to vehicles with only front and rear suspensions since the filtering effect will be increased by adding to the total contact area.

V. SIMULATION RESULTS

Verification of Modeling Assumptions

The representation of the suspension dynamics in Equation (2) as being equivalent to the secondary suspension in series with the finite pad length filter is verified by comparison with simulations of complete vehicle/suspension systems. For example, Figure 4 shows a unit step response of two adjacent air cushions coupled through their feeding system on a 134 m/s (300 mph) TACV. Both primary and secondary suspension dynamics are simulated for different pad lengths.

The responses consistent with Equation (2) for this simulation would be two ramps with slopes of V/L_p, the second delayed by (pad separation/V) seconds. The ramp results from the time averaging of the active volume by the cushion as it encounters the step. The simulated responses differ from the approximate representation only in their high frequency content, which is filtered by the secondary suspension. The decoupling of the high frequency primary suspension dynamics and the low frequency secondary suspension/vehicle displacements is demonstrated in Figure 5, again verifying the approximation in (2).

Parametric Effects

The effects of body inertia I, suspension damping b_h, and c.g. location η on ride quality as described in Section IV are demonstrated in Figures 6, 7, and 8. The design example considered is described in Table 1. The vehicle is traversing a guideway with surface roughness spectral density $\pi VC/\omega^2$. Shown for comparison is the U.S. DOT UTACV ride quality specification [17].

The degradation of ride quality with mass concentrated near the vehicle center ($r^2 = 0.5$) is shown resulting from a high pitch natural frequency. Similar degradation can be shown to result from distributing suspensions to vehicle ends by eliminating the middle suspensions, also raising ω_p. Also demonstrated is the dependence of optimum damping ratio ξ_h on pitch and heave natural frequencies, and on the specific ride quality criterion used.

TABLE 1

VEHICLE PARAMETERS - DESIGN EXAMPLE

Mass, M, kg (lb)	54 440 (120 000)
Length, L_v, m (ft)	36.6 (120)
Number suspensions along vehicle length, N	6
Heave natural frequency, Hz	0.75
Forward speed, V, m/s (mph)	134 (300)
Guideway surface roughness, C, m (ft)	36.6×10^{-6} (1.2×10^{-6})

The cumulative effects of variable c.g. location are illustrated in Figure 8, in which $\omega_p < \omega_h$, for values of η equal to -0.2, 0, and +0.2. The shifting of the heave natural frequency from 0.75 Hz to 0.87 Hz for $\eta = \pm 0.2$ is evident, as are the adverse results of an aft location for the c.g. on front acceleration, and front c.g. location on rear acceleration.

From these examples it is clear that the pitch-heave model is necessary, as a minimum level of sophistication, for predicting ride quality.

VI. IMPLICATIONS FOR VEHICLE AND SUSPENSION DESIGN

The primary variables to be considered in TLV design include the vehicle weight, shape, and mass distribution, the operating speed, primary suspension characteristics including its unsprung mass and pad length L_p, the secondary suspension dynamics (if used), the guideway geometry, and the external force inputs. A practical design must meet passenger comfort criteria, such as the ISO specifications, with minimal suspension power consumption when operating over economically feasible guideways.

The overall suspension design (secondary, if used, and primary) must support and guide the vehicle along its prescribed path while ignoring local irregularities or errors in the guideway. In the dominant frequency range for vehicle-suspension systems the suspension may be approximated as a stiffness k_b and damping b_b. Lowering k_b lowers the dominant frequencies and improves ride comfort by reducing the acceleration resonant peak. However, limits are placed on low values of k_b by allowable dynamic secondary suspension and gap displacements. Minimizing unsprung mass is beneficial since it reduces dynamic loads on the guideway and usually improves ride comfort by causing the unsprung mass resonance to occur at a frequency that is filtered by the secondary suspension and finite cushion area.

The dominant dynamic motions of TLV in pitch and heave occur with characteristic frequencies ω_p and ω_h which are typically in the range of 0.5 to 2 Hz. Since vibrational accelerations are proportional to frequency squared, it is desirable to make both ω_p and ω_h as low as possible. In most cases ω_p should be made less than ω_h, in which case the worst accelerations will occur at the front of the vehicle. To lower the pitch natural frequency for a given heave stiffness and damping, the following measures can be taken:

(a) Distribute the vehicle mass toward the ends to increase I. Heavy equipment such as axial fans, electric-power conditioning equipment and LIM's are examples of components that could be so located. The upper bound on I is given by $r^2 = 3$ (all mass concentrated at ends) with a realistic limit probably of $r^2 = 2$. (The prototype UTACV recently developed by Rohr Industries has an r^2 equal to about 1.6, with the center of gravity almost exactly at the middle of the vehicle length.)

(b) Reduce rotational stiffness. This would be accomplished by concentrating stiffness near the vehicle center. However, a trade-off between pitch stiffness and allowable endpoint excursions exists, limiting the concentration of stiffness near the center. Added data on transient aerodynamic moments is needed to quantify the bound on pitch stiffness.

Optimum locations of the c.g. will be dependent on the specifications of the vehicle body (mass, flexibility), suspension, and shape of the comfort criteria. At this level of analysis, however, no clear advantage is evident for locating the c.g. away from the geometric center, while the shifting of the natural frequencies and lengthening of the distance to one vehicle end tend to adversely affect passenger ride quality for non-zero values of η. In contrast to these guidelines, a common vehicle analytical model is one consisting of a uniform mass distribution with discrete suspensions at the front and rear only. In this case the pitch frequency ω_p is greater than the heave frequency ω_h by a factor of $\sqrt{3}$; the resulting accelerations in the 4 to 30 Hz ride quality sensitive range may be up to an order of magnitude greater than for a similar vehicle designed according to the above guidelines. A poorly chosen vehicle configuration can lead to costly overspecification of required guideway smoothness and stiffness.

Optimal damping ratios for each suspension are determined by comparison of acceleration spectral densities with frequency dependent ride quality criteria. Considering only heave motion yields optimal values of ξ_h between 0.2 and 0.3 [10]. The reason for the existence of an optimum is that as damping increases the height of the resonant peak at ω_h is reduced, but at the same time more power is transmitted through the suspension to the vehicle at high frequencies. When both pitch and heave motion are considered, optimal suspension damping ratios may range from 0.1 to 0.5, depending on the pitch natural frequency.

When active feedback is used to control a suspension, improved performance can be obtained by sensing absolute vehicle accelerations and vehicle-guideway displacements and using the results to control suspension force. Acceleration feedback increases the effective mass and rotational inertia of the vehicle, reducing the pitch and heave natural frequencies without lowering the respective stiffnesses in these two modes. Displacement feedback is used to alter the stiffness and damping characteristics of the passive suspensions.

The design of support and guidance suspensions differ in several respects. The guidance suspensions support no equilibrium load; the preload is thus a free parameter which can be advantageously used with suspensions with non-linear force-deflection characteristics. Guidance suspensions act in push-pull, effectively doubling the suspension stiffness and damping. Finally, available ride quality data [5] indicates that comfort sensitivity is higher in the lateral plane, resulting in stricter comfort requirements.

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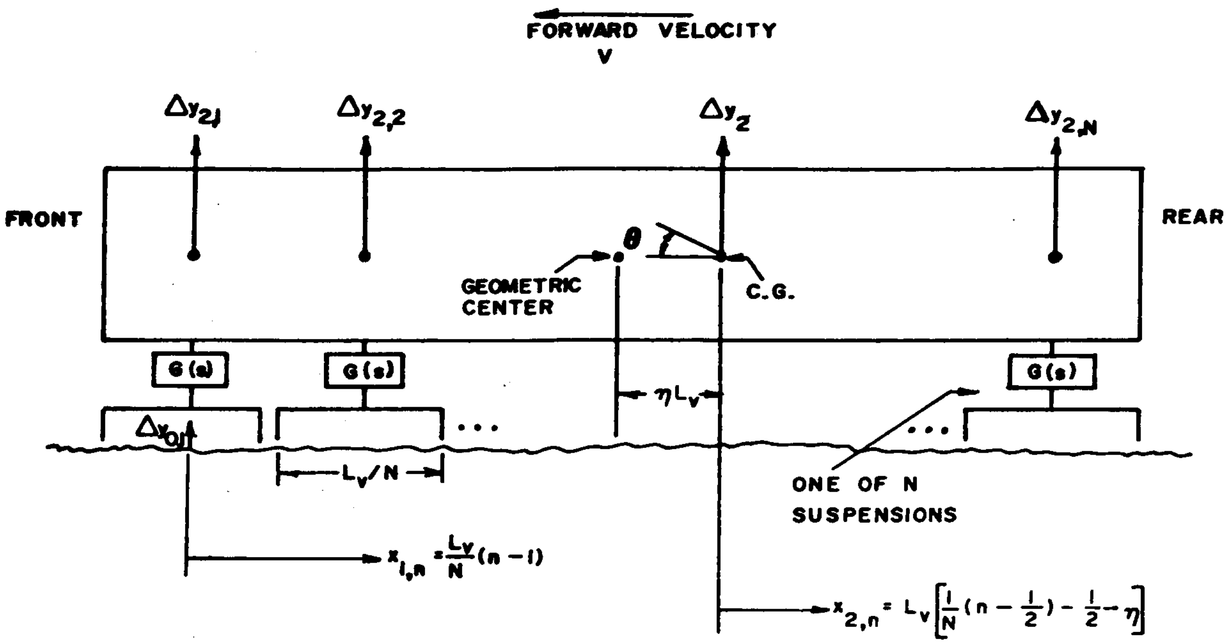

Figure 1.- Schematic of general pitch-heave model.

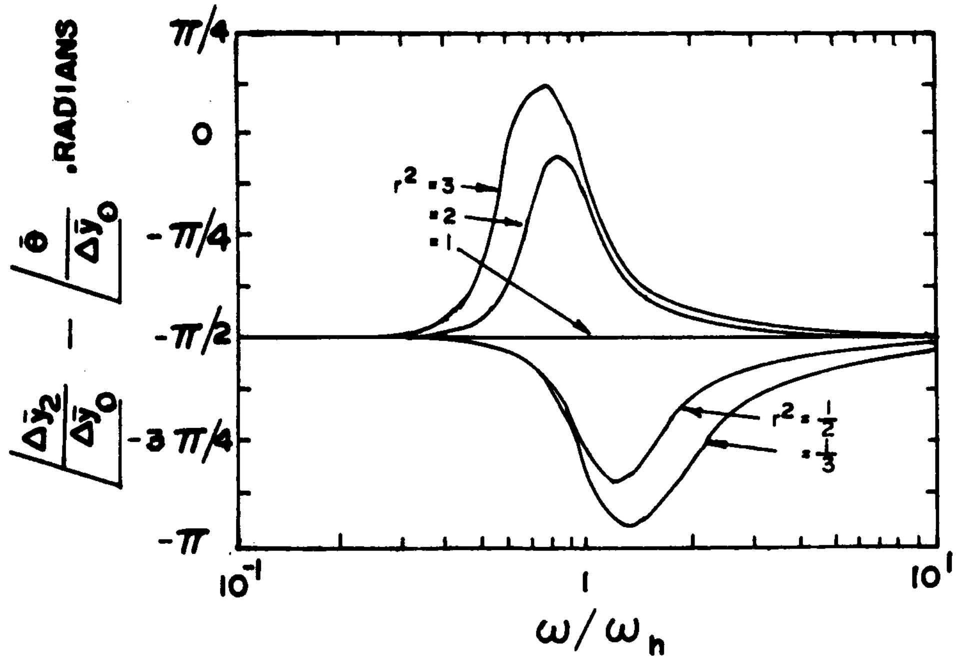

Figure 2.- Phase angle between heave and pitch motions.

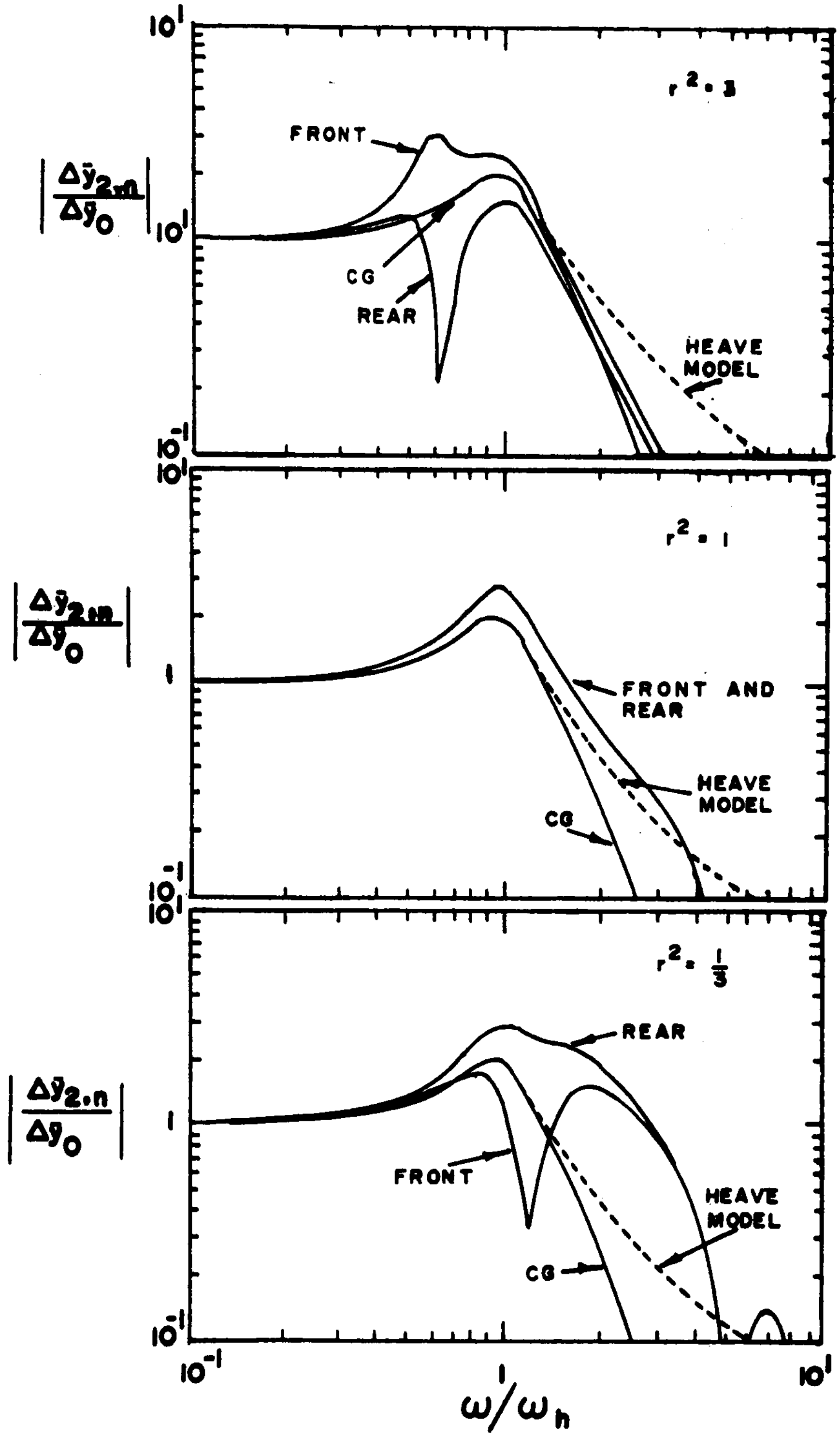

Figure 3.- Magnitudes of transfer functions relating vehicle displacement to guideway input amplitude using pitch-heave models.

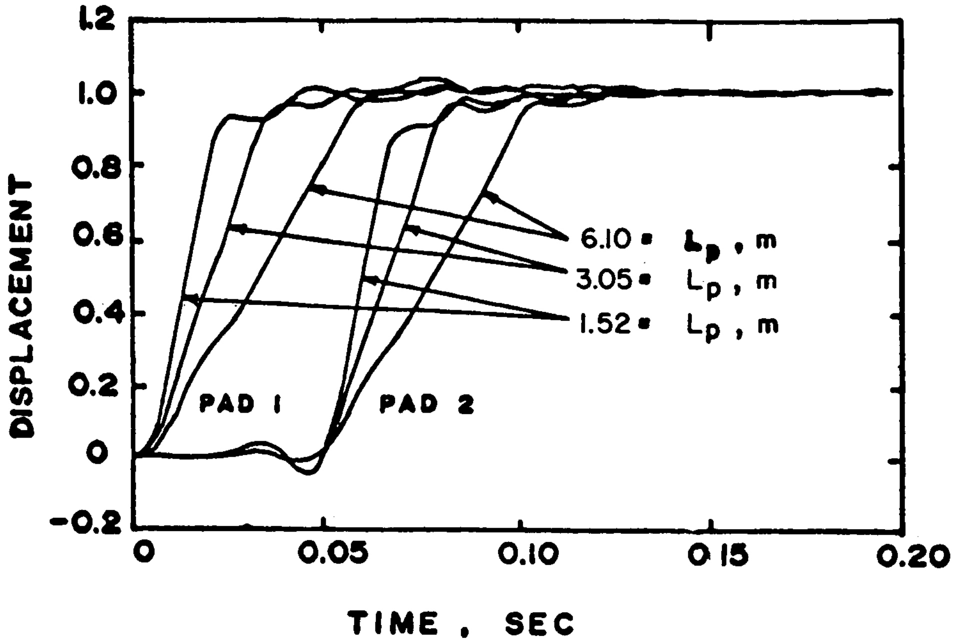

Figure 4.- Unit step response of two cushions, twenty feet apart, for different pad lengths.

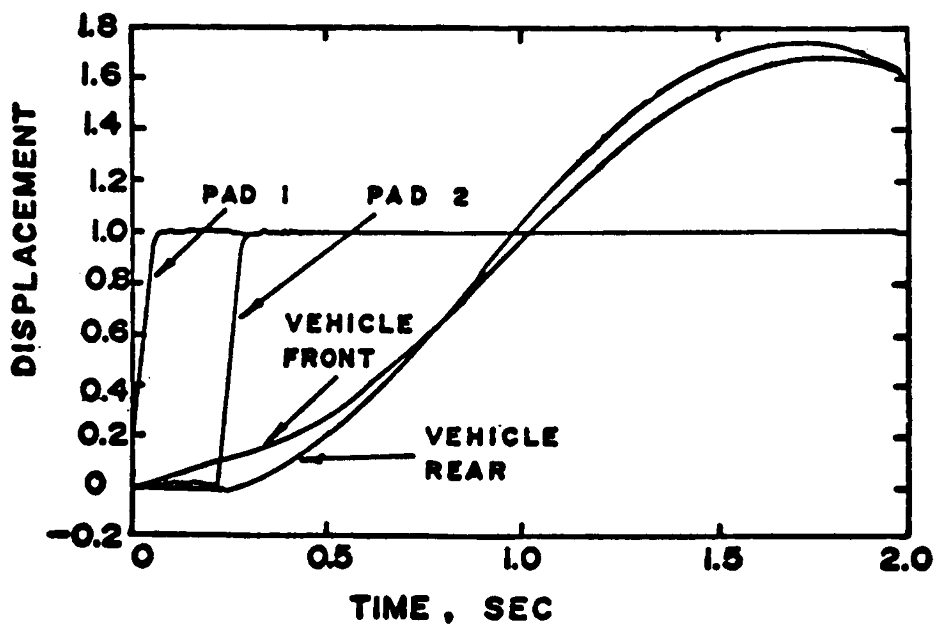

Figure 5.- Unit step response of two cushions, pitch-heave vehicle model.

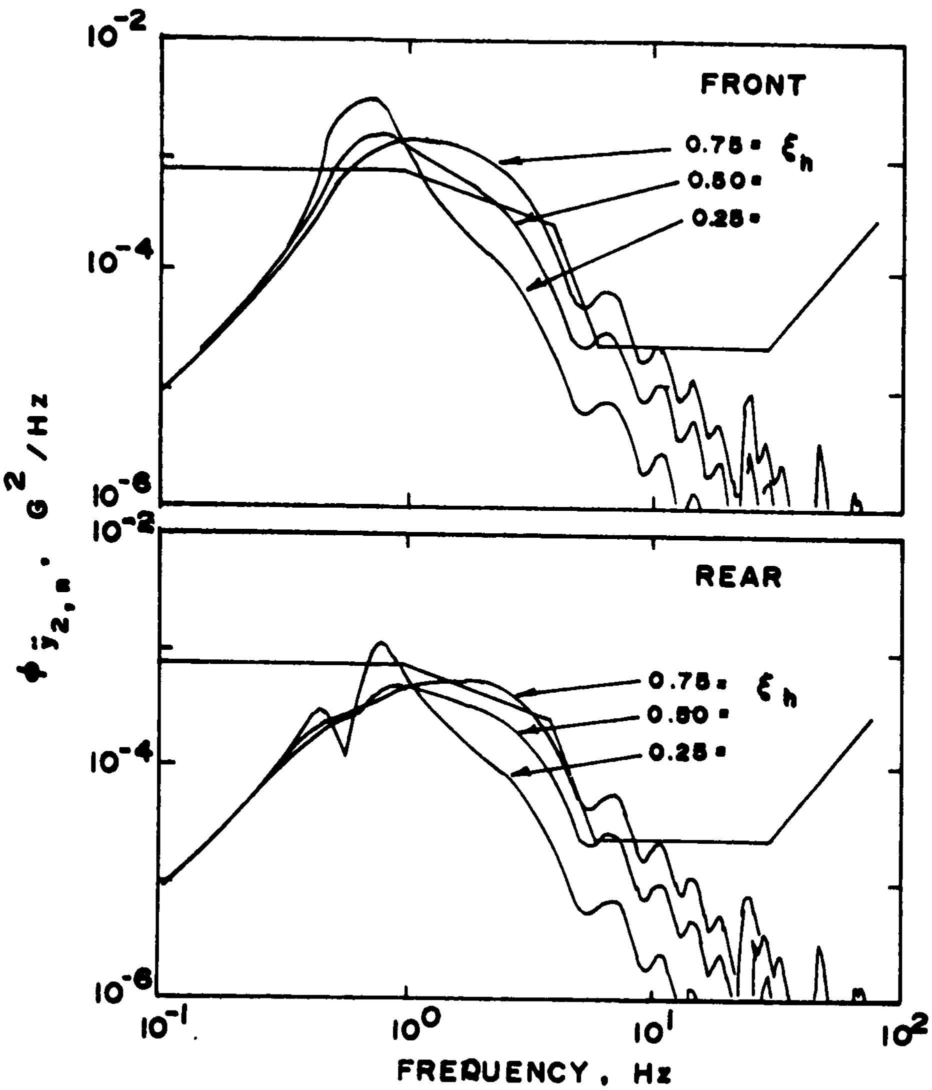

Figure 6.- Pitch-heave model acceleration spectral density as a function of suspension damping for $r^2 = 2$.

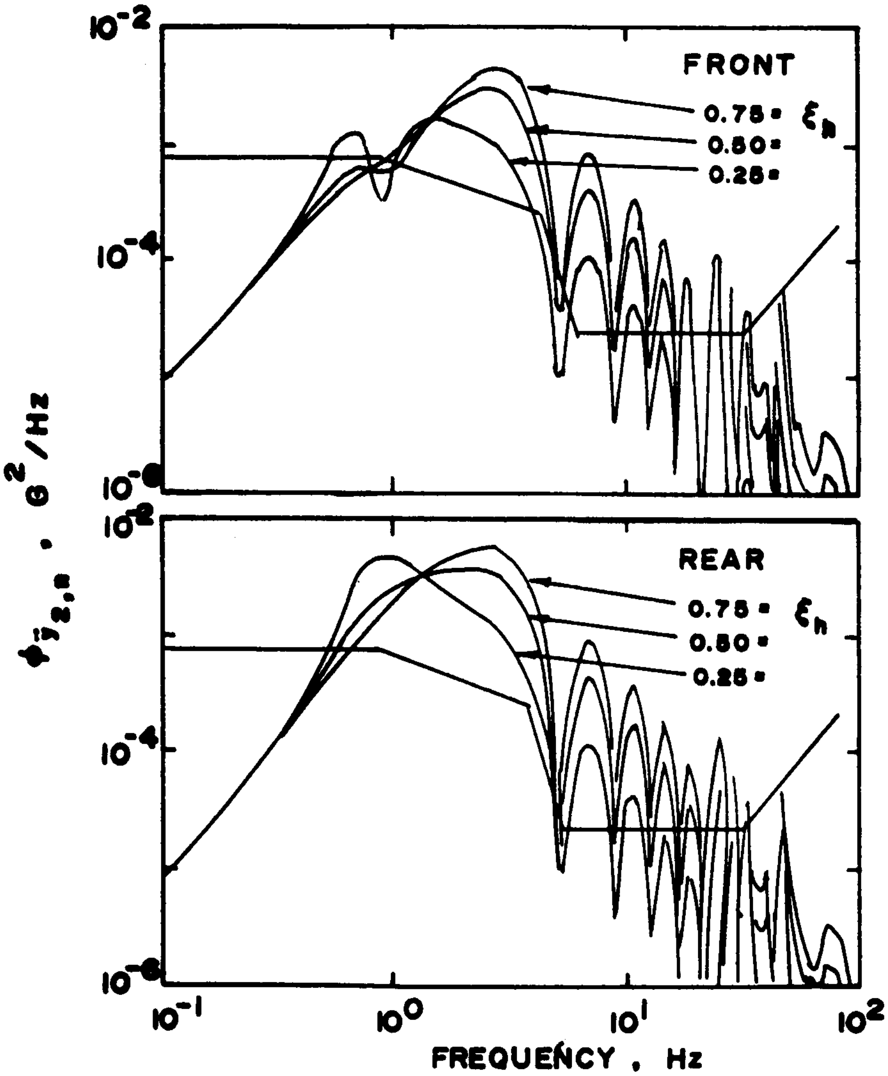

Figure 7.- Pitch-heave model acceleration spectral density as a function of suspension damping for $r^2 = 0.5$.

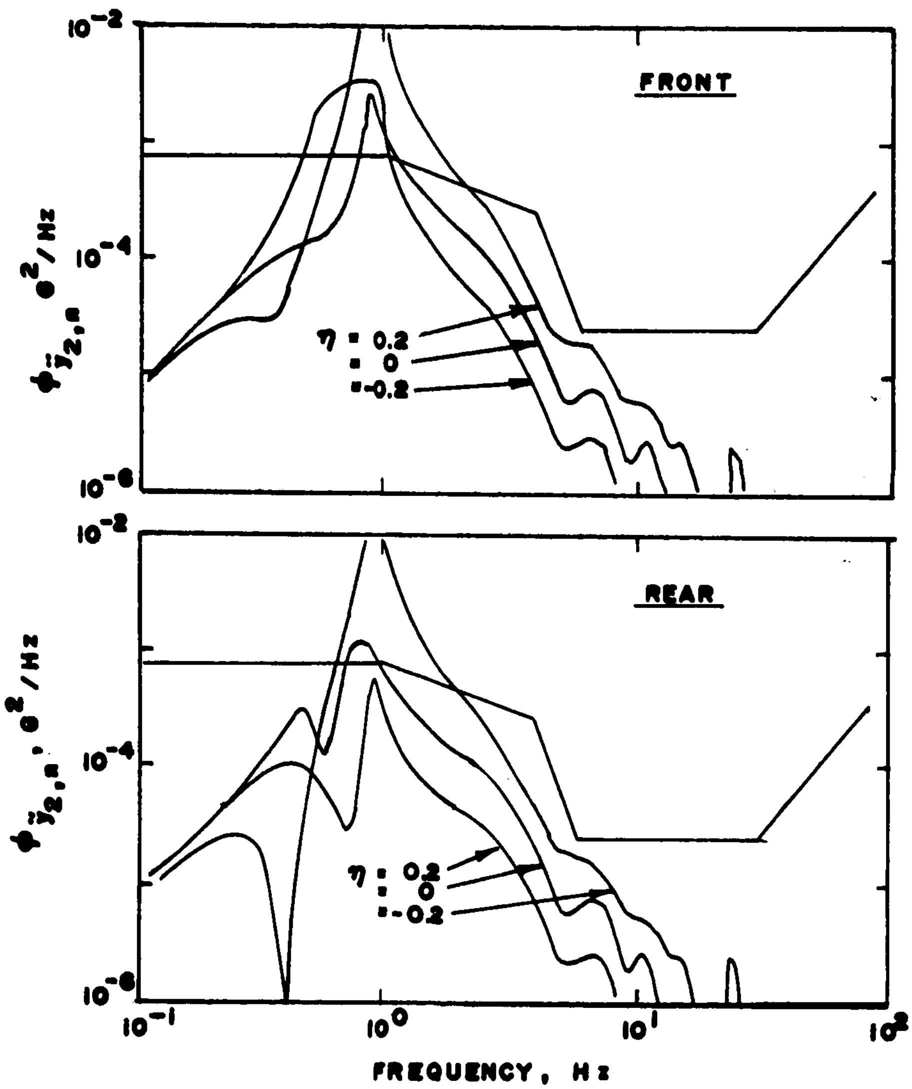

Figure 8.- Effect of variable c.g. location on ride quality $\left(r^2 = 2,\ \xi_h = 0.25\right)$.

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AN APPROACH TO HIGH SPEED SHIP RIDE QUALITY SIMULATION

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SUMMARY

The high speeds attained by certain advanced surface ships result in a spectrum of motion which is higher in frequency than that of conventional ships. This fact along with the inclusion of advanced ride control features in the design of these ships has resulted in an increased awareness of the need for ride criteria. Such criteria can be developed using data from actual ship operations in varied sea states or from clinical laboratory experiments. A third approach is to simulate ship conditions using measured or calculated ship motion data.

Recent simulations have used data derived from a math model of Surface Effect Ship (SES) motion. The model in turn is based on equations of motion which have been refined with data from scale models and SES of up to 101 600-kg (100-ton) displacement.

Employment of broad band motion emphasizes the use of the simulators as a design tool to evaluate a given ship configuration in several operational situations and also serves to provide data as to the overall effect of a given motion on crew performance and physiological status. It additionally averts to a degree the more clinical problem of predicting reaction data from single frequency experiments. The long term exposure (currently up to 48 hours per simulation) was chosen to evaluate any cumulative effects of fatigue or stress that might be induced by the motion.

The particular motion simulated to date is especially interesting because its spectrum of 0.1 to 5 Hz covers both the classical motion sickness region and the mechanical interference region. The tendency of the low frequency motion to induce kinetosis and the transient nature of kinetosis leads to special problems in experimental design and to the interpretation of data as required for fine tuning of ride control.

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INTRODUCTION

Ship Motion, or ride quality, is the result of excitation of ships' response characteristics by energy contained in the wave train through which the ship passes. The major factors influencing ride quality in any given hull form are:

(a) Wave height and period

(b) Distribution of energy within an encountered sea condition

(c) Relative speed between the ship and the sea surface

(d) The ships response characteristics in all six degrees-of-freedom

These elements interact to alter the magnitude and frequency of the ship motion and as a consequence to effect personnel aboard the ship. The "effects" either manifest themselves as discomfort (in severe cases leading to extreme nausea and vomitting) or performance degradation (or both). Compared to other external factors affecting human behavior such as noise, temperature, and vibration, minimal quantative data is available on ship motion either in respect to acceptable levels or sensitive frequencies. Discomfort has been accepted, at least militarily, as part of the cost of operation at sea while little or no account has been taken of crew performance degradation (other than in extreme conditions).

Thus, with the advent of new ship forms, there is little or no basis upon which to judge possible crew problems arising from the ship motion environment - not even from that part of the predicted motion spectra which is similar to conventional hulled ships, let alone from that part of the spectra which is new.

Various means of achieving the desired knowledge are available. The approach taken to assessing the motion predicted for the SES has been to simulate the ride environment with observation and measurement of the effects on volunteer subjects. However, before proceeding with the selection of a suitable simulator, it is necessary to understand something of the characteristics of the motion environment to be reproduced.

THE FORCING FUNCTION

The distribution of wave amplitude as a function of frequency for a fully wind developed sea is described by the Pierson-Moskowitz distribution (ref. 1), $S(\omega)$, in terms of dimensionless empirical constants, α and β, the gravitational constant g, the wind velocity u, and the angular frequency of the wave, ω, as

$$S(\omega) = \frac{\alpha g^2}{\omega^5} \exp\left[-\beta\left(\frac{\omega_u}{\omega}\right)^4\right]$$

where $\alpha = 0.0081$
$\beta = 0.74$
$\omega_u = g/u$

According to this distribution, the energy peak of the sea occurs at a frequency depending only on the wind velocity:

$$\omega_p = \frac{g}{u} \left(\frac{4}{5}\beta\right)^{\frac{1}{4}}$$

Since a ship traveling across the surface of the sea experiences a wave encounter frequency, ω_e, which is related to the actual wave frequency, ω, ship velocity, V, and ship heading angle with respect to the wave velocity vector, χ, by

$$\omega_e = \omega - \omega^2 \frac{V}{g} \cos \chi$$

it follows that the ship will be driven by a forcing function with apparent spectral distribution:

$$S(\omega_e) = S(\omega)\frac{\partial\omega}{\partial\omega_e}$$

$$= \frac{S(\omega)}{\left[1 - \frac{2\omega V}{g} \cos \chi\right]}$$

and energy peak whose frequency varies with sea state and ship speed as indicated in figure 1.

The significance of this fact is that ships traveling at speeds in the range of 20 knots routinely experience this peak in the energy spectrum at encounter frequencies of the order of 0.16 Hz to 0.6 Hz while high speed ships currently under design and potentially capable of speeds on the order of 100 knots can be expected to experience these energy peaks at encounter frequencies as great as 1.9 Hz. These ships, of which the SES is an example, will thus operate in a motion region which falls well above that of conventional ships and for which neither extensive practical or laboratory experience exists.

THE SURFACE EFFECT SHIP

The SES itself is unique. Its general features are depicted in figure 2. An SES travels across the surface of the water supported by a cushion of air. The air is contained on two sides by the ship's rigid side walls and at the bow and stern by the ship's flexible bow and stern seals. Air escapes around these surfaces and through controlled openings in the form of valves or louvers in the deck or sidewalls of the ship.

Forces on both the seals and sidewalls affect the quality of the SES ride, but the predominant force and nature of the ride results from the confined air cushion. The nature of the cushion is in turn determined by the system of fans which supply pressure to the plenum, the variable deck openings which vent air from the plenum, and the surface of the sea whose rough contour results in a pumping action as the SES traverses its surface.

The general nature of the SES has been modeled extensively (for example, see ref. 2 and ref. 3.) The modeling starts by developing the basic physics of the individual forces alluded to above and by then coupling them into a central mathematical equation of motion. The equation is then driven by an irregular wave forcing function and the resulting time varying 6-degree-of-freedom (DOF) motion of the ship is used to study the ship characteristics.

The previously described Pierson-Moskowitz distribution has been used to describe the irregular wave driving function in all of our simulations to date. (Any forcing function can be used to drive the equation. The Pierson-Moskowitz distribution has been used because it is considered a good general representation of a fully developed sea.) The continuous distribution is approximated with a discrete series by dividing the wave spectrum into logarithmic intervals such that:

$$\ln \omega_i - \ln \omega_{i-1} = \frac{1}{N} (\ln \omega_N - \ln \omega_o)$$

where

N = total number of frequency intervals.

According to this approximation, the time varying wave amplitude y(t) can be represented as an 8-element trigonometric series:

$$y(t) = \sum_{i=1}^{8} A_i \cos \overline{\omega}_{ei} t$$

$$\overline{\omega_i}^2 \equiv \omega_i \omega_{i-1}$$

where the encounter frequency, $\overline{\omega}_{ei}$, explicitly takes into account the shift in the apparent wave energy spectrum $S(\overline{\omega}_{ei})$ due to ship speed. The coefficients A_i define the peak amplitude at frequency elements ω_i and are determined from:

$$A_i^2 \approx 2S(\overline{\omega}_i) \; [\omega_i - \omega_{i-1}]$$

THE RESULTANT MOTIONS

When this discrete representation of $S(\omega_e)$ is utilized in the equation of motion and the time varying solution of the 6-DOF motion is analyzed in the frequency plane, the one-third octave heave acceleration spectra depicted in figure 3 results. These spectra represent the motion at the center of gravity of an early conceptual SES model (configuration A) traveling in a bow sea in various speed and sea state conditions. It is evident that the lower speed and higher sea state conditions produce a shift in the peak motion to lower frequencies and greater peak accelerations with the predominant energy of the motion falling in a spectral region which is midway between that of conventional ships and conventional surface vehicles. Figure 4 (data courtesy of Bell Aerospace Co.) indicates the predicted motions at the center of gravity of a more recent design. Note that this ship is predicted to have a better ride quality in terms of total acceleration and that the acceleration spectra undergo a major redistribution as a result of the use of a Ride Control System (RCS). (The term Ride Control System refers to those general features of the SES that are used to control the ride quality. They may be either active or passive in nature and are exemplified by the valves and louvers mentioned previously.)

The statistics of the motions are summarized in table 1. The expected frequency, f_e, the predicted number of maxima per unit time, N_1, and the spectral broadness factor, ε, are computed from the power spectral density for acceleration, $\phi(f)$, the one-third octave acceleration amplitude $A_j(1/3)$ and one-third octave center frequency f_{cj} by

$$m_k = \int_o^\infty \phi(f) f^k df \approx \sum_{i=1}^{N} A_j^2(\tfrac{1}{3}) f_{cj}$$

$$f_e = (m_2/m_o)^{\frac{1}{2}}$$

$$N_1 = (m_4/m_2)^{\frac{1}{2}}$$

$$\varepsilon = [1-(f_e/N_1)^2]^{\frac{1}{2}}$$

Note the broad band nature of the motion as made evident by the relatively large value of ε. The heave motion of the SES when excited by a sea with Pierson-Moskowitz distribution is also predicted to have a reasonably Gaussian amplitude distribution despite the high degree of non-linearity present in the equations of motions. (See fig. 5.)

A further feature of the motion is indicated in Table 2 which compares the Root Mean Square (RMS) acceleration in heave surge, and sway for configuration A traveling in a bow sea. As is the case for most other operating conditions, the vertical acceleration (the combination of heave and pitch motion at the point undergoing motion) greatly exceeds the other motion components. This is a result of two conditions: (1) the sidewalls and seals of the SES have a minimal immersion and consequently very small side forces are generated in surge and sway; (2) as the SES begins to pitch or roll extensively the sidewalls or seals begin to vent air and are quickly restored to the water surfaces.

MOTION SIMULATION

The initial objective of developing a motion simulation program was to test for the presence of any gross physiological or performance changes attributable to exposure to the "new" high speed ship environment. At the planning stage, certain minimum requirements were identified and certain constraints were recognized which are worth some discussion before proceeding to a description of the simulations run to date. Included are

(a) The Simulator

(b) The Subjects

(c) The Experimental Design

(d) The Task Battery

The Simulator. The first requirement of the simulator (or Motion Generator) was that it should faithfully reproduce either real world or predicted motion in as many degrees of freedom as possible. This was by no means a simple requirement to meet. The absence of data on the comparative importance of subsets of motion within the total bandwidth and the significance of cross coupling effects between the various axes suggested a machine having a broad bandwidth and flat response characteristic in all six degrees of freedom, but no such machine existed. A compromise was, therefore, immediately necessary. Since initial concern was with high speed operation, a relatively small displacement (in heave) 6-DOF machine with good high frequency (0.1 to 10 Hz) characteristics was chosen and is described in more detail later. As it became evident that slower speed higher sea states posed problems similar to those experienced in conventional ships, the need for a "rough water" simulator (larger displacement, lower bandwidth) was also identified. Such a machine is also described in more detail later.

Whatever the specific physical limitations of any particular machine, it was rapidly evident that the ability to faithfully reproduce the commanded input was essential when dealing with broad band multi-axis motions. A subtle reason for placing emphasis on high fidelity is that it soon became evident that human response appeared to be very sensitive to certain characteristics (e.g. wave crests and troughs) and any tendency to "wash out" such characteristics rapidly removed realism from the simulation. In the same vein, it is of interest to know whether the motion character can be described simply in RMS terms or whether some weight needs to be given to the ratio of peaks to average values, etc. Interpretation is discussed in more detail in a later section and is mentioned here simply to underline the requirement for a "quality" simulation.

The simulator was also required to support a load representing a ship compartment which ideally would include at least two crewmen, a variety of tasks and life support facilities.

Since human volunteers were to be used, considerable emphasis was placed on safety features of the chosen machine(s) although time and space do not permit further discussion here.

The Subjects. The use of human volunteers for work of the proposed nature is strictly controlled to ensure the safety of the individual whether or not he appreciates the potential hazards of the position to which he is exposed. The protocol includes rigorous medical screening prior to acceptance as a volunteer, pre and post exposure medical examination, medical observation whenever in motion, and complete freedom to leave the simulation at any time without cause or explanation.

Subjects to be "scientifically" acceptable should be either carefully selected average people or part of a sufficiently large sample size to represent the population at large. Again, compromise has been necessary and the various simulations have used some 35 subjects at one time or another ranging from naive to experienced seamen.

Motivation is a major consideration. Motion sickness if experienced is not a minor event. The freedom to leave the simulation at any time makes it extremely difficult to ensure that volunteers "live through" the experience as they would in the real world. Significant emphasis is therefore placed on maintaining crew morale by (among others) having a two man crew, providing a busy, realistic work schedule and scenario, allowing considerable choice of food and drink, and maintaining an informal relationship between subjects and test administrators.

Subjects are constrained not to drink alcoholic beverages during time out of the simulator, to maintain a defined sleep cycle and to avoid any pastime which may interfere with their ability to maintain a positive attitude to the simulation.

The constraints imposed by confinement, the latent fear of vomitting, and the artificial nature of the motion generator's mechanical driving system are frequently commented upon by volunteers and are judged to produce the most difficulty in maintaining a smooth and orderly simulation series.

The Experimental Design. SES motion simulations to date can best be summarized as "exploratory" rather than "experimental". As stated earlier, the initial objective has been to assess the effects of SES motions in a gross manner related to physiological and performance changes. More recently an attempt has been made to establish ride quality criteria at least to a level of confidence which assures that a ship having a RMS acceleration less than some given value will have no major problems resulting from ship motions.

The simulation of high speed ship motion as currently undertaken is highly complex. It uses broad band, quasi-random motion, human volunteers, a battery of real world related and scientific tasks all of which come together within the limits of a 2.4 m by 2.4 m (8 ft by 8 ft) cabin. Refinements continue to evolve at every stage to improve the acceptability of data collected but it should be understood that the current program involves many variables and constraints which are difficult to filter out with total confidence. Simulations are planned ahead of their actual execution; therefore, they have certain fixed aspects: duration, conditions to be tested, measurements and observations to be made, etc. The arrangement attempts to follow a balanced design of motion and control conditions; however, while the most recent series has a set of protocols governing contingencies for various deviations from the test plan, structure is still fairly loose and provides for opportunities to explore targets of opportunity. The overall plan calls for exploration of extended periods of exposure (currently out to 48 hr continuous in one condition) and for comparison of effects in a variety of sea state/speed conditions. (Simulated conditions are chosen to bound the speed and sea state parameters set for a 2000-ton SES.) Therefore, when control conditions are added, the simulation program becomes extensive and difficulty in maintaining crew motivation and morale can become significant due to their confinement in "unreal" surroundings and the repetitiveness of the daily routine.

The Task Battery. At the outset the primary objective of including crew tasks was to provide meaningful employment for the volunteer crews. Tasks were scored or commented on by crewmen as to their realism and difficulties encountered in their execution. Crewmen also completed questionnaires on such matters as the degree to which they were affected by the motion both personally and in their ability to carry out specified tasks.

As the program developed, a more sophisticated array of tasks and tests was produced. While always trying to maintain the cooperation and understanding of the volunteers by ensuring that tasks or tests do not become too esoteric, the battery (see table 3 for full details) currently includes measurement of sleep performance and measurement of head movement by means of a special mouth mounted 6-DOF accelerometer package (originally developed by the Naval Aerospace Medical Research Laboratory, Detachment, New Orleans) as well as the more real world (and popular) navigation plotting, and missile directing (XY tracking) tasks. Volunteers still complete questionnaires and considerable emphasis is placed on briefing, debriefing, and interaction between volunteer crews and the directing staff. Much valuable insight has been gained by observation - e.g., variation of head movement with motion states with and without headgear - and by subjective discussions with volunteer crews - e.g., techniques learned for accommodating mechanical interference, etc.

The lack of totally controlled conditions using a minimum number of variables presents difficulties when attempting to achieve maximum knowledge from task data; however, tasks and their scores have generally served the program well. Remarkable consistency has been seen in some scores; strong trends in others. Gross questions are being answered: crews can sleep, can perform life support functions, do experience kinetosis in some conditions and not in others, do have more difficulty performing fine motor tasks and so on.

NASA MARSHALL SPACE FLIGHT CENTER (MSFC) SIMULATION

As is evident from the preceding discussion, the predominant interest at the onset of the program was centered on the high speeds predicted for the SES and the corresponding high frequency motion as compared to that of conventional ships; accordingly, the motion generator at MSFC was selected for the first simulation of the 6-DOF motion of the SES. This work was performed in the fall of 1973 and has been described briefly in reference 4 and more extensively in reference 5. The MSFC motion generator is an early version of the "large-stroke" simulators used for flight training for large jet aircraft. The facility includes a closed circuit television system for simulation of external terrain viewing and, as configured for our test, the four-place cabin depicted in figure 6 (adapted from ref. 4).

The purposes of this initial simulation were fourfold: (1) to test for the presence of any gross physiological effect such as extreme fatigue or stress that might be correlated to the motion, (2) to test for the presence and nature of any gross performance decrement, (3) to assure that a simulation of SES ride quality could be provided which was subjectively similar to that of an actual SES, and (4) to determine the relative importance of the SES motion associated with a given DOF.

The first two objectives were realized by means of general medical examinations before and after each motion exposure and by a battery of tasks administered during the exposure. The third objective was achieved by exposing the subjects to motions reproduced from recorded operations of the SES-100B and obtaining their opinion of the ride quality. (The SES-100B is one of two 101 600-kg (100-ton) SES test craft.) The final objective was achieved by exposing the subjects to the 6-DOF motion predicted for the 2000-ton SES and selectively deactivating one or more DOF. As indicated in the description of SES motion, the magnitude of the heave acceleration significantly exceeds that of the other DOF. As a result of this fact and on the basis of the MSFC results it has been deemed sufficiently realistic to restrict future tests to 3 DOF, at least until our knowledge of motion effects has increased considerably.

As it turns out, the fact that a 3-DOF simulation satisfies primary requirements is fortunate since the MSFC motion generator introduced an artifact into the high sea state simulations. The originally predicted capability of the motion generator operating with a cabin of approximately the same mass as used in our simulations is indicated in figure 7. The motion generator was limited at low frequency by the stroke of the simulator and at high frequency by the load capacity. In the intermediate region, the capability was expected to be limited by the flow rate of the motion generator's hydraulic system. This would have resulted in a "soft" limiting occuring for any motion approaching 0.61 m/sec (2 ft/sec).

In fact, one of the system's safety features actuated a pressure surge valve at any cabin velocity approaching 0.61 m/sec (2 ft/sec), resulting in an impulse exceeding 1g amplitude and 0.10 second duration. In order to avoid these impulses it was necessary to limit the motion more greatly than had originally been intended. Because of these limitations, and based on motion criteria available at that time, it was judged that no motion effects were to be expected for the longest periods of motion exposure used in the simulations (4 hours) and indeed no major effects were noted. Accordingly, plans were initiated to carry out future simulations on the Office of Naval Research (ONR) motion generator at Goleta, California.

THE ONR MOTION GENERATOR

The ONR motion generator has three DOF (heave, pitch, and roll). The 1358 to 1814 kg (3000 to 4000 lb) cabin is driven along the heave axis by an 8.9-cm (3.5-in.) diameter ram piston and in pitch and roll by two independent piston systems (ref. 6) mounted on the base of the cabin. (See fig. 8.) The general servo system (since modified) is indicated in figure 9 (drawing courtesy of Systems Technology Incorporated). Pressure to drive the ram was developed by a constant displacement, pressure-compensated hydraulic pump operating against a servo valve controlled variable restriction in the drain. The hydraulic servo valve was in turn controlled by a pneumatic transducer. Upward motion was produced by the servo valve closure and the corresponding increase in ram fluid pressure. Downward motion was generated by the cabin's own weight, the rate of fall being controlled by the servo valve and ultimately by back pressure in the drain line.

The pitch and roll servos were identical. They consisted of a constant pressure, variable volume pump providing 190 liters/min (5 gal/min) flow at 11 MPa (1600 psi) pressure. The pump drove a double acting Hana hydraulic cylinder which was in turn controlled by a Moog servo valve. Individual chain driven potentiometers provided the analog voltages corresponding to the respective displacements of the 3 DOF.

The original version of the motion generator suffered from several deficiencies with respect to our desired simulation. The output response was linear only to approximately 0.35g and demands for more acceleration resulted in greater lag through the system and an ever-increasing disparity between the phase of the heave motion and the phases of the pitch and roll motion. Structural resonances were present in both the pitch and roll axes resulting in cross-coupling between the heave motion and the pitch and roll motions. The heave motion excited these resonances at about 2.2 to 2.6 Hz depending on the weight of the cabin (ref. 7). Finally, a stiction-like motion was present which resulted in a deadband or region of insensitivity to drive commands whenever the heave motion crossed through zero velocity. The minimum sinusoidal command to which the heave servo would respond once the system had come to rest was approximately ±0.06g.

The motion generator has since been upgraded in two series of modifications. The first series of modifications occurred prior to the first two rounds of testing at Goleta (Phases I and IA), and consisted of the addition of phase compensations to match the pitch and roll servo control response to the heave response and the addition of a further compensation network to flatten the heave servo response.

Prior to the initial modification (ref. 6), the transfer function (ratio of angular rate command to angular rate realized) for pitch and roll could be approximated by

$$H_{po}(S) \simeq [(1+\tau_{p1}S)(1+\frac{2\zeta_p S}{\omega_p} + \frac{S^2}{\omega_p^2})2]-1$$

$$\simeq [1+\tau_{p1}S]^{-1} \text{ for } \omega<\omega_p$$

where τ_{p1} = 0.49 seconds

ζ_p = 0.35

ω_p = 15.7 radian second^{-1}

and the heave transfer function (ratio of acceleration commanded to acceleration realized) by

$$H_{Ho}(S) \simeq [(1+\frac{2\zeta_H S}{\omega_H} + \frac{S^2}{\omega_H^2})(1+\tau_1 S)(1+\tau_3 S)]^{-1}$$

$$\simeq e^{-\tau S}(1+\frac{2\zeta_H S}{\omega_H} + \frac{S^2}{\omega_H^2})^{-1} \text{ for } \omega<\tau_1^{-1}$$

where ζ_H = 0.707

ω_H = 2.5 radian second^{-1}

τ_1 = 0.08 second

τ_3 = 0.06 second

τ = 0.14 second

After addition of the compensator networks, the pitch transfer function became

$$H_p(S) \simeq \left[(1+\tau_{cp}S)\left(1+\frac{2\zeta_p S}{\omega_p}+\frac{S^2}{\omega_p^2}\right)\left(1+\frac{2\zeta_c S}{\omega_c^2}+\frac{S^2}{\omega_c^2}\right)\right]^{-1}$$

where $S_c = 0.707$

$\omega_c = 12.6$ radian second^{-1}

$\tau_{cp} = 3.2$ second

and the heave transfer function became

$$H_H(S) \simeq \left[\left(1+\frac{2\zeta_o S}{\omega_o}+\frac{S^2}{\omega_o^2}\right)(1+\tau_3 S)(1+\tau_{cH}S)\right]^{-1}$$

where $\zeta_o = 0.707$

$\omega_o = 0.31$ radian-second^{-1}

$\tau_{cH} = 0.019$ second

The second series of modifications occurred following the Phase I and IA simulations. The pitch and roll servos were modified by the addition of non-linear feedback networks to suppress the effects of structural resonances in pitch and roll. The compensation networks were also modified by changing the break point of the second order filter from 2 to 4 Hz. The heave servo was modified extensively. The capacity of the main hydraulic reservoir was increased from 1041 to 3785 liters (275 to 1000 gal), the flow capacity from 1041 to 2271 liters/min (275 to 600 gal/min), the hydraulic pressure capability from 4.5 to 6.9 MPa (650 to 1000 psi), and the capacity of the heave drive pump from 56 to 149 kW (75 to 200 hp) by substitution of two pumps operating in parallel. The electropneumatic command transducer was replaced by a hydraulic controller and the servo control was modified to include both a pressure and position feedback as indicated schematically in figure 10.

These changes have resulted in a significant increase in the performance capability of the system as indicated in figures 11 and 12 and further summarized in table 4. The non-linear feedback network has reduced cross-coupling to a rather negligible value and the system coherence* has been improved from about 0.6 to 0.98 (the latter value corresponds to 2 percent harmonic distortion). Finally, the deadband has been decreased from ±0.06 to ±0.04g.

The primary results of the artifacts present in the pre-modification motion were to limit the magnitude of acceleration peaks to 0.6 instead of 1.0g and to introduce an unwanted high frequency component into the motion. These effects manifested themselves as a modification to the commanded amplitude distribution as indicated in figure 13. These effects are further indicated in figure 14 which compares the acceleration spectra for the output and commanded motions corresponding for the simulated 80 knot/sea state 3 running condition.

GOLETA SIMULATIONS

Despite the limitations inherent in the pre-modification simulator, it was possible to obtain certain tests on the partially upgraded simulator; accordingly, two rounds of testing (ref. 8) were initiated in August 1974 (Phase I) and October (Phase IA). The cabin used in these tests and the general layout of the test battery are indicated in figure 15. Testing was continued in a manner analogous to the MSFC tests.

During the August 1974, series four volunteer crewmen were used in two crew pairs. Each crew pair was subjected to an identical series of exposures, commencing with 30 minutes in each of three conditions (0.154, 0.238, 0.25g RMS across a frequency band of approximately 0.1 to 2 Hz.) The series culminated one 4 hour ensemble of the above conditions and one 3.5 hour continuous exposure to 0.25g RMS. While three out of four subjects suffered from motion sickness when first exposed to 0.25g only one did so during the 3.5 hour exposure. This fact together with other generally encouraging results led to a decision to expand the series to 48 hr exposure periods to be run during October 1974.

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*The system coherence, ρ^2, is defined for each DOF in terms of mean square power associated with that DOF and the degree of correlation between the commanded motion and the resultant motion as

$$\rho^2 = \frac{\text{total power - uncorrelated component}}{\text{total power}}$$

Once again, 4 subjects were used and both crew pairs commenced with 48 hr at 0.154g RMS. For their second period of exposure, the first crew received 0.12g RMS (approximately 50% of the 0.25 case used in August) while the second received 0.18g RMS. The results of these tasks are still under evaluation but are expected to be issued shortly in a consolidated report from the various participating members of the simulation team.* The results that have been reduced were encouraging. However, confirmation of the trends indicated is required and, therefore, another round of testing (Phase II) commenced on 7 July 1975 using the fully modified simulator. The testing pattern will be basically the same as in Phase I and IA with the inclusion of a few new tasks and slight variations on some of the previous ones.

ON THE APPLICATION OF RIDE CRITERIA TO BROAD BAND MOTION

Although the immediate concern of this project is to gain first hand experience with predicted SES motion, it is highly desirable that a procedure be established for treating broad band motion in a general way. As a first step in achieving this goal, it is necessary to develop a method for establishing the equivalency of motion conditions with equal RMS value but different amplitude distributions. Jex and Allen (ref. 10) have indicated some of the problems involved in establishing this equivalency.

The importance of this issue centers on the effects of intermittent large amplitude accelerations and the degree of interaction or cross-coupling between motion effects resulting from different regions of the motion spectrum. As an example, one might evaluate the effects of the motion depicted in figure 4 against a particular motion criteria by considering the RMS spectra in any one-third octave spectral band and comparing it to the motion criteria for each corresponding one-third octave band. The motion could be judged acceptable or not depending on whether the motion of any given band exceeded the motion criteria for that band. This amounts to neglecting any interaction between the effects associated with other bands.

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*Members of the motion simulation team include personnel from
- PMS304 - Surface Effect Ship Project
- NAMRLD - Naval Aerospace Medical Research Laboratory Detachment
- SESTF - Surface Effect Ship Test Facility
- NASA - Science and Engineering Division (MSFC simulation only)
- NSRDC - Naval Research and Development Center
- ONR - Office of Naval Research (Goleta simulations only)
- STI - Systems Technology Incorporated
- HFR - Human Factors Research Incorporated (Goleta simulations only)

The other extreme is to consider coupling between all bands as in the following example* which weights the power spectral density of acceleratic against the square of the allowable heave acceleration, $\ddot{Z}_L^2(f)$:

$$\int_{f_1}^{f_2} \frac{\phi(f)df}{\ddot{Z}_L^2(f)} \simeq \sum_{i=1}^{N} W_i^2(\tfrac{1}{3})A_i^2(\tfrac{1}{3}) \leq C_j$$

where

$W_i(\frac{1}{3})$ = weighting function evaluated at the one-third octave center frequency

$W_i^2(\frac{1}{3}) \simeq \ddot{Z}_L^{-2}(f_{ci})$

C_j = a constant, usually less than 1.

Such a criteria may well be overly stringent. Consider the case where $A_i(1/3)$ is approximately zero except in two frequency bands, where

$$A_i(\tfrac{1}{3}) = 0.71\ \ddot{Z}_L(f_{ci})$$

The above form of evaluation (with C_j=1) would indicate the motion exposure to be unacceptable despite the fact that the acceleration in both bands is 30 percent less than allowed with narrow band data.

*An alternate interpretation of this evaluation rule is that it takes into account the additive nature of the motion and makes allowance for the extremely large amplitudes that could result if all of the low amplitude spectra were momentarily to add constructively.

The lack of well-defined physiological criteria and such apparent inadequacies of existing evaluation criteria for broad band motion as well as the desire to have a single number which evaluates or scores a given motion condition have led to experimentation with a variety of "figures of merit" (FOM) in trying to predict the effects of motion exposure in advance. The tendency to date has been to develop these FOM in terms of two spectral regions which bound the motion region of primary interest to the SES (the spectral range from about 0.1 to 10 Hz). The first range (about 0.1 to 0.8 Hz) involves primarily kinetosis while the second range (about 0.8 to 10 Hz) involves primarily what is referred to in the context of the SES as mechanical interference, (or more commonly in general as "the whole-body motion regime" (ref. 11).

The latter region is the most well studied and SES efforts with the FOM approach have consisted primarily of evaluating our own motion exposure results in terms of existing and proposed single frequency motion criteria and those methods proposed by various organizations for extension of this criteria to broad band motion.

In the kinetosis region, evaluation has proceeded in much the same way with the primary effort directed toward the extension of the work of O'Hanlon, et al. (ref. 12). This group has been working for some time under the sponsorship of ONR on an empirical model of motion sickness incidence. The result of their work is indicated in figure 16 (drawing courtesy of Human Factors Research, Inc.) which graphs the motion sickness incidence (MSI)*, as a function of the RMS acceleration when exposed to single frequency sinusoidal motion. These data give a good fit to a log-normal cumulative distribution:

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*The representation of MSI describes the cumulative percentage of frank emesis expected from young unadapted adult males within two hours after initial exposure to motion. More recent but preliminary work at HFR presents a dynamic model of MSI in terms of the asymptotic proportion of sick individuals, P_A, and the time dependent proportion, P_T, as

$$MSI = 100\ P_A P_T$$

where $$P_j = \frac{1}{\sigma_j \sqrt{2\pi}} \int_{-\infty}^{X_j} \exp\left[\frac{-(X-\mu_j)^2}{2\sigma_j^2}\right] dX$$

$j = A, T$

$\mu_A = -0.80 + 2.73\ (\log_{10} f + 0.77)^2$

$\mu_\beta = 2.00 - P_A$ $\qquad X_A$ = common logarithm of acceleration (RMS g's)

$\sigma_A = 0.46$ $\qquad X_T$ = common logarithm of time (minutes)

$\sigma_T = 0.36$

$$MSI = \frac{100}{\sigma\sqrt{2\pi}} \int_{-\infty}^{X} \exp\left[\frac{-(X-\mu)^2}{2\sigma^2}\right] dX$$

where $X = \log_{10} 0.901\ a_{RMS}$

$\sigma = 0.43$

$\mu = 1.032 + 5.132 \log_{10}f + 3.562(\log_{10}f)^2$

a_{RMS} = acceleration (RMS g's)

from which it can be determined that the curves of constant MSI (fig. 17) have a maximum at a frequency of 0.190 Hz.

These curves in turn have been normalized by J. George and H. Donnelly at the Applied Physics Laboratory, Johns Hopkins University, to form the single weighting function depicted in figure 18.

If now a weighted acceleration, a_w, is formed from this weighting function according to:

$$a_w = \left\{\sum_{i=1}^{N} [W_i(f_{ci})A_i(\tfrac{1}{3})]^2\right\}^{\frac{1}{2}}$$

and substituted for a_{RMS} in O'Hanlon's MSI, a FOM for kinetosis, D_k, can be developed which gives an intuitive feeling for the quality of the motion. The use of D_k as a rating of one motion condition relative to another seems quite justified; however, it is to be emphasized that D_k is not to be given a quantitative interpretation since insufficient data have been taken for such an assessment (the HFR group tested almost 600 subjects to develop their single frequency data) and also because the data do not adequately take into account the process of adaptation. The adaptation to sea motion is an accepted fact and preliminary work by the same group at HFR has noted definite trends in this process as a function of the amplitude and time of exposure. The significance of this fact is that a designer of passenger ships, which normally carry unadapted passengers, might strive to achieve a very small value of D_k while a designer of a military ship which carries only adapted personnel might find the effects of higher frequency motion (which might, for instance, interfere with operation of electronic equipment) to be much more important. This situation is depicted in figure 19 which compares iso-kinetosis curves to hypothetical fatigue criteria developed by fitting contours of equal sensations (ref. 13) to curves of Fatigue Decreased Proficiency (ref. 11). It is apparent that a criterion such as D_k which is based strictly on kinetosis in unadapted males might be impractical for a military ship.

It is hoped that continuing work will help to clarify some of these issues. In the meantime, the newly modified ONR motion generator represents a significant new capability for investigating these and other effects of motion exposure.

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TABLE 1.- SUMMARY OF THE MOTION STATISTICS PREDICTED TO OCCUR AT THE CENTER OF GRAVITY OF A 2000-TON SES (CONFIGURATION A) OPERATING WITHOUT RIDE CONTROL

[Data describe the acceleration, a, expected frequency, f_e, number of maxima per unit time, N_1, and broadness factor, ε]

Tape No.	Speed/Sea State	a (RMSg)	f_e (Hz)	N_1 (second^{-1})	ε
JR21	80/3	0.194	0.88	1.56	0.82
JR19	60/4	0.248	0.78	1.27	0.79
JR12	40/5	0.278	0.72	1.16	0.79

TABLE 2.- COMPARISON OF THE RMS ACCELERATION (g) IN HEAVE, SURGE, AND SWAY AT A POSITION 23.5 m (77 ft) FORWARD OF THE CENTER OF GRAVITY FOR A 2000-TON SES (CONFIGURATION A) WITH RIDE CONTROL IN A BOW SEA

Sea State/Speed	5/40	4/60	3/80
Component			
Surge	0.036	0.016	0.01
Sway	0.033	0.02	0.01
Heave	0.24	0.14	0.09

TABLE 3.- SUMMARY OF TASKS AND TESTS

PHASES	NAME	ACTIVITY (C = Primarily Cognitive; M = Primarily Motor)		SCORING MEASUREMENTS
I, IA II	Navigation	C	Plotting own ship's and radar target positions and courses from verbal information	Fraction of radar contacts not plotted
I, IA II	Cryptography	C	Manual decoding of written messages	Time to completion or fraction of message decoded at mandatory termination
II	Radar Task I	C/M	Monitor PPI radar detect incoming missile and provide discrete motor response	Fraction of targets missed; fraction of targets in error
II	Radar Task II	C/M	Monitor PPI radar, detect collision hazards and provide discrete motor response	Fraction of targets missed; fraction of targets in error
IA, II	Visual Acuity	C	Read optometric near-point and far-point material	Acuity levels, list reading
II	Dual Axis Weapon Tracking	M	Maintain control over simulated weapon flight by initiating commands via two axis electrical joy stick	Vertical control signal; vertical display error Horizontal control signal; horizontal display error Zero crossings for all of above
I, IA II	ECM Tracking	M	Antijam Frequency Meter tracking, MK VIII first-order autopaced critical task, dial display, unrestrained knob control	Critical instability score (median of 3 trials)
I, IA II	Equipment Handling	M	Take 59 kg (13 lb) case from rack and reinstall in rack; perform in both sitting, standing positions	Time to completion (table to table) and subjective rating
I, IA II	Fine-Motor Lock	M	Combination lock opening with one hand	Time to completion
IA II	Keyboard Operation	C/M	Calculating own ship's course and speed from timed samples of position using mini-calculator	Fraction of incorrect results and time to completion
II	Maintenance Task	M	Strip typical electro-mechanical circuit board using standard tools	Time to complete; number of components damaged during removal
I, IA II	Questionnaires	C/M	Complete selected sections of questionnaires when directed	Subjective rating
I	Eating, Drinking	M	Eating sandwiches, drinking milk, cola	Subjective rating
IA II	Complete Housekeeping	C/M	Food preparation, cleanup, personal hygiene, sleeping, R & R	Subjective rating
II	Head Motion Measurement	-	Using head mounted 6-DOF accelerometer package measure head motion	Correlation of head motion with commanded motion and with other motion effects
I, IA II	Stress Hormone Analysis	-	Regular, periodic urine sampling and analysis for stress hormones	Levels of stress hormones present at periods throughout simulation
II	Blood Pressure and Oral Temperatures	M	Interactive/self administered checks of B.P. & body temperatures	Regular record plot to show any unusual trends
IA II	Sleep Data Measurement Analysis	-	Automatic collection of EES EMG data whenever crewmen are at rest or sleeping	Comparison of sleep performance control/motion conditions by hand scoring and computer scoring techniques

TABLE 4.- PERFORMANCE CAPABILITY OF ONR MOTION GENERATOR

PRE- AND POST-MODIFICATION

	Units	Unmodified	Modified
Heave Performance			
a. Amplitude	m (ft)	±3.4 (±11)	±3.1 (±10)
b. Velocity	$m\text{-}sec^{-1}$ ($ft\text{-}sec^{-1}$)	±2.4 (± 8)	±5.5 (±18)
c. Acceleration	g's	±0.6	±1.2 ±0.9
d. Compensated Bandwidth (3db)	Hz	0.5 to 5	-
e. Linearity, accelerator		0.6 to 0.7	±2db to 5 Hz
f. Coherency		0.6 to 0.7	0.98
g. Deadband	g	0.06	±0.04
Pitch and Roll			
a. Amplitude	deg	±15	±15
b. Velocity	$deg\text{-}sec^{-1}$	±25	±25
c. Acceleration	$deg\text{-}sec^{-2}$	±180	±180
d. Compensated Bandwidth (3db)	Hz	0.06 to 2	0.06 to 2
e. Phase Matching to Heave	deg	see test	<36°
f. Coherency	-	0.96	0.96

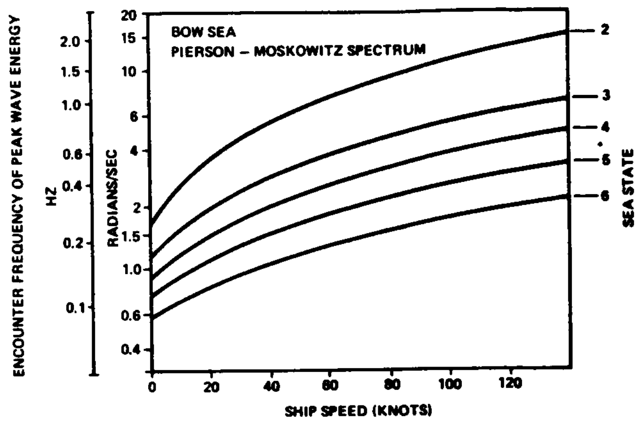

Figure 1. Encounter frequency of peak wave energy as a function of ship speed and sea state for a bow sea and a Pierson-Moskowitz sea spectrum.

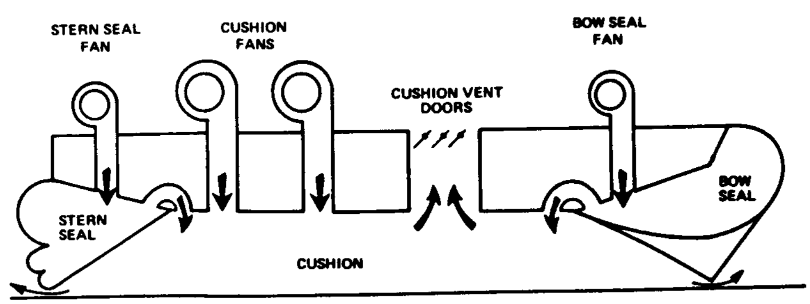

Figure 2. The nature of SES Lift and Ride Control System elements.

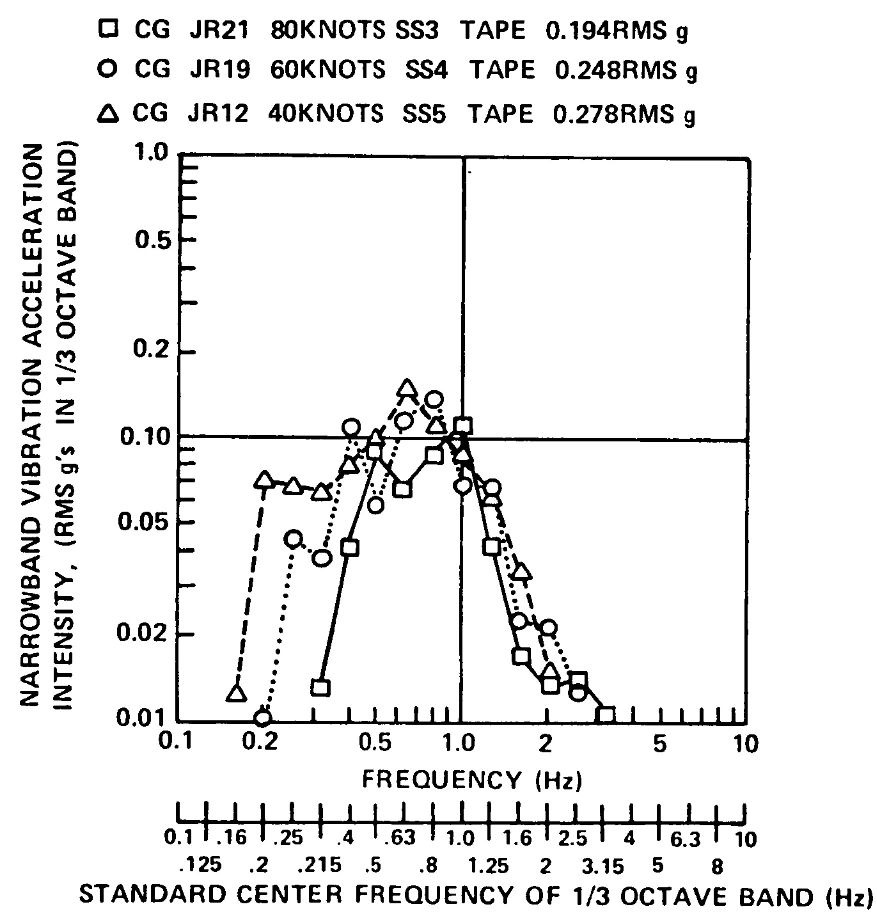

Figure 3. Mathematical prediction of acceleration intensity at the center of gravity of a generic SES (Configuration A) without Ride Control.

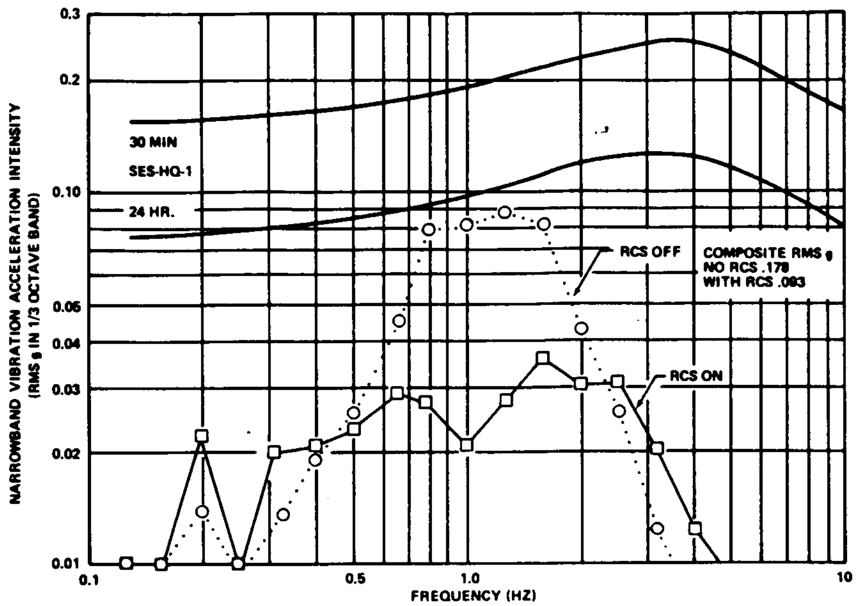

Figure 4. Math model predictions of the ride quality of a more recent SES design with and without Ride Control System activated. Condition is 65 knots and sea state 4.

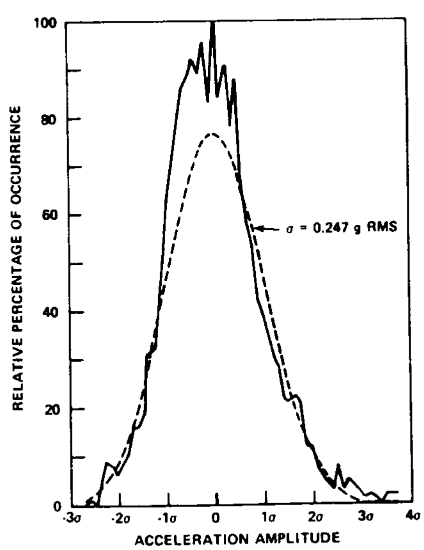

Figure 5. The acceleration amplitude distribution for Configuration A at 60 knots in Sea State 4.

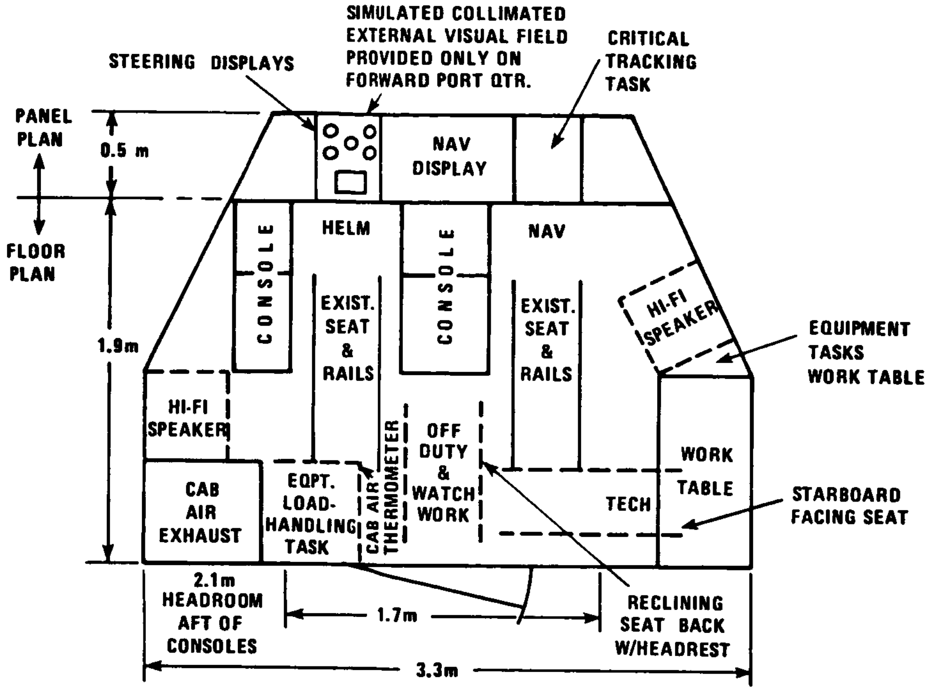

Figure 6. The configuration of the SES cabin during the MSFC simulation; Helm, Nav, Tech, and Off Duty indicates positions of subjects during motion exposure.

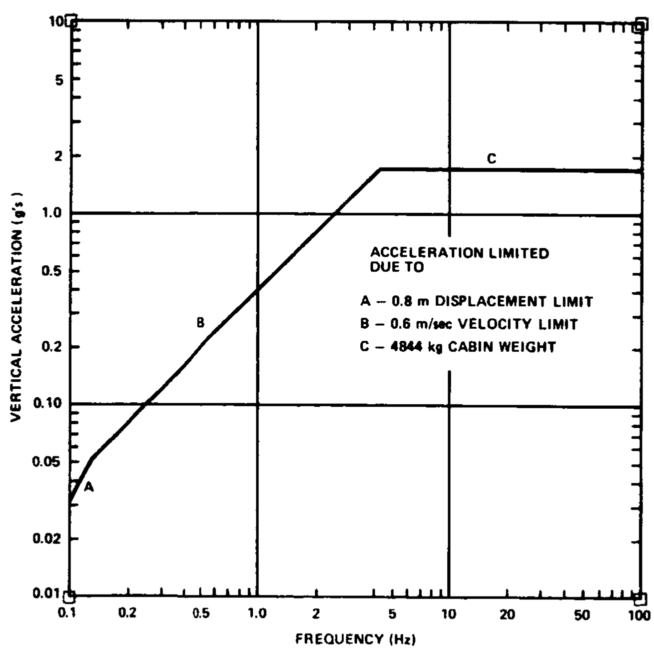

Figure 7. Performance of MSFC motion generator in vertical acceleration for sinusoidal input.

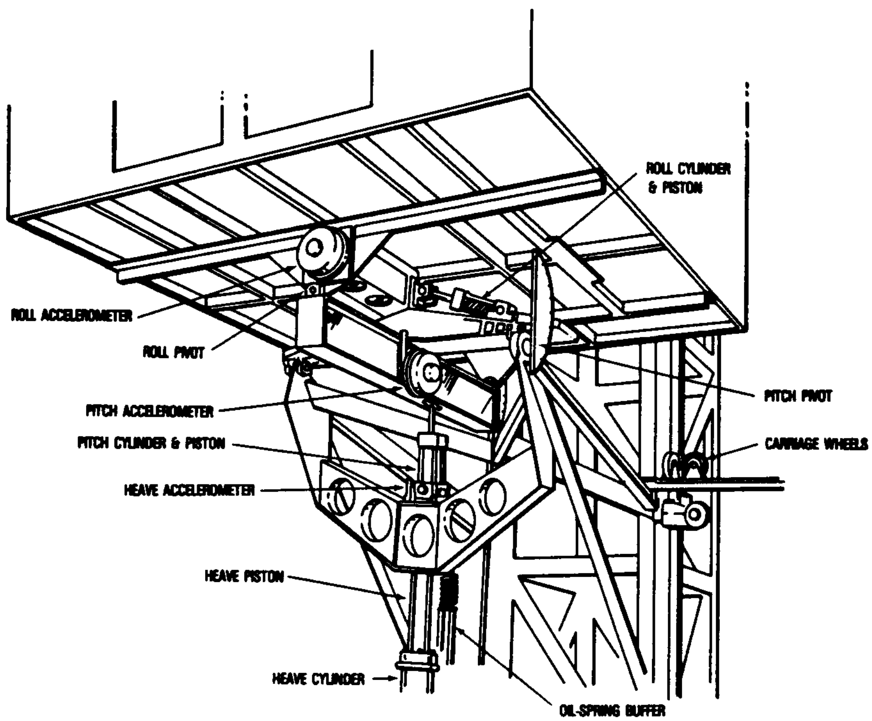

Figure 8. The moving carriage, gimbal, and associated structures of the ONR Motion Generator.

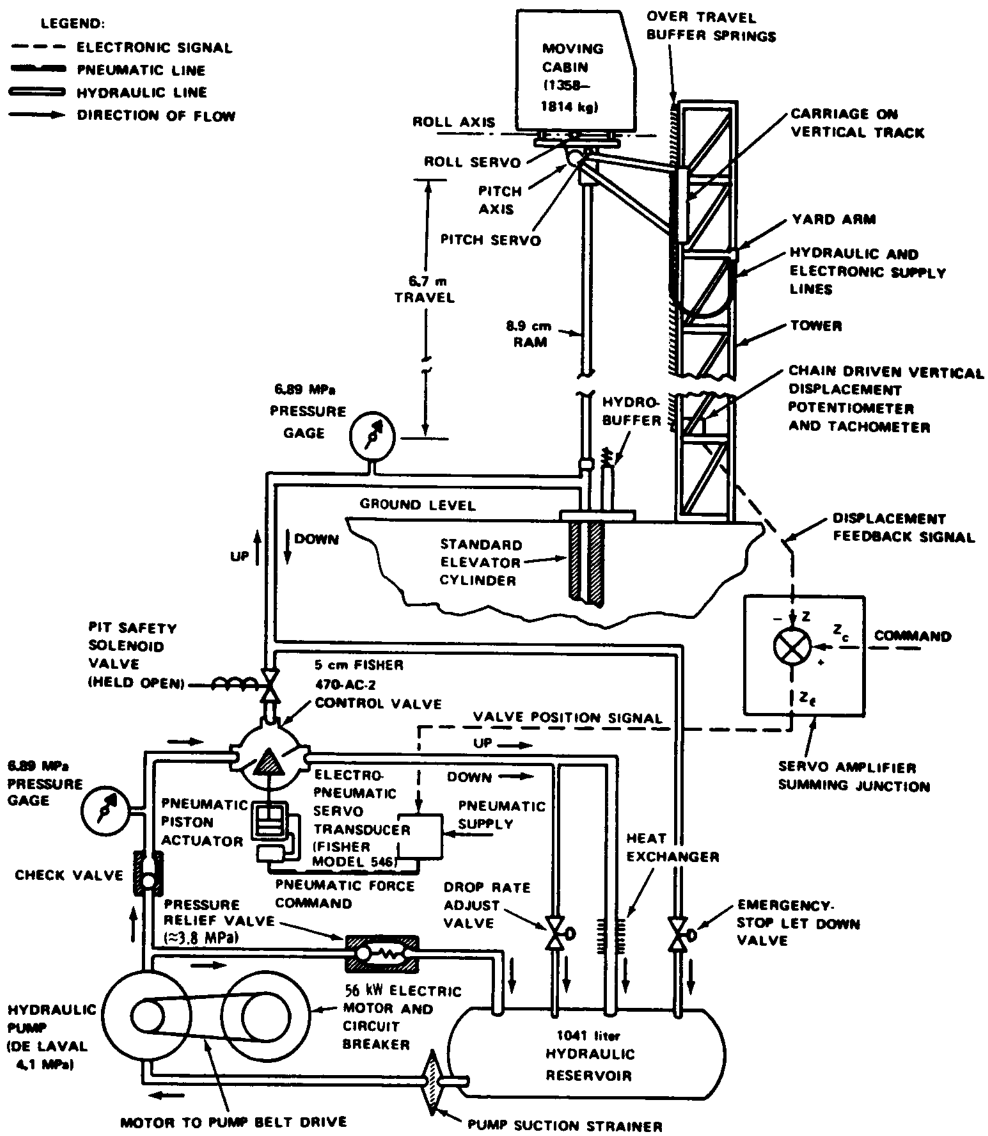

Figure 9. ONR Motion Generator heave drive hydraulic system functional schematic.

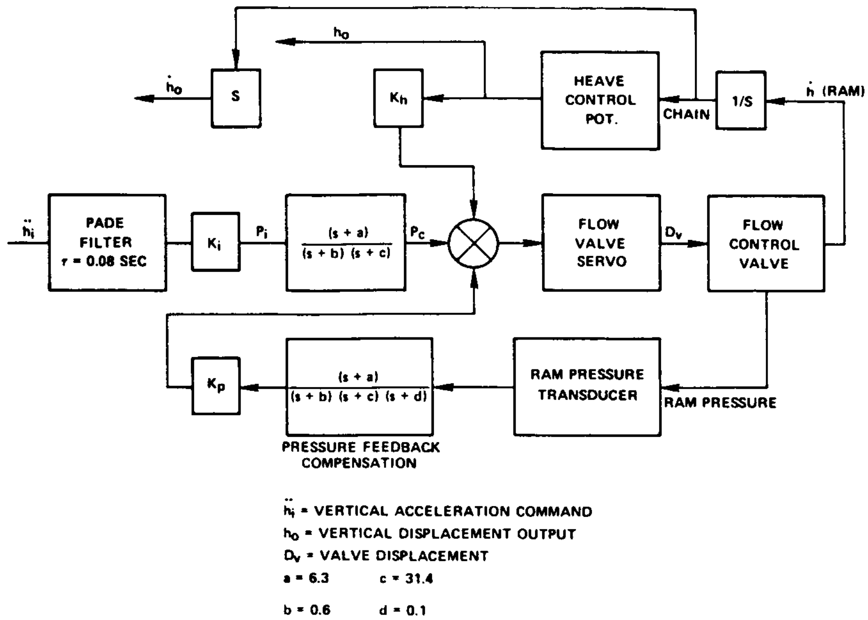

Figure 10. Control loop for modified heave servo system.

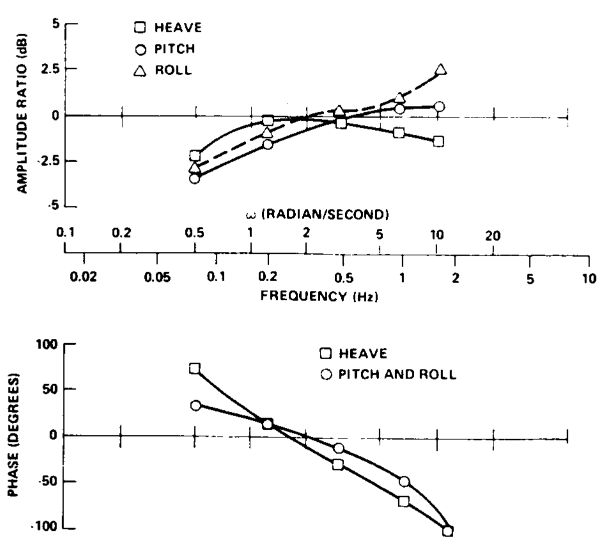

Figure 11. Bode comparison of angular rate and heave response.

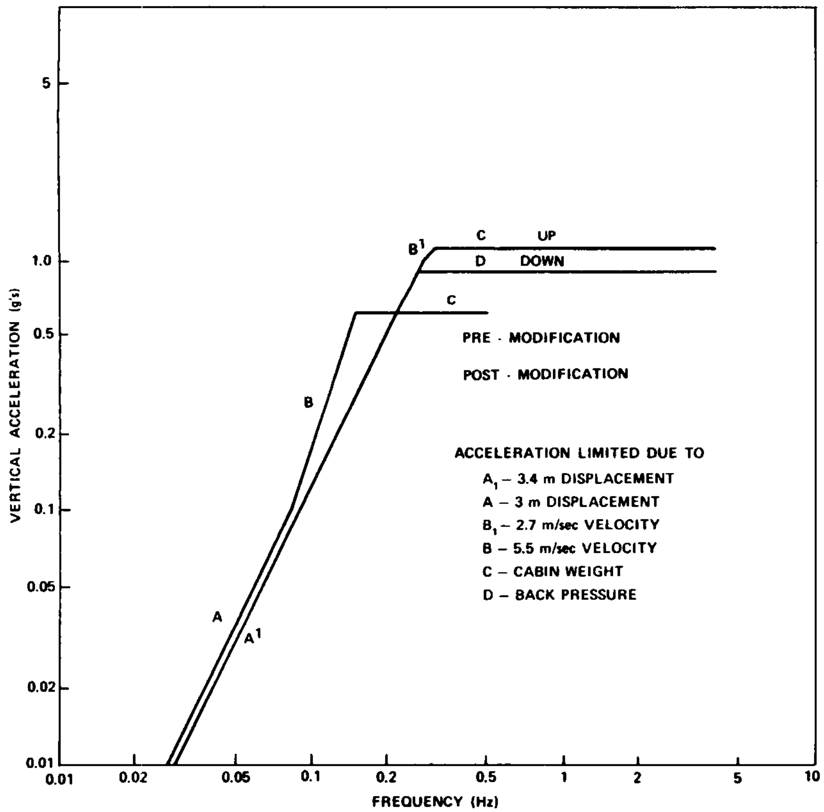

Figure 12. Performance of the ONR Motion Generator pre- and post-modification with 1724 kg (3800 lb) cabin.

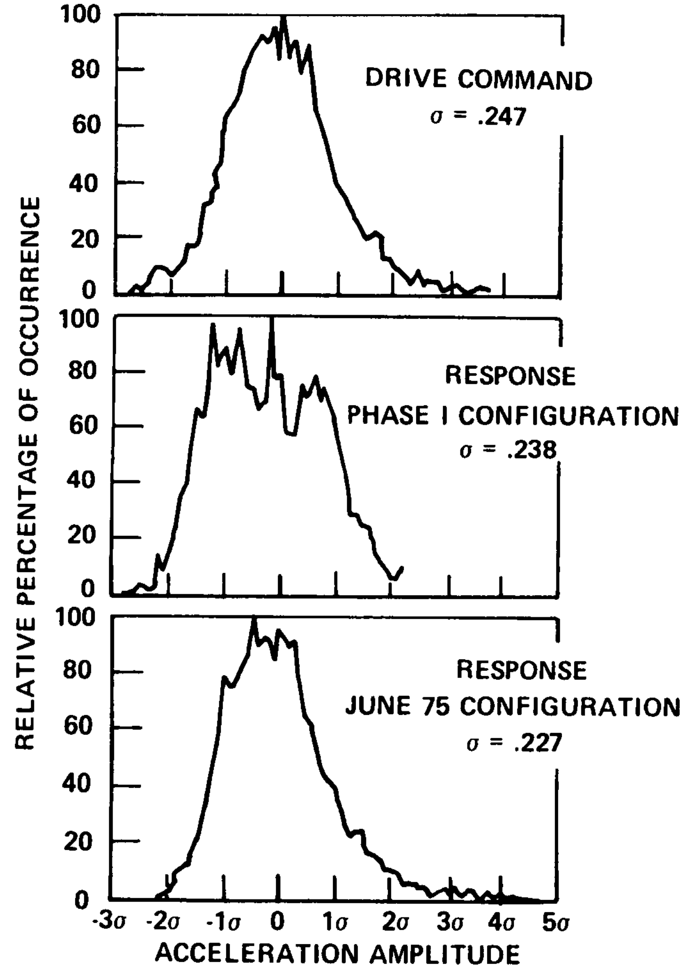

Figure 13. Alteration of the acceleration amplitude distribution by the ONR motion generator for 60 knot/ sea state 4 case.

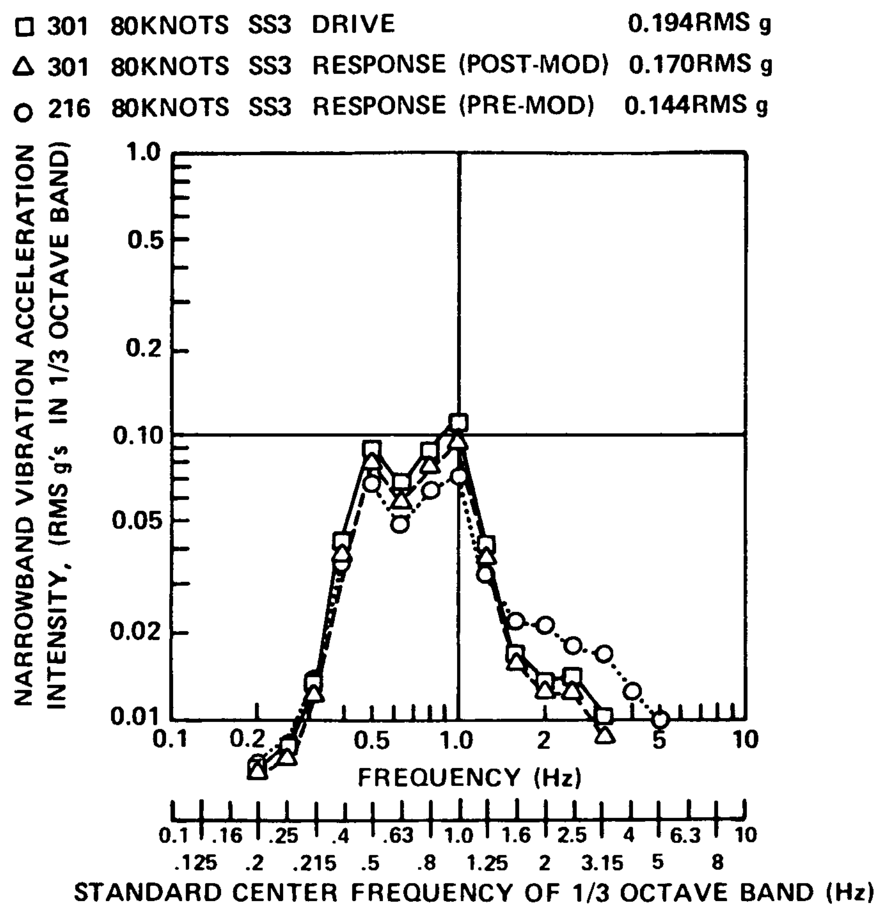

Figure 14. Pre- and post modifications response of ONR Motion Generator to 80 knot/sea state 3 motions.

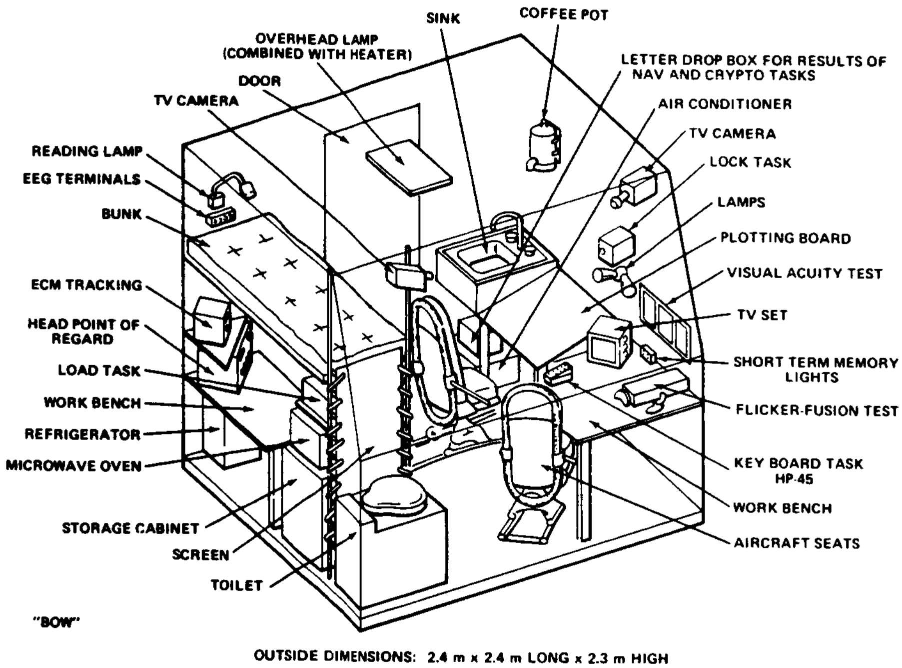

Figure 15. Large cabin for Phase IA.

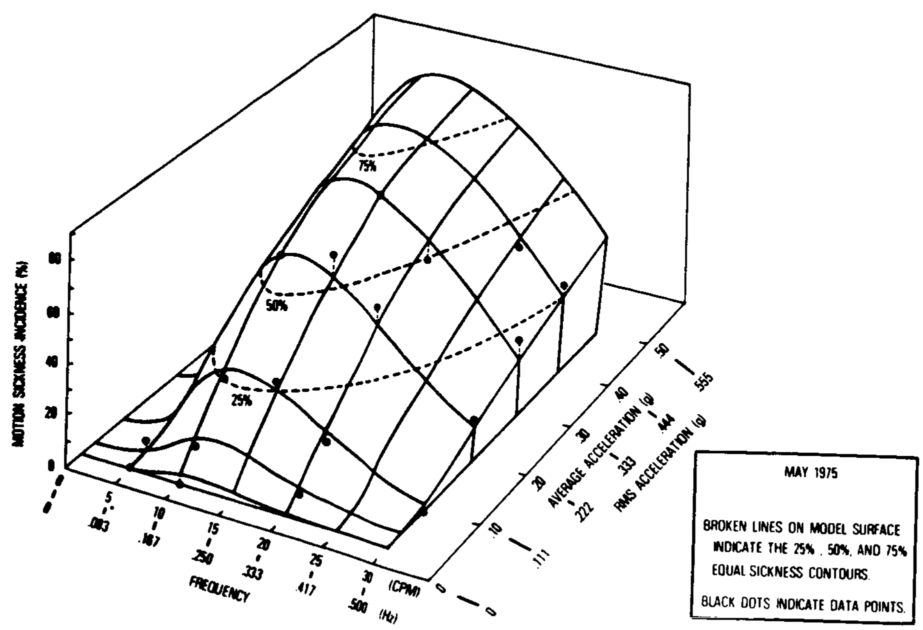

Figure 16. Empirically derived model of motion sickness incidence.

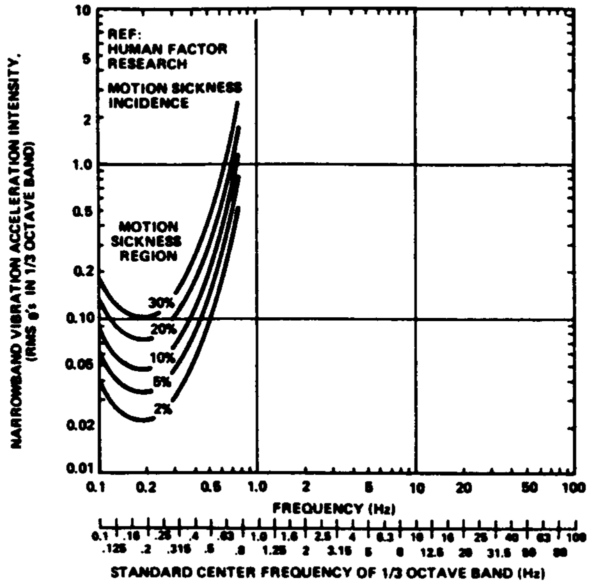

Figure 17. Curves of constant MSI as a function of frequency.

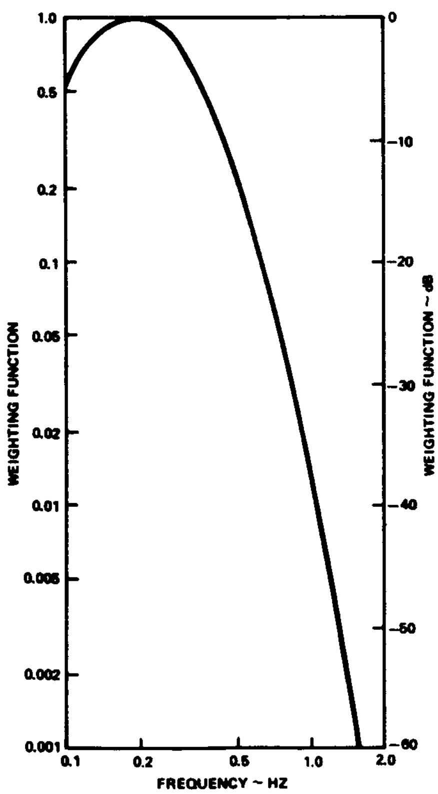

Figure 18. Broadband acceleration weighting function for kinetosis.

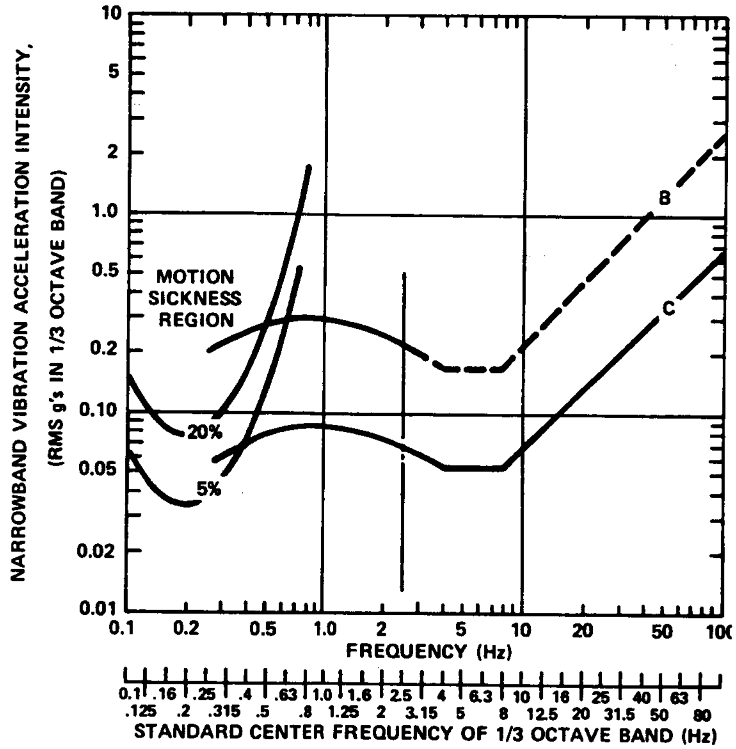

Figure 19. A hypothetical criteria for fatigue developed by fitting an Equal Sensation Contour to a Fatigue Decreased Proficiency curve at 2.5 Hz.

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EVALUATION OF RIDE QUALITY MEASUREMENT PROCEDURES BY SUBJECTIVE EXPERIMENTS USING SIMULATORS*

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Summary

For the purposes of vehicle design and procurement, well-defined procedures are needed for measuring ride quality. A number of more or less different Ride Quality Measurement Procedures (RQMP's) have been proposed and/or used in the past, e.g., ISO, ISO alternate, or Shaevitz exceedance counts.

Since ride quality is, by definition, a matter of passenger response, there is need for a Qualification Procedure (QP) for establishing the degree to which any particular RQMP does correlate with passenger responses. Once established, such a QP will provide very useful guidance for optimal adjustment of the various parameters which any given RQMP contains.

The present paper proposes a QP based on use of a ride motion simulator and on test subject responses to recordings of actual vehicle motions. Test subject responses are used to determine simulator gain settings for the individual recordings such as to make all of the simulated rides equally uncomfortable to the test subjects. Simulator platform accelerations vs. time are recorded with each ride at its equal discomfort gain setting. The equal discomfort platform acceleration recordings are then digitized. A computer is used to apply a prospective RQMP to each of the equally uncomfortable simulator motions and to determine the scatter among the ride index values which the RQMP assigns to these motions. The best RQMP will be taken to the one for which the scatter is smallest.

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This program has been carried out on a pilot basis using the Passenger Ride Quality Apparatus at NASA Langley Research Center, using recordings of 19 passenger railcar ride motions (vertical and lateral), and working with subjective responses from a panel of four subjects.

The present paper includes a discussion of various RQMP's which are available, a description of the experimental procedure, and preliminary results illustrating the extent to which several particular RQMP's deviate from ideal correlation with passenger response.

1. The Role of Ride Motion Measurement in Vehicle Specifications

This article is motivated to a large extent by the needs of the engineer who is responsible for drawing up specifications for railroad or rail-transit cars and who seeks to insure that the cars will "ride" well.

The engineer can use either or both of the following two basic approaches:

1) he can set forth a prescription for measuring the ride motion of the new cars at stated speeds on stated trackage and require that the measured motion not exceed stated limits, or

2) he can rely on analysis and/or experience as a basis for requiring that the new car suspension incorporate specific ride quality related features he believes will help to secure a satisfactory ride.

One weakness of the second approach is that it limits the manufacturer's control over running gear design and may reduce the likelihood that the manufacturer can be held responsible for the ride quality consequences of the many other features which he himself must contribute to the suspension. Thus, for specifications on which there is to be competitive bidding, the engineer is likely to be more interested in specifying upper limits for measured motion of the resulting ride than in specifying details of suspension design.

A satisfactory specification of the manner in which the ride motion of a new car is to be tested must include a prescription for converting the vehicle's actual ride motion (e.g. vertical, lateral, and longitudinal acceleration as functions of time) into a number (or set of numbers) which can serve as a "measure" of the amount of motion as far as ride quality is concerned. A quantitative prescription of this type will be referred to as a measure of ride motion, or simply as a <u>ride measure</u>.

Section 2 below reviews the nature of the empirical data on human sensitivity to some particular motions. Section 3 reviews some ride measures which are available. Section 4 proposes a method for characterizing the extent to which any given ride measure represents discomfort as it is actually perceived by passengers. Section 5 describes an experimental procedure for obtaining the necessary empirical data. Section 6 describes some recent experimental work using the ride motion simulator at NASA Langley Research Center. Section 7 presents results of a preliminary evaluation of several ride measures.

2. Data on Passenger Sensitivity to Specific Motions

A number of investigators have published results of empirical studies on human sensitivity to sinusoidal motion and a few workers have reported on sensitivity to vibratory motion composed of randomly varying contributions having frequencies within a narrow band about a nominal central frequency. (See for example ref. 1.) The results of these studies are normally expressed via contours of vibration amplitude as a function of frequency with the contours being drawn so that the discomfort experienced by the average test subject is constant along any one contour. The contours are sometimes approximated via straight lin segments for ease of representation.

It will be convenient to have a name for referring to these contours. While the term isocomfort has been used, we will refer to each contour of equal discomfort as an isobother. Empiri research will presumably reveal that isobothers which differ in discomfort also show some variation in shape, analogous to that of the Fletcher-Munson curves for aural sensitivity. However, we will ignore such dependence and will denote the r.m.s. amplitude of the acceleration as a function of frequency along an isobother simply as $I(f)$.

The main appeal of sinusoidal motions is that the number of distinct sinusoidal motions (e.g., distinct combinations of frequency and amplitude) which are likely to be important in a given passenger environment is only about 300 (20 one-third octaves from 0.5 to 50 Hz , 5 amplitude levels for each one-third octave, and three directions of motion). This makes it practical to gather empirical data which will cover any sinusoidal motion which might be encountered.

When attention is turned to motions of a more general character, it becomes difficult even to find a way to ennumerating a set of distinct representative motions, and if a comprehensive ennumeration could be devised, testing of all of the representative would be a staggering task. On account of the foregoing, more general motions are not approached with the assumption that all possible types can be ennumerated. Instead, they are approached with the assumption that it will be possible to devise quantitative prescriptions (ride measures) for converting recorded acceleration histories directly into numerical measures of discomfort.

3. Examples of Ride Quality Measurement Procedures

The term ride measure was introduced at the end of Section 1 as a means of referring to a prescription for converting a record of acceleration as a function of time into a number which is intended to be a measure of the discomfort produced by the corresponding motion. The present section discusses a few examples of specific ride measures which have been formulated in the past and a few ways in which they can be generalized.

A. Exceedance Count Measures

These measures are based on counting the number of times that the acceleration crosses each of several acceleration thresholds. Prior to the development of modern electronic equipment, it was a standard railroad practice to have the acceleration recorded in an approximate manner on a strip chart by pens actuated mechanically by suspended masses. The thresholds were represented by grid lines printed on the charts, and the number of times that the signal crossed a grid line was counted by hand. With modern instrumentation, these functions can be accomplished electronically, and at least one firm (Schaevitz Engineering Co.) has marketed a ride recording instrument package set up on this basis.

If there is a need to determine which of two given rides is to be considered the more comfortable, and if the exceedance counts are selected as the basic measured data, then a formula must be chosen for converting each set of recorded exceedance counts into a single number which is to be the measure of the corresponding ride motion.

A formula used by the Pennsylvania Railroad to reduce exceedance counts from mechanical recorders was as follows: give each count a weight proportional to the square of the associated acceleration level and form the weighted average number of counts per unit time. Or, expressed in symbols,

$$\text{RMEC3} \propto \frac{\Delta a}{D} \sum_{\ell} \frac{3}{2} a_{\ell}^{2} C_{\ell}$$

where RMEC stands for "Ride Measure- Exceedance Count", the suffix 3 is included in preference to a suffix 2 (the exponent) for reasons which will appear later, D is the duration of the time of counting, a_{ℓ} is the acceleration at the ℓ th threshold, C_{ℓ} is the count for that threshold, the summation is over all of the thresholds, and the factor Δa, which is the spacing between adjacent thresholds, is included so that the whole expression will approach a finite limit if the spacing between thresholds approaches zero. The factor of 3/2 is included for later convenience. The symbol $\propto$ denoting "is proportional to" will be used for the time being, and a specific normalization will be suggested at the end of this section.

Having introduced this measure, we will now explore some of its features.

For conceptual purposes it is convenient to work with the limit in which the spacing between adjacent acceleration thresholds does approach zero. Thus we will use

$$\text{RMEC3} \propto \lim_{\Delta a \to 0} \left[\frac{\Delta a}{D} \sum_{\ell} \frac{3}{2} a_\ell^2 \, C_\ell \right]$$

$$\propto \frac{1}{D} \int_{-\infty}^{\infty} da \, \frac{3}{2} a^2 \, C(a)$$

When it is helpful to be more explicit, we can express the value obtained when this ride measure is applied to the acceleration signal $a(t)$ as

$$\text{RMEC3} \left[a(t) \right] \propto \frac{1}{D} \int_{-\infty}^{\infty} dx \, \frac{3}{2} x^2 \, C_{\left[a(t)\right]} (x)$$

where $C_{[a(t)]}(x)$ is the number of times that the signal $a(t)$ passes the threshold x during the interval D .

The general properties of C are $C(x) \geq 0$ for all x and $C(-\infty) = C(\infty) = 0$.

Applying the foregoing ride measure to a sinusoidal motion,

$$a(t) = A \sin (2\pi ft),$$

one has

$$\text{RMEC3} \left[A \sin (2\pi ft) \right] \propto \frac{1}{D} \int_{-A}^{A} da \, \frac{3}{2} a^2 \quad f D$$

$$\propto \quad f A^3$$

where f and A are respectively the frequency and amplitude of the sinusoidal motion. The fact that the result is proportional to the third power of the amplitude provides the motive for use of the suffix 3 .

As there is likely to be interest in a ride measure which, when applied to a sinusoidal motion, will give a value proportional to $f A^2$, we may note two ways of arriving at such a measure.

From the preceding exercise with RMEC3 it is easy to see that the measure defined by

$$\text{RMEC2} \quad \propto \quad \frac{1}{D} \int_{-\infty}^{\infty} da \, |a| \, C(a)$$

will vary as the square of the amplitude when it is applied to a sinusoid.

Another definition with this feature may be obtained in a somewhat more intuitive manner as follows. Thinking in terms of the sum over discrete levels and using the a^2 values as weights, we want to increment the count for a given level only when that level is the highest (or lowest) one reached by a local peak (or valley) of the wave form. As that idea can be expressed in terms of differences between the counts which have been defined already we can write

$$\frac{1}{D} \left[\sum_{\ell=-\infty}^{-1} \frac{a_\ell^2}{2} (C_\ell - C_{\ell-1}) + \sum_{\ell=1}^{\infty} \frac{a_\ell^2}{2} (C_\ell - C_{\ell+1}) \right]$$

Putting $C_\ell - C_{\ell-1} = \Delta C_\ell$ and going to the limit of zero spacing between levels, this becomes an integral over acceleration, namely

$$\propto \frac{1}{D} \left[\int_{-\infty}^{0} \frac{a^2}{2} \, dC(x) - \int_{0}^{\infty} \frac{a^2}{2} \, dC(x) \right]$$

$$\propto \frac{1}{D} \left[- \int_{-\infty}^{0} da \; a \; C(a) + \int_{0}^{\infty} da \; a \; C(a) \right]$$

$$\propto \frac{1}{D} \int_{-\infty}^{\infty} da \; |a| \; C(a)$$

Thus we find that the two approaches give the same result.

We observe next that various forms of weight function can be tried in order to see which weight functions lead to ride measures which correlate best with passenger judgements. In this vein, let $w(a)$ represent an arbitrary weight function, and denote the corresponding exceedance count ride measure by

$$\text{RMECw} \quad \propto \quad \frac{1}{D} \left[\int_{-\infty}^{0} \frac{w(x)}{2} \, dC - \int_{0}^{\infty} \frac{w(x)}{2} \, dC \right]$$

In the interests of a simple notation, we take it as an axiom that the zero point on the axis of acceleration values is located at the point of minimum discomfort and that we will always have $w(0) = 0$ so that all measures will give the value zero when $a(t) = 0$ for all time t.

Then, integrating by parts, we have

$$\text{RMECw} \propto \frac{1}{D}\left[-\int_{-\infty}^{0} C \frac{w'}{2} \, dx + \int_{0}^{\infty} C \frac{w'}{2} \, dx\right]$$

If from physical symmetry it can be assured that $w(-x) = w(x)$, then the foregoing becomes

$$\propto \frac{1}{D}\int_{0}^{\infty}\left[C(x) + C(-x)\right]\frac{w'(x)}{2} \, dx$$

However, the original expression in terms of dC will usually be the more convenient one.

As an example of the application of the general definiti the value obtained when it is applied to a sinusoid is

$$\text{RMECw}\ [A_0 + A_1 \sin(2\pi ft)]$$

$$\propto \begin{cases} (f/2)[w(A_0 + A_1) + w(A_0 - A_1)], & \text{if } A_0 < A_1 \\ (f/2)[w(A_0 + A_1) - w(A_0 - A_1)], & \text{if } A_0 > A_1 \end{cases}$$

Whereas the above definitions assumed counts based on preset absolute acceleration values, one can also define counts based on thresholds whose locations are dependent on the recent past behavior of the acceleration.

The following is one simple way of obtaining counts based on moving thresholds. Namely, look at the local peaks and local valleys of the acceleration waveform and treat the wave form as a sequence: $a_1, a_2, \ldots, a_n$ where all the odd members are local peaks and the even members are local valleys (or vice versa). Then apply one of the previously described exceedance count ride measures as though the ride consisted of a sequence of unconnected segments:

from $-|a_1-a_2|/2$ to $+|a_1-a_2|/2$

then

from $-|a_2-a_3|/2$ to $+|a_2-a_3|/2$

etc.

An indication of the magnitude of the change in results which will follow from use of moving thresholds may be obtained by applying the formulae given above for:

$$a(t) = A_0 + A_1 \sin(2\pi ft)$$

to the case that $w(x)$ is x^2 or x^3 .

With static thresholds we have

$$\text{RMEC2}\left[A_0 + A_1 \sin(2\pi ft)\right] \propto \begin{cases} f(A_0^2 + A_1^2), & \text{if } A_0 < A_1 \\ f(2A_0A_1), & \text{if } A_0 > A_1 \end{cases}$$

and

$$\text{RMEC3}\left[A_0 + A_1 \sin(2\pi ft)\right] \propto f(A_1^3 + 3A_1A_0^2)$$

The results which apply if the static thresholds are replaced by moving ones are $f A_1{}^2$ and $f A_1{}^3$ respectively. Comparison with the preceding results indicates that the choice of the type of threshold can have a pronounced effect on the results.

While general discussion of the criteria of ride measure validity is reserved for Section 4, one criterion will be introduced here. Namely, if it were to be completely satisfactory, a ride measure ought, among other things, to yield the same value for all points on any one isobother (isobother being the term used in Section 2 to refer to a sinusoidal motion amplitude vs. frequency contour along which the average person judges annoyance to be constant).

In the limit that the acceleration discrimination level spacing tends to zero, any reasonable exceedance count ride measure can be made to satisfy this particular criterion exactly. All that is required is that the acceleration signal pass through a suitably chosen filter prior to counting of the exceedances.

Let $I(f)$ denote the isobother's amplitude as a function of frequency, and let $K(f)$ denote the magnitude of the transfer function of the filter. Then referring to the earlier expression for the value obtained when RMECw is applied to a sinusoid and denoting the even part of w by w_e , we have

$$\text{RMECw}\left[K_w(f)\ I(f)\ \sin(2\pi f\, t)\right] \propto f\, w_e\left[K_w(f)\ I(f)\right]$$

and requiring that this expression have a constant value, B, independent of f , we find

$$w_e\left[K_w(f)\ I(f)\right] = B/f$$

Thus the desired filter characteristic is determined to be

$$K_w(f) = w_e^{-1}(B/f) / I(f)$$

where w_e^{-1} denotes the function inverse to w_e. The inverse will exist because $w_e(x)$ must be monotonically increasing function of x if it is to be physically reasonable.

If the criterion of constant value at all points on any one isobother were assumed to be a sufficient test of ride measur validity, then the preceding consideration would settle the question of the relative manner in which any exceedance count ride measure should treat different frequency components in the signal it receives. However, the foregoing consideration will be regarded here as a motivation for introducing the filter rather than as a basis for deciding what characteristic the filter shoul have.

Stephens (reference 2) has given interesting data characterizing vehicle motions in terms of the maximum value of $a(t)$ in each motion recording. That ride measure can be regarded as a representative of a group of measures which can be written in terms of the function inverse to $C_{[a(t)]}(x)$. Namely, letting $A(c)$ be the acceleration at the largest threshold which is crossed c times by the signal $|a(t)|$, one can write a measure in the form

$$\sum_{c=1}^{\infty} w[A(c)]\ V(c)$$

The specific example used by Stephens has $w[A] = A$, $V(1) = 1$, and $V(i) = 0$ for $i>1$.

B. Exceedance Time Measures

When reliance had to be placed on mechanical means, exceedance counts were used because it was easier to count the number of times that the acceleration crossed each of several thresholds than it was to determine the cumulative time spent above each one of them. However, as the development of electroni has made it easy to determine exceedance times, exceedance time measures have become of interest. United Aircraft Corp. was an early user of this approach.

Let $T_{[a(t)]}(x)$ be the cumulative time during which the acceleration

$$a(t) > x \quad \text{if} \quad x > 0$$

and during which

$$a(t) < x \text{ if } x < 0$$

Then the common exceedance time ride measure may be defined as

$$\text{RMETw} \propto \frac{1}{D} \left[\int_{-\infty}^{0} w(x)\ dT(x) - \int_{0}^{\infty} w(x)\ dT(x) \right]$$

This measure may be more familiar in the guise,

$$\text{RMETw} \propto \frac{1}{D} \int_{0}^{D} dt\ w\left[a(t)\right]$$

The latter form calls attention to the fact that this exceedance time measure is the same as the time average of the corresponding function of the acceleration. It is usually also the more convenient form when $a(t)$ is a mathematical function, such as a sinusoid. (To show the equivalence of the two forms, one may express the second form in terms of a series based on division of the acceleration range into a number of equal sized small segments and then let the segment size tend to zero so that the series becomes an integral over the acceleration range.)

The weighting function which has generally been used in past work is $w(a) = a^2$, in which case the exceedance time measure is the mean square value of the acceleration (for example, see ref. 3).

Taking $w(x) = x^2$, and applying the measure to a sinusoid, one obtains

$$\text{RMET2}\left[A_0 + A_1 \sin(2\pi f t)\right] \propto A_0^2 + \frac{1}{2} A_1^2$$

Thus, with the static thresholds which have been assumed, a constant term in the acceleration appears to affect exceedance time measures more strongly than it affects the corresponding exceedance count measures.

One other specific form of exceedance time measure which has occasionally been used in procurement specifications is that based on the weighting function

$$w(x) = \text{stepA}(x) \equiv \begin{cases} 0 & \text{if } |x| \leq A \\ 1 & \text{if } |x| > A \end{cases}$$

In this case it is convenient to integrate the first definition by parts to obtain

$$\text{RMETw} \propto \frac{1}{D}\left[- \int_{-\infty}^{0} T\ dw + \int_{0}^{\infty} T\ dw \right]$$

from which we have

$$\text{RMETstepA} = [T(-A) + T(A)]/D$$

Thus this measure is seen to give the fraction of the time that the magnitude of the acceleration exceeds the value A. The only virtues this measure possesses are that it is easy to understand and easy to implement.

Since the values obtained when exceedance time measures are applied to a sinusoid are independent of the frequency of the sinusoid, every exceedance time measure will be consistant with the isobother data if the acceleration signal is passed through a filter with transfer function magnitude proportional to $1/I(t)$ prior to determination of the exceedance times.

Another measure used by Stephens (reference 2) is defined as the value A such that $|a(t)|>A$ for 10 percent of the duration of the ride. This measure may be treated as being of the form

$$- \frac{1}{D} \int_{0}^{\infty} w(x)\ v(T(x))\ dT(x)$$

with $T(x)$ defined as $T\left[\,|a(t)|\,\right](x)$, with $w(x) = x$, and with $v(T) = \delta(T - .9D)$ (where $\delta(x)$ is the symbol commonly used for the derivative of the unit step function with step at $x = 0$). The additional freedom which can be introduced by varying the weighting function $v(T)$ may turn out to be useful.

C. Spectral Measures

Whereas the measures discussed above deal directly with the acceleration as a function of time it is also possible to deal with the Fourier transform of the acceleration. To simplify the discussion, we will assume that suitable weighting of the various spectral components (such as might be needed for consistency with isobother data) has already been accomplished via filtering prior to the Fourier transformation or via numerical scaling of each of the spectral components after the transformation

There are two forms of spectral measure which have been discussed extensively in the past. One is something like an exceedance count measure and the other is analogous to an exceedance time measure.

The former is the prescription recommended in International Standard 2631 (ref. 4). The prior scaling of the various frequency components is specified based on isobother data.

The prescription requires ascertaining the r.m.s. value of each standard 1/3 octave band contribution in the spectrum. The value assigned by this measure is the largest of the r.m.s. values obtained in that manner. While this measure is quite adequate for dealing with sinusoidal motions, it is not a plausible approach to more general motions. (For example, if two sinusoidal motions which are separated in frequency by an octave or so are valued equally by this measure, the motion obtained by superposing them will receive the same value as either one alone.)

The other spectral measure which has been discussed frequently in the past (refs. 3, 4, 5) is that obtained by integrating the square of the magnitude of the transform with respect to frequency. By Parcival's theorem, this particular measure is equivalent to the corresponding exceedance time measure, namely the mean square value of the acceleration. However, integration of functions of the magnitude of Fourier transform other than the square will lead to measures which do not have simple exceedance time measure equivalents.

Mention may also be made of the interesting hybrid measure introduced by Brickman, Wambold, and Zimmermann (refs. 6 and 7). This measure is based on obtaining the spectra of a succession of short samples of motion, tabulating transform amplitude threshold exceedance counts, and forming an average weighted both with respect to amplitude and frequency.

D. Scaling and Normalization

The specific sample ride measures discussed above incorporate weighting functions which are proportional to a power of the acceleration. Thus, they are homogeneous in the sense that

$$RMn[b\ a(t)] = b^n\ RMn[a(t)]$$

where RM denotes the measure, b is an overall factor by which the acceleration function is multiplied, and n is the exponent of acceleration in the weighting function. Taking the case of power law exceedance count measure as an example we have

$$\mathrm{RMECn}\left[b\ a(t)\right] \propto \frac{1}{2D}\left\{\int_{-\infty}^{0} |x|^{n}\ dC_{\left[b\ a(t)\right]}(x) - \int_{0}^{\infty} |x|^{n}\ dC_{\left[b\ a(t)\right]}\right\}$$

where it will be recalled that $C_{[b\ a(t)]}(x)$ is the number of times that the signal $b\ a(t)$ crosses the threshold x during the interval D.

It follows from the definition of the exceedance count function that

$$C_{\left[b\ a(t)\right]}(bx) = C_{\left[a(t)\right]}(x)$$

so that

$$C_{[b\ a(t)]}(x) = C_{[a(t)]}(x/b)$$

Thus,

$$\mathrm{RMECn}[b\ a(t)]$$

$$\propto \frac{1}{2\ D}\left\{\int_{-\infty}^{0} |x|^{n}\ dC_{\left[a(t)\right]}(x/b) - \int_{0}^{\infty} |x|^{n}\ dC_{\left[a(t)\right]}(x/b)\right\}$$

$$\frac{b^{n}}{2\ D}\left\{\int_{-\infty}^{0} |y|^{n}\ dC_{\left[a(t)\right]}(y) - \int_{0}^{\infty} |y|^{n}\ dC_{\left[a(t)\right]}(y)\right\}$$

$$= b^{n}\ \mathrm{RMECn}[a(t)]$$

Any measure which is homogeneous may be rescaled so as to be linear. That is, defining the rescaled measure as the nth root of the original measure, we have

$$\mathrm{LRMn}[b\ a(t)] \equiv \left[\mathrm{RMn}[b\ a(t)]\right]^{1/n}$$

$$= b\ \mathrm{LRMn}[a(t)]$$

A linear measure can be normalized so as to assign the r.m.s. value to sinusoidal motion at some reference frequency. Then to the extent that the measure correlates well with comfort, the value which it assigns to any other motion will be the r.m.s. amplitude of an equally uncomfortable sinusoidal motion with frequency equal to the reference frequency.

Homogeneity is convenient because it permits a measure to be interpreted in the simple manner indicated above. However, it may be found that the ride measures which correlate best with subjective judgements of ride quality are not homogeneous.

The nonhomogeneous examples which come most easily to mind are those obtained when the simple power of acceleration which occurs in one of the homogeneous measures is replaced by some more general function of the acceleration such as a polynomial or a combination of exponentical functions.

One example using the hyperbolic cosine is

$$\text{RMET } \cosh\left[a(t)\right] \propto \frac{1}{D}\int_0^D dt \cosh\left[k\ a(t)\right]$$

where k is an adjustable parameter.

Looking at the example

$$a(t) = A \cos(2\pi f t)$$

one has

$$\begin{aligned} \text{RMET } \cosh\left[A \cos(2\pi f t)\right] &\propto \frac{1}{D}\int_0^D dt \cosh\left[k\ A \cos(2\pi f t)\right] \\ &\propto \frac{1}{\pi}\int_0^\pi d\theta \cosh\left[k\ A \cos\theta\right] \\ &\propto I_0\ (k\ A) \end{aligned}$$

where $I_0(x)$ is a modified Bessel function (reference 8) whose behavior is somewhat like that of the exponential function.

This measure may be rescaled so that (ignoring the effect of preliminary filtering) it assigns the r.m.s. value to any sinusoidal motion. Namely, writing $I_0^{-1}(x)$ for the function inverse to $I_0(x)$, the rescaled measure is

$$\text{LRMET} \quad \cosh\,[a(t)] \propto \frac{1}{k}\, I_0^{-1} \left\{ \frac{1}{D}\int_0^D dt\, \cosh\left[k\ a(t)\right] \right\}$$

Rescaling of the type just illustrated may be applied to any measure of this general sort, but whereas with a homogeneous measure the result would be a linear measure, here the result is a measure which is linear only so long as the motion being measured is sinusoidal.

In order to facilitate exchange of information, it might be desirable for all ride measures to have their outputs scaled so as to assign the r.m.s. value to the motion consisting of sinusoidal vertical oscillation at a chosen frequency such as 1 or 6.0 Hz.

4. A Method for Testing and Development of Ride Measures

The need which engineers have for a means of specifying ride comfort was discussed in Section 1. Section 3 has indicated that there are many different measures available for this purpose. Supposing that two such measures are under consideration, we come now to the question of how to decide which one is better. We will argue that this question has a reasonably definite answer and that that answer suggests a practical program for ride measure development and validation.

We take it as a postulate that a ride measure will be completely satisfactory only if it correlates fully with discomfort as perceived by the average passenger. (Here, as elsewhere, we assume that it is meaningful to talk about an "average passenger" and that the average passenger perceives discomfort due to ride motion as a scalar quantity. Naturally, the average passenger's response can be expected to vary depending on duration of exposure, type of seat, activity during travel, etc.) Expressed symbolically our postulate is that a ride measure, RM , will not be completely satisfactory unless it has the property that $RM(R_1) = RM(R_2)$ for every pair of ride motions R_1 and R_2 such that R_1 and R_2 are equally annoying to the average passenger.

This postulate suggests two different ways of determining how satisfactory a given ride measure is. The first way is to look at the scatter in the values assigned by the ride measure to a number of rides which are equally uncomfortable to the average passenger. That is the method which we propose. The other way is to look at the variation in perceived discomfort for a number of rides all of which are assigned the same value by the ride measure. Since difference in discomfort is somewhat ambiguous from an experimental point of view, we regard the proposed approach as the proper one in principle.

The first step in conducting either kind of correlation deficiency test is to make or select recordings of the ride motions on which the test is to be based. One might seek to develop a ride measure which could be applied to any motion environment. However, the specific ride measure which correlates best with comfort for one mode of travel and range of speeds may not be the same as the specific ride measure which correlates best with comfort for a larger group of modes and speeds. To the extent that this is so, development and testing of a ride measure should be based on ride recordings exhibiting the kinds of motion that might actually be produced by the equipment in whose specification the ride measure is to be used.

The proposed approach (i.e. determine the scatter of the values which the measure assigns to the members of a group of equally uncomfortable rides) may be carried out by: 1) using a dynamic ride simulator to reproduce each of the chosen ride motions, 2) adjusting the overall motion amplitude of each ride until the test subjects sitting in the simulator judge its discomfort to be equal to that of each of the other rides, and 3) determining the value assigned to each of the equal discomfort motions by the ride measure under test.

This method of testing has a feature which makes it very convenient for the purpose of ride measure development and optimization. Namely, since the necessary empirical data consists just of recordings of ride motions which have all been normalized to a common level of perceived discomfort, the data may be gathered without reference to any particular ride measure. Once the normalized ride motions have been recorded in digital form, the task of testing and optomizing a prospective measure (with respect to that library of normalized rides) becomes one of computation alone.

The other method of testing would require that the ride measure under test be known and in operation for the gathering of the emperical data and would make the data specific to the ride measure used. Thus it is not only inferior in principle but would be very inconvenient in practice as well.

The indicated advantage of the proposed method of testing is a reflection of the fact that it treats discomfort as the independent variable and the corresponding ride measure values measured as dependent variables. Thus, results obtained using the proposed method are convenient from the point of view of the engineer who begins with a design goal for comfort and who wishes to know what limit he must place on the measured value of the motion.

Jacobson and Kuhlthau (ref. 3) have described an alternative approach to testing and development of ride measures which has an advantage of greater realism of motion environment due to gathering of test subject responses in actual vehicle travel but in which the bases for test subject judgements cannot be as clearly defined.

5. A Symmetrized Experimental Procedure

The authors' thinking in the area of experimental method was stimulated by a paper by C. Ashley (ref. 9). Ashley determined isobother curve amplitudes at various frequency points by adjusting the amplitude until the test subject judged the sinusoid to be equal in discomfort to a quasi-constant random reference signal to which the test subject was alternately exposed. Ashley's procedure constitutes a significant improvement over procedures which seek to have subjects compare ride motions which differ in discomfort, and it could be used for the program outlined in Section 4 above. However, it may be feared that singling any one motion out as the standard of reference for all of the others could cause some undetectable bias. (For example, repeated exposure to the reference motion could cause test subjects to become unduly sensitive to other motions which were similar to it.)

Partly from fear of bias, and partly because of aesthetic dissatisfaction with the lack of symmetry if one motion is singled out as a standard, the authors have employed a symmetrical procedure as follows:

Let the number of ride motion samples to be used be n. Imagine that ride i is fed to the simulator with variable gain and that it is compared to ride j which is fed to the simulator with the gain at which it is recorded. Let g_{ij} denote the gain value which makes ride i's discomfort equal to that of ride j. Note that g_{ij} is defined in terms of a "true" equality and is not meant to be effected by inconsistancies in test subject responses. While there are n(n-1)/2 different (i j) combinations, the set of g_{ij}'s possesses only (n-1) degrees of freedom; namely they may all be determined from the values g_{n1}, g_{n2}, $g_{n,n-1}$ via the relations,

$$g_{ij} = g_{in}\, g_{nj}$$

$$= g_{nj}/g_{ni}$$

On the other hand, let r_{ij} denote the corresponding gain settings as determined from test subject responses during a particular set of comparisons using a ride motion simulator. Because of experimental error the r_{ij} values will not be transitive (i.e. $r_{ij}r_{jk}$ will not equal r_{ik}).

We, therefore, seek the set of g_{ij} values which provides the best fit to the experimental r_{ij} values.

The variables to be determined are g_{n1}, g_{n2}, . . . $g_{n,n-1}$, which we will abbreviate as g_1, g_2, . . . g_{n-1}. For the error function which is to be minimized we take

$$E = \frac{1}{2} \sum_{i \neq j}{}' \left[r_{ij}/g_{ij} - 1 \right]^2 = \frac{1}{2} \sum_{i \neq j}{}' \left[g_i r_{ij}/g_j - 1 \right]^2$$

where the prime over the summation symbol is to indicate that a given (ij) pair is not to be included in the sum if the corresponding r_{ij} was not measured. We presume that the error function given above is the best choice. However, we are not aware of any theorem to that effect, and there are other simple positive definite functions which could be used.

The g_i values which minimize E are found with the help of a simple computer code which uses Newton's method and iterates until the partial derivatives, $\partial E/\partial g_i$, are all close to zero.

The level of discomfort to which all of the rides are to be adjusted is chosen to be that of ride n when its gain is multiplied by

$$g_{n,mean} = \left[g_1 \, g_2 \cdots g_{n-1} \right]^{1/n}$$

The comfort of ride i is brought to that level by multiplying its gain by the factor

$$S_i = g_{n,mean} / g_{ni}$$

This choice of settings has the desirable property that the product

$$S_1 \, S_2 \, . \, . \, . S_n = 1$$

and thus that the passenger reponses can not cause any rise or fall in the geometric mean of all of the settings.

Determination of the S_i 's should be done in two or three stages with the first one serving to bring all of the ride samples close to a common level of discomfort so that adjustments in subsequent stages will be small. The motive here is to minimize errors which would arise from nonlinearity in simulator and test subject responses.

As a further detail of procedure, the ride i - ride j pairs are presented to the test subjects in a random order.

6. Production of a Library of 19 Equal Discomfort Rides

The authors have carried out the steps set forth above on a pilot basis as follows:

A) Selection of Sample Motions

Seventeen samples of passenger rail car ride motion were selected so as to include a number of distinctly different types of disturbing motion as well as several "good" rides. Each rail car sample included vertical acceleration and lateral acceleration as sensed by accelerometers located on the floor of the car over one of the trucks. Two sinusoidal samples were added to the collection so as to facilitate comparison with work by others.

The numbers of segments from the various sources were:

Car Type	Truck Type	# of segments
Metroliner	G70	5
St. Louis Silver Liner	G70	4
Penn Central E5	Commonwealth inside S.H.	2
DOT Test Car	Pioneer	2
Santa Fe High Level	Commonwealth outside S.H.	2
Budd Silverliner	Pioneer	1
GE Silverliner	G70	1
Sine Wave, 6 Hz.	1 lateral, 1 vertical	2
	TOTAL	19

The disturbing motions which are represented were described when they were recorded by terms such as, brake shudder, chafing, grinding, resonance, bounding, growling, lurching, and bottoming.

B) Presentation of Pairs of Rides to the Test Subjects

The ride motion simulator used in this work was the Passenger Ride Quality Apparatus (PRQA) at NASA Langley Research Center. Data were gathered on the basis of responses from three men and one woman seated in aircraft "tourist class" type seats.

Let A and B denote two ride motions being compared. The two rides were fed to the PRQA in accordance with the following protocol:

ride A	,	10 sec
pause	,	2 sec
ride B	,	10 sec
pause	,	2 sec
ride A	,	10 sec

stop tape drive

have subjects say which ride was more annoying

manually adjust the separate gain controls provided for rides A and B so as to reduce the difference in annoyance

ride B	sample and pause durations as before
ride A	
ride B	

stop tape drive.

have subjects say which ride was more annoying

manually adjust gain settings so as to further reduce the difference in annoyance.

The above sequence was repeated until the test subjects indicated that the two rides were equally annoying. At that point the gain settings for both rides were recorded and the test tape was run forward to the next pair of rides.

The 10 sec and 2 sec durations appeared to be satisfactory. The ordering of pairs on the test tapes was randomized. Independent control of the gains for rides A and B was accomplished by means of an electronic control module located between the tape drive and the PRQA and controlled by timing and switching signals on tape channels 7 and 8. The person conducting the test was kept informed of the identities of the individual rides via a digital read out operated by coding on tape channels 9 through 14. That module also operated a pair of lights for keeping the test subjects informed as to whether the ride in progress was A or B.

C) Determination of Gain Settings for Equal Discomfort

The testing accomplished to date has consisted of only one cycle. Thus the gain setting ratios, r_{ij} , which have been measured are fairly large. However, the procedure set forth in Section 5 has been carried out and a recording of simulator platform motion has been made for each ride with gain setting equal to 0.496 times the S_i value defined in Section 5. (The extra factor of 0.496 was introduced to assure that the PRQA would not be over driven.) The final gain settings based on 76 pair comparisons were:

RIDE NO.	GAIN SETTING
1	0.621
2	0.525
3	0.504
4	0.362
5	0.306
6	0.509
7	0.429
8	0.482
9	0.572
10	0.362
11	0.531
12	0.384
13	0.302
14	0.800
15	0.609
16	1.500
17	0.860
18	0.360
19	0.300

The characteristics of the signals fed to the gain control module and of the accelerations of the PRQA platform pursuant to the final gain settings are both illustrated by the computer generated oscillograms reproduced in figures 1 through 19. Figures 20 and 21 show the r.m.s. values of the vertical and horizontal components of each of the rides both by half octave band and overall.

For the recording of the PRQA motions in response to the rides at their final settings, the PRQA was ballasted with 3 passengers and 68 kg (150 lbs) of bagged sand. Rides 1 through 19 were played in sequence with brief pauses between rides. As a matter of curiosity, each passenger was asked to rate each ride on a numerical scale from 0 (no discomfort) to 8 (maximum discomfort). No further verbal instruction was given. The results were as follows:

DISCOMFORT RATINGS BY "BALLAST" PASSENGERS

(ratings shown for each subject have been divided by the mean value of the ratings which that subject assigned)

RIDE	SUBJECT 1	SUBJECT 2	SUBJECT 3	MEAN $\bar{X}$	SAMPLE STANDARD DEVIATION S
1	1.28	.57	1.02	.96	.36
2	1.06	.75	1.02	.94	.17
3	1.17	.94	1.02	1.02	.12
4	1.40	1.64	1.10	1.38	.27
5	1.01	.50	1.02	.84	.30
6	.84	1.01	1.00	.95	.10
7	1.01	1.13	1.02	1.05	.07
8	.67	1.07	1.02	.92	.22
9	1.01	1.38	1.05	1.15	.20
10	1.51	1.57	1.30	1.46	.14
11	1.89	1.19	1.02	1.03	.15
12	.61	.88	.97	.82	.19
13	1.17	1.32	1.02	1.17	.15
14	.50	157	.90	.66	.21
15	1.06	1.19	1.10	1.12	.07
16	.84	.75	1.02	.87	.14
17	1.01	.31	.27	.53	.42
18	1.56	1.13	1.05	1.25	.27
19	.45	1.31	1.05	.88	.37
S	.31	.36	.19	.23	

The "ballast" subjects appear to find some significant differences in discomfort among the final rides. The following may be noted as possible sources of difference:

- the "ballast" subjects rode in the simulator for a much shorter time than the original subjects
- the empirical gain ratios were larger than one would wish because circumstances have not yet allowed for a second stage of comparisons with the starting gains equal to the final gains from the first cycle.
- the judgements of the ballast subjects may have included some extra randomness due to ambiguity as to frame of reference.

7. Preliminary Evaluation of Several Ride Measures

We have begun to carry out the program of section four using the library of 19 rides described in section six. Work to date has been limited to an initial scrutiny of the family of ride measures given by the formula

$$RM = \sum_{k=1}^{4} \sum_{i}^{\text{band } k} \left[A_k + B_k \log(f_i) \right] V_i^{\,n} + \sum_{k=6}^{7} \sum_{i}^{\text{band } k} \left[A_k + B_k \log(f_i) \right] H_i^{\,n}$$

where V_i and H_i are the magnitudes of the vertical and lateral acceleration Fourier components at frequency f_i, where i is summed over the frequency points in each of the bands into which the frequency range is divided, and where the disposable parameters of the measure are the exponent n, the constants A_k and B_k which define the semi-log straight line weighting function in frequency band k, and the locations of the boundaries of the bands in the frequency range. The A's and B's are constrained so as to make the weighting function continuous at the band boundaries. Thus, for any fixed choice of frequency band boundaries, the weighting curves offer eight disposable parameters. Overall normalization effectively reduces that number to seven. This ride measure is convenient for purposes of exploration because it depends linearly on the weighting function height parameters.

A least squares fitting routine was used to find the weighting curve height parameters which minimize the error function

$$\text{Error} = \sum_{i=1}^{19} (RM_i - 1)^2$$

where RM_i is the value assigned to the i_{th} ride.

This fitting was done with the exponent, n, and the frequency band boundary points fixed and was repeated for several combinations of exponent and frequency band boundaries.

For the purpose of comparing measures with different exponent values we use the sample standard deviation of the linearized form of each measure, namely

$$\text{Deviation} = \left\{ \frac{1}{18} \sum_{i=1}^{19} \left[RM_i^{1/n} - 1 \right]^2 \right\}^{1/2}$$

Table 1 shows some sample results with the weighting curve heights scaled so that each curve has height unity at the beginning of the fourth band of the vertical spectrum. An exponent value of four was also tried but was found to give residual errors larger than those obtained with the exponent value three.

One may note that some of the weighting curve heights are negative. While it is clear that the occurrance of negative weighting values can be legitimate relative to a fixed set of ride motions, it is also clear that a ride measure with some negative spectral weights will fail badly if it is applied to a sinusoidal motion with a frequency such that the corresponding weighting is negative. Thus for results which are to be used in practice, the weighting would need to be made everywhere positive, either by constraint, or by augmenting the library of equal discomfort rides with rides having appreciable energy at frequencies where negative weights had been obtained.

While the specific results obtained to date must be considered tentative because of the limitations of the equal discomfort ride data base both as to number of rides and as to likelihood of scatter in actual discomfort, they suggest the following three conclusions.

First, to obtain parameter optimization results which are not unduly sensitive to minor variations in the structure of the model, the empirical data base of equal discomfort ride motions will need to be a good deal larger than the one discussed here.

Second, for rail car comfort the upper portion of the frequency range appears to be more important than the isobother type data would suggest.

Third, the square of the acceleration appears to provide a better measure than does either the first or the third power.

8. Acknowledgements

The ride recordings and test tapes referred to in Section 6 were prepared by R. Arendt of Calspan Corporation with assistance from C. A. Woodbury of Louis T. Klauder and Associates, and constituted an essential part of the experimental work reported here.

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TABLE 1 – VALUES FOR WEIGHTING CURVE HEIGHTS AS DETERMINED BY LEAST SQUARES FIT AND CORRESPONDING RESIDUAL DEVIATION, AS DEFINED IN SECTION 7

FREQUENCY BAND BOUNDARIES: VERTICAL	FREQUENCY BAND BOUNDARIES: HORIZONTAL	EXPONENT VALUE	DEVIATION	RELATIVE HEIGHTS OF RESULTING WEIGHTING CURVE AT FREQUENCY BAND BOUNDARIES: VERTICAL					HORIZONTAL		
.5 1.5 4 12 30	.5 6 30	1	0.21	–1.49	.59	–.19	1.0	–1.30	–1.81	1.27	5.82
		2	0.17	– .16	.02	–.77	–1.0	–18.05	– .41	2.80	9.19
		3	0.21	– .01	–.01	–.19	–1.0	–4.52	– .19	1.85	6.28
.5 1 2 6 30	.5 6 30	1	0.21	–3.68	.10	4.09	1.0	12.15	–4.26	3.59	5.16
		2	0.163	– .11	.32	.89	1.0	5.61	– .36	.78	2.08
		3	0.19	.57	.34	.87	1.0	4.13	– .38	3.64	12.30
.5 1 2 6 30	.5 4 30	1	0.21	–4.80	.34	4.61	1.0	17.15	–4.37	3.83	–3.61
		2	0.165	– .10	.31	.87	1.0	5.68	– .34	.44	1.74
		3	0.19	.82	.41	1.06	1.0	4.09	– .50	3.20	25.27

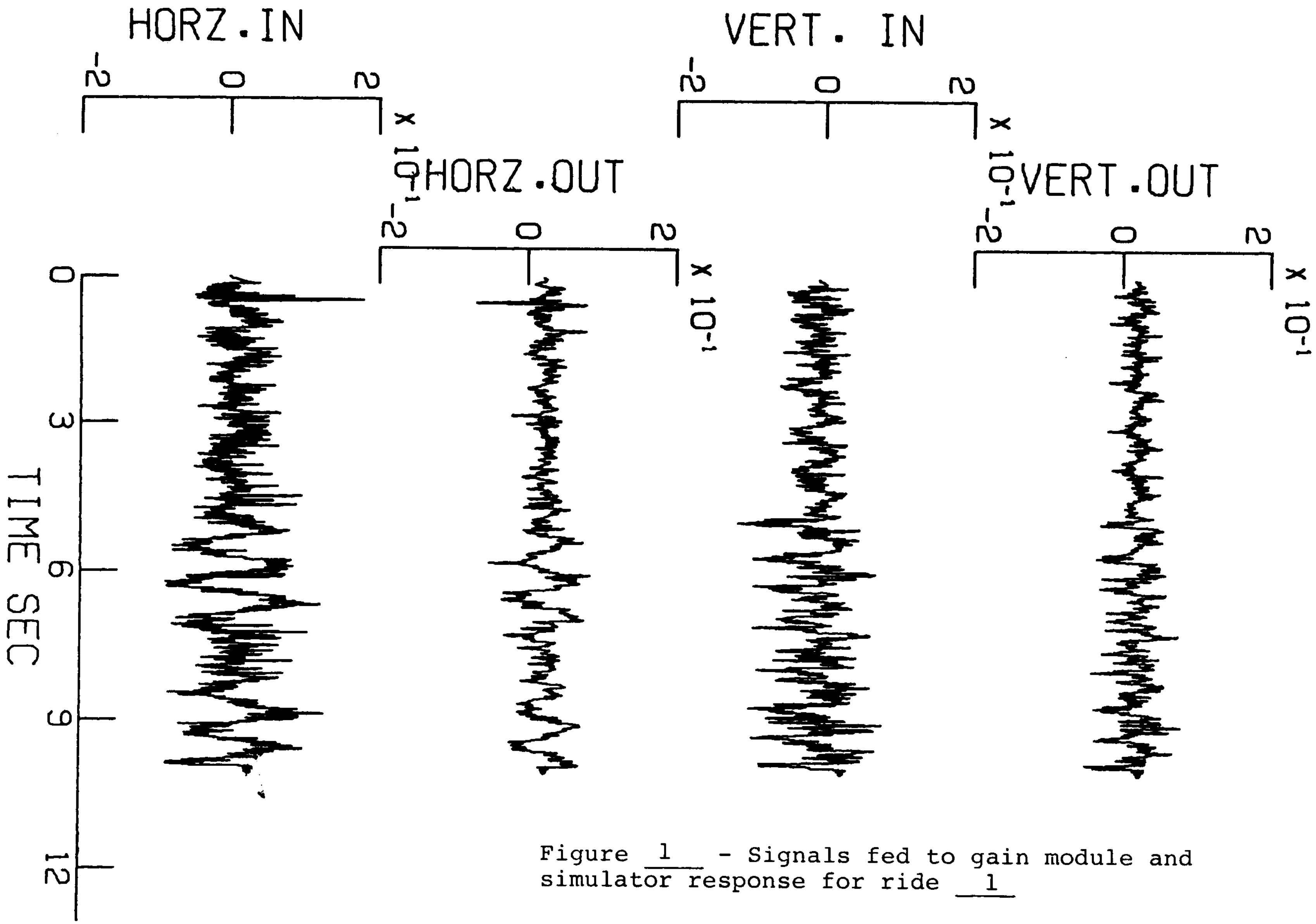

Figure 1 - Signals fed to gain module and simulator response for ride 1

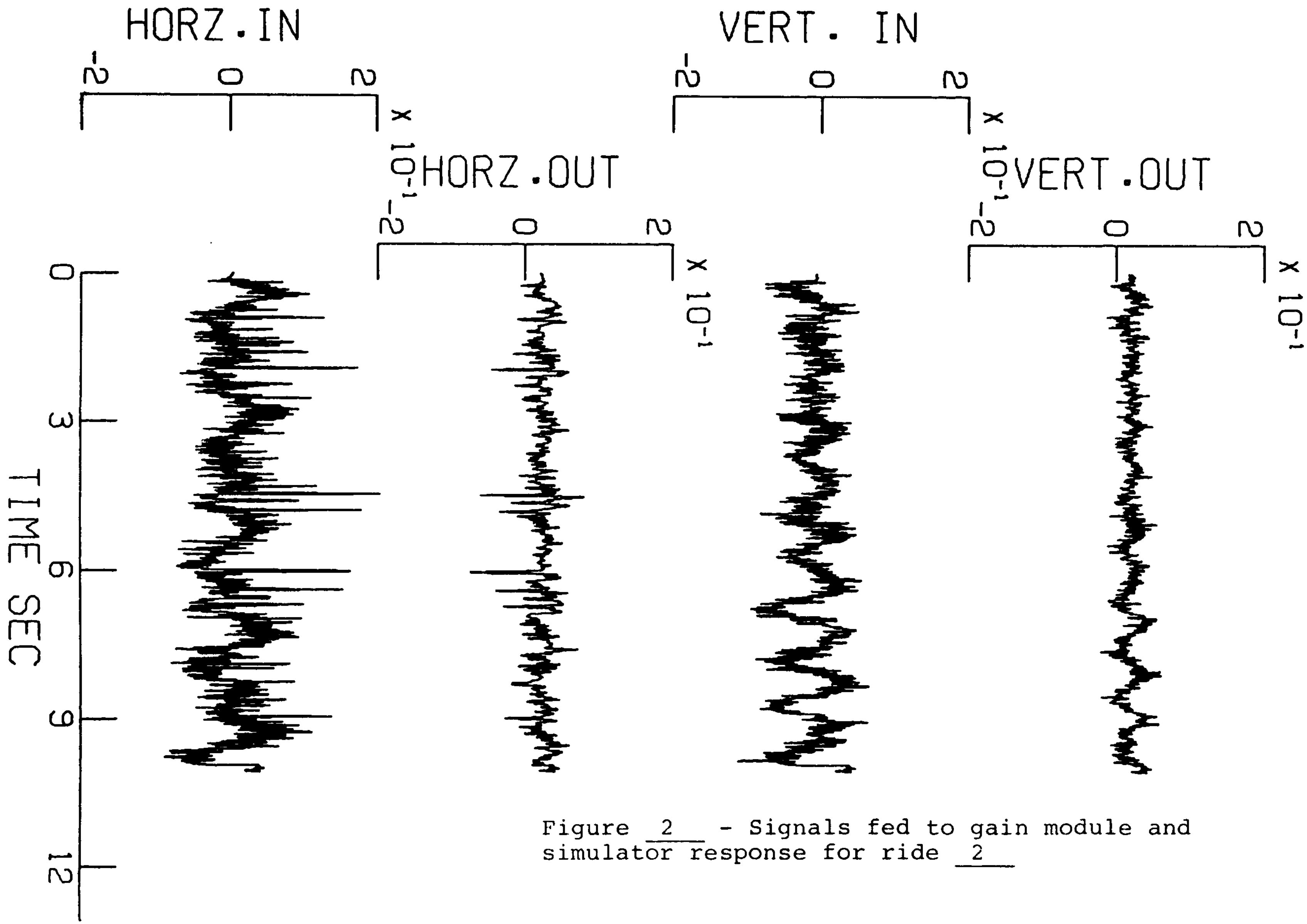

Figure 2 - Signals fed to gain module and simulator response for ride 2

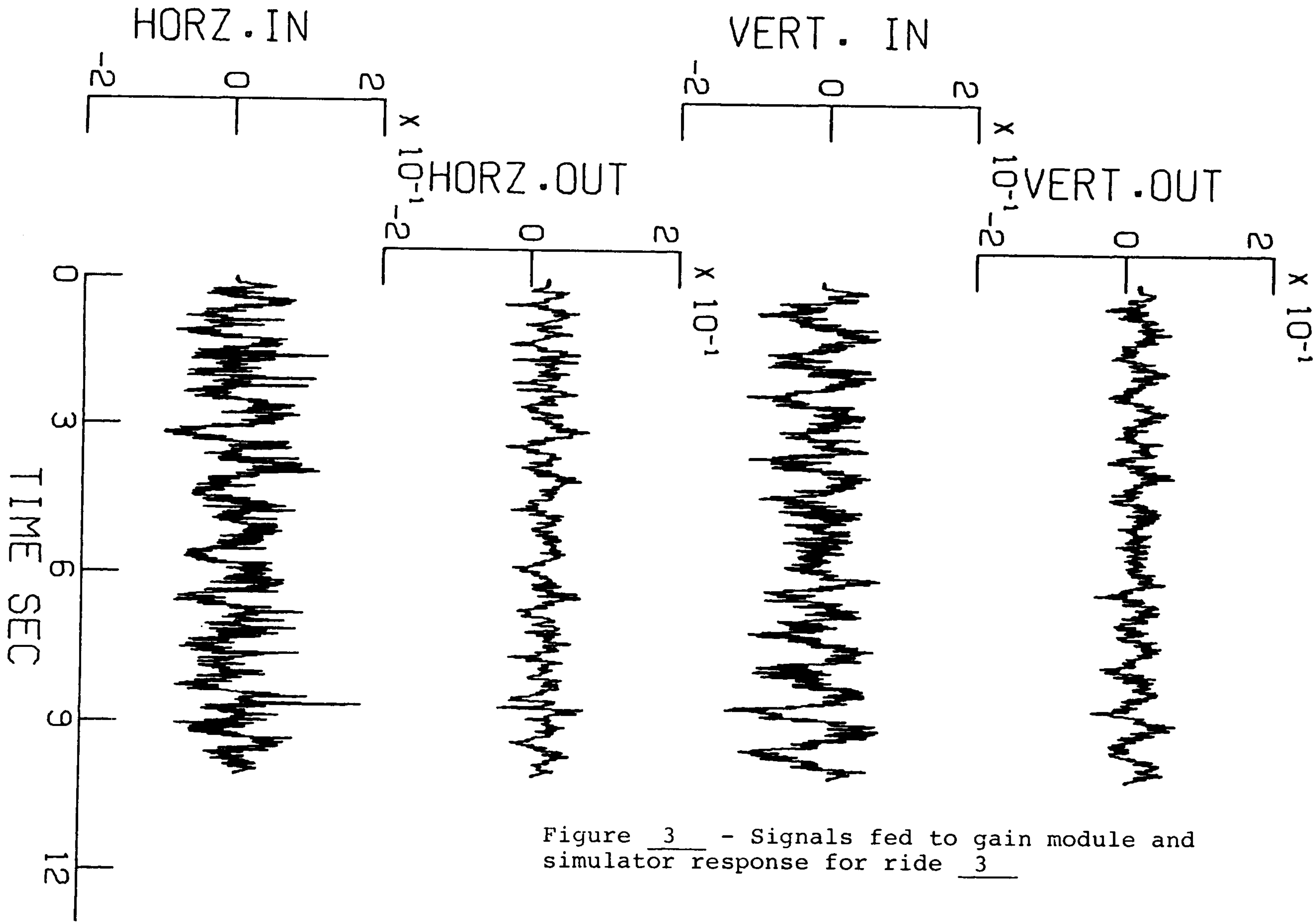

Figure 3 - Signals fed to gain module and simulator response for ride 3

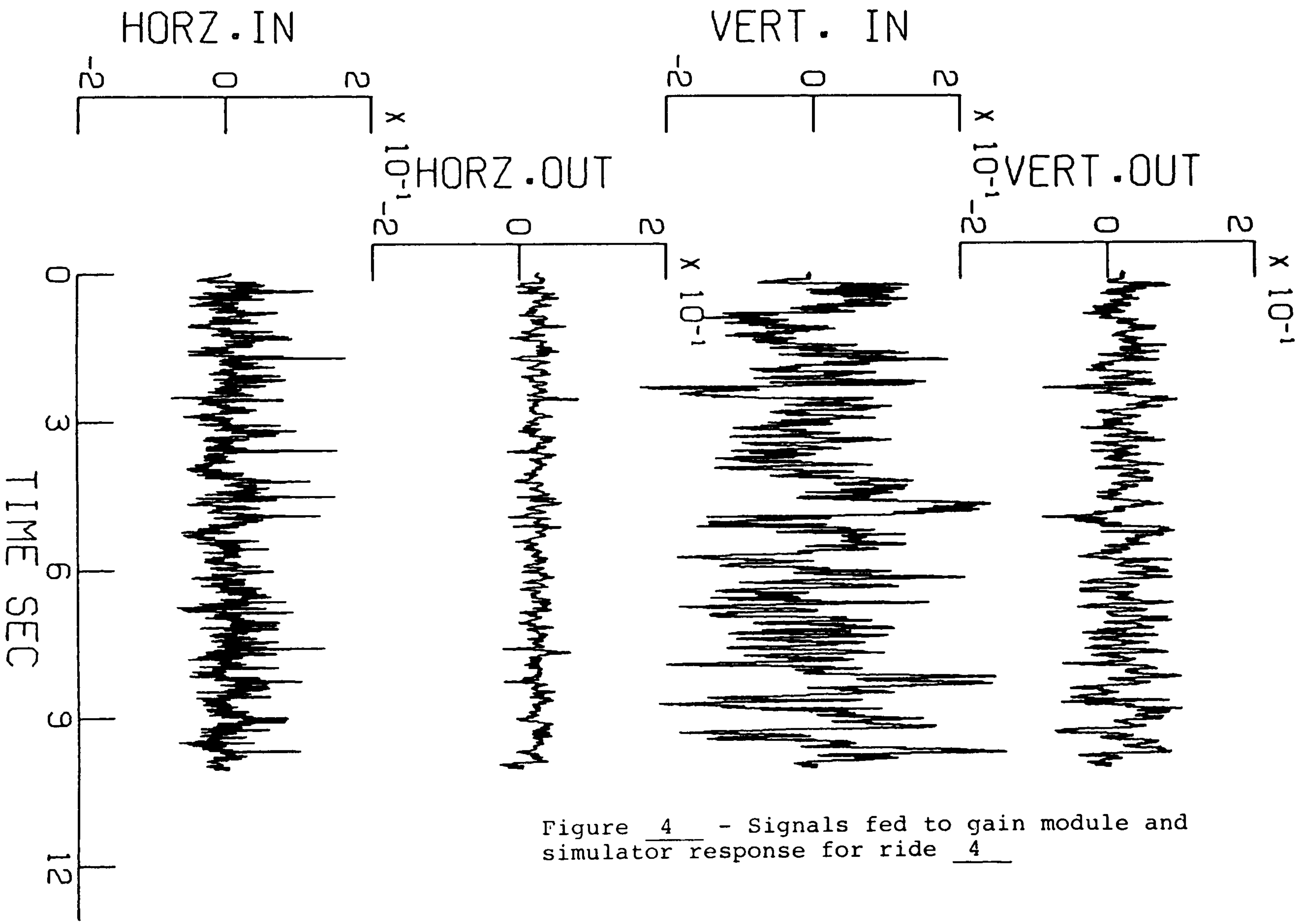

Figure 4 - Signals fed to gain module and simulator response for ride 4

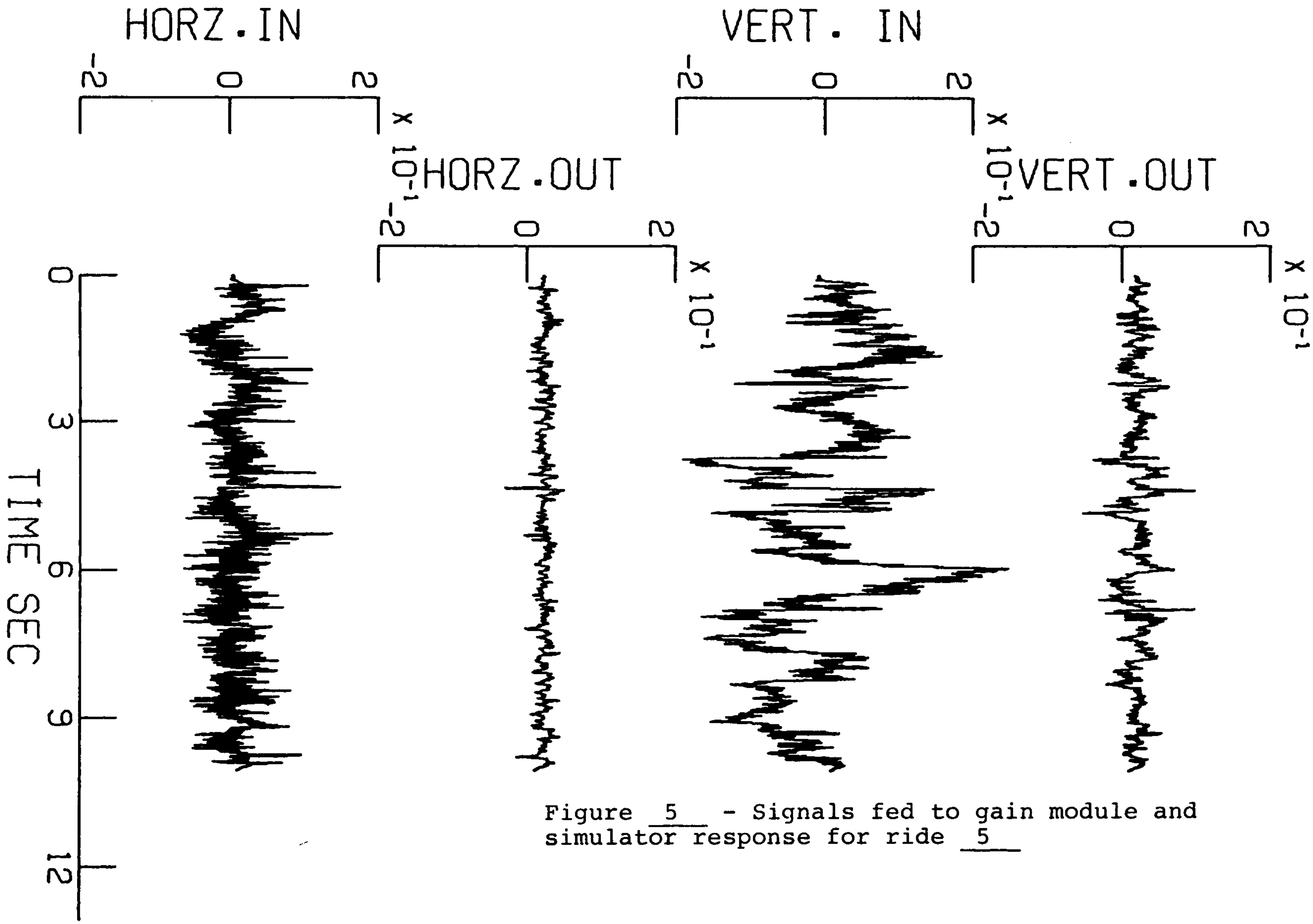

Figure 5 - Signals fed to gain module and simulator response for ride 5

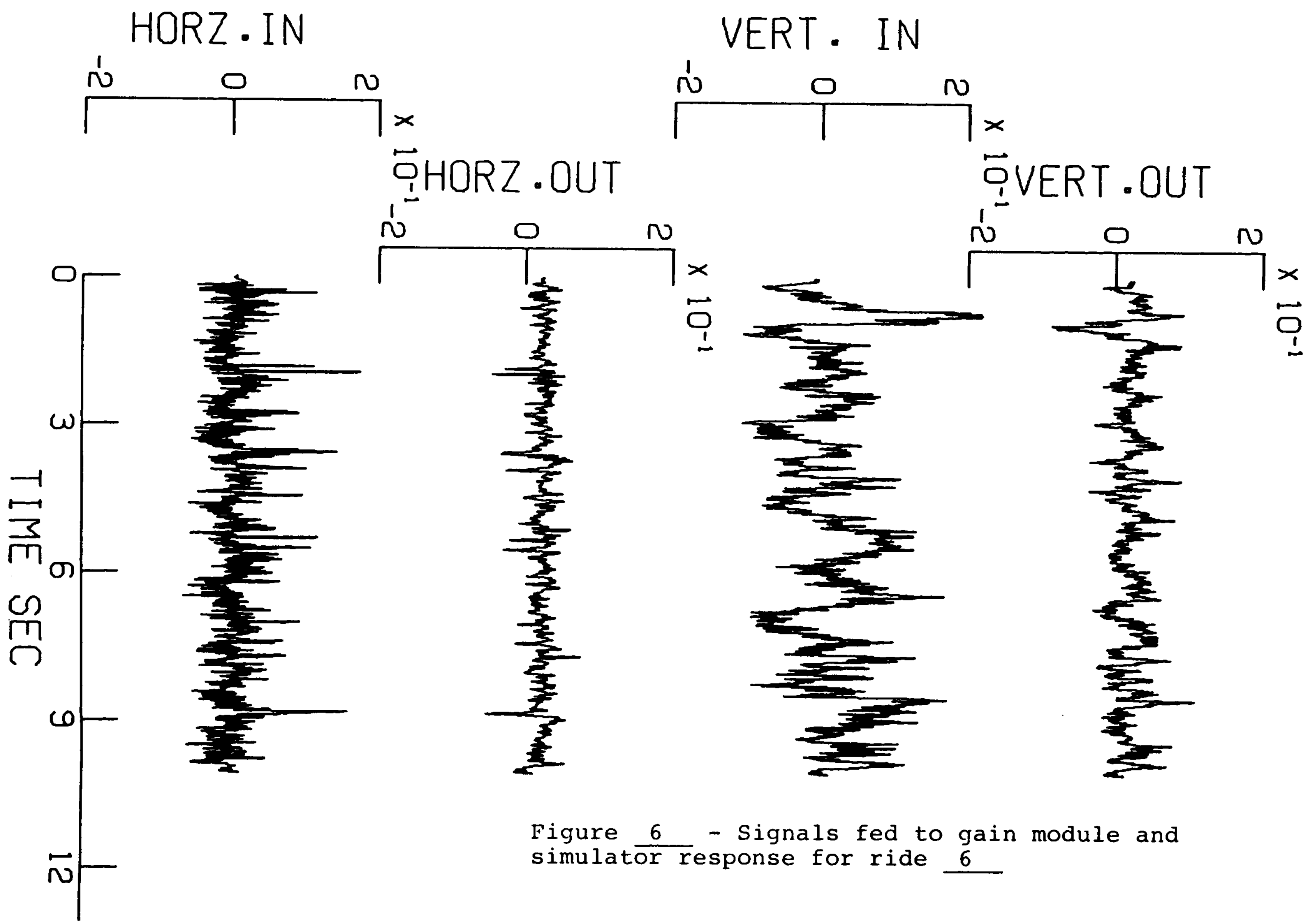

Figure 6 - Signals fed to gain module and simulator response for ride 6

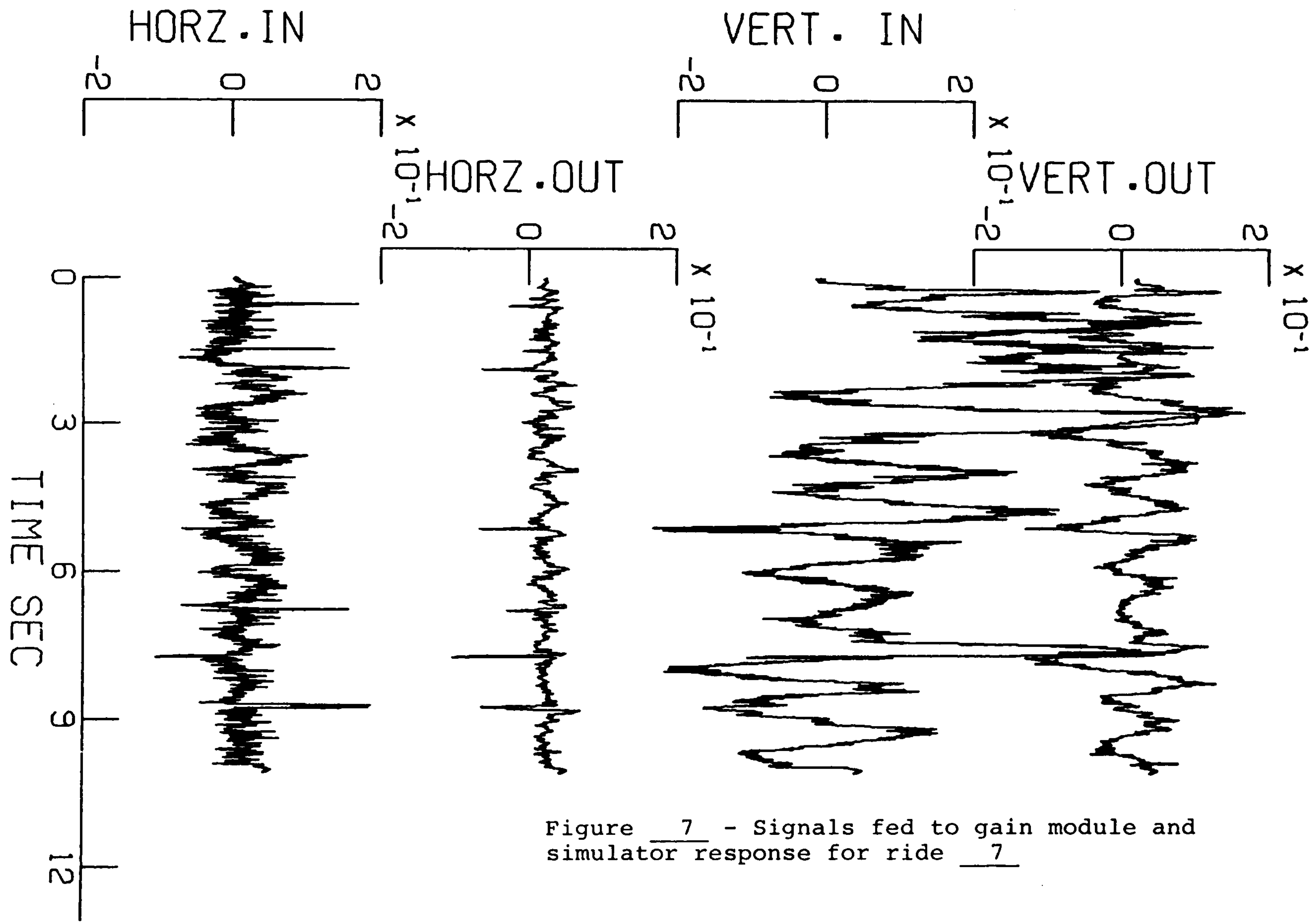

Figure 7 - Signals fed to gain module and simulator response for ride 7

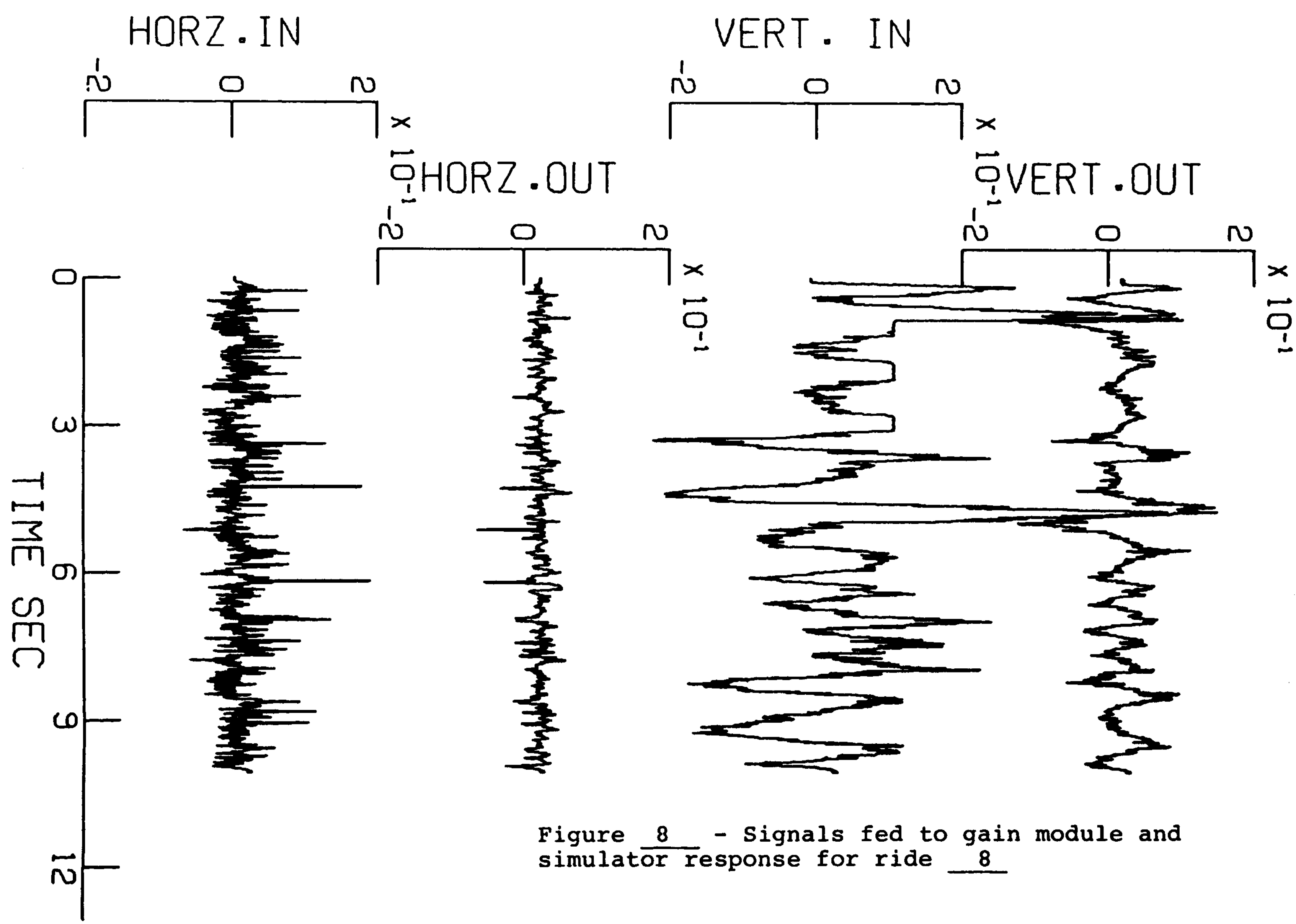

Figure 8 - Signals fed to gain module and simulator response for ride 8

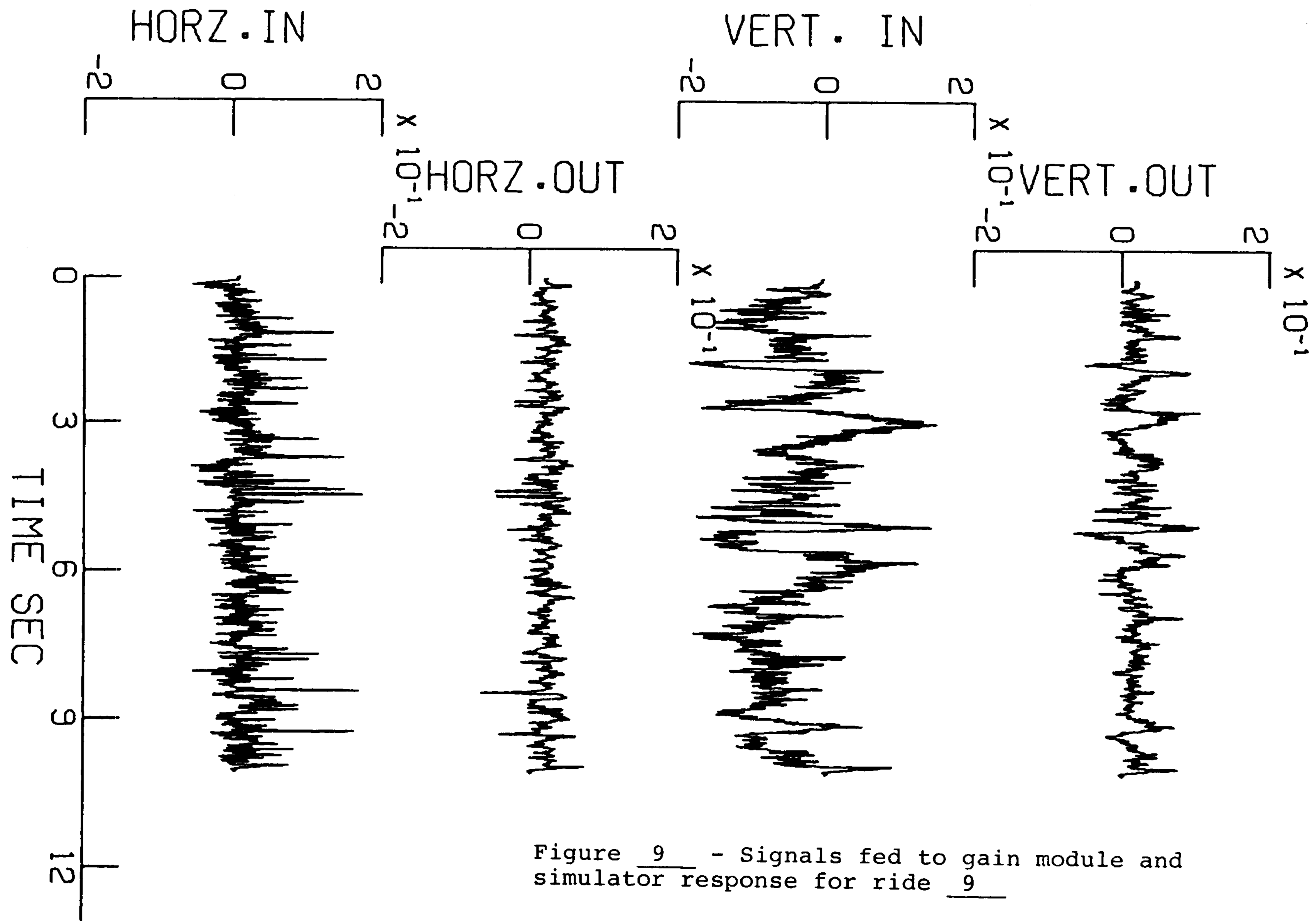

Figure 9 - Signals fed to gain module and simulator response for ride 9

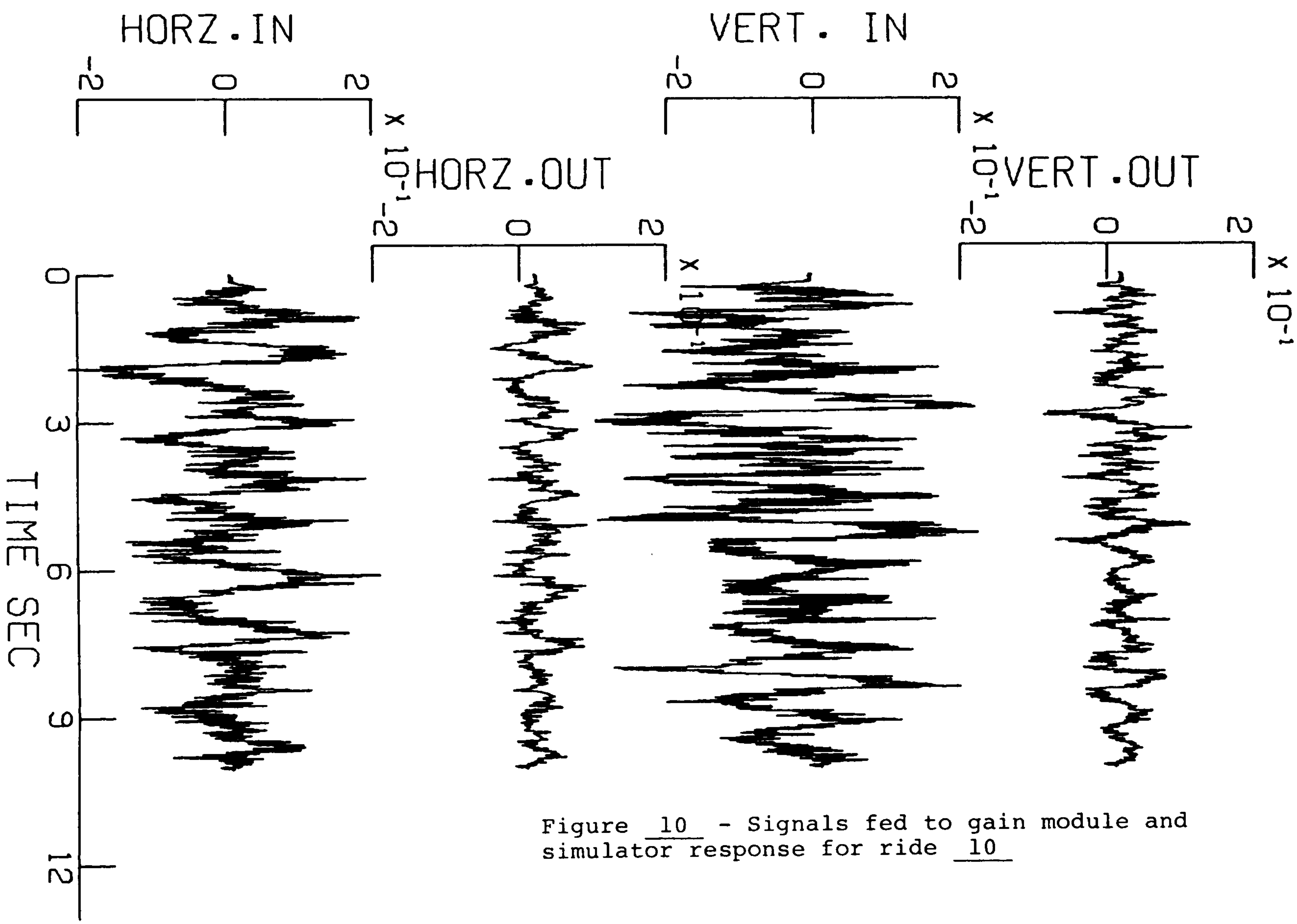

Figure 10 - Signals fed to gain module and simulator response for ride 10

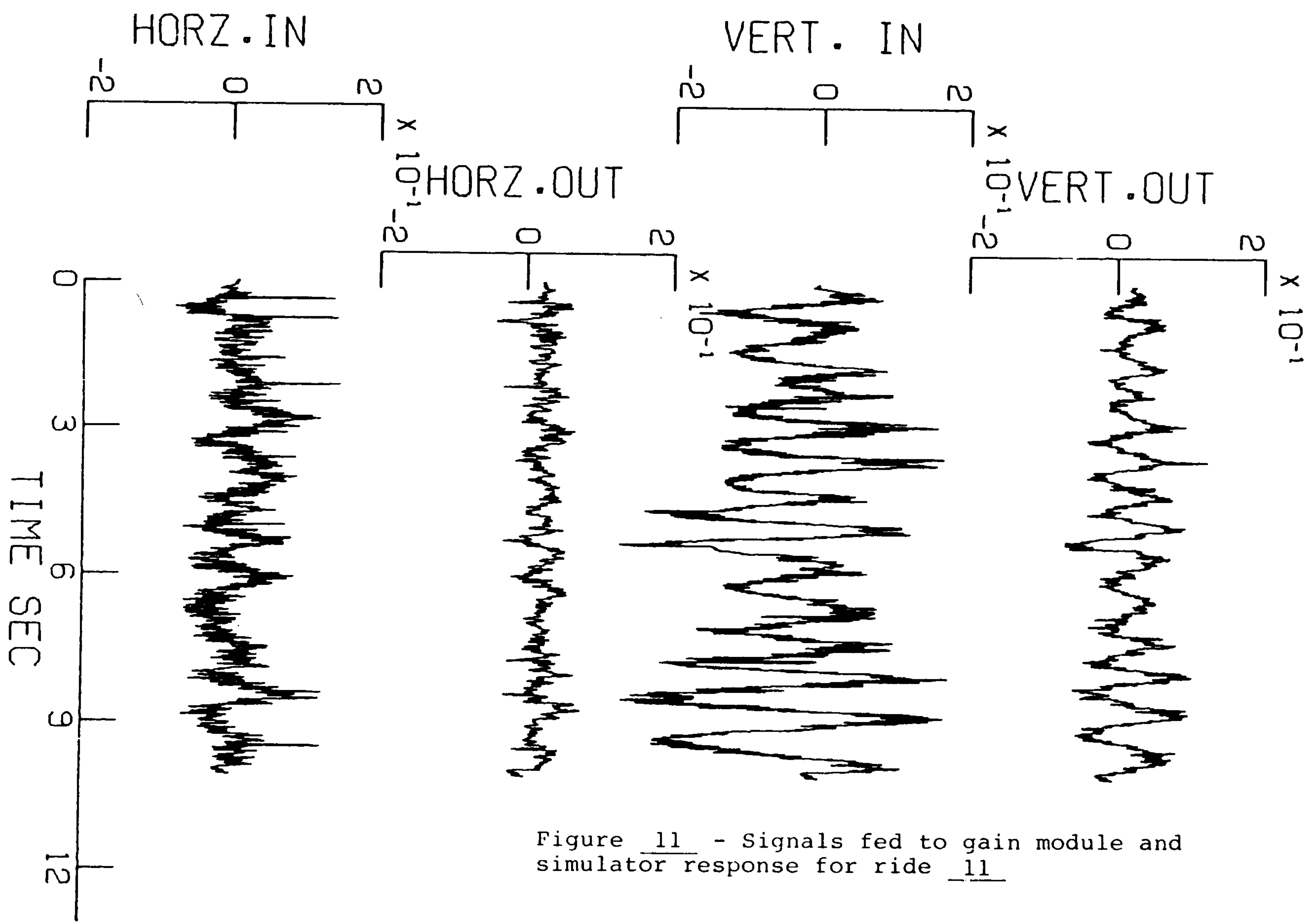

Figure 11 - Signals fed to gain module and simulator response for ride 11

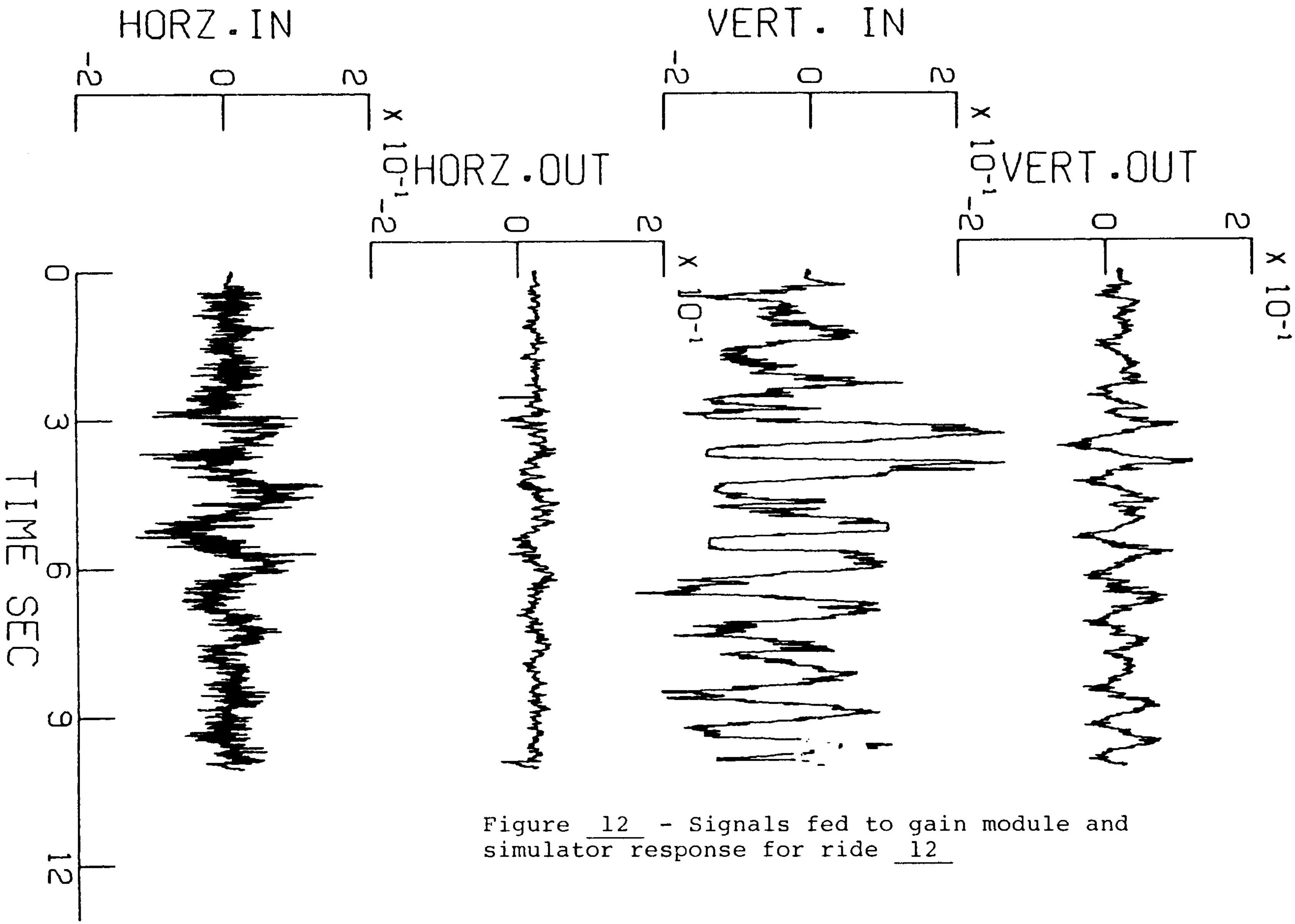

Figure 12 - Signals fed to gain module and simulator response for ride 12

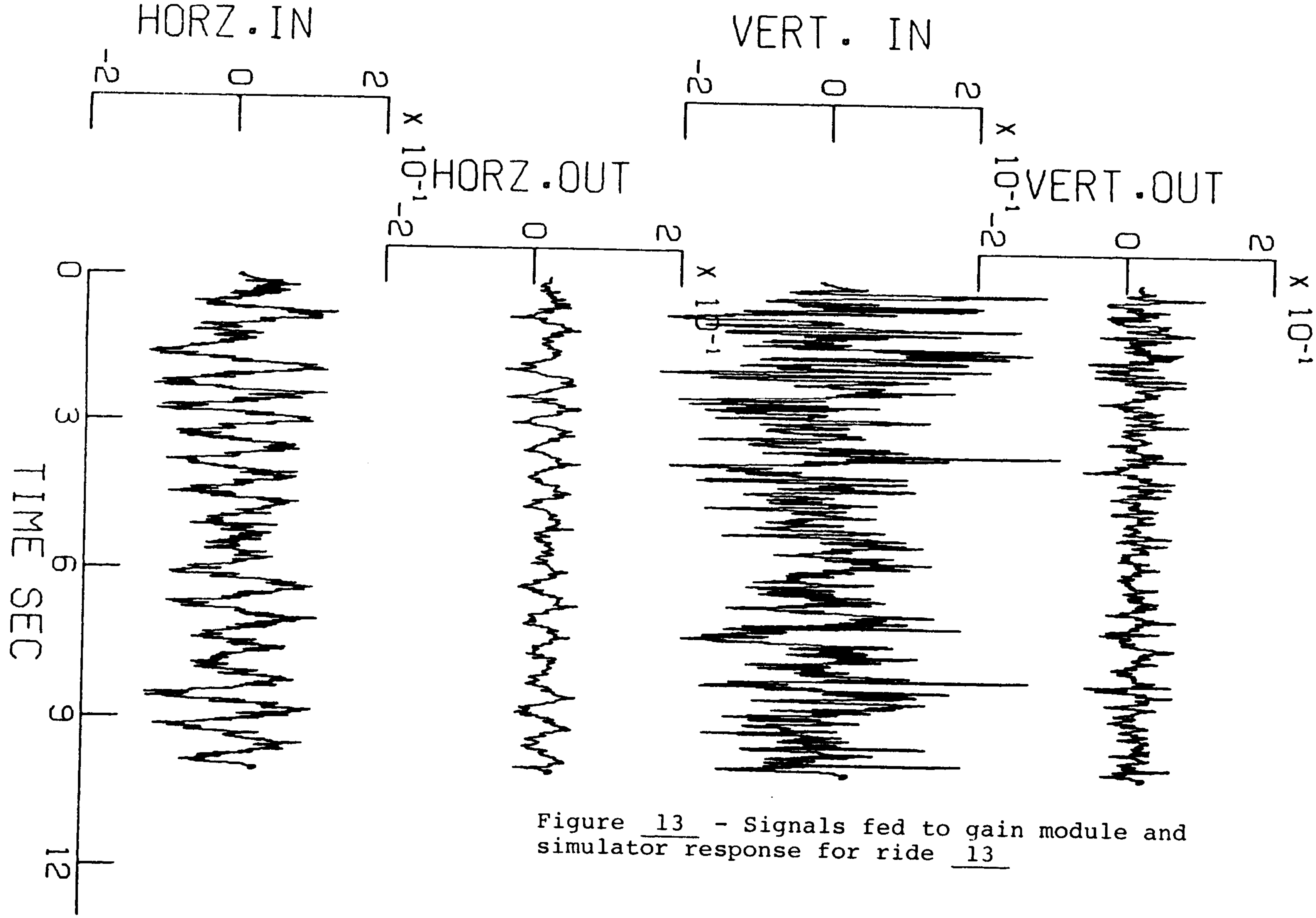

Figure 13 - Signals fed to gain module and simulator response for ride 13

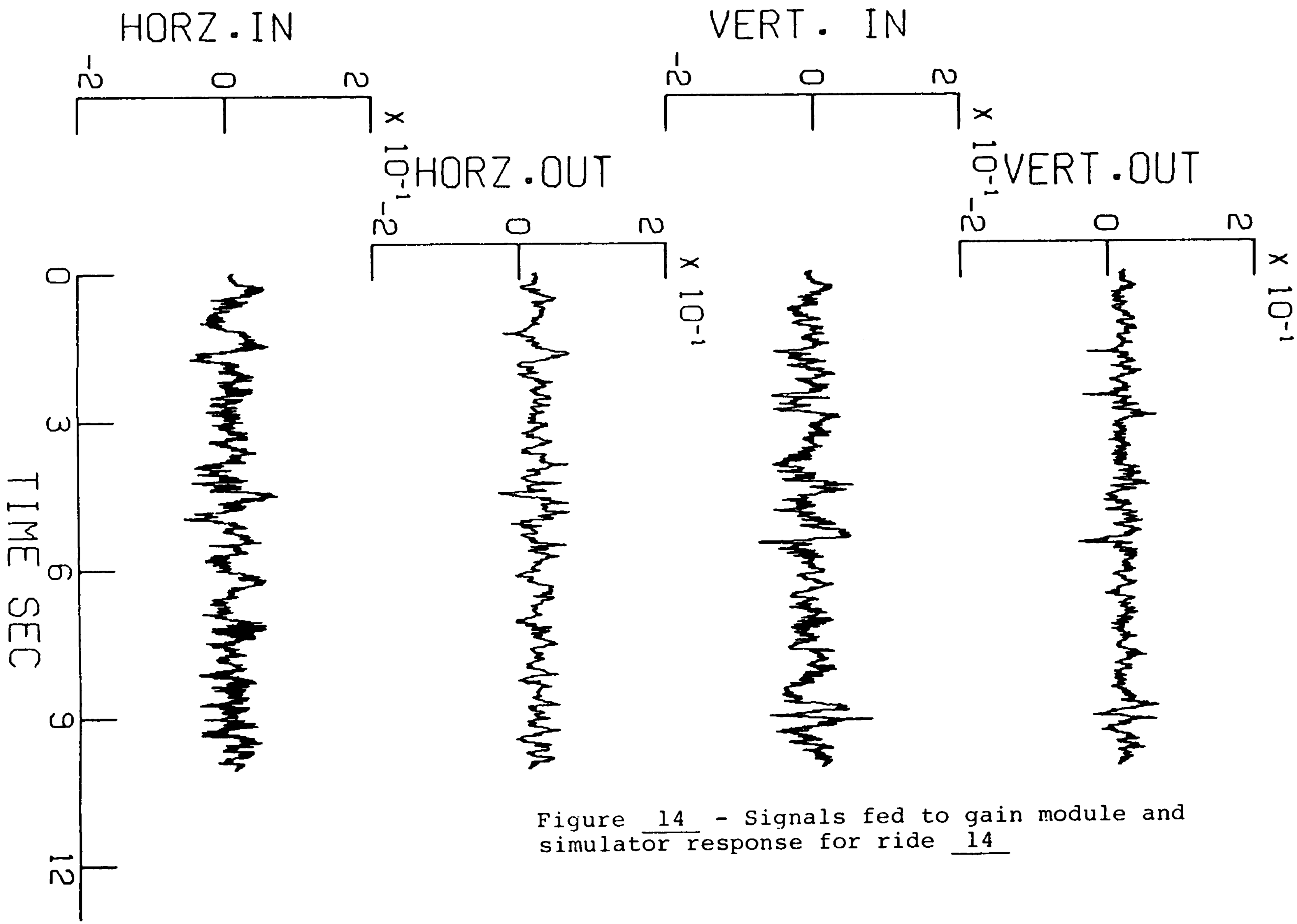

Figure 14 - Signals fed to gain module and simulator response for ride 14

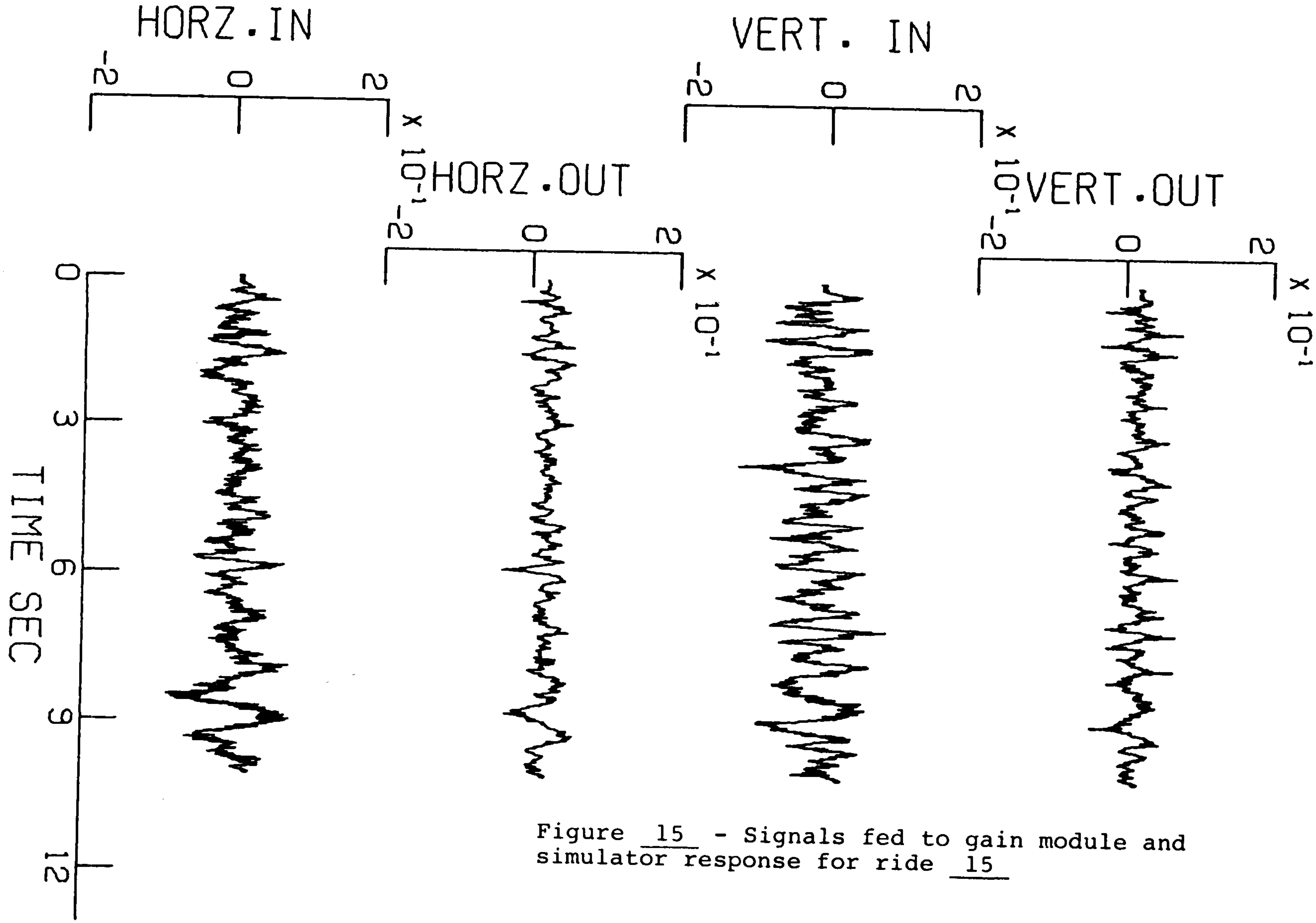

Figure 15 - Signals fed to gain module and simulator response for ride 15

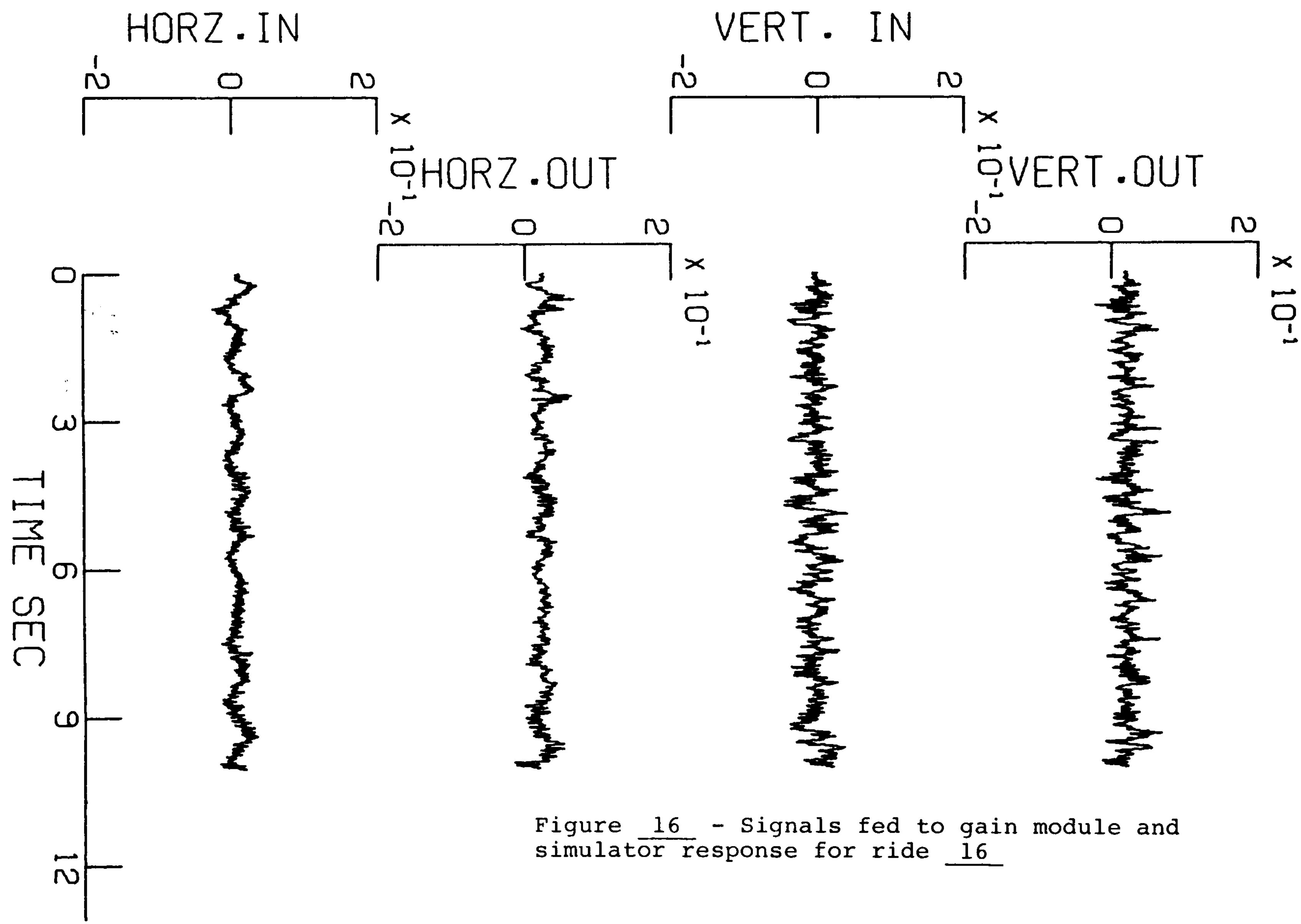

Figure 16 - Signals fed to gain module and simulator response for ride 16

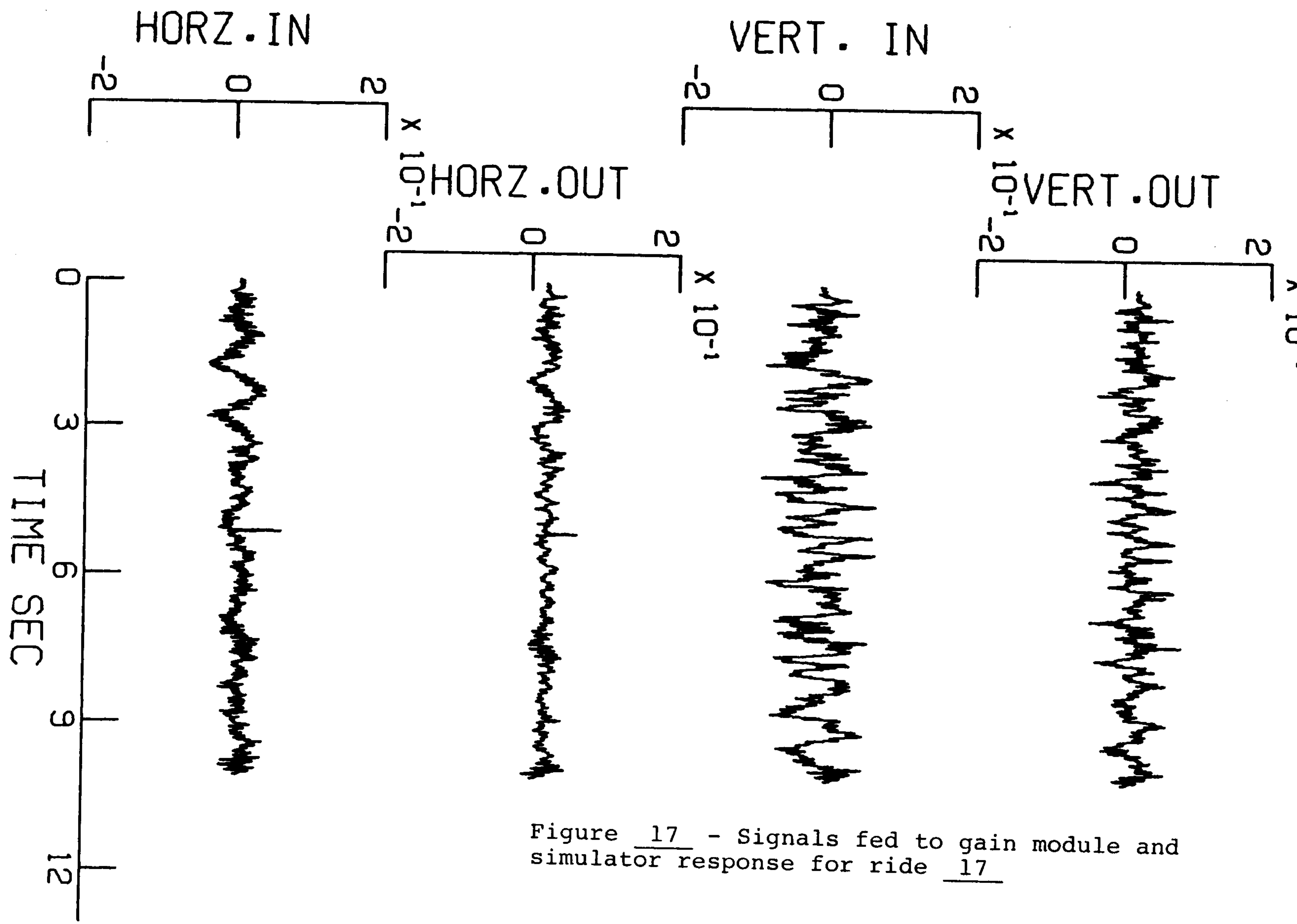

Figure 17 - Signals fed to gain module and simulator response for ride 17

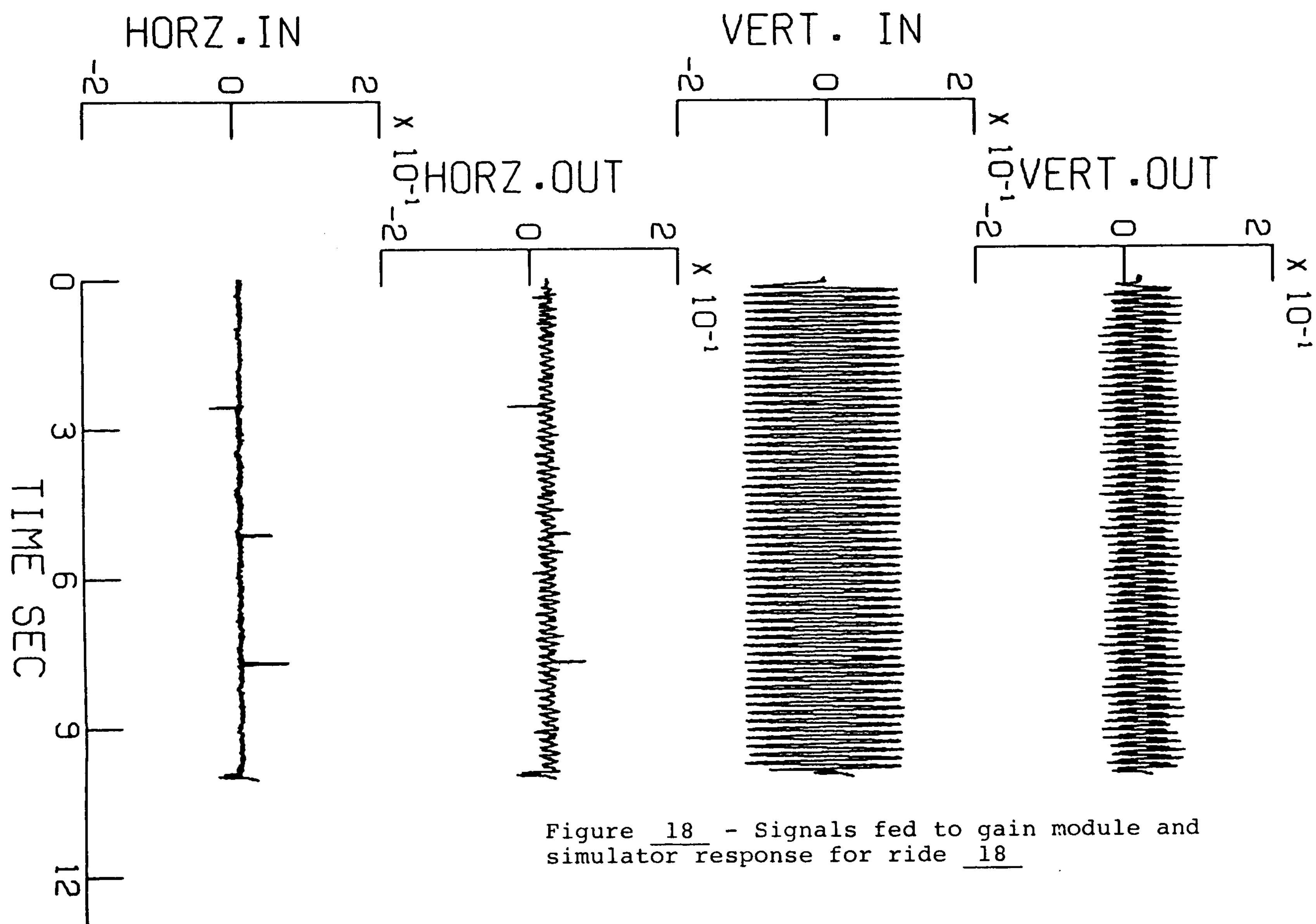

Figure 18 - Signals fed to gain module and simulator response for ride 18

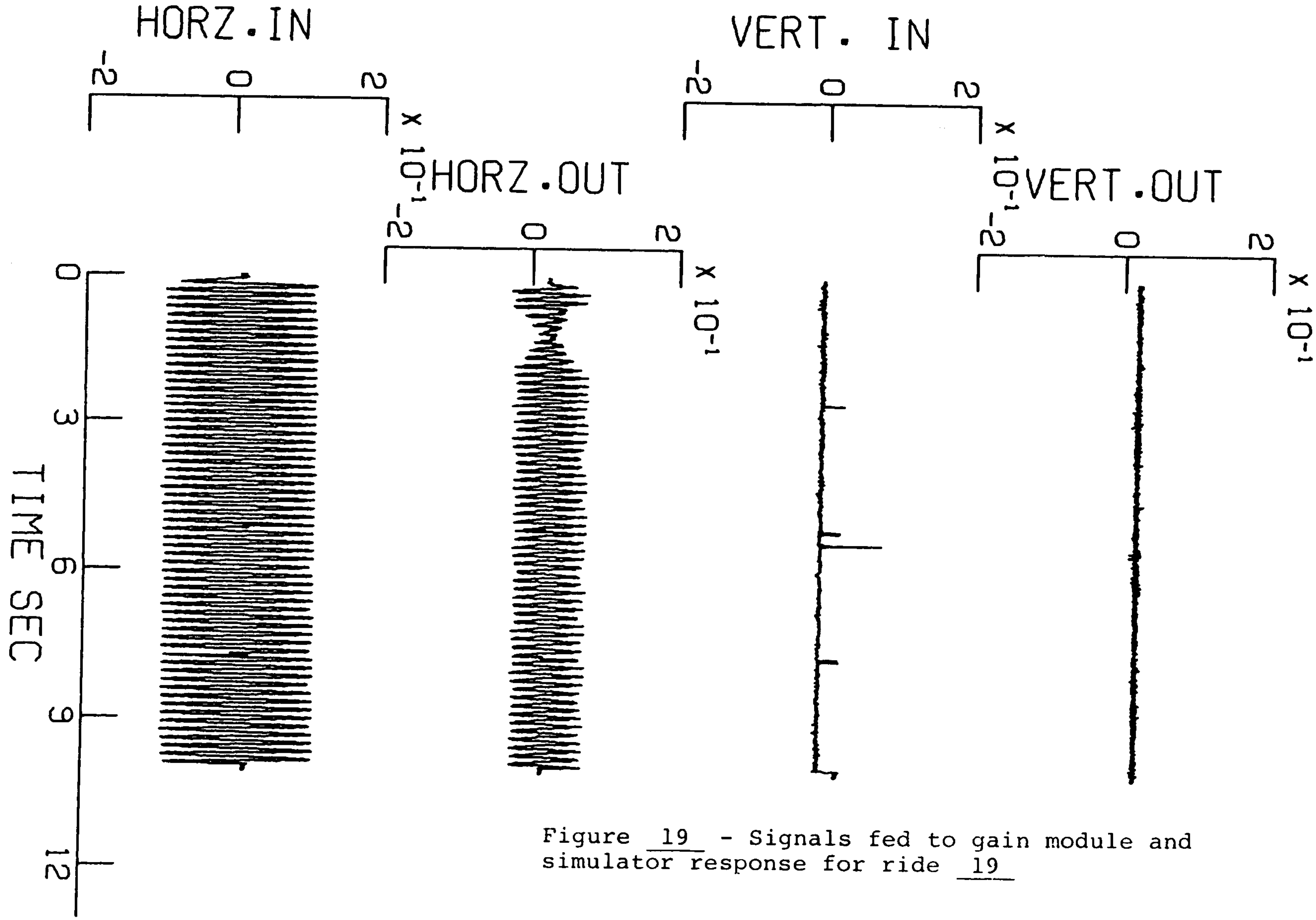

Figure 19 - Signals fed to gain module and simulator response for ride 19

RMS VERTICAL CONTRIBUTIONS BY HALF OCTAVE

HALF OCTAVE BOUNDARY FREQUENCIES

0.25 0.35 0.50 0.71 1.00 1.41 2.00 2.83 4.00 5.66 8.00 11.31 16.00 22.63 32.00 45.25 64.00

RIDE																
1	.00016	.00020	.00047	.00132	.00393	.00817	.00449	.00530	.00670	.00402	.00663	.00391	.00477	.00398	.00129	.00146
2	.00008	.00026	.00054	.00177	.00758	.00295	.00234	.00216	.00167	.00299	.00275	.00344	.00363	.00419	.00111	.00156
3	.00037	.00050	.00047	.00191	.00772	.00999	.00418	.00299	.00427	.00418	.00346	.00429	.00471	.00408	.00116	.00158
4	.00024	.00083	.00232	.00539	.00831	.01492	.00912	.00702	.00642	.01645	.00719	.00713	.00675	.00433	.00201	.00174
5	.00010	.00059	.00241	.00626	.00568	.00506	.00527	.00489	.00473	.00576	.00397	.00455	.00402	.00302	.00155	.00165
6	.00018	.00099	.00137	.00537	.01134	.00873	.00706	.00602	.00775	.00677	.00562	.00497	.00464	.00318	.00138	.00157
7	.00023	.00163	.00520	.01172	.03135	.02142	.01183	.00834	.00740	.00821	.00662	.00627	.00596	.00487	.00239	.00192
8	.00029	.00178	.00326	.00687	.01940	.02027	.01198	.00584	.00473	.00635	.00578	.00489	.00520	.00383	.00168	.00163
9	.00023	.00103	.00207	.00360	.00912	.00921	.00803	.00682	.00707	.00595	.00584	.00420	.00466	.00366	.00171	.00161
10	.00003	.00037	.00075	.00233	.00829	.01715	.00527	.00725	.00786	.00562	.00349	.00566	.00748	.00479	.00247	.00164
11	.00031	.00062	.00063	.00393	.01129	.02527	.00719	.00458	.00291	.00499	.00345	.00450	.00517	.00400	.00218	.00168
12	.00046	.00086	.00227	.00314	.01550	.01505	.00542	.00506	.00415	.00437	.00327	.00340	.00406	.00313	.00154	.00160
13	.00005	.00034	.00111	.00142	.00510	.00742	.00527	.00811	.00458	.01008	.00697	.00669	.00579	.00351	.00161	.00157
14	.00007	.00007	.00035	.00161	.00489	.00336	.00311	.00581	.00279	.00398	.00275	.00298	.00267	.00210	.00100	.00147
15	.00017	.00041	.00049	.00175	.00532	.00581	.00669	.00784	.00303	.00398	.00245	.00319	.00320	.00251	.00120	.00150
16	.00018	.00027	.00092	.00161	.00400	.00365	.00533	.00576	.00432	.00410	.00408	.00565	.00307	.00246	.00123	.00148
17	.00008	.00017	.00047	.00180	.00749	.00531	.00468	.00908	.00369	.00503	.00409	.00624	.00344	.00270	.00131	.00154
18	.00009	.00013	.00016	.00033	.00058	.00047	.00046	.00059	.00330	.02330	.00080	.00160	.01476	.00614	.00128	.00162
19	.00003	.00002	.00003	.00005	.00007	.00013	.00025	.00012	.00017	.00018	.00049	.00067	.00097	.00042	.00055	.00137

RIDE	OVERALL RMS VALUE
1	0.01718
2	0.01206
3	0.01729
4	0.03074
5	0.01666
6	0.02312
7	0.04574
8	0.03454
9	0.02180
10	0.02597
11	0.03116
12	0.02505
13	0.02114
14	0.01167
15	0.01527
16	0.01406
17	0.01778
18	0.02860
19	0.00203

Figure 20 - RMS value of vertical component of each ride by half octave band and overall

RMS LATERAL CONTRIBUTIONS BY HALF OCTAVE

RIDE	0.25	0.35	0.50	0.71	1.00	1.41	2.00	2.83	4.00	5.66	8.00	11.31	16.00	22.63	32.00	45.25	64.00
	HALF OCTAVE BOUNDARY FREQUENCIES																
1	.00035	.00053	.00076	.00155	.00660	.01322	.00438	.00449	.00474	.00508	.00559	.00605	.00401	.00228	.00093	.00072	
2	.00011	.00041	.00243	.00293	.00529	.00297	.00271	.00407	.00512	.00802	.00589	.00753	.00400	.00182	.00086	.00074	
3	.00007	.00032	.00042	.00268	.00746	.00643	.00880	.00362	.00414	.00594	.00418	.00627	.00412	.00187	.00089	.00075	
4	.00007	.00019	.00051	.00154	.00263	.00382	.00418	.00333	.00425	.00545	.00360	.00445	.00284	.00160	.00090	.00074	
5	.00004	.00013	.00082	.00193	.00174	.00268	.00111	.00203	.00318	.00346	.00289	.00339	.00239	.00128	.00066	.00070	
6	.00015	.00017	.00046	.00218	.00238	.00349	.00381	.00438	.00469	.00605	.00509	.00495	.00412	.00169	.00084	.00069	
7	.00003	.00024	.00151	.00221	.00153	.00703	.00289	.00459	.00524	.00532	.00424	.00484	.00329	.00173	.00089	.00074	
8	.00006	.00014	.00069	.00176	.00169	.00336	.00223	.00419	.00476	.00515	.00408	.00439	.00325	.00165	.00091	.00075	
9	.00010	.00012	.00053	.00235	.00353	.00226	.00422	.00453	.00602	.00565	.00612	.00622	.00445	.00174	.00091	.00078	
10	.00015	.00075	.00061	.00430	.01184	.01218	.00532	.00518	.00452	.00475	.00605	.00777	.00466	.00156	.00088	.00073	
11	.00008	.00031	.00140	.00290	.00668	.00702	.00430	.00361	.00359	.00513	.00443	.00552	.00427	.00173	.00082	.00073	
12	.00005	.00024	.00068	.00564	.00286	.00190	.00252	.00240	.00316	.00339	.00275	.00265	.00284	.00182	.00068	.00071	
13	.00013	.00046	.00085	.00135	.00267	.01313	.00620	.00266	.00236	.00314	.00360	.00283	.00238	.00149	.00077	.00072	
14	.00003	.00023	.00054	.00152	.00770	.00376	.00427	.00692	.00296	.00355	.00304	.00330	.00237	.00139	.00067	.00065	
15	.00023	.00050	.00051	.00343	.00738	.00414	.00391	.00738	.00297	.00230	.00260	.00247	.00227	.00161	.00071	.00063	
16	.00017	.00042	.00114	.00216	.00635	.00154	.00224	.00340	.00164	.00166	.00329	.00379	.00216	.00119	.00069	.00065	
17	.00016	.00042	.00106	.00304	.00444	.00135	.00181	.00332	.00217	.00239	.00259	.00416	.00173	.00117	.00056	.00063	
18	.00007	.00005	.00004	.00007	.00017	.00039	.00035	.00041	.00074	.00656	.00176	.00208	.00148	.00156	.00105	.00074	
19	.00006	.00018	.00022	.00019	.00041	.00043	.00029	.00090	.00401	.02657	.00232	.00400	.00238	.00518	.00225	.00097	

RIDE	OVERALL RMS VALUE
1	0.02000
2	0.01667
3	0.01805
4	0.01194
5	0.00838
6	0.01366
7	0.01417
8	0.01183
9	0.01517
10	0.02300
11	0.01574
12	0.01004
13	0.01657
14	0.01386
15	0.01376
16	0.01015
17	0.00927
18	0.00760
19	0.02798

Figure 21 - RMS value of lateral component of each ride by half octave band and overall

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TECHNIQUES FOR OBTAINING SUBJECTIVE RESPONSE TO VERTICAL VIBRATION

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SUMMARY

Laboratory experiments were performed to validate the techniques used for obtaining ratings in the field surveys carried out by the University College of Swansea. In addition, attempts were made to evaluate the basic form of the human response to vibration. The paper describes some of the results obtained by different methods and compares them.

INTRODUCTION

In the summer of 1972 a series of tests was planned for a vertical vibrator then newly installed in the Mechanical Engineering Department, the University College of Swansea. The aim of these tests was to investigate the nature of the subjective response to human beings to vertical vibration over the frequency range from 1 to 70 Hz and over a range of vibration amplitudes corresponding roughly to those recorded in public transport vehicles. In parallel with the laboratory tests a series of questionnaire surveys was being carried out on a variety of transport vehicles in which passenger reaction to the vibratory motion of these vehicles was obtained using rating-line techniques.

Over the years many tests have been carried out to determine human reaction to vibration over all or part of the relevant frequency range. However, the published results show what can only be described as a remarkable inconsistency. The literature up to 1970 has been reviewed by R. M. Hanes (ref. 1) and demonstrates wide variations in reported results. For example, values of sensation threshold for vertical vibration reported by different authors covered a range 1 to 1000 at some frequencies. The differences appear to arise largely from the use of inappropriate, imprecise and sometimes inadequate methods and equipment.

At the time several other workers were known to be beginning research programmes covering similar ground to that proposed by the authors. It was

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felt appropriate to spend time investigating in some detail the experimental methods which would be used, rather than to select a method, carry on with the tests, and possibly produce yet another questionable series of data. In addition a need had arisen for a laboratory survey of rating methods which could be used realistically in questionnaire studies of fare-paying passengers in public service vehicles. This also pointed to a detailed study of methods.

This paper describes some of the experiments conducted and the conclusions which have been drawn about the usefulness of the techniques. It is effectively in two sections with two subsections.

The first section describes in some detail the validation of the rating-line technique which has been used extensively in the questionnaire studies. A brief section then follows describing some experiments designed to test the usefulness of a cross-modality approach, using noise as the matching sensation, for field studies and discusses the reasons why it was abandoned.

The second main section discusses the findings of a short series of experiments in which reaction to vertical motion was investigated using three different psychophysical methods. These were magnitude estimation, fractionation (halving) and multiplication (doubling). The three methods were used to determine exponents for a power-law expression, and the differences in the results obtained are described and discussed. The final section discusses some of the difficulties met in equating sensations across frequencies and indicates a possible way by which they may be overcome.

EQUIPMENT

The equipment used for all laboratory experiments, except the cross-modality check, was a Unidyne electrohydraulic actuator capable of achieving ± 0.115m (± 4.5 in.) displacement along a vertical axis at low frequencies. The operating boundary was set by a velocity limit of about 0.5 m/sec (20 in./sec) at higher frequencies.. The usable frequency range extended up to about 100 Hz. For the experiments discussed here the operating conditions were kept well within the capabilities of the system.

For the majority of the experiments subjects were standing on a flat plate mounted directly on the top of the actuator. The acceleration of the plate was sensed by a piezoelectric accelerometer mounted directly to the plate beneath the subject's right foot. The accelerometer, calibration table and measuring amplifiers were all Bruel and Kjaer equipment.

For the brief series of cross-modality tests the subjects were seated on an ordinary, adjustable-height, office chair with the balls of their feet resting on a plate driven by a small electromagnetic vibrator. The main weight of the subjects feet and legs was taken on a stationary platform surrounding the plate driven by the shaker.

For all tests the input acceleration waveforms were sinusoidal.

RATING-LINE METHOD

Several varieties of rating line were used in the questionnaire surveys and are discussed in Oborne and Clarke (refs. 2 and 3). Broadly speaking it was found that the way in which the rating line was divided hardly influenced the results, whereas the effect of the scale ends could be classified in a meaningful way which followed the predictions of intuition or "common sense". A summary of the investigations of these aspects of rating-line techniques is presented elsewhere in this compilation (ref. 4).

The present discussion centers on an attempt to discover just how much information could be obtained from results obtained using an unstandardised rating line. The main points of interest were the repeatability of such ratings and how closely they match similar results obtained by other approaches. Full details of the experiments are given in Oborne and Clarke (ref. 5). This section summarizes that paper.

The rating line used in the experiments is shown in figure 1, together with some information about the test subjects and the test conditions used for the first part of the experiment. The rating lines were presented to the subjects in book form, one line per page, so that each rating was made on a fresh page. Each subject was given the test conditions in a random order. No attempt was made to define the scale ends "smooth" and "rough", nor were any of the test conditions designated as standards. Thus, any standards used in rating judgements were based entirely on the individual subject's background and history and any preconceived ideas he or she may have had about the experiment.

The rating was taken as the distance of the mark in cm from the "smooth" end of the scale. For each test stimulus a mean rating was determined. Figure 2 shows some of the plots of rating against root-mean-square acceleration of the input vibration for specific frequencies. It can be seen that for all stimuli at the same frequency the points plot close to a straight line. The figure includes the best and the worst of the frequency plots, the best being at 11 Hz with a product moment correlation coefficient of 0.999 and the worst being at the ends of the frequency range with correlation coefficients of 0.972 and 0.979 at 3 Hz and 80 Hz respectively. Table I shows the values obtained for the coefficients in the regression equation for the frequencies tested, together with the correlation coefficients.

Figure 3 is obtained from the regression lines, using the argument that equal ratings indicate equal subjective effects. The contours, which are actually equal rating contours, can then be regarded as equal comfort contours or equal reaction contours. The parameters of the contours are specific distances along the rating line. Some indication of the validity of this assumption is gained from examination of figure 4 which shows the curves of figure 3 compared with typical results obtained by two other investigators, Chaney (ref. 6) and Jones and Saunders (ref. 7). It is seen that the shapes of the contours obtained using the rating lines agree fairly well with both of these other curves. It is useful to note that the Chaney curve was obtained by asking subjects to indicate when a vibration had reached a "mildly annoying" level, whereas Jones and Saunders asked subjects to adjust variable stimuli to a reference vibration at 20 Hz. The agreement looks good, at least good enough to justify further development of the unstandardised rating-line approac. For interest's sake the ISO 1 minute reduced-comfort boundary is also indicated (ref. 8).

COMBINATION OF LINE RATINGS WITH SEMANTIC SCALES

The next stage in the test process was to relate the rating contours to other measures of subjective reaction, bearing in mind the fact that the intention was to use such rating lines in field studies. It was decided to investigate the relationship between such line ratings and ratings obtained on a six-point category scale. Figure 5 shows the categories chosen for the test, together with some basic information about the subjects and stimuli used.

Each subject gave a rating of each stimulus on the six-point comfort scale. Each stimulus was allocated the label given most frequently. Fifteen of the stimuli received nearly equal numbers of votes for two categories. These were left out of the subsequent analysis.

The stimuli were then grouped into the appropriate categories. For all the stimuli in a particular category the mean and standard deviation of the line ratings were found. Each category was then arbitrarily assigned a range of line ratings corresponding to one standard deviation on either side of the appropriate mean. The original rating line was thus divided into regions corresponding to the six comfort descriptors. The results are shown in figure 5.

The first point to note is that the two central categories ("fairly comfortable" and "fairly uncomfortable") can be coalesced into one since the overlap is so great. Second, but not unexpectedly in view of the small number of subjects used, there are overlaps and gaps in the groups. These were closed by halving the overlap or underlap to give a contiguous set of categories on the line. Since the boundaries of the categories were defined in terms of distances along the rating line, it was possible to obtain equal-sensation contours which indicated the boundaries of the descriptive zones. The final boundaries arrived at are shown in figure 6 with the positions of the indecisive stimuli indicated.

The final test in the sequence repeated the last stage except that the subjects were given definitions for the categories. Instead of being given just the bare list of categories, they were given a sentence or two of explanation for each category. Since the basic idea of the series of tests was to investigate a technique for use in vehicle studies in the field, the definitions were related to some aspect of travel, and travel time was the one that was actually achosen. Table 2 gives the category descriptors and summaries of the definitions used, and figure 7 shows the final contours obtained. In view of the overlap in the two central categories of the six-point word scale in part II, the scale was collapsed into a five-point scale which suited the choice of definitions very nicely. It can be seen that the use of defined categories has reduced the number of indecisive inputs by 50%, and by comparison with figure 6 it can be seen that the category boundaries are almost the same, the readjustments being relatively minor.

The work just described, which was conducted entirely in a laboratory, was followed by the use of the rating scales and techniques discussed here in a field study. This is described in some detail, together with some of the results, in another paper in this Symposium (ref. 4).

CROSS-MODALITY MATCHING OF NOISE AGAINST VIBRATION

Several times in the literature the suggestion has been made that an ideal way of obtaining the reaction of subjects to ride vibration is to use a method which involves the matching of an adjustable noise stimulus with the sensation of the ride perceived by the subject or passenger. This approach has a lot to recommend it since it is relatively easy to apply a noise stimulus using, for example, a standard audiometer such as is used for audiological screening.

It was decided that tests were necessary to discover how well subjects could match noise to vibration in relatively quiet laboratory situations and in circumstances similar to those which would be encountered in a typical passenger vehicle. Since some recordings of noise had been made in a passenger train, it was decided to use these to provide the background noise for the tests which required this.

The noise to be used for matching was provided by a standard audiometer. To test the effect of different sounds as matching media, three different types of acoustic noise were used. These were pure tones at 250 Hz and 1000 Hz and a broad band low frequency noise with all frequencies up to 200 Hz present. The noise levels could be adjusted in steps of 5dB, which was the standard step for the screening audiometer used.

Individual thresholds of hearing were determined for each of the three noise types. These were used as the appropriate datum points, so that comparisons were being made between vibration levels and matched noise levels above individual thresholds. Individual thresholds were used for correction since a degree of intersubject variation was present.

Figure 8 shows the results for the three noise types. A reasonable regression line was obtained from these results, bearing in mind that only twelve subjects were used. There is no significant difference between the results for the three types of noise used. Thus, provided corrections are made for individual subject thresholds, a good match can be obtained subjectively by using noise as a medium for rating vibration.

A test was then carried out in background noise. The noise used was provided by a recording of train noise played back to give a level of 75 dB(A) at the subject's head. For comparison the level at the subject's head during the "no noise" trials was 55 dB(A). Since there was no difference between the results obtained using three different stimuli in the no background noise situation, only the 1000 Hz tone was used for this trial.

Figure 9 shows the results. Two points are immediately obvious. The first is that the results with and without background noise are similar in shape, but that the results with background noise lie below those without background noise. However, the difference between the two levels is not constant, nor is it particularly consistent, so that corrections would be difficult, particularly in view of the fact that the corrections are of the same order of magnitude as the measured values.

It was decided at this stage to abandon the use of cross-modality matching against noise as a field technique because of these inconsistencies. It would appear, however, that this technique would work well in a controlled laboratory situation.

DETERMINATION OF THE EXPONENT IN A POWER-LAW REPRESENTATION OF SUBJECTIVE REACTION TO VIBRATION

For many years investigators have been obtaining curves which show how the physical measures of vibration intensity vary across the frequency range to produce equality of sensation. The different equal-sensation contours were generally described by using relatively vague semantic terms. Some attempts have been made to provide a combination formula to enable both frequency and level of vibration to be taken into account, but these had provided largely inconsistent results.

Stevens (ref. 9) has proposed that for all sensations which may be described in terms of their intensity, rather than quality, a very simple power law related the subjective sensation to the physical excitation. This law is expressed as $\psi = k\phi^n$ where ψ is the subjective magnitude, ϕ is the physical magnitude, n is the power-law exponent and k is a constant, which depends on the units used. He has determined values of the exponent n for many sensory modalities but not for whole-body vibration.

It was to be expected that if this law was valid for such sensations as light brightness, light colour, loudness of noise, touch and tactile vibratory sensation, then it would hold for whole-body vibration. Here ϕ would conveniently be taken as the root-mean-square acceleration of the vibration excitation, and in parallel with well-documented results from other modalities, it could be expected that the exponent n would vary somewhat with frequency.

Stevens has used various techniques for establishing the appropriate values of the constants in the power-law expression but had largely concentrated on magnitude estimation or on fractionation/multiplication methods. In using the magnitude estimation method the subject is exposed first to a standard stimulus and instructed to consider it as having a magnitude of 10. The subject is then given the test stimulus and is asked for a numerical assessment. For the fractionation method the subject is provided alternately with a standard stimulus and a variable one and is asked to adjust the variable one to provide half the sensation of the standard.

Shoenberger and Harris (ref. 10) carried out the first well-documented, systematic attempt to evaluate the effect of change in physical level of whole-body vibration at a constant frequency on subjective sensation. They produced some reasonably consistent results by using the technique of magnitude estimation. However, their group of subjects was made up entirely of physically fit U.S. Air Force personnel, all of whom had previous experience of vibration experiments. In addition, the method of magnitude estimation had recently been attacked by Poulton (ref. 11) on the grounds, among others, that results obtained by this method appeared to be influenced heavily by the intensity range of variables used in the experiment. Although Shoenberger and Harris listed Poulton's paper in their references, they did not appear to take his comments into account.

An experiment was designed to see whether the results shown in reference 10 could be reproduced consistently using a range of subject types, including some women, and to see whether Poulton's strictures were valid. For this purpose the exponents for a range of frequencies of vibration were to be established using magnitude estimation, fractionation (halving) and multiplication (doubling) techniques. The results are shown in table 3 and in figures 10 and 11.

Table 3 shows the results of using the three methods to determine exponents for six of the frequencies investigated. The subject group sizes were 12 for the magnitude estimation experiments and 8 for the halving and doubling experiments. The exponents quoted are appropriate mean values of the individual results. It can be seen that there are clear differences between the exponents obtained by the three methods. The results using halving and doubling, although different from each other are consistently higher than those obtained using a magnitude estimation technique. The questions then arose as to which was the appropriate value to take or how one averaged the different results to obtain an adequate result.

Figures 10 and 11 show in more detail the results obtained for the frequencies of 7 Hz and 50 Hz, respectively. Figures 10(a) and 11(a) show plots of assessed magnitude against root-mean-square acceleration on log-log scales, so that the exponent is the slope of the best straight line through the data. Figures 10(b) and 11(b) show plots of the standard stimulus against the stimulus assessed by the subject as providing half the sensation provided by the standard. From the slopes of these plots the exponents can be determined using the expression $n = \log 2/\log$ (slope).

Looking first at figures 10(a) and 11(a) several interesting points stand out. First, the mean values of each set of data appear to lie consistently on a curve,one which is admittedly not too distant from a straight line. Second, the standard stimulus, if plotted, lies consistently below the best straight line in such a way that if the standard is included as a data point, the curve is a cusped curve showing two loops. Third, the scatter of the data points increases consistently as the stimulus moves away from the standard. Closer investigation of the lines and data points provided by Shoenberger and Harris in their paper and insertion of their standards on the figures indicate exactly the same trends, save that they quote no value for scatter. These effects are, roughly speaking, those predicted by Poulton and they form the basis of his assertion that determined exponents can be made to have values over a fairly wide range by careful choice of the range of experimental values.

Looking at figures 10(b) and 11(b) two indications appear. First, the median values for each stimulus lie almost exactly on a straight line for the two examples, as is the case with all sets of results. Second, the scatter is significantly less than that obtained with the magnitude estimation results. In this connection it should be noted that very precise exponents can be obtained by fractionation, the range obtained for a given frequency covering a ratio of 1.25 to 1.00 for individual exponents. This compares favorably with a ratio of 2 to 1 or 3 to 1 for exponents for individuals obtained by magnitude estimation.

Similar considerations apply to the results obtained using the doubling method except that the scatter is greater than the scatter for results from halving while still being considerably less than that for results obtained by magnitude estimation.

Part of the difference between exponents obtained by halving and by doubling could be explained by a time effect. This effect is such that in many psychophysical experiments in which two stimuli are matched one against the other, there is a consistent bias in the level of the second stimulus. The indications here are that the second stimulus is consistently matched high; some experiments in which a subject was asked to equate two vibrations at the same frequency produced matching responses consistently about 10 percent higher than the standard. This would account for a large part of the difference between the slope obtained by halving and by doubling and would also account for the standard in magnitude estimation results being consistently below the line.

It seems appropriate to take the mean of the halving and doubling results to correct for this effect. If this is done the resulting exponent is still higher than that obtained from magnitude estimation.

An appropriate way of checking the correctness of the exponents is to select from a set of equal-sensation contours one to be used as a datum, and to use the power law with appropriate exponents to predict the other equal-sensation curves. Using this procedure Shoenberger and Harris obtained fair agreement between predicted and determined contours. Even better agreement is obtained if the exponents produced by averaging the halving and doubling results, as discussed here, are used. Checks on some other experimental data give the same result. Unfortunately the work had to be discontinued at this point, and time and manpower have not yet enabled it to be picked up again.

DEFINITION OF EQUAL-SENSATION CONTOURS FOR SINUSOIDAL INPUTS

The last section referred to equal-sensation contours for vibration responses at different frequencies. Although many investigators have been producing such contours in recent years, the tragedy is that there is still disagreement as to the basic shapes of the curves in that there appear to be two groups of results. These will not be discussed here since they have been extensively discussed elsewhere (see, for example, ref. 12).

Some experiments were carried out at Swansea to obtain equal-sensation contours to be used with results for exponents such as those discussed in the previous section. The method used was the now standard technique whereby vibration at an appropriate level at some arbitrarily chosen standard frequency is matched by sensations from vibrations over a range of frequencies. Reasonable results were obtained, but it became increasingly obvious that subjects did not enjoy matching vibrations at 2 Hz with vibrations at 10 Hz for example, because of the differences in the sensations. Accordingly, it was felt that matching vibration sensation would be easier if a sequence of standards was used, stepping across the frequency range in short steps.

A student in the Department of Psychology at Swansea was given the task of investigating this method whereby the matching is always done using adjacent frequencies in the range of values being investigated (ref. 13). The result of one rating becomes the "standard" for the next, and so on until by this monotonic stepping process the frequency range of interest is covered. As a check the process is then reversed and the frequency range traversed back to the beginning. Ideally, the curves obtained from increasing and decreasing frequencies should coincide and the final point should be the same as the starting point.

Figure 12 shows two sets of test data from the 14 obtained. It can be seen that the match between the curves for increasing and decreasing frequencies is fairly good. The two curves shown are typical of all the results in that the 14 curves obtained fall roughly into two groups. In one there is fairly marked frequency dependence, particularly for the higher frequencies. In the other the curves are fairly flat in terms of frequency dependence. These two types of individual response curves match roughly the two types of group data which have been published.

Finally figure 13 shows a plot of the curve of the means of the 14 subjects (each providing two points at each frequency), together with the scatter band.

CONCLUDING REMARKS

Experiments are described which were designed to test the efficiency of methods which could be used to obtain human response to vibration inputs, particularly in passenger vehicle situations.

The rating-line method, which had been found to be one of the most easily used in field situations, was used to generate equal sensation contours whose shapes matched those obtained by other methods. The numerical values of line ratings have not, so far, been linked directly with numerical estimates of intensity obtained by, for example, magnitude estimation methods.

The use of magnitude estimation techniques was examined in some detail in view of criticisms made by Poulton and others and in view of the fact that many investigators were beginning to use these techniques for obtaining subjective reactions to vibration. It was found that, for situations where they could be used, the methods of halving and doubling gave consistently different results with higher exponents for a power-law expression than those provided by magnitude estimation. From an engineering point of view the indication is that magnitude estimation results are probably not conservative and should, therefore, be viewed with caution.

ACKNOWLEDGEMENTS

Thanks are due to the Science Research Council of Great Britain for financial support, to Dr. M. McCullough for his work on the psychophysical aspects of the study, to those who consented to act as subjects, to D. A. Humphries for permission to quote from his thesis and to Professor C. E. M. Hansel and Professor F. T. Barwell of the University College of Swansea for advice and support.

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TABLE 1. - VALUES OF REGRESSION EQUATION COEFFICIENTS AND CORRELATION COEFFICIENTS OBTAINED FOR FREQUENCIES TESTED

[Rating = aX + b   X = rms acceleration in m/sec^2]

Frequency, Hz	a	b	Correlation coefficients
3	5.525	5.167	0.972
5	6.592	5.864	0.985
7	5.676	5.530	0.996
9	5.493	5.044	0.997
11	6.224	4.647	0.999
15	5.771	4.536	0.998
20	5.126	4.155	0.996
30	5.655	3.124	0.989
40	5.579	2.279	0.990
60	3.930	1.763	0.988
80	3.880	1.699	0.979

TABLE 2. - DEFINITIONS USED FOR DESCRIPTORS FOR PART III OF STUDY

[Seven undergraduates (different from those of part II) were used as test subjects for part III; seventy-five stimuli (identical to those of part II) were used]

Descriptor	Definition
Very comfortable	For a long journey
Comfortable	For a journey of about 1 1/2 hours
Just comfortable	For a journey of not more than 1/2 hour
Uncomfortable	Only if the journey was very short
Very uncomfortable	Would not use this form of transport

TABLE 3. - POWER-LAW EXPONENTS BY EXPERIMENT

Frequency, Hz	Magnitude Estimation	Fractionation Method	
		Halving	Doubling
3	1.08	1.24	1.51
5	1.08	0.99	1.46
7	0.94	0.98	1.28
20	0.90	0.79	1.12
30	0.78	0.78	1.54
50	0.82	0.96	—

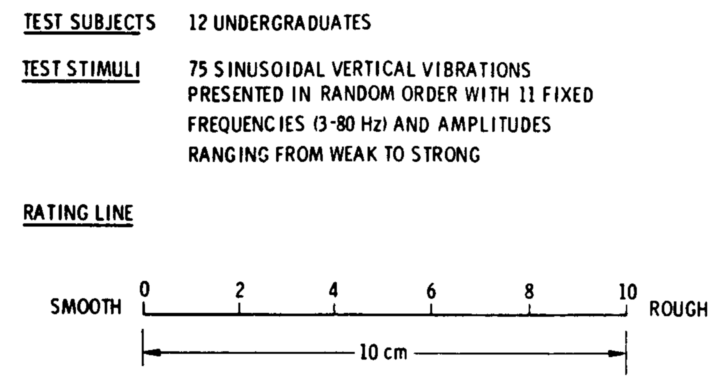

Figure 1.- Rating line used, together with details of subjects and stimuli for part I of present study.

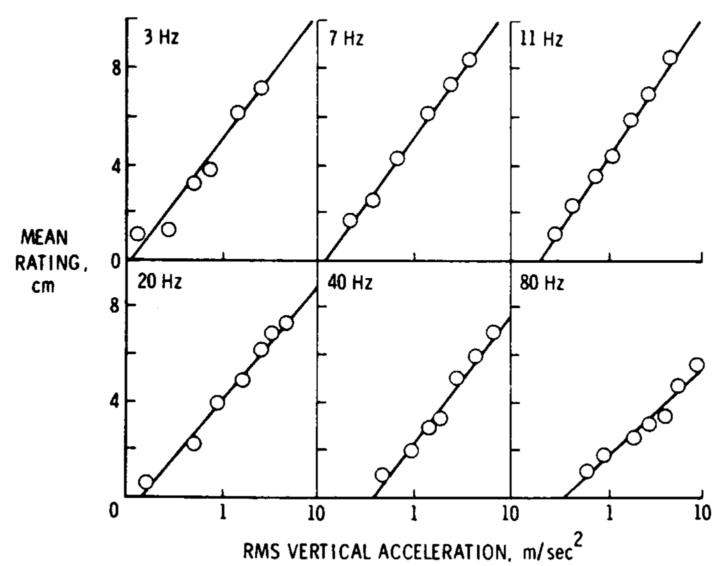

Figure 2.- Plots of mean rating against vibration level for six of the frequencies used, together with relevant regression lines.

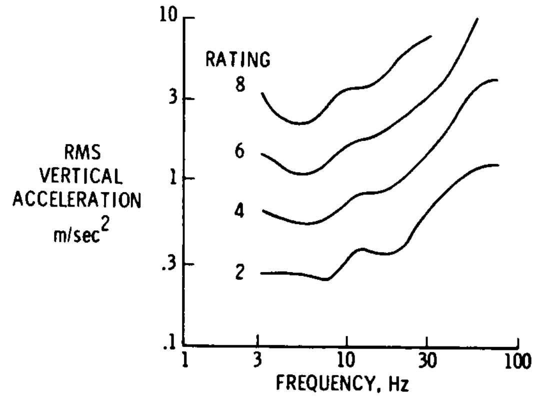

Figure 3.- Equal rating contours (equal comfort contours).

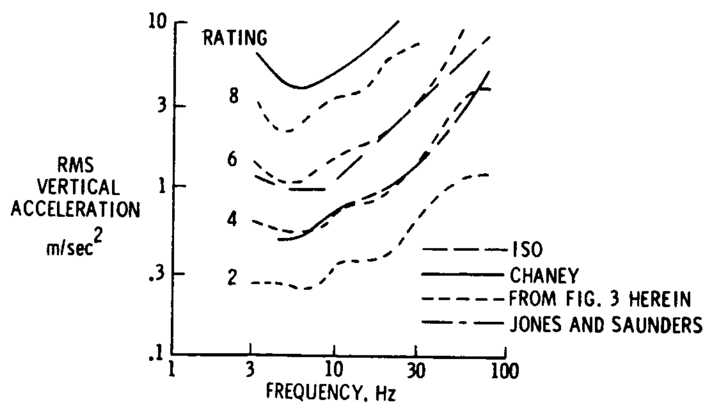

Figure 4.- Comparison of equal rating contours of present paper with results of Chaney and Jones and Saunders and with the ISO 1 minute reduced-comfort boundary.

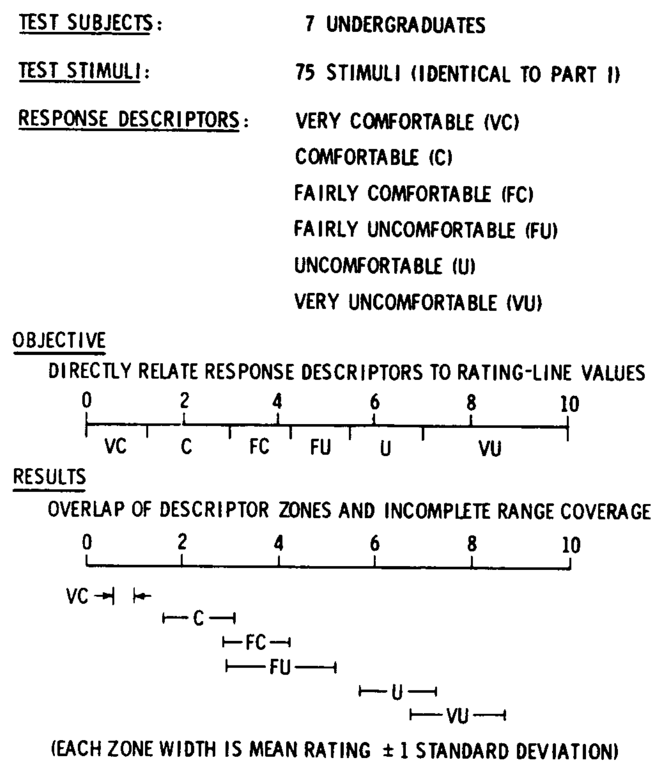

Figure 5.- Results of matching semantic descriptions of vibration stimuli with ranges of rating-line responses, together with details of subjects and stimuli, for part II of present study.

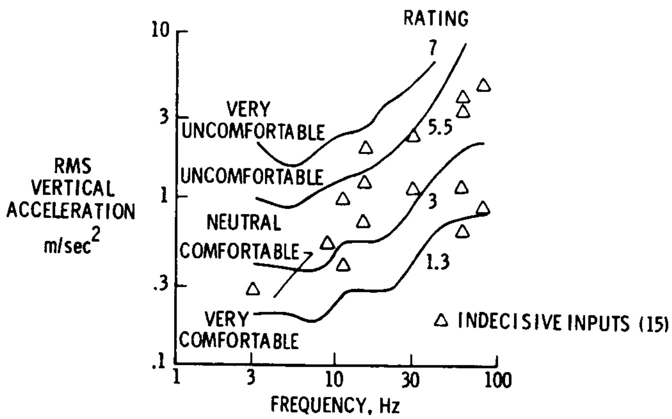

Figure 6.- Equal comfort zones obtained using undefined descriptors.

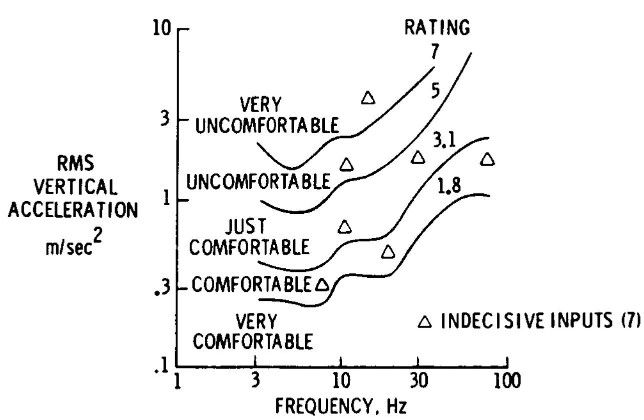

Figure 7.- Results of matching semantic descriptions of vibration stimuli, using defined categories, with ranges of rating-line responses.

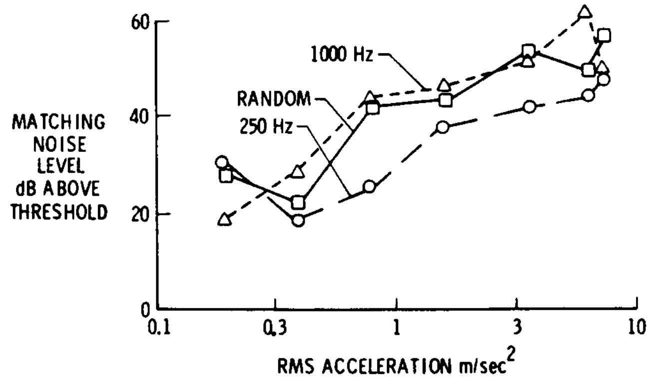

Figure 8.- Cross-modality match of noise level with vibration. (Average of 12 test subjects.)

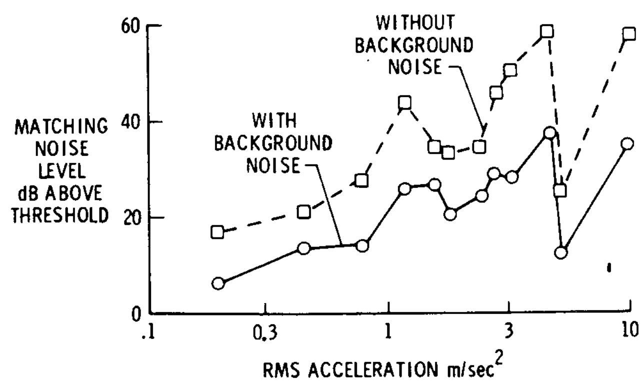

Figure 9.- Effect of background noise on cross-modality matching of vibration with 1000 Hz tone.

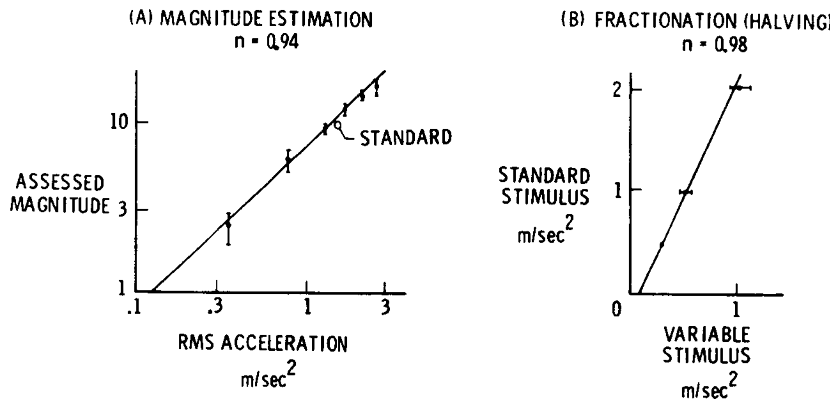

Figure 10.- Results for tests at 7 Hz with magnitude estimation and fractionation.

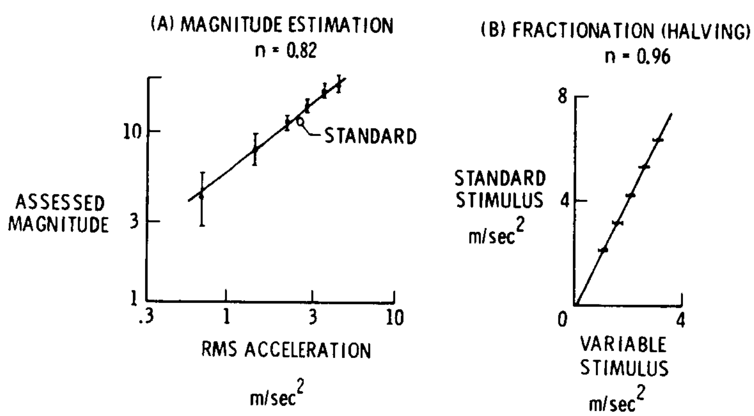

Figure 11.- Results for tests at 50 Hz with magnitude estimation and fractionation.

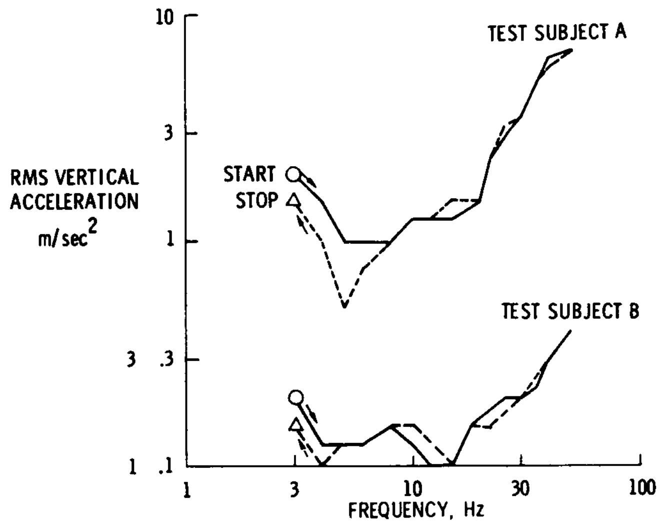

Figure 12.- Typical individual results obtained by progressive matching. (Leftmost scale is for subject B.)

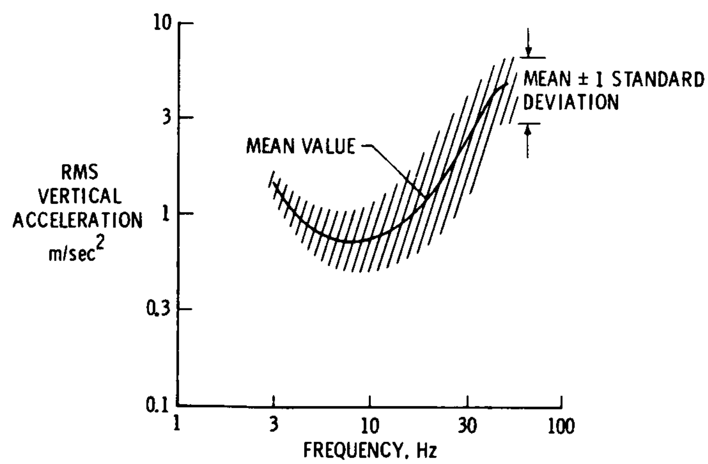

Figure 13.- Average results for 14 subjects obtained by progressive matching.

N76-16767

DEMOGRAPHIC AND PSYCHOLOGICAL VARIABLES AFFECTING TEST SUBJECT EVALUATIONS OF RIDE QUALITY

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SUMMARY

Two ride-quality experiments, similar in objectives, design, and procedure were conducted, one using the U.S. Air Force Total In-Flight Simulator and the other using the Langley Passenger Ride Quality Apparatus, to provide the motion environments. Large samples (80 or more per experiment) of test subjects were recruited from the Tidewater Virginia area and asked to rate the comfort (on a 7-point scale) of random aircraft motion typical of that encountered during STOL flights.

Test subject characteristics of age, sex, and previous flying history (number of previous airplane flights) were studied in a two by three by three factorial design. Correlations were computed between one dependent measure, the subject's mean comfort rating, and various demographic characteristics, attitudinal variables, and the scores on Spielberger's State-Trait Anxiety Inventory.

An effect of sex was found in one of the studies. Males made higher (more uncomfortable) ratings of the ride than females. Age and number of previous flights were not significantly related to comfort ratings. No significant interactions between the variables of age, sex, or previous number of flights were observed.

Of the demographic and attitudinal variables, the only ones which correlated to the mean comfort ratings were attitude toward flying and the state anxiety score (a measure of the person's anxiety level during the test flight or ride).

In both experiments there was a high degree of reliability between the ratings of the same motion, when these motions were repeated after a relatively short time interval.

INTRODUCTION

Most investigations of the human response to, or sensitivity to, motion have used as subjects a small number of people selected primarily because of their availability, not because they represented a population of interest (ref. 1). The results of the studies reviewed by Hanes showed that threshold values of even one-degree-of-freedom sinusoidal motion differed considerably.

Hanes suspected a relationship between individual (subject) characteristics and responses to motions. If identifiable subject variables, such as age, sex, flight experience, are significantly related to subjective comfort ratings of motions, then these variables would have to be considered when conducting tests to determine ride comfort levels. The Hampton Institute researchers decided to test for such relationships by initiating a series of tests with the following objectives:

(1) To determine the relationship of the age, sex, and previous flying experience of the test subject to his comfort ratings of aircraft motion via an experimental design

(2) To determine the effect of other demographic and attitudinal variables via a correlation design

(3) To assess the anxiety level of each participant and its contribution to his reported comfort

In order to accomplish these objectives two experiments were conducted. One involved the U.S. Air Force Total In-Flight Simulator (TIFS) and the other, the Langley Passenger Ride Quality Apparatus (PRQA), a ground-based simulator. These two experiments provided the opportunity to test a wide range of frequencies and various degrees of freedom of motion. A detailed description of the TIFS and the PRQA and their performance characteristics is found in references 2 and 3, respectively.

The two experiments discussed in this paper had, in addition to common objectives, similar design and procedure which are described.

SUBJECTS

Paid volunteers were recruited from the Tidewater Virginia area which consists of the cities of Hampton, Newport News, Norfolk, and Virginia Beach. The ages of the subjects ranged from 18 to 75 and the number of previous flights from 0 to over 50. The subjects also represented a relatively large variation in income, occupation, and education level.

Due to limitations of the time and cost involved, the subjects were not "trained" in the use of the scale used to rate motion. One consideration in selecting subjects is whether they should be trained in the use of the scale used to rate the motion environment, for it is possible that people make major changes in the way they use the scale over the first few experimental sessions. If the subjects then become quite consistent in the way they use the scale, it is advantageous to the researchers since it increases the reliability and decreases the variability of the data obtained using these subjects.

References 4 and 5, for example, have used a small number of trained subjects to collect data on the passenger acceptance of the motion of commuter airlines.

EVALUATION PROCEDURE

All subjects were informed of the importance of basing their ratings on the comfort or discomfort of the vibrations and not variations or changes in vibration, or other factors such as temperature and noise.

Individual, subjective comfort ratings were recorded by means of a hand-held paper scoring sheet attached between a revolving cardboard disc and clip-board. The disc was designed to prevent the subjects from seeing their ratings of previous ride segments.

A 7-point rating scale, with associated numerical integers, as well as semantic labels, was used by each subject to indicate his level of comfort or discomfort.

For the purpose of analysis, however, the 7-point rating scale was converted back to a 5-point scale in order to make direct comparisons of subject responses across simulators, since other simulator experiments had been or were being conducted by means of a 5-point scale, and also to have the data available to other experimenters who were using only the 7-point scale.

A preliminary study confirmed our hypothesis that the test subjects' frequency of using the two extreme values on either end of the 5-point scale would not change if the rating scale were enlarged to a 7-point scale which includes the categories of somewhat comfortable and somewhat uncomfortable. A comparison of the two scales follows:

5-point scale	Rating	7-point scale	Converted 5-point scale
5	Very uncomfortable	7	5
4	Uncomfortable	6	4
	Somewhat uncomfortable	5	3.5
3	Acceptable (neutral)	4	3
	Somewhat comfortable	3	2.5
2	Comfortable	2	2
1	Very comfortable	1	1

EXPERIMENTAL DESIGN AND PROCEDURE

Subject characteristics (variables) which were thought more likely to contribute to different ratings of the same motion were selected and the experiments designed to detect any such effects. Consequently, the variables of age, previous flying experience, and sex were studied using an experimental design, that is, a two by three by three factorial experiment in which people were selected to fit into the following cells:

		Previous Number of Airplane Flights		
		0 to 3	4 to 9	10+
	18 to 25			
Age	26 to 45			
	46+			

Approximately equal numbers of males and females were placed in each group.

Prior to the simulator experience each participant in the study filled out a questionnaire which asked for demographic information (age, height, weight, education, income, occupation, sex), previous flying history (number of flights, type of plane, frequency per year, susceptibility to motion sickness), and attitude about flying (is it enjoyable, is it preferred over other means of transportation, is it safe). The responses to these questions were used to determine whether any of these demographic, attitudinal, or experiential variables were significantly correlated to the comfort rating of random aircraft motion. The questionnaire was designed to be quite similar to that used by the University of Virginia research team to survey users of commercial short-haul airlines (passengers filled out the survey while on board a flight) as well as potential users in a ground-based survey (ref. 6).

Since our test subjects were given no practice trials or other experience in the use of the rating scales, subjects were exposed to two "rides," the second of which was identical in whole or in part to the first and separated by a 30- to 60-minute interval. This procedure allowed us to check for consistency of responding.

A post-ride questionnaire provided an overall evaluation of each subject's reaction to the simulator experience. The third questionnaire, Spielberger's (ref. 7) State-Trait Anxiety Inventory (STAI) was administered to determine the amount of anxiety experienced while in the simulator (state-induced anxiety), and the amount of anxiety generally experienced by the subject (trait anxiety).

TIFS STUDY

Test Subject Profile

Eighty people participated in the Hampton Institute experiments on TIFS. Although the primary criteria for selection of test subjects were those of age, sex, and previous flight experience, our large subject pool allowed us to include people from many income and education levels and from a wide variety of occupations. The responses to the pre-questionnaire were used to compile a profile of the test subjects which included 41 females and 39 males. Figures 1 to 6 give the distributions for the demographic variables of age, sex, education, number of previous airplane flights, occupation, and income.

The distribution for each of these variables approximates that for the general flying public except for sex and income. (See ref. 6.) The general flying public is comprised of 75 percent, not 50 percent, males and has a median income of $22 000.

In response to the question about attitude toward flying, 76 percent said they enjoy it, 14 percent feel uneasy, 4 percent fear flying, and 6 percent were not sure (fig. 7). A ground-based sample of over 500 regular users of commercial airlines (ref. 6) had the following distribution: 60 percent enjoy flying, 35 percent have no strong feelings, and 4 percent dislike it. Responses to a questionnaire handed out during commercial flights (ref. 6) showed that 45 percent of that sample of 750 like flying. The TIFS test subjects consisted of a higher proportion of people who enjoy flying than either of these samples, a result to be expected when using people who volunteer to be part of a research program of this type.

TIFS Ride Environment

Investigators at the University of Virginia have measured the motion environment of a variety of STOL aircraft used by commercial airlines (ref. 4). They recorded 2-minute segments of the aircraft's motion at random intervals throughout a flight. The segments ranged from smooth, straight-and-level flight through extreme turbulence. Two investigators rated each segment for comfort with a 5-point rating scale (very comfortable to very uncomfortable). This data base was used to provide the motion environment for the TIFS aircraft.

Since the interest was in simulating the whole range of comfort conditions, the inflight comfort ratings were used to construct the motion environment for the flights. "Typical" (as determined quantitatively from g_{rms} and deg/sec values) segments were selected from those which had been given a subjective rating of 1, 2, 3, 4, and 5, that is, from very comfortable to very uncomfortable. The segments were then strung back-to-back to produce an uninterrupted flight approximately 50 minutes in duration.

Previously, a 16-minute tape (constructed as described in the preceding paragraph) had been used to drive a 6-degree-of-freedom (DOF) Langley-based simulator, the Visual-Motion Simulator (VMS), and comfort ratings of the motion had been collected. This 16-minute "standard" tape was also included as part of the TIFS study to see if responses to it would be the same whether the subject was on the ground or in the air. The motion environments of the TIFS flights were as follows:

Takeoff and climb to altitude	"Standard" 6-DOF tape from VMS (eight 2-min segments)	Rest (4 min)	Various 6-DOF motions (twelve 2-min segments)	Return to Langley
Motion of plane is now driven by the tapes			Off computer-driven system	

Flight 1

The first 16 minutes of flight 2 consisted of single-, 2- or 3-DOF motions in an attempt to determine the way in which people integrate multiple-DOF motions. The remaining 24 minutes of programed motion (segments 9 to 20) were the same as that of flight 1.

Takeoff and climb	Single-, 2- or 3-DOF motions (eight 2-min segments)	Rest (4 min)	Same motions as flight 1 (twelve 2-min segments)	Return to Langley
On computer system			Off computer system	

Flight 2

The g_{rms} values for the linear DOF and deg/sec values for the angular DOF actually produced by TIFS are given in tables 1 and 2. Table 1 presents the means and standard deviations of the motion components of the 6 first flights. Table 2 gives the same values for segments 9 to 20 of the 4 second flights. Data from the first 8 segments containing 1, 2, or 3 DOF are now being analyzed.

Segments 9 to 20 of flights 1 and 2 were programed to be identical, but, as tables 1 and 2 reveal, the mean values for each DOF for each segment were close but not exactly the same. A Spearman's ρ was calculated for each DOF by comparing the means of segments 9 to 20 of flight 1 to those of flight 2. As table 3 shows, there was a high positive correlation for each of the 6 DOF, indicating that the motion of flights 1 and 2 were indeed quite similar.

Test Procedure

Subjects assembled at the NASA facilities at Langley were told the purpose of the experiment and instructed how to rate the motion. This part of the procedure, as were as many others as possible, was standardized so that all subjects were given the same information and/or experience. After being informed that they might be video taped, subjects were seated on the plane according to a prearranged seating plan which randomly assigned people to seats.

Ten to twelve test subjects participated in each flight. Subjects were selected so that an equal number of males and females, an equal number of inexperienced (0 to 9 previous flights) and experienced (10+ previous flights) air travellers, and all ages were represented on each flight. The purpose of this selection procedure was to make certain that differences in the motion of the aircraft due to natural turbulence or any change in procedure due to weather conditions were equally distributed across the subject variables of age, sex, and previous flying history.

After the airplane had climbed to altitude and begun straight and level flights, the test tape was engaged. Subjects were instructed when to begin an evaluation interval and when to record the comfort rating over the airplane's public address system by one of the experimenters. Twenty-second portions of each 2-minute segment were selected for rating by the test subjects. The interval between ratings thus varied from 90 to 120 seconds. During the 4-minute rest period of flight 1, the state section of the STAI was passed out to the test subjects with instructions to answer it according to how they were feeling at that time. Post-questionnaires were distributed and answered as the plane taxied in after each flight. The trait side of the STAI was administered after the second flight.

Although 80 subjects participated in the TIFS experiments, only 58 were used in the experiments reported in this paper. Changes in scheduling due to weather conditions permitted only 40 of these to ride a second flight.

Two passengers on each flight were video taped so that nonverbal cues of anxiety could be measured to see if these cues correlated with the passengers' self-report, that is, did they appear to be anxious even if they would not admit to so feeling.

Results

The means and standard deviations of the comfort ratings of the 58 people who experienced flight 1 and the 40 who took a second flight are given in table 4. In order to obtain a second (repeated) measure of the subjects' rating of a stimulus, segments 9 to 20 on the second flight were programed to be identical to the corresponding segments on flight 1. Such a procedure provides a test for reliability of responding (ref. 8), a test not previously reported in the ride-quality literature. The data were first analyzed for the group as a whole. A Spearman's ρ correlation coefficient was calculated which compared the means of the ratings of segments 9 to 20 of flight 1 to those of flight 2 ($\rho = 0.937$, significant at the 0.01 level), which indicated that mean comfort ratings were consistent across flights, at least for the relative rankings of the segments.

Reliability of responding was also measured by using the mean comfort rating for each subject (SCR). The SCR is the mean of a subject's rating of segments 9 to 20. For those 40 people who experienced both flights, the Kendall correlation coefficient for the SCR of flight 1 to that of flight 2 was 0.53, significant at the 0.001 level, again demonstrating that untrained people can and do make consistent judgements of the comfort levels of motion.

The variables of age, sex, and previous number of airplane flights were analyzed two variables at a time by using an unweighted analysis of variance (ANOVA) for unequal N's (ref. 9). The dependent measure used for this analysis was the SCR for flight 1. The results of the ANOVA, which are presented in table 5, show that there were no significant main effects of the variables of age, sex, and previous number of flights or significant interactions of these variables. If there is a relationship between these variables and the subjective assessment of the motion encountered in flight, it is a more subtle effect than can be detected by using a mean of 12 responses (the SCR). For example, younger people might rate turbulent segments as more uncomfortable than older ones do but rate the smooth-and-level flight as less comfortable. The use of the SCR which averages the response to all segments would cancel ou such an effect. A more detailed analysis of the data which will look for such effects is currently underway.

Inspection of figures 8 to 10, which show the means for each of the 20 segments of flight 1 for males and females for the different age groups and fo those with different amounts of flying experience, respectively, demonstrates the most striking characteristic of the results of our TIFS investigation: the consistency with which the test subjects rated the ride quality of our test motions. The same segments, for example, were rated as being less than comfor table or acceptable (9, 12, 15, 17) by all the various subject classifications

Correlation coefficients of various demographic and attitudinal variable with the SCR of flight 1 were computed by using the SPSS statistical package (ref. 10) which gives both the Spearman's ρ and the Kendall's τ values. (See table 6.) The only significant correlation was that of SCR and feelings about flying (possible feelings were enjoy, uneasy, dislike, fear, and will not). Negative feelings about flying are therefore significantly associated with a higher mean rating of the comfort of the ride.

The Spielberger State-Trait Anxiety Inventory is a two part questionnair designed to measure (1) a person's present level of anxiety (his state of anxiety) and (2) his usual or typical level of anxiety (trait). Possible scores for each part range from 20 (very low anxiety) to 80 (very high anxiety). The mean trait score for the 58 people who rode the first TIFS flight was 32.38 (SD = 7.87), and the mean state score was 28.32 (SD = 8.13). The state score is lower than the trait score indicating that the subjects as a group were <u>less</u> anxious during flight 1 than they usually are. It seems likely that the novelty of being paid to take an airplane ride and of being a part of an NASA experiment were positive factors which reduced anxiety for most subjects. Observation of the nonverbal behavior of the 2 subjects per flight who were video taped confirmed their self reported lack of anxiety.

Although the test group as a whole had low anxiety levels on board fligh 1, some of the people did report moderate to high state anxiety levels. The Kendall test of significance did indeed show a significant, positive correlati

($\tau = 0.193$, $p < 0.05$) between state anxiety and the SCR. There was no correlation between the trait score for each subject and his SCR ($\tau = 0.007$).

PRQA STUDY

Subjects

A total of 85 subjects provided data for the PRQA study. As for the TIFS study, age, sex, and flight experience were the primary subject variables investigated. A comprehensive breakdown of these is presented in figures 11, 12, and 13.

Apparatus

The PRQA is capable of reproducing 3 DOF of the ride motion recorded from an actual vehicle. These three motions are vertical, lateral, and roll with high frequency and low amplitude capability. See reference 3 for detailed characteristics.

Procedure (PRQA Ride Environment)

Six subjects were tested simultaneously on the PRQA. Each subject was exposed twice (ride 1 and ride 2) to a 15-minute motion tape with 10 consecutive "ride segments" of selected motions. These motions were input with 2 DOF - lateral and vertical - obtained from recordings of random Taxi, Takeoff, Inflight, and Touchdown motions from actual STOL flights.

These motions were recorded by engineers in the Noise Effects Branch of the Langley Research Center; thus, they were not the same motions as used in either the VMS or the TIFS study. Consequently, a "one-to-one" comparison may not be made between the two studies regarding "ride comfort levels." In addition, it is important to note that the accompanying airplane sounds were not used with the PRQA study.

Each segment consisted of 60 seconds of motion with the middle 20 seconds serving as the "test portion." Segments 3 and 8 were the only segments with inputs of identical motion. A taped command of the words "Begin" and "Rate" signaled these 10 test portions per ride. A 30-second section of smooth flight preceded the first segment and separated all other segments.

There was an approximate 30-minute interval between ride 1 and ride 2, during which time a post-flight questionnaire and Anxiety Rating Scale were completed. The subjects were not informed that rides 1 and 2 were identical. Neither were they informed of the type of motion nor the sequencing.

Results and Discussion

Figure 14 shows the mean g_{rms} (acceleration at seat) value per segment. This value was obtained by averaging the g_{rms} values for all six seats per segment.

Figure 15 presents the mean comfort rating for ride 1 and ride 2 for eac segment. The corresponding standard deviation values are located to the right of each bar. In six of the segments the mean comfort rating was higher for ride 2 than for ride 1. The mean increase for these six segments was 0.128. The remaining 4 segments had a lower mean comfort rating for ride 2, amounting to a mean decrease of 0.062. Thus, the overall change in mean comfort rating from ride 1 to ride 2 amounted to a mean increase of 0.052. This consistency in rating between segments of ride 1 and ride 2 yields a Spearman's Rank Correlation of 0.94, which is significant at less than the 0.001 level. As these differences were relatively small and probably due (in this experimental procedure) to random factors, the mean for all 20 segments was used as the mean comfort rating.

Figure 16 represents the mean comfort rating per g_{rms} value. Again, a Spearman's Rank Correlation between these two variables yields a ρ of 0.94. It should be noted early in the discussion that only two of these mean ratings even reached the "somewhat uncomfortable" level; thus these motions were collectively experienced as being not uncomfortable.

Figure 17 shows the mean comfort rating for the three flight experience groups. In all ten segments the 4 to 9 flight experience group had the highest mean comfort rating. Figure 18 displays the mean comfort rating for the three age groups. In all segments, except 6 and 8, the 18 to 25 age group had the highest mean comfort rating.

Figure 19 displays the mean comfort rating for both sexes. In all segments the males have a higher mean comfort rating. It should be noted, howeve that this mean difference becomes very small on segments 2, 5, and 10. Figure 14 earlier presented these as the segments with the three highest g_{rms} value A comparison of the "difference" or "similarity" in ratings by the sexes conti gent upon g_{rms} values is shown in figure 20. When the RMS values are below 0.04 the males have considerably higher mean ratings than the females. Howeve when the RMS values are above 0.05 the mean ratings are only slightly higher for the males than the females. It is highly possible that females are more tolerant of certain vibrations than males. Thus, in this study, a certain RMS value -- or some other value -- had to be reached before a sex difference in responding to motion was negated.

Figure 21 shows the absolute difference in mean comfort rating between the sexes for RMS values. The 7 segments below the 0.04 RMS value have a mean difference rating of 0.41 between the sexes, whereas, the corresponding mean difference for the three segments with RMS values above 0.05 is 0.04. A Spearman's Rank Correlation yields a ρ of -0.76 between these RMS values an differences in ratings according to sex.

Figure 22 displays the mean comfort rating for both males and females pe flight experience category. Flight experience is divided into three categorie for the reduction of data. The three categories are (1) 0 to 3, (2) 4 to 9, and (3) 10+ actual airplane rides. In all three flight experience categories, the males had a higher mean comfort rating than the females. A two-way unweighted analysis of variance with unequal N's yields a significant main effect for sex, but not for flight experience. Neither was there a significant interaction effect. See table 7.

Figure 23 shows the mean comfort rating for males and females for the three age groups. A two-way unweighted analysis of variance yields neither a significant main effect nor interaction effect. Again, however, the male mean comfort rating was higher on all three categories.

Figure 24 presents the mean comfort rating for the three flight experience categories per age group. Again, the ANOVA yields no significant effects. It is interesting to note here that the subjects with ten or more flights, regardless of age category, were highly similar in their ratings.

Correlation coefficients between the mean subject comfort rating and (1) various demographic variables as well as (2) anxiety score measures are found in table 8. The only two significant correlations are between mean subject comfort rating and (1) weight and (2) state anxiety score. Closer scrutiny of the data could show that there is no independent relationship between weight and comfort rating. It is highly probable that the underlying relationship is between sex and comfort rating.

CONCLUDING REMARKS

Two experiments were conducted to determine whether age, sex, and/or flight experience, along with other demographic and attitudinal variables, in addition to anxiety levels, play a significant role in influencing a person's "comfort rating" of typical STOL aircraft motions.

It is again important to note that in all cases, the data were analyzed by using only mean values. Consequently, the conclusion must be considered in this frame of reference. When large and small differences are averaged, the resulting average yields only a moderate difference. Thus, with the relatively wide range of motions, actual subject differences in comfort ratings per segment may have been cancelled out when averaged over the 20 segments.

The only primary subject variable to significantly influence mean comfort ratings was the sex of the subject. This, however, was found only in the PRQA study, and only with motions having g_{rms} values (acceleration at seat) less than 0.04. These results could indicate that males are more sensitive to minimal RMS values, whereas females are more tolerant of these same motions. The TIFS study had 8 out of 20 segments with vertical g_{rms} values greater than 0.04. Thus, those motions may have obscured a sex difference, since the mean comfort rating of all 20 segments (the SCR) was used as the dependent measure.

Because it is possible that interactions exist between the demographic variables, it is recommended that factors such as education, occupation, and income should not be neglected when selecting subjects for ride quality studies.

A significant correlation between attitude toward flying and mean comfort rating may indicate that those subjects who have a positive attitude were more tolerant of typical STOL aircraft motion than those having negative attitudes toward flying.

There is an indication that a person's anxiety level at the time of flying, that is, anxiety generated as a result of being in an aircraft, affects his SCR to aircraft motion. This is supported by the significant correlation between a person's state anxiety score and his respective SCR. No significant correlation was found between a person's usual anxiety level (trait) and his SCR.

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TABLE 1. MOTION ENVIRONMENT OF FIRST FLIGHTS (N = 6)
MEANS AND STANDARD DEVIATIONS OF EACH RATED SEGMENT

SEGMENT	*LONG ACC g's	LAT ACC g's	VERT ACC g's	ROLL RATE deg/sec	PITCH RATE deg/sec	YAW RATE deg/sec
1	.0018 .0007	.0024 .0003	.0069 .0008	.0960 .0238	.0424 .0051	.0251 .0040
2	.0015 .0003	.0242 .0010	.0161 .0025	.8447 .0222	.1054 .0148	.2137 .0124
3	.0014 .0007	.0029 .0008	.0087 .0043	.1114 .0487	.0571 .0323	.0267 .0071
4	.0029 .0021	.0214 .0020	.0229 .0142	.9427 .1625	.2398 .0428	.2343 .0501
5	.0027 .0010	.0160 .0032	.0160 .0066	.4007 .0606	.1586 .0266	.1387 .0334
6	.0014 .0007	.0082 .0006	.0081 .0019	.6973 .1377	.0635 .0162	.0745 .0637
7	.0020 .0011	.0032 .0014	.0088 .0024	.1990 .1816	.0523 .0262	.0578 .0739
8	.0015 .0001	.0235 .0092	.0162 .0046	.9732 .3100	.1085 .0307	.1345 .0522
9	.0045 .0004	.0196 .0017	.0928 .0107	.9490 .0327	.2663 .0155	.2879 .0094
10	.0017 .0016	.0039 .0026	.0082 .0046	.1330 .0682	.0411 .0145	.0352 .0251
11	.0034 .0014	.0100 .0007	.0498 .0049	.8749 .0232	.1475 .0085	.1405 .0128
12	.0054 .0008	.0388 .0056	.1086 .0106	1.3942 .2039	.3647 .0344	.5172 .1057
13	.0031 .0019	.0064 .0017	.0122 .0059	.3322 .1780	.0676 .0163	.1108 .0751
14	.0048 .0010	.0093 .0009	.0505 .0052	1.3727 .0479	.2439 .0096	.3378 .0171
15	.0038 .0007	.0660 .0974	.0559 .0071	1.7393 .0773	.3037 .0046	.4001 .0159
16	.0017 .0009	.0032 .0009	.0075 .0022	.1180 .0276	.0475 .0215	.0251 .0052
17	.0080 .0025	.0434 .0035	.1281 .0108	1.3820 .3677	.3511 .2068	.6288 .0530
18	.0026 .0013	.0055 .0007	.0171 .0016	.2907 .0087	.0634 .0054	.0667 .0034
19	.0045 .0013	.0081 .0009	.0583 .0075	.5194 .0267	.2137 .0321	.1330 .0109
20	.0042 .0008	.0206 .0015	.0945 .0117	.9388 .0061	.2773 .0235	.2869 .0101

* Upper value is mean value. Lower value is the standard deviation.

TABLE 2. MOTION ENVIRONMENT OF SECOND FLIGHTS (N = 4)
MEANS AND STANDARD DEVIATIONS OF EACH RATED SEGMENT

SEGMENT	*LONG ACC g's	LAT ACC g's	VERT ACC g's	ROLL RATE deg/sec	PITCH RATE deg/sec	YAW RATE deg/sec
9	.0052 .0011	.0199 .0015	.0925 .0110	.9305 .0572	.2705 .0152	.3029 .0097
10	.0021 .0016	.0046 .0048	.0067 .0031	.2323 .2847	.2264 .3883	.0672 .0614
11	.0034 .0006	.0090 .0010	.0475 .0060	.8417 .0313	.1454 .0049	.1430 .0079
12	.0064 .0008	.0376 .0077	.1108 .0175	1.4185 .0538	.3538 .0268	.5145 .0970
13	.0015 .0005	.0050 .0011	.0104 .0011	.2461 .0901	.0511 .0064	.0883 .0249
14	.0048 .0004	.0089 .0004	.0495 .0057	1.3483 .0775	.2540 .0163	.3430 .0237
15	.0045 .0006	.0263 .0022	.0560 .0068	1.6425 .0718	.2927 .0068	.4053 .0132
16	.0017 .0005	.0049 .0028	.0092 .0064	.2308 .1727	.0421 .0109	.0479 .0375
17	.0084 .0019	.0399 .0055	.1168 .0189	1.3438 .4599	.3335 .2015	.6543 .2040
18	.0025 .0007	.0063 .0015	.0182 .0029	.3241 .0492	.0567 .0077	.0734 .0164
19	.0050 .0007	.0086 .0015	.0604 .0069	.5389 .1009	.2075 .0071	.1345 .0131
20	.0050 .0002	.0210 .0023	.0927 .0125	.9021 .0323	.2741 .0136	.2955 .0078

* Upper value is mean value. Lower value is the standard deviation.

TABLE 3. CORRELATIONS OF THE MOTION COMPONENTS OF FLIGHTS 1 AND 2

	ρ Value	Significance Level
Mean Vertical Acceleration	.993	.01
Mean Lateral Acceleration	.965	.01
Mean Longitudinal Acceleration	.909	.01
Mean Roll Rate	.986	.01
Mean Pitch Rate	.888	.01
Mean Yaw Rate	1.000	.01

TABLE 4. COMFORT RATINGS FOR FLIGHTS 1 AND 2

	FLIGHT 1		FLIGHT 2	
SEGMENT	MEAN	SD	MEAN	SD
1	1.92	0.80		
2	2.77	0.74		
3	1.98	0.71		
4	2.98	0.71		
5	3.12	0.76		
6	1.82	0.67		
7	1.85	0.74		
8	3.06	0.80		
9	3.66	0.95	3.36	0.71
10	1.86	0.80	1.72	0.76
11	2.70	0.82	2.22	0.74
12	3.76	0.87	3.65	0.68
13	1.96	0.84	1.79	0.75
14	2.48	0.77	2.66	0.64
15	3.47	0.81	3.44	0.83
16	1.72	0.70	1.81	0.87
17	3.83	1.03	3.72	0.98
18	1.84	0.75	1.97	0.78
19	2.03	0.72	2.00	0.71
20	3.09	0.85	3.24	0.74

TABLE 5. SUMMARY OF 3 BREAKDOWNS OF TWO-WAY UNWEIGHTED ANALYSIS OF VARIANCE FOR MEAN COMFORT RATINGS (UNEQUAL N'S PROCEDURE)

SOURCE	SS	df	MS	F
Sex	0.014	1	0.014	0.044
Flights	0.355	2	0.177	0.548
S × F	0.876	2	0.438	1.352
Error	16.866	52	0.324	
Sex	0.227	1	0.227	0.662
Age	0.292	2	0.146	0.426
S × A	0.292	2	0.146	0.426
Error	17.816	52	0.343	
Flights	0.240	2	0.120	0.346
Age	0.208	2	0.104	0.299
F × A	1.127	4	0.282	0.810
Error	17.057	49	0.348	

TABLE 6. CORRELATIONS BETWEEN MEAN SUBJECT COMFORT RATING OF FLIGHT 1 AND SOME DEMOGRAPHIC VARIABLES

Variables	N	Kendall's τ	Spearman's ρ	Significance Level
Age	58	0.047	0.062	NS
Income	55	0.031	0.036	NS
No. Previous Flights	58	0.019	0.021	NS
Feelings About Flying	57	0.224	0.279	<.05
No. of Flights/Year	46	0.015	0.018	NS
Education	58	-0.068	-0.088	NS

TABLE 7. SUMMARY OF 3 BREAKDOWNS OF TWO-WAY UNWEIGHTED ANALYSIS OF VARIANCE FOR MEAN COMFORT RATINGS (UNEQUAL N'S PROCEDURE)

SOURCE	SS	df	MS	F
Sex	1.913	1	1.913	5.335*
Flights	2.205	2	1.102	3.074
S × F	0.376	2	0.188	0.525
Error	28.331	79	0.359	
Sex	1.316	1	1.316	3.549
Age	1.083	2	0.541	1.461
S × A	0.171	2	0.086	0.231
Error	29.279	79	0.371	
Flights	1.582	2	0.791	2.240
Age	1.593	2	0.796	2.255
F × A	2.778	4	0.694	1.966
Error	26.845	76	0.353	

*$p < 0.05$

TABLE 8. CORRELATIONS BETWEEN MEAN SUBJECT COMFORT RATING AND SOME DEMOGRAPHIC VARIABLES AND ANXIETY MEASURES

Variables	N	Kendall's τ	Spearman's ρ	Significance Level
Age	85	-0.072	-0.092	NS
Flight Experience	85	-0.020	-0.022	NS
Education	85	-0.135	-0.171	NS
Occupation	85	0.018	0.027	NS
Weight	85	0.159	0.226	<.05
State Anxiety Score	80	0.225	0.301	<.01
Trait Anxiety Score	85	0.034	0.052	NS

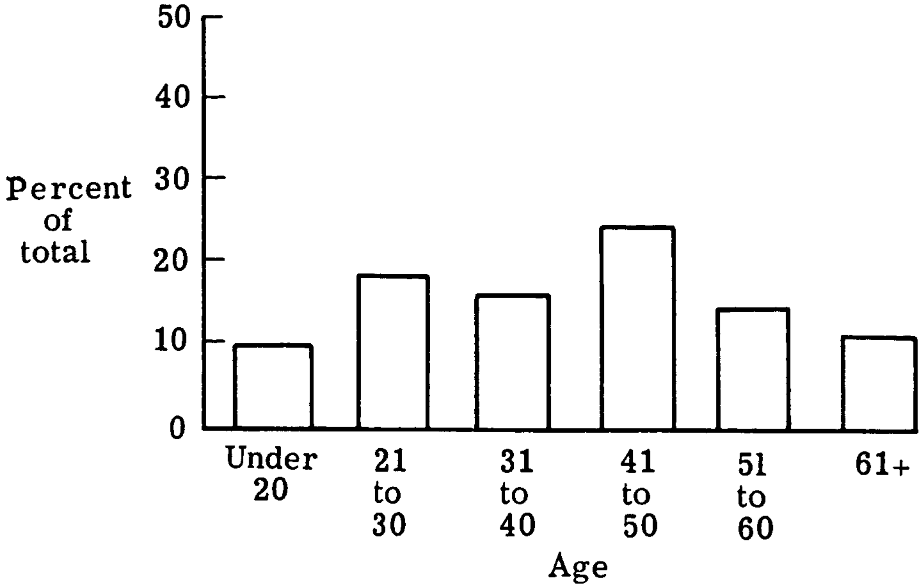

Figure 1.- Age distribution (TIFS).

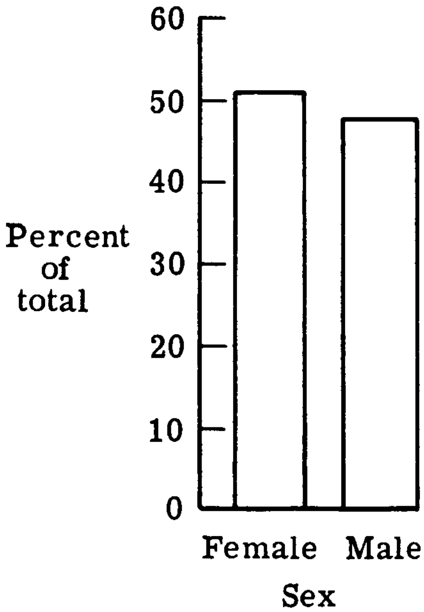

Figure 2.- Sex distribution (TIFS).

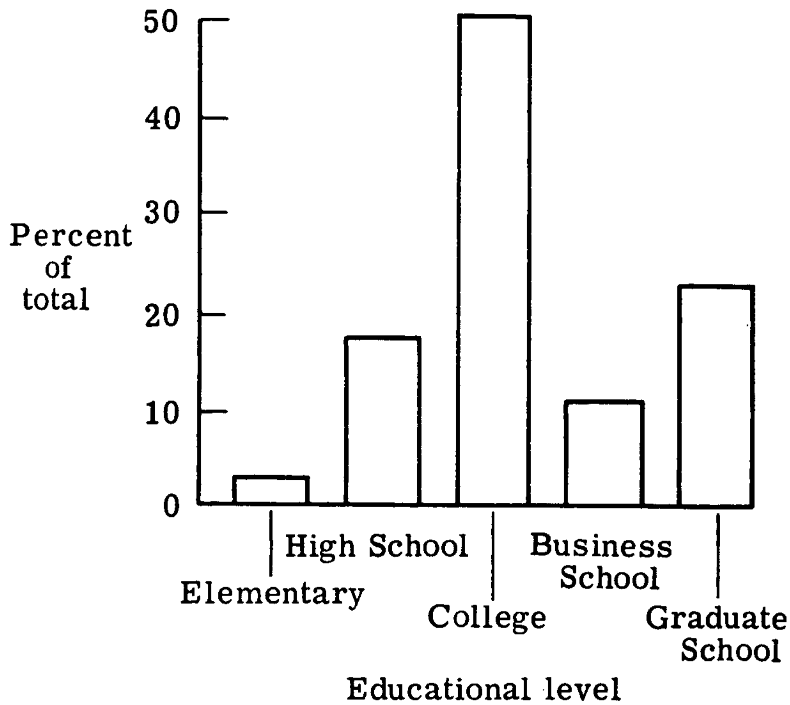

Figure 3.- Education distribution (TIFS).

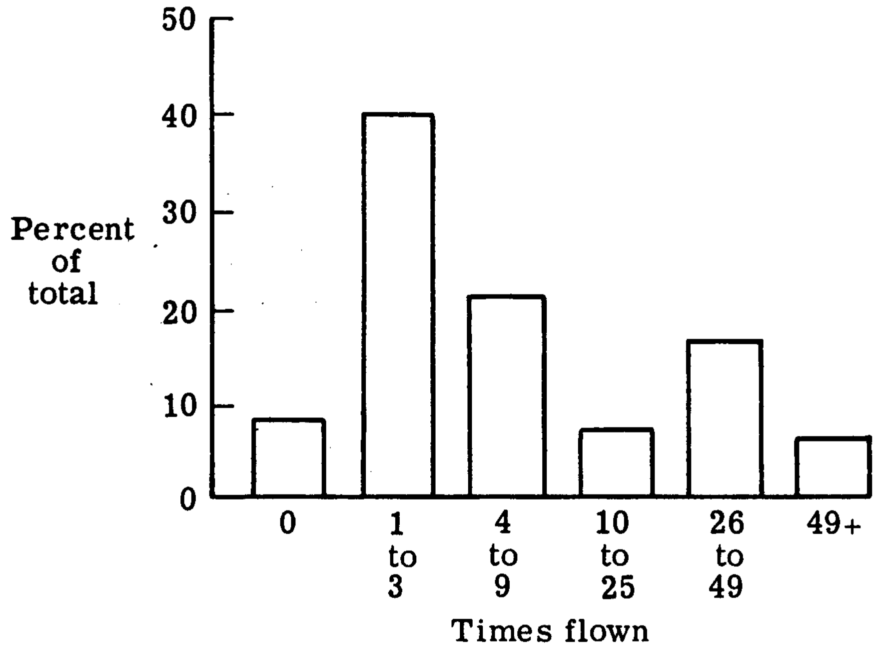

Figure 4.- Flight experience (TIFS).

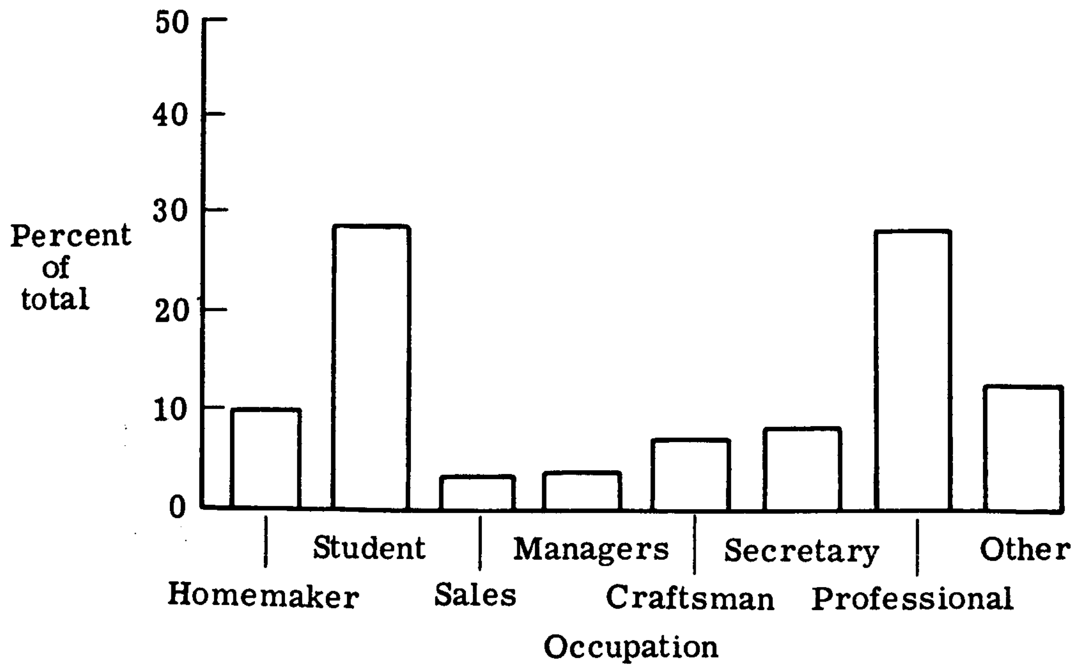

Figure 5.- Occupation distribution (TIFS).

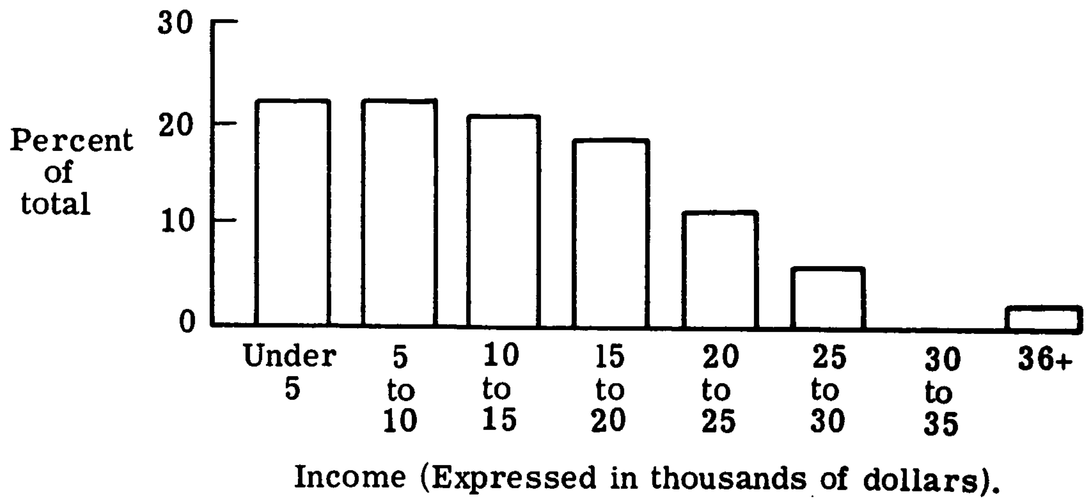

Figure 6.- Income distribution (TIFS).

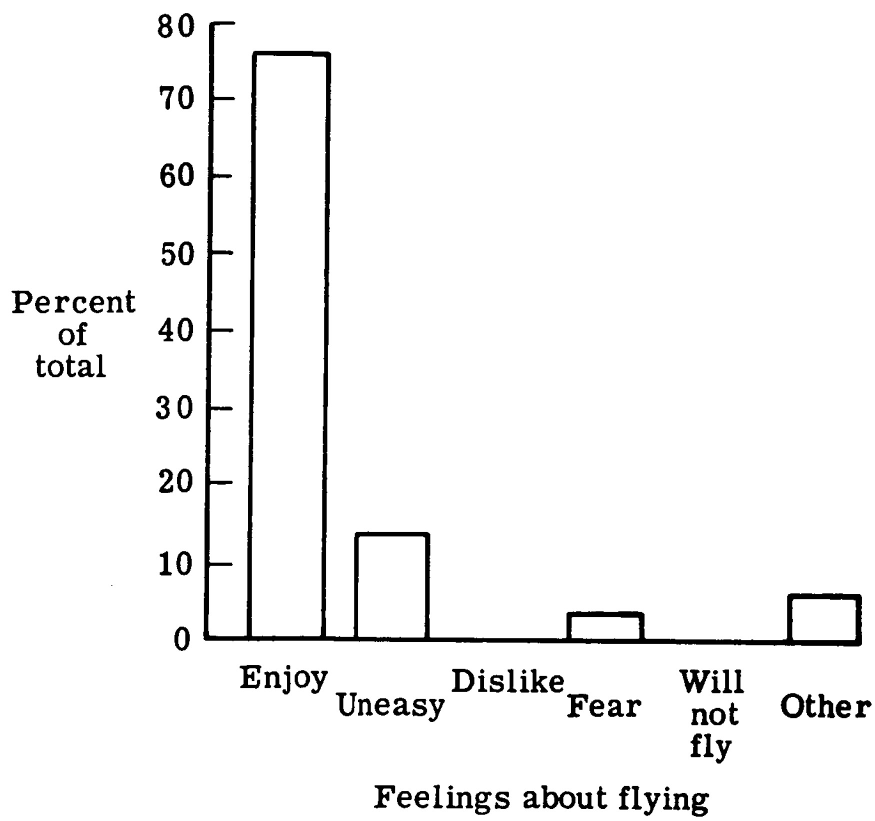

Figure 7.- Attitudes towards flying distribution (TIFS).

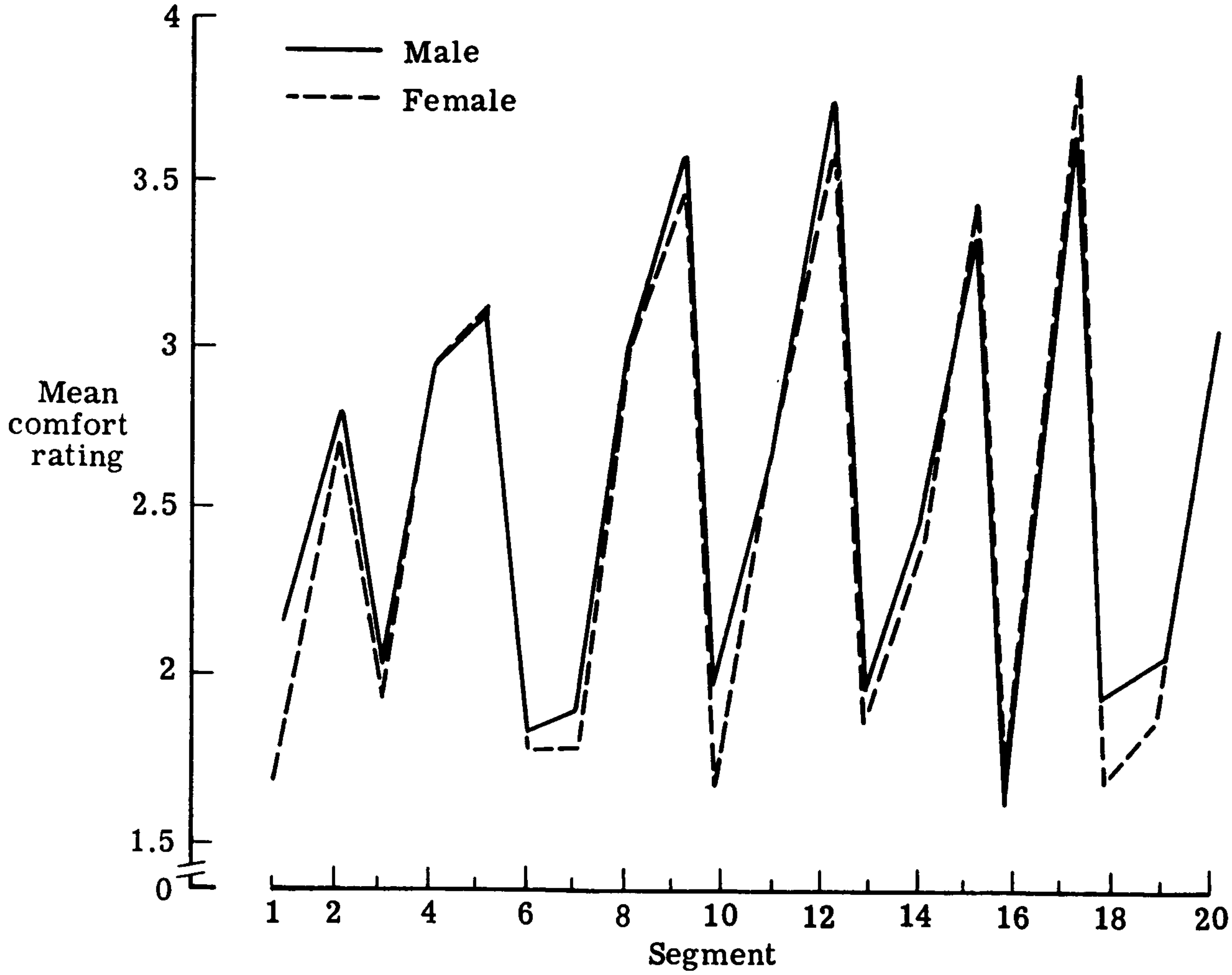

Figure 8.- Mean comfort rating as a function of sex (TIFS - flight 1).

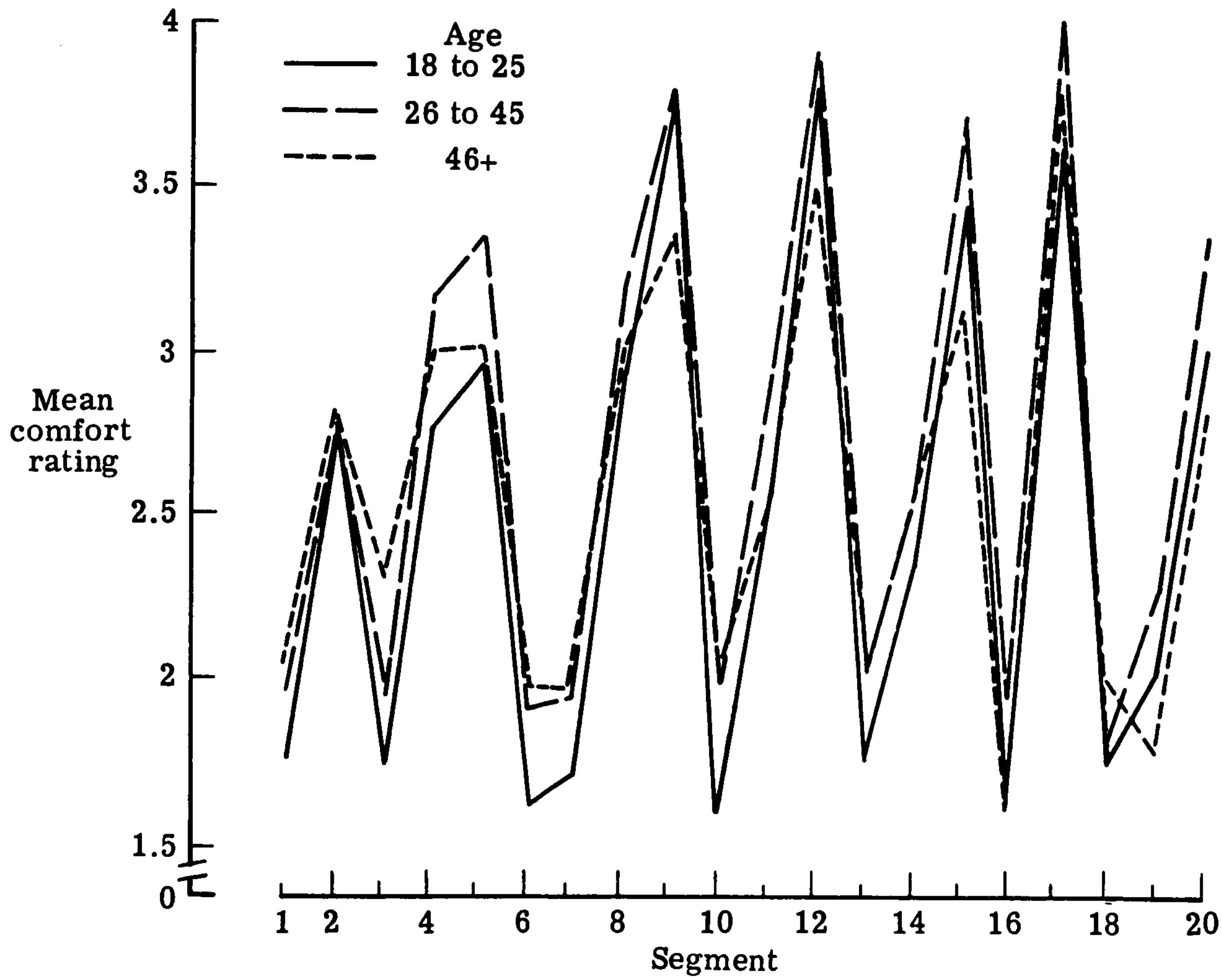

Figure 9.- Mean comfort rating as a function of age
(TIFS - flight 1).

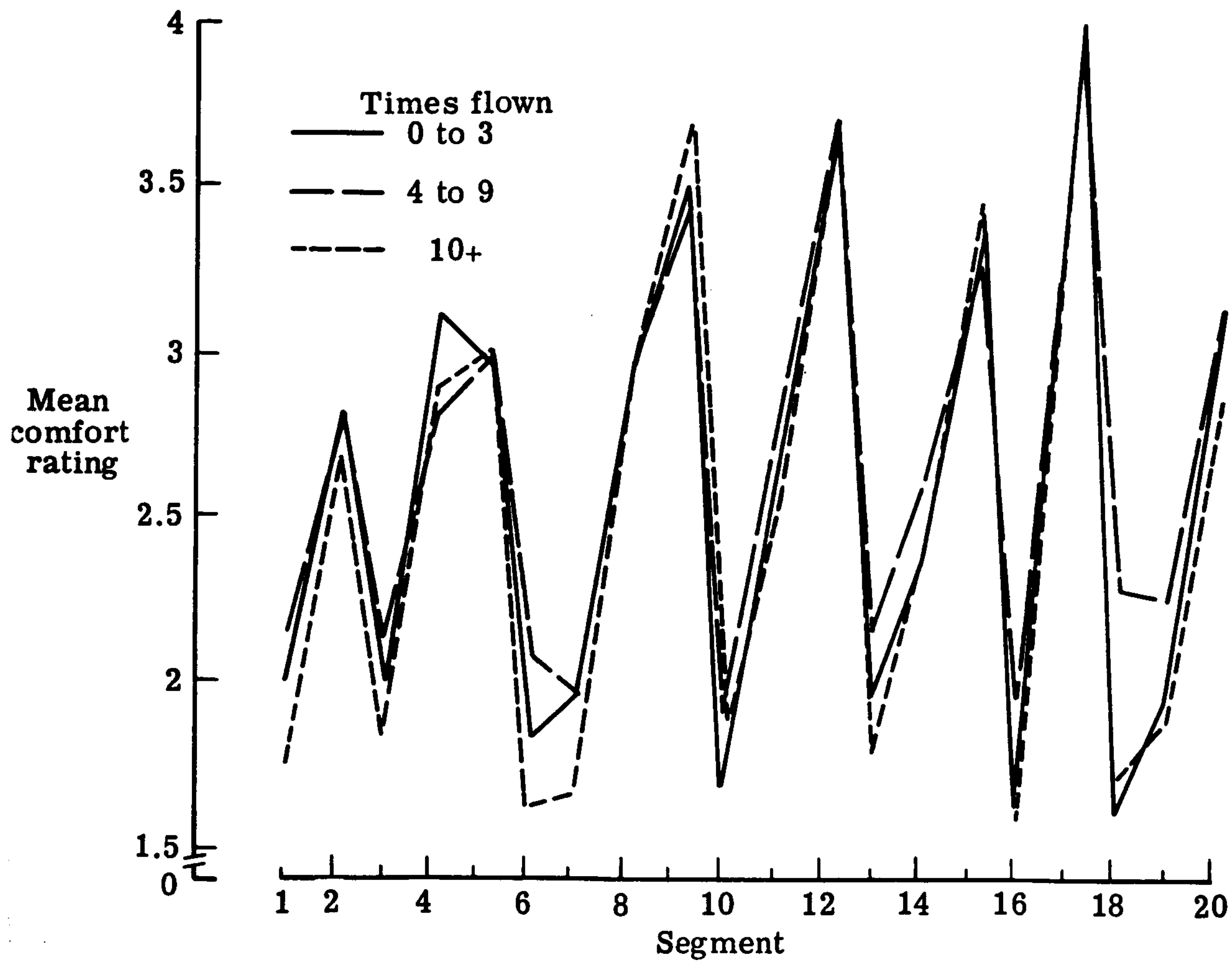

Figure 10.- Mean comfort rating as a function of flight experience (TIFS - flight 1).

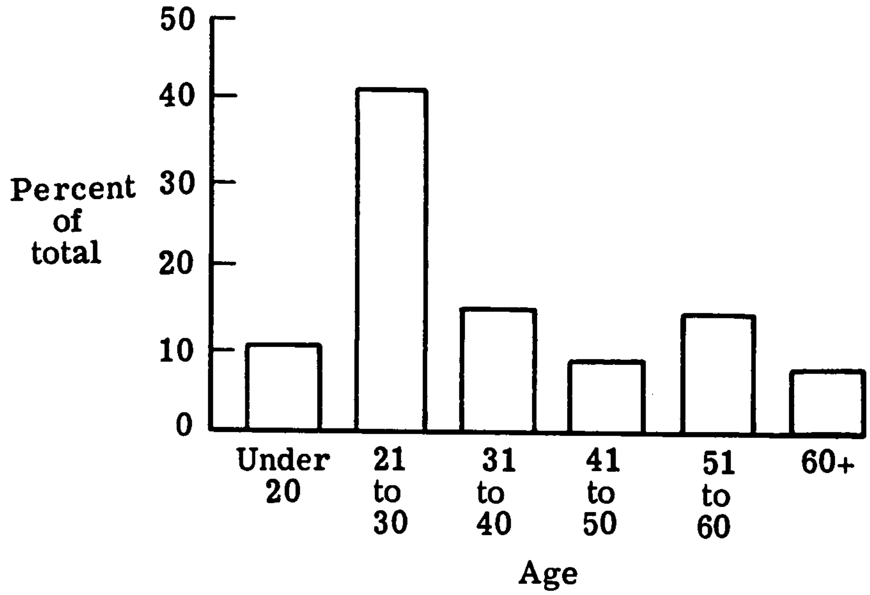

Figure 11.- Age distribution (PRQA).

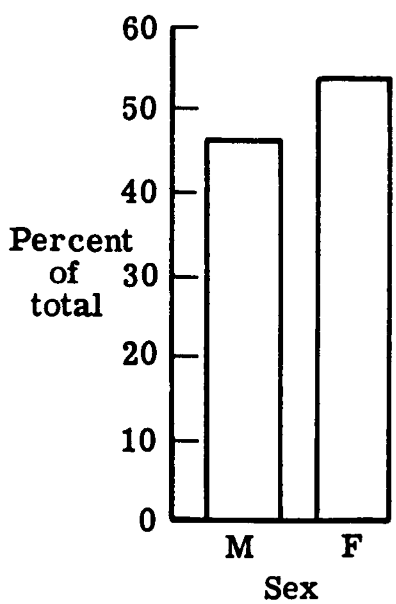

Figure 12.- Sex distribution (PRQA).

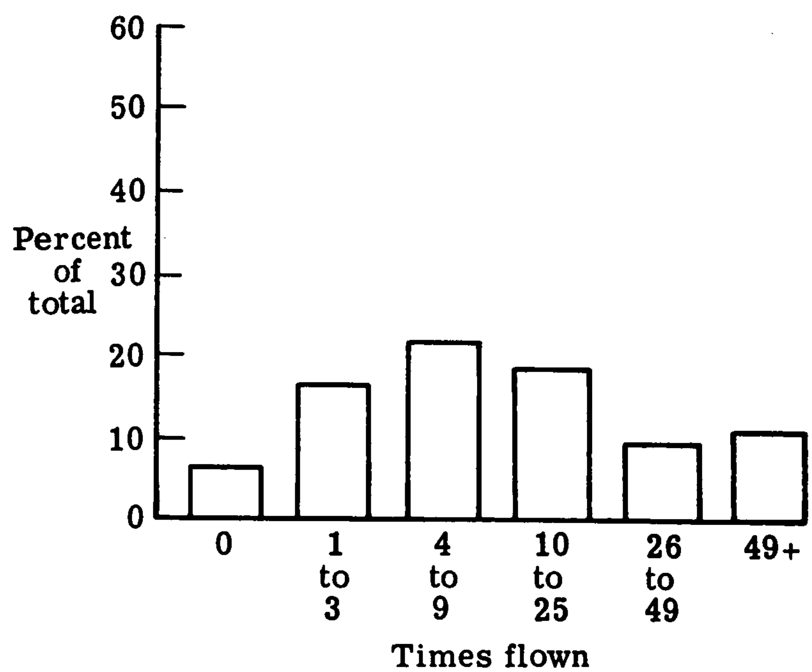

Figure 13.- Flight experience (PRQA).

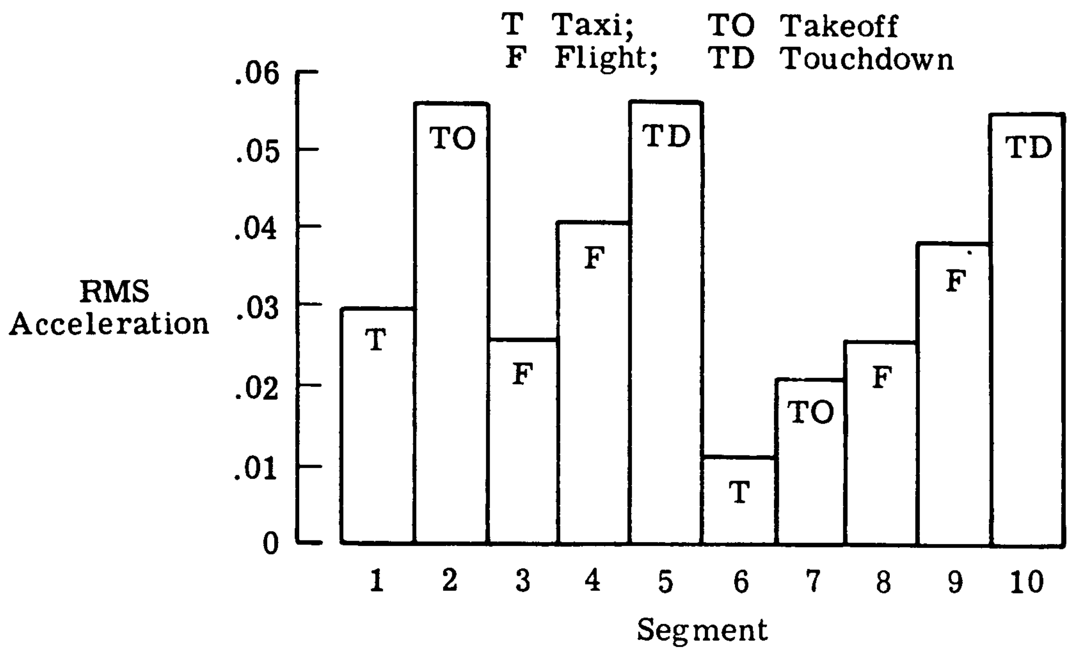

Figure 14.- Mean RMS acceleration.

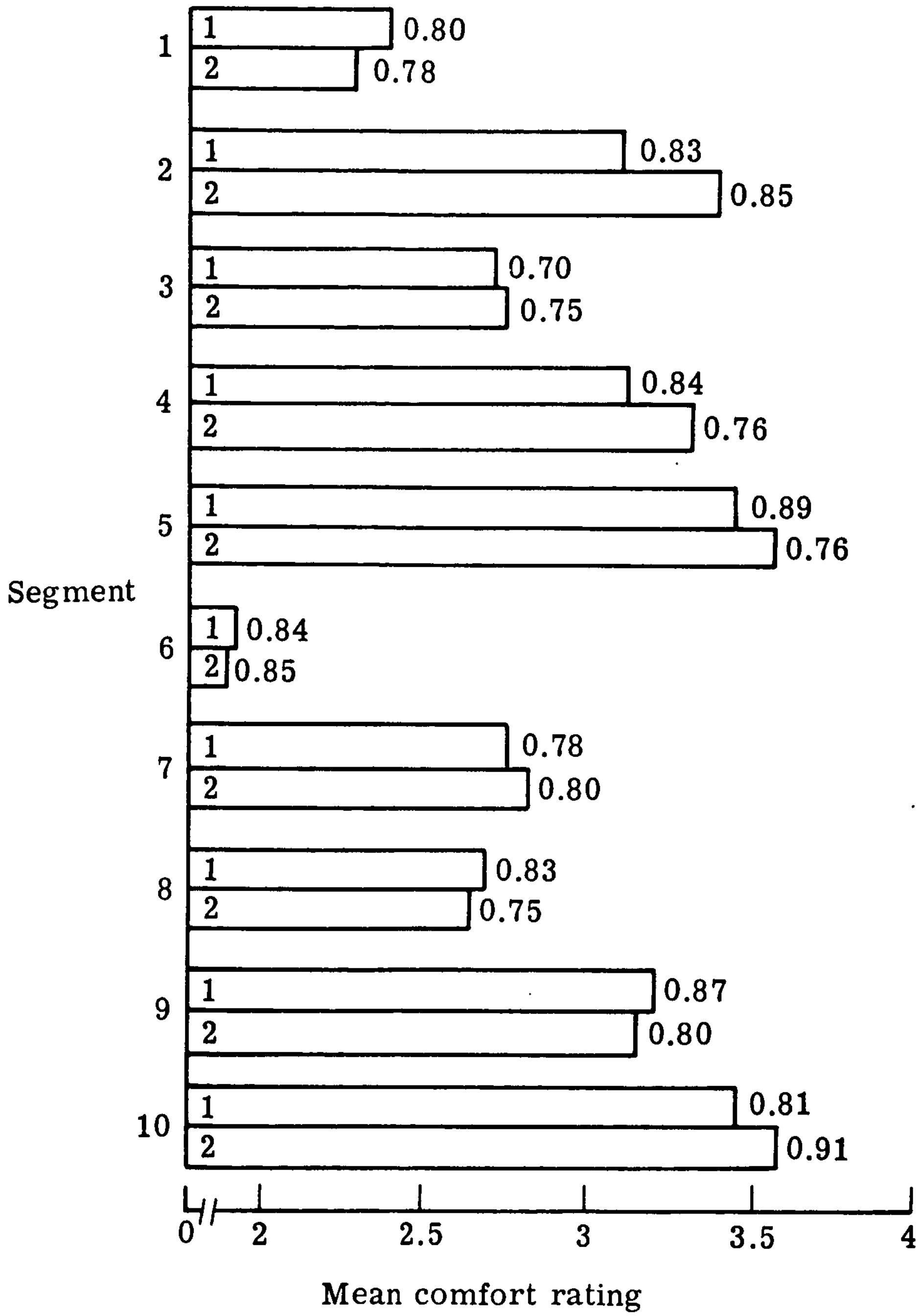

Figure 15.- Mean comfort rating for ride 1 and ride 2, and the corresponding standard deviation.

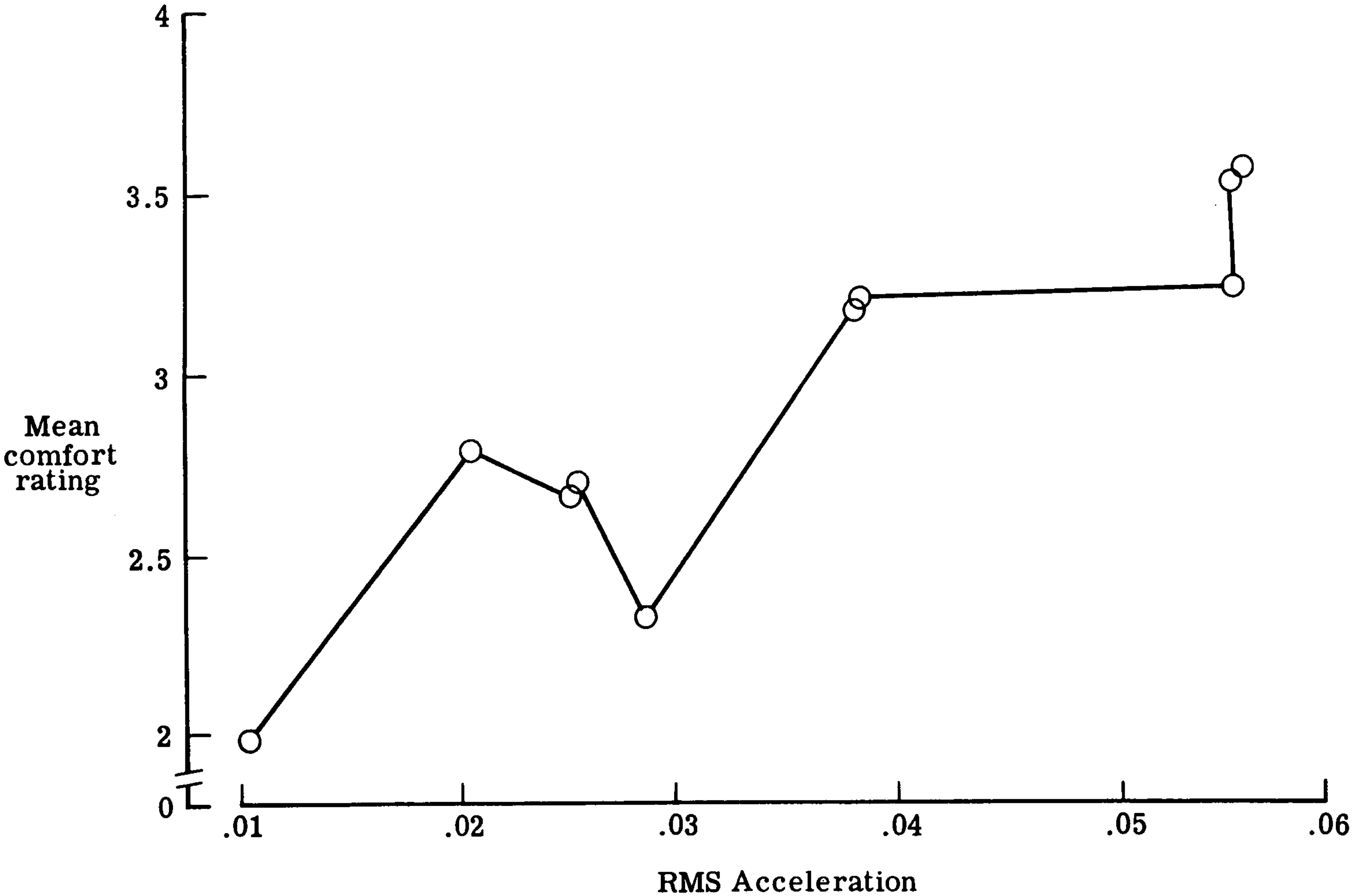

Figure 16.- Mean comfort rating as a function of RMS acceleration.

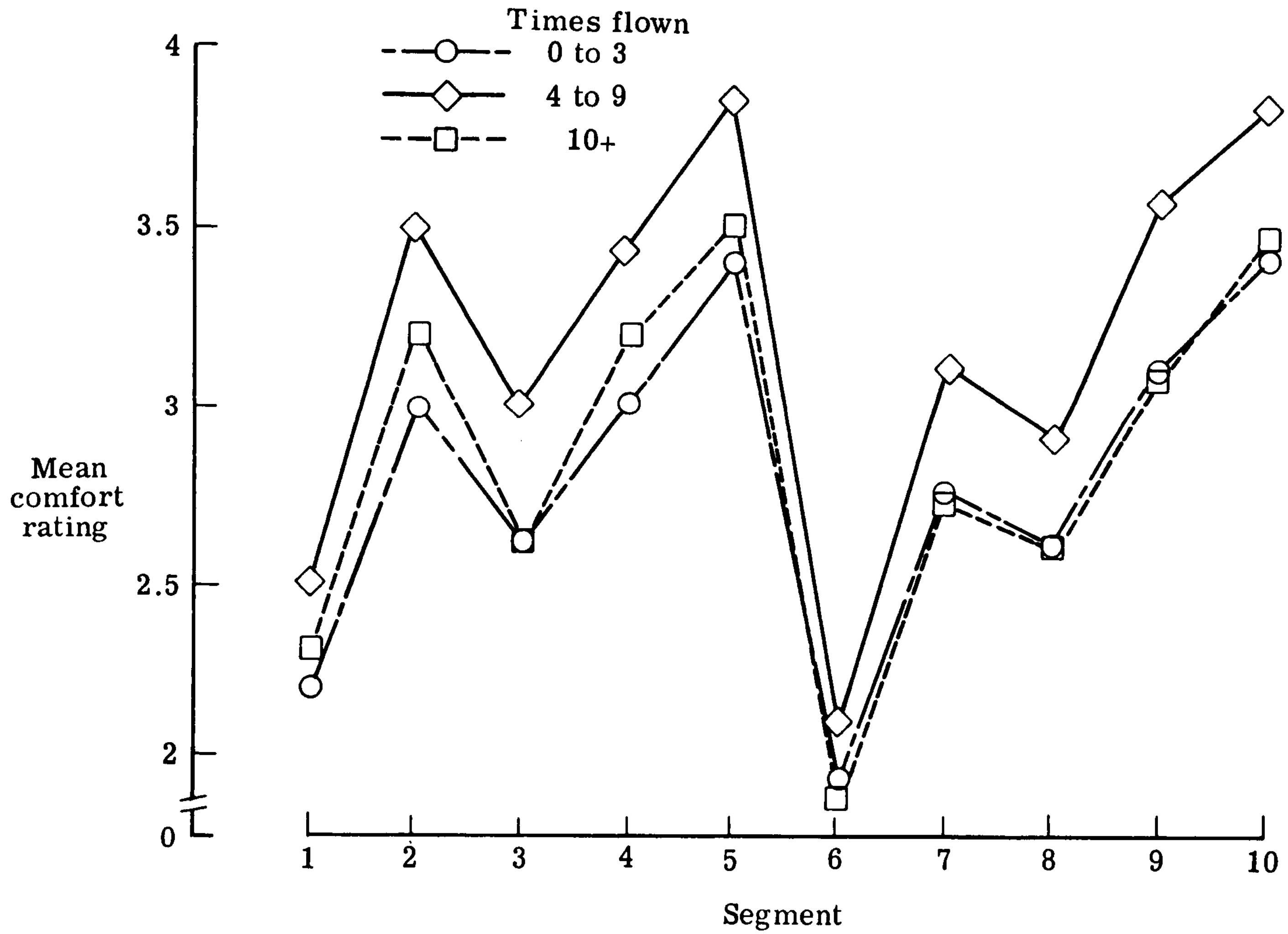

Figure 17.- Mean comfort rating as a function of flight experience.

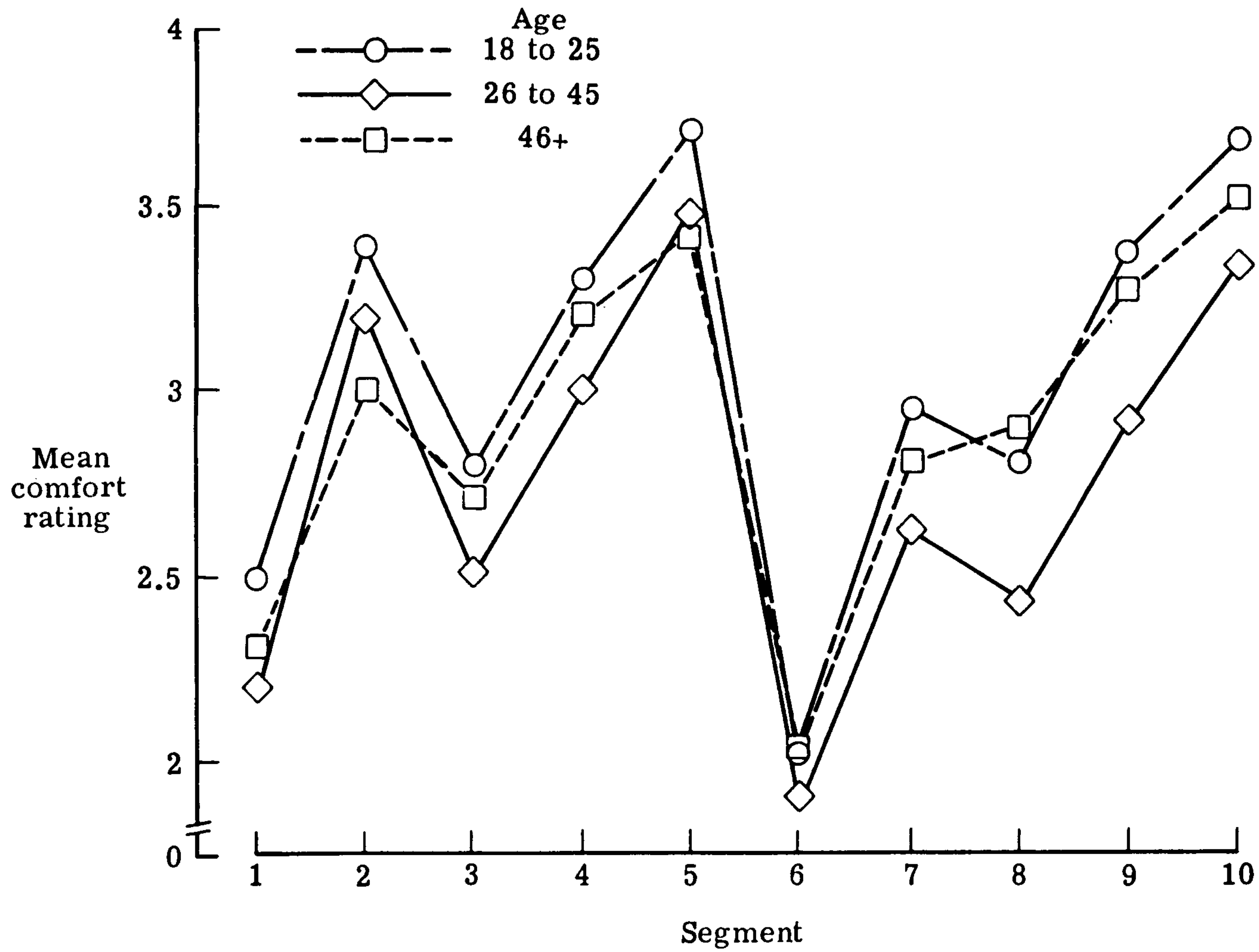

Figure 18.- Mean comfort rating as a function of age.

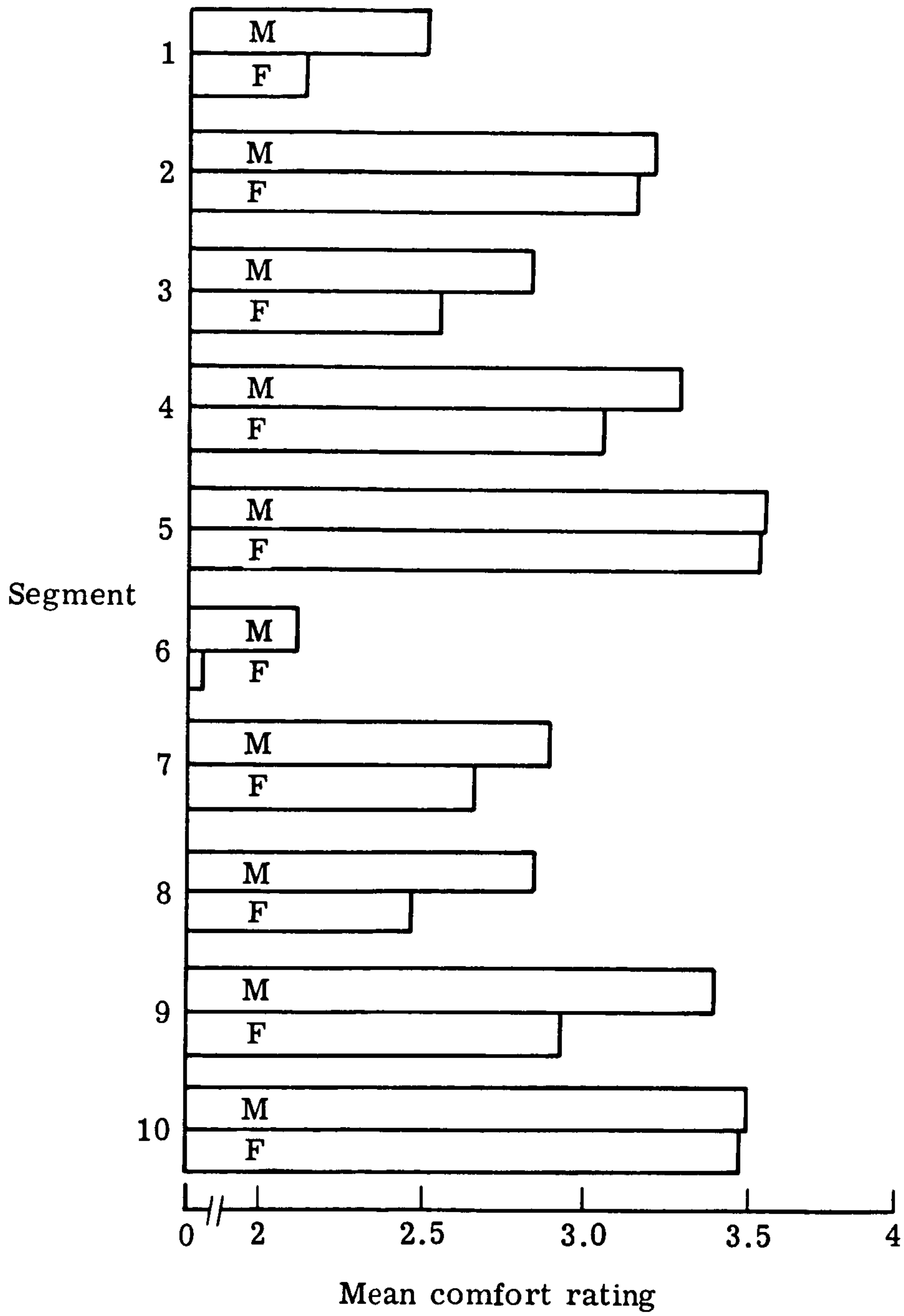

Figure 19.- Mean comfort rating as a function of sex.

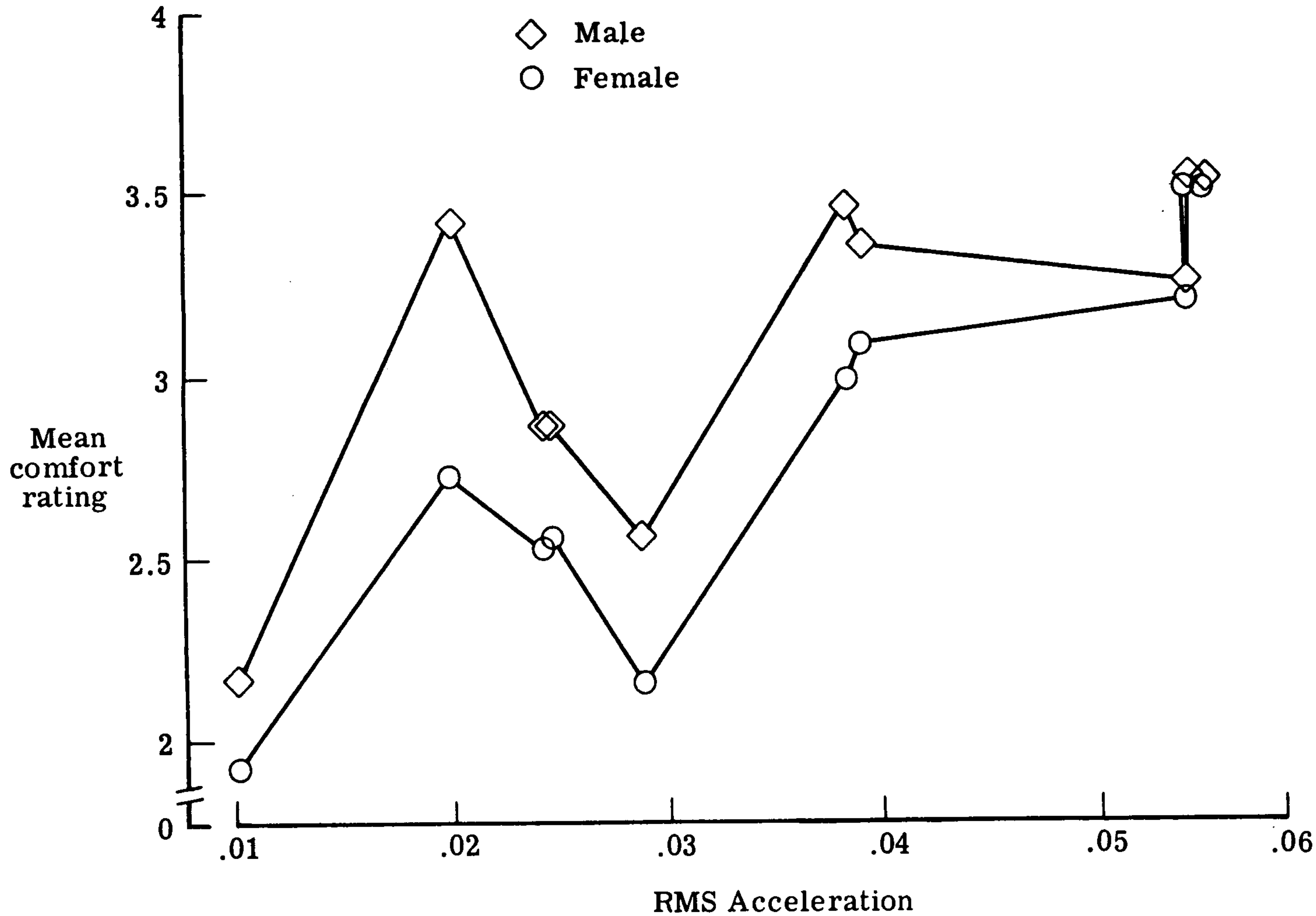

Figure 20.- Mean comfort rating as a function of sex and RMS value.

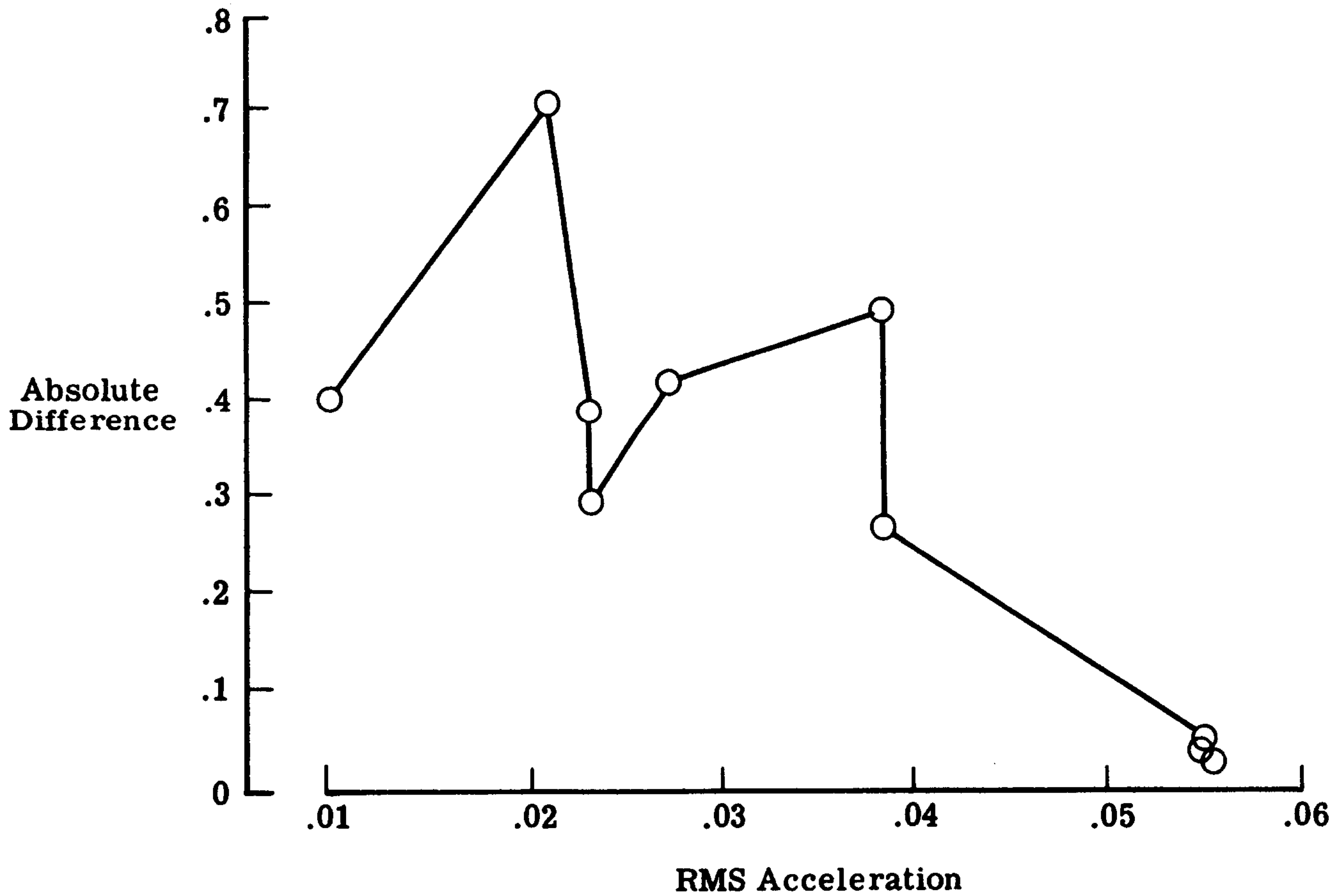

Figure 21.- Absolute difference in mean comfort rating between the sexes for RMS values.

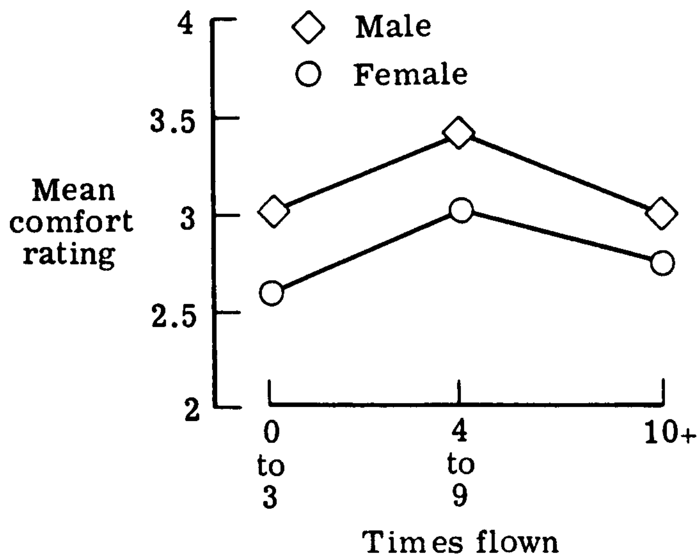

Figure 22.- Mean comfort rating as a function of sex and flight experience.

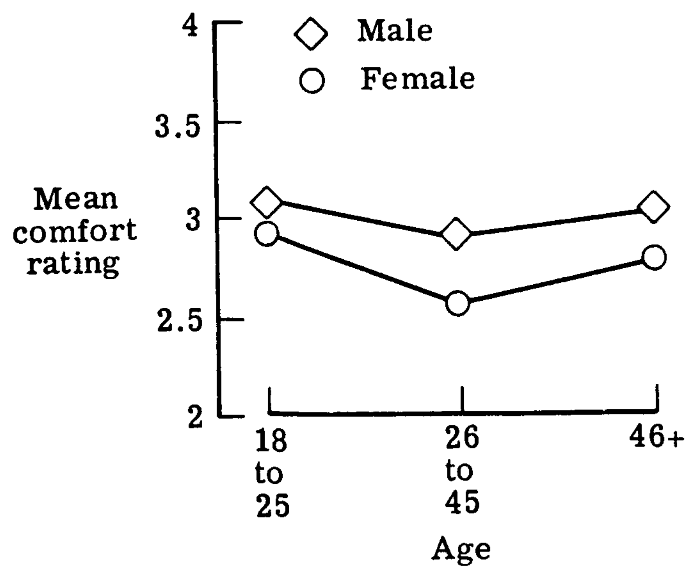

Figure 23.- Mean comfort rating as a function of sex and age.

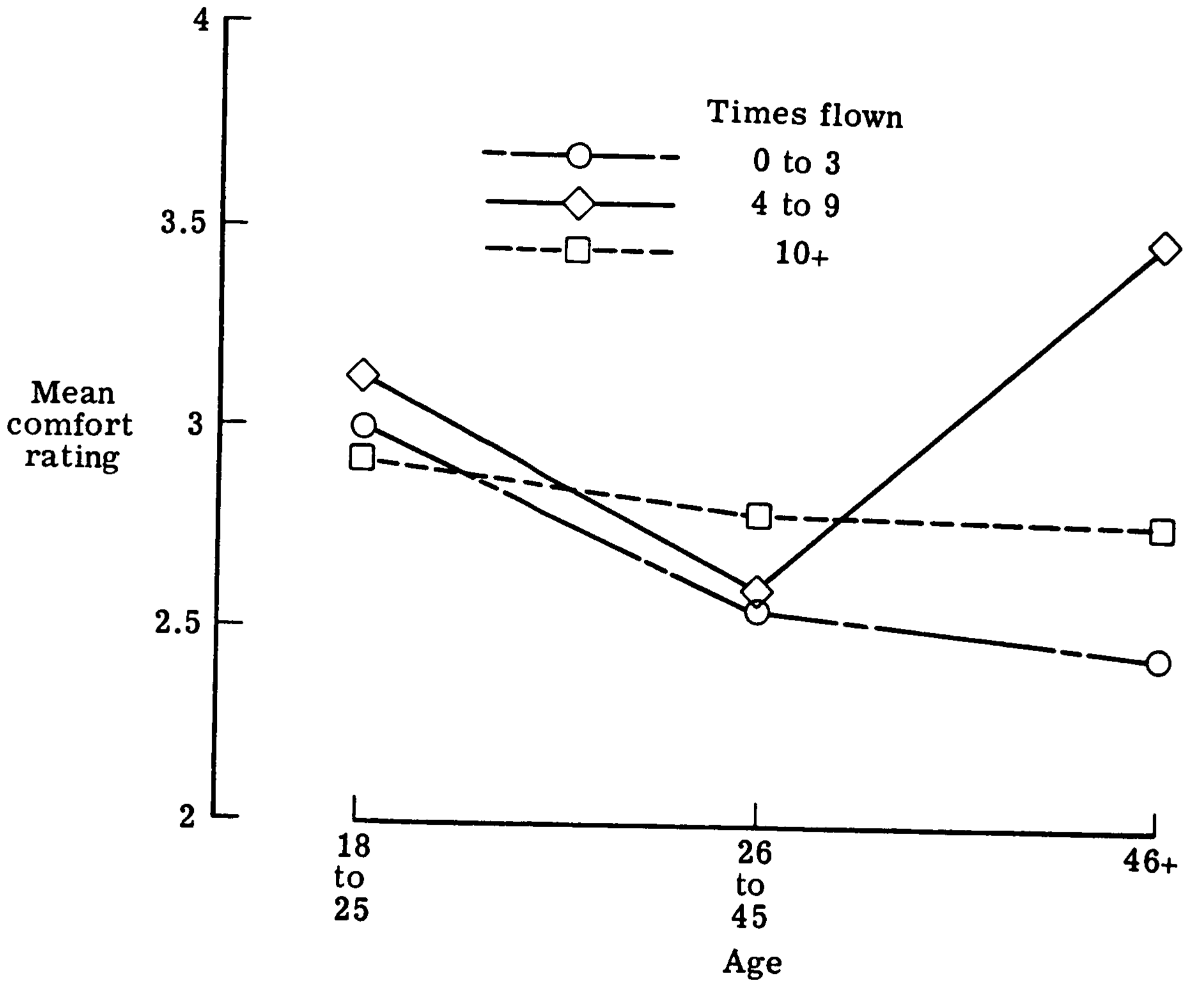

Figure 24.- Mean comfort rating as a function of flight experience and age.

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HUMAN COMFORT IN RELATION TO SINUSOIDAL VIBRATION

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SUMMARY

An investigation has been made to assess the overall subjective comfort levels to sinusoidal excitations over the range 1 to 19 Hz using a two axis electrohydraulic vibration simulator. Exposure durations of 16 minutes, 25 minutes, 1 hour, and 2.5 hours have been considered. Subjects were not exposed over such durations, but were instructed to estimate the overall comfort levels preferred had they been constantly subjected to vibration over such durations.

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INTRODUCTION

Meister and Reiher in 1931 (ref. 1) were some of the first research workers to examine the problem of human comfort in relation to sinusoidal vibration. Since then a wealth of information has been presented by various organisations. Recently, an ISO committee has attempted to define criteria, prescribe limits of exposure, and suggest methods of measurement with respect to comfort, performance, and safety, over the range 1 to 80 Hz. The resulting international standard "Guide for the Evaluation of Human Exposure to Whole-Body Vibration" identifies three main criteria of human reaction to vibration and defines the limits accordingly. These are:

(1) The preservation of working efficiencies, with the limiting fatigue-decreased proficiency (FDP) boundary

(2) The preservation of health or safety, with the 'exposure limit' boundary

(3) The preservation of comfort, with the limiting 'reduced comfort' boundary

The values for the reduced comfort boundary are based upon various studies relating to the transportation field, and the relationship between exposure time and frequency is shown in figures 1 and 2. The proposed comfort limits are related to a three-degree-of-freedom orthogonal coordinate system centred in the heart, and illustrated in figure 3. The decline in human tolerance presumed to occur with increasing exposure duration is clearly reflected in figures 1 and 2. It must be emphasised, however, that the proposals are tentative since exposure duration as a factor affecting comfort has received very little study and firm data suitable as the basis for standardisation are limited.

The reported investigation relates to the following objectives:

(1) To estimate the overall subjective comfort levels (in weighted rms g)

(2) To evaluate the shape of the comfort contours (using ISO's weighting networks)

(3) To determine and compare the percentage deviations in rms g levels between the estimated comfort contours and the corresponding ISO proposals at various frequencies and exposure durations

TWO AXIS ELECTROHYDRAULIC VIBRATION SIMULATOR

The simulator used for the investigation is that of the RAE, Farnborough, and utilises a flat platform (183 x 122 cm), weighting around 200 kg and supported by three trunnion mounted hydraulic actuators. Two actuators support the table in the vertical axis and the third is attached horizontally to the table in the same plane. Each vertical actuator has a piston area of 11.3 cm with a stroke of ±25.4 cm and is controlled by three electrohydraulic servo valves. The horizontal actuator has a piston area 22.6 cm with a stroke of ±25.4 cm and is controlled by six servo valves.

Closed loop control of each actuator utilises position and piston differential pressure feed-back. Each servo valve has a maximum flow capability of 655.6 cm^3/s, giving a linear velocity limit of 152.4 cm/s in either axis.

The simulator performance permits a maximum acceleration of ±2 g and a frequency range of 0.5 to 50 Hz, on either or both of the axes, with a payload of 273 kg. Considerable off-centre loads are permitted.

Built-in oscillators provide the necessary displacement input and the frequency of the signals applied to the two axes can either be the same, with adjustable phase angle, or independently variable over the range 0.5 to 50 Hz. Facility is provided for input of external displacement or acceleration signals.

A piston differential pressure feed-back signal is used to reduce 'stiction' effects at the extremes of actuator motion and results in improved acceleration waveforms. Displacement feed-back is based upon a combination of the output from resolvers fitted to the bearing trunnions and a displacement transducer signal. This results in displacement feed-back proportional to true horizontal or vertical motion.

The performance of the simulator conforms to the following specification (ref. 2):

(a) Frequency response of 0 to -1.0 dB over the range 0.5 to 50 Hz.

(b) Phase lag not exceeding 20° at 5 Hz.

(c) Distortion of the fundamental acceleration sine wave less than 15%, over the range 1 to 10 Hz, measured at the platforms when fully loaded.

(d) The response of the system is stable over a 3-hour period when operating at maximum endurance.

(e) The platform is constructed so that the vertical forces applied at its centre can be reproduced at the extremities. The resonant frequency when unloaded is not less than 500 Hz.

(f) A safety feature allows the operator or the test subject to shut down the system with a maximum retardation of 10 g in the event of an emergency. Means are also provided to absorb and contain the kinetic and potential energy at the extremes of motion in the event of a control system malfunction.

SUBJECTS AND POSTURE

A total of seven subjects participated in the investigation and all relevant details are given in table 1. All subjects were considered to be normal and wore normal clothing and footwear. Before being subjected to the test programme all participants were requested to sign a declaration form in accordance with the draft guide on the safety aspects relating to human vibration experiments (ref. 3).

The experimental facilities and associated safety features were explained to each subject and the general purpose of the investigation indicated. Each subject was given detailed instructions (appendix A) of his/her specific role in the experiment.

Once the test programme was fully understood the subject was seated on a hard wooden seat mounted on the vibrator platform. A birdseed cushion pad was provided that gives a 1:1 transmission ratio up to 30 Hz. The dynamic response characteristics of this and other cushion materials are given in figure 4. A standard lap seat belt was used by all subjects, adjusted to a loose position in order to minimise restraining effects and still provide adequate safety precautions. Plate 1 illustrates the posture adopted throughout the investigation and figure 3 illustrates the orthogonal coordinate system adopted.

INSTRUMENTATION

It is clear from the ISO standard (ref. 4) that human whole-body response to vibration is frequency dependent (figures 1 and 2). The standard recognise the use of instrumentation for measurement of ride or vibration severity, using frequency weighting networks conforming to the standardised human frequency response embodied in the limits and specifies the precision required

Two frequency weighting networks corresponding to human response to vibra tion in the a_x (or a_y) and a_z axes were employed to measure comfort levels (figure 5). It will be observed that the ISO standard does not indicate limits for vibration frequencies below 1 Hz owing to the scarcity of data and lack of agreement in this region. However, the 0.1 to 1 Hz region is of some significance for evaluation of suspension performance and human postural sway in the a_x and a_y axes. Hence it becomes necessary to tentatively extrapolate the ISO characteristics. A single weighting function so designed can be made to apply to any amplitude and duration, since the boundaries recommended for various exposure durations, for a specific axis, follow the same amplitude-frequency relationship. The filter output coupled to a true rms digital voltmeter yields the normalised rms value of the input acceleration signal.

The set-up adopted consisted of measuring the acceleration level on the seat and very close to the subject's buttocks. The acceleration signal generated by a piezo-resistive accelerometer was processed through a carrier amplifier, weighting filter, and rms digital voltmeter.

GENERAL TEST PROCEDURE

Two people were required to operate the simulator. One operator, stationed at the control panel, monitored safety levels and controlled frequency and level of vibration. The investigator acted as general test supervisor and as such directed the test programme and monitored the required data. An intercom system provided the necessary communication links between the subject, investigator, and simulator operator during the test sessions.

Any relevant information volunteered by the subjects during the experimen was recorded and subjects were asked to comment on the nature of the experimen at the end of each session. Subjects were free to discuss the experiment throughout the investigation.

A general ambient noise level of 62 dB(A) was recorded at head level with the simulator operating, with earphones producing some attenuation. Room temperature varied from 20.6 to 21.7°C (69 to 71°F) with a relative humidity of 55 to 60%. Since the vibrator was enclosed within a walk-in chamber, a nondistracting environment was available for the test programme.

TEST PROGRAMME

Each subject was exposed to eight sinusoidal vibrations per axis, selected within the range from 1 to 20 Hz. Exposure durations of 16 minutes, 25 minutes, 1 hour, and 2.5 hours were considered. The order of stimulus presentation was randomised for each subject. Each experimental session per subject per axis consisting of eight frequencies and four exposure durations lasted just over 40 minutes. Each subject completed two sessions covering two axes on the same day, with at least a 30-minute interval between sessions. The remaining axis was covered after a lapse of at least 24 hours.

RESULTS

Vertical Mode Response (a_z Axis)

The estimated mean rms g comfort levels and the standard deviation for male and female subjects are presented in table 2. The shape of the comfort contours and the estimated overall mean rms g levels are indicated in figure 6.

It is significant that the contours bear little resemblance to the ISO 'reduced comfort boundary' contours, particularly in the low and high frequency regions. All contours indicate a maximum sensitivity at 1 Hz, decreasing to a minimum in the region 2 to 3 Hz, increasing to a maximum in the region 5 to 7 Hz, falling away in the region 8 to 15 Hz, and finally increasing at higher frequencies.

The trend observed below 2 Hz correlates well with the observations of Dupuis (ref. 5), Dupuis, Hartung and Louda (ref. 6), Ashley (ref. 7), and Ashley and Rao (ref. 8), although the techniques and objectives differed from those currently employed. The increase in sensitivity in the high frequency region has also been reported by Ashley (ref. 7), Ashley and Rao (ref. 8), Jones and Saunders (ref. 9), Miwa (ref. 10), Shoenberger and Harris (ref. 11), and Oborne and Clarke (ref. 12).

It is also noted that the estimated overall comfort levels appear to be significantly higher than the corresponding ISO 'reduced comfort boundary' standards. Examination of table 2 and figure 6 indicates that human beings in a seated position can comfortably tolerate relatively high g levels in the frequency range 2 to 3 Hz, thus suggesting that seats and suspension systems should be based around a natural frequency of this order. It should be noted that Rao and Jones (ref. 13) and Simic (ref. 14) have observed that this frequency corresponds to the natural frequency of normal walking and that as a result humans possess a high tolerance to rms g levels at this frequency.

The data also indicate that, in general, the male can comfortably withstand higher rms g levels than the female. This tentative conclusion is based upon a small number of subjects and must be viewed with caution.

The increase in sensitivity at high frequencies compared to the ISO standards suggests that from the point of view of comfort, human beings do not prefer high frequency to the body. This aspect was emphasised during the test programme by comments made by subjects that at frequencies greater than 10 Hz, cramp sensations were experienced in the feet and thighs, fluttering sensation in the face and lower back, and speech modulation and blurred vision at around 20 Hz.

Finally, the relationship between estimated rms g level and estimated exposure time appears to be much less exaggerated than expected. (See tables 3(a) and 3(b).) It should be noted that this observation is based upon the mean values. Furthermore, subjects were required to extrapolate their comfort judgement of a short term vibration experience to a long term exposure, which proved extremely difficult for durations exceeding 25 minutes.

Side-to-Side Vibrational Mode (a_y Axis)

The test results are presented in table 4 and figure 7 shows the estimated overall mean rms g levels against frequency for both male and female subjects.

Sensitivity approaches a minimum towards 1 Hz and above 11 Hz tends to increase. It is interesting to note below 10 Hz the contours tend to follow the threshold of perception contours of Meister (ref. 15), Von Bekesy (ref. 16), Kanazawa (ref. 17), citing Ishimoto and Ootsuka, and Loach (ref. 18). In relation to the ISO the contours exhibit a higher comfort threshold below 7 Hz and a lower threshold above 7 Hz.

A number of contributory features reported by test subjects relate to the increased sensation above 7 Hz:

(a) 'Pins and needles' sensation in legs (11 to 13 Hz)

(b) Increased vibratory sensations in stomach, legs, and feet (8 to 9 Hz), causing difficulty in keeping the feet still

(c) 'Pins and needles' sensations in the calf muscles, thighs and buttocks (15 to 20 Hz)

Test subjects reported that at frequencies below 7 Hz, the head, shoulders, hips, knees, and feet were out of phase with each other.

Front-to-Rear Vibrational Mode (a_x Axis)

The test results are presented in table 5 and figure 8 shows the estimated overall comfort levels. In general terms the contours are of a similar form to those for the a_y axis and agree with the threshold of perception characteristics reported by Meister (ref. 15), Von Bekesy (ref. 16), and Kanazawa (ref. 17), citing Ishimoto and Ootsuka.

Below 9 Hz the comfort levels are higher than expected and above 9 Hz the levels are below those of the ISO 'reduced comfort boundary' values. These findings are in good agreement with those relating to the a_y axis. It is interesting to note that females exhibit a higher tolerance to acceleration level than do males.

A comparison of the estimated overall comfort levels between a_x and a_y axes indicates that subjects, in general, can tolerate higher acceleration levels in the a_x axis; a fact which is at variance with the ISO standards.

As expected large standard deviation values have been obtained due to the small sample size and the extrapolation involved.

The percentage deviation in comfort levels between the 16-minute and 2.5-hour exposure durations have been compared with the related ISO comfort levels and the results given in table 6.

CONCLUSIONS

The current investigation was mainly concerned with the object of estimating the overall subjective comfort levels (in weighted rms g) in response to sinusoidal vibrations applied separately to the a_x, a_y, a_z axes. The estimated comfort levels were extrapolated to 16 minutes, 25 minutes, 1 hour, and 2.5 hours. The results have indicated the following broad conclusions:

(1) A significant variation in the form of the contours for all three axes has been observed in comparison with the ISO standards.

(2) Generally, a much higher comfort level is exhibited for the vertical vibration mode (a_z axis). Regarding the a_x and a_y axes the comfort levels are higher in the range 1 to 9 Hz and lower in the range 9 to 20 Hz.

(3) The relationships between comfort levels and exposure duration relating to the a_x and a_y axes differ from the corresponding ISO contours.

While accepting that the sample size is small and the study is nonexhaustive, the a_z axis results support available evidence that the ISO standard requires some modification below 2 Hz. One such modification has recently been proposed by Allen (ref. 19) and is shown in figure 9. In addition the present study suggests that there is need for modification of the contours above 8 Hz.

The a_x and a_y axes data also suggest that the ISO standard requires some modification. Figure 10 indicates a contour profile more in line with the results of the present investigation.

ACKNOWLEDGEMENTS

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APPENDIX A

SUBJECTS' INSTRUCTIONS

The object of this investigation is to assess the overall subjective comfort levels to various sinusoidal vibrations.

Sit straight but relaxed on the cushioned seat mounted on the vibrator platform with your palms on the knees and your feet flat on the vibrator platform. Sit as still and erect as possible without swaying or moving your body unnecessarily. Do not lean against the backrest. Wear the seat belt loosely. Put on the headset for voice communication with experimentor and vibrator operator. A "panic button" switch is conveniently positioned near your right hand. If during the investigation you feel not too happy about the vibration condition due to any reason, you may at any time, stop the functioning of the vibrator by pressing the "panic button."

In this experiment you will be subjected to a certain sinusoidal vibration. Imagine that if you are continuously exposed to this vibration for a prescribed duration of time (say, 16 minutes, 25 minutes, 1 hour or 2.5 hours) what acceleration level would you prefer to be exposed for an overall comfortable ride?

For the purposes of this experiment, the term 'overall comfort level' is defined as the level that you can comfortably tolerate over the prescribed duration while doing the routine tasks (such as reading, writing, sleeping, eating, etc.) during travelling. Give clear instructions to the vibrator operator through the intercom system to adjust the acceleration level you prefer to be comfortably exposed. You may take your own time to reach your decision. After you have positively decided about the preferred comfort level, let the Experimentor know of your decision, so that he can take a few readings before proceeding further. The above procedure will be repeated many times for different frequencies and in different axis of reference.

This investigation is solely dependent on your skill and keenness of your judgement. Please maintain constant alertness throughout the experiment.

If you have any questions please ask them now.

Thanks for your cooperation.

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TABLE 1

Subject Details and Statistical Analysis

Subject Identification Male- M Female-F	Age (years)	Height (cm)	Weight (Kg)	Nationality
M. 1	21	175.3	73.2	British
M. 2	28	173.7	70	British
M. 3	24	170	66.8	British
M. 4	32	178.8	63.6	British
F. 1	26	181.2	70	British
F. 2	21	167.6	59.5	British
F. 3	18	160	52.3	British

Statistical parameters	Subject	Age (years)	Height (cm^{2})	Weight (Kg)
Mean	Males	26.3	174.45	68.4
	Females	21.7	169.6	60.6
Standard Deviation	Males	4.79	3.65	4.13
	Females	4.04	10.74	8.9

TABLE 2

Comfort Evaluation Studies of Males and Females.
Comparison of Mean RMS 'g' and SD Results.
a_z axis excitation.

Time	Freq. (Hz)	Reduced Comfort Boundary Levels from ISO (RMS 'g')	Mean RMS 'g' Values of Males	Mean RMS 'g' Values of Females	Mean RMS 'g' Values of Males + Females	SD Values of Males	SD Values of Females	SD Values of Males + Females
16 MIN	1	0.14	0.17	0.10	0.14	0.04	0.06	0.06
	3	0.076	0.21	0.17	0.19	0.08	0.16	0.10
	5	0.066	0.14	0.09	0.11	0.01	0.06	0.04
	6	0.066	0.13	0.10	0.11	0.05	0.06	0.05
	8	0.066	0.16	0.15	0.16	0.08	0.06	0.07
	10	0.082	0.16	0.20	0.17	0.08	0.17	0.10
	15	0.12	0.20	0.13	0.17	0.06	0.11	0.09
	19	0.155	0.16	0.08	0.13	0.05	0.05	0.06
25 MIN	1	0.115	0.10	0.07	0.09	0.06	0.02	0.05
	2	0.08	0.16	0.13	0.15	0.04	0.03	0.04
	4	0.056	0.13	0.11	0.12	0.02	0.04	0.03
	7	0.056	0.14	0.11	0.13	0.03	0.04	0.03
	9	0.062	0.16	0.13	0.15	0.04	0.01	0.03
	11	0.076	0.10	0.11	0.10	0.03	0.01	0.02
	13	0.09	0.13	0.07	0.11	0.07	0.01	0.06
	17	0.117	0.12	0.05	0.09	0.02	0.02	0.04

Continuation.....

Table 2. Comfort Evaluation Studies of Males and Females. Comparison of Mean RMS 'g' and SD Results.

Time	Freq (Hz)	Reduced Comfort Boundary Levels from ISO (RMS 'g')	Mean RMS 'g' values of			SD Values of		
			Males	Females	Males + Females	Males	Females	Males + Females
	1	0.076	0.12	0.06	0.10	0.05	0.01	0.05
	3	0.044	0.11	0.15	0.13	0.04	0.10	0.07
	5	0.037	0.10	0.10	0.10	0.02	0.07	0.04
1	6	0.037	0.07	0.09	0.08	0.05	0.05	0.05
HOUR	8	0.037	0.11	0.09	0.11	0.07	0.04	0.06
	10	0.047	0.10	0.12	0.11	0.06	0.01	0.05
	15	0.070	0.09	0.10	0.09	0.08	0.03	0.06
	19	0.088	0.08	0.07	0.08	0.04	0.02	0.03
	1	0.046	0.08	0.06	0.07	0.06	0.01	0.05
	2	0.032	0.11	0.14	0.12	0.04	0.05	0.05
	4	0.022	0.12	0.08	0.10	0.02	0.01	0.03
	7	0.022	0.06	0.05	0.05	0.05	0.03	0.04
	9	0.025	0.10	0.05	0.08	0.05	0.02	0.05
	11	0.031	0.10	0.08	0.09	0.08	0.04	0.06
2.5	13	0.037	0.11	0.05	0.08	0.10	0.02	0.08
HOURS	17	0.048	0.08	0.05	0.07	0.08	0.02	0.06

TABLE 3 (a)

a_z axis Excitation. Percentage Deviation in RMS 'g' Level Between 16-minute and 1-hour 'Reduced Comfort Boundary' Contours

Frequency, Hz	1	3	5	6	8	10	15	19
ISO 2631 (1974) (in %)	44	43	43	44	44	43	43	43
Current Findings (in %)	28	31	9	27	31	35	47	38

TABLE 3 (b)

a_z axis Excitation. Percentage Deviation in RMS 'g' Level Between 25-minute and 2.5-hour 'Reduced Comfort Boundary' Contours

Frequency, Hz	1	2	4	7	9	11	13	17
ISO 2631 (1974) (in %)	60	60	60	60	60	60	60	60
Current Findings (in %)	22	20	16	61	46	10	27	22

TABLE 4

Comfort Evaluation Studies of Males and Females.

Comparison of Mean RMS 'g' and SD Results

a_y axis excitation

Time	Freq (Hz)	Reduced Comfort Boundary Levels from ISO (RMS 'g')	Mean RMS 'g' values of			SD Values of		
			Males	Females	Males + Females	Males	Females	Males + Females
16 MIN	1	0.048	0.32	0.13	0.24	0.20	0.07	0.18
	3	0.071	0.28	0.12	0.21	0.14	0.11	0.15
	5	0.12	0.26	0.10	0.19	0.12	0.09	0.13
	7	0.164	0.27	0.08	0.19	0.20	0.08	0.18
	9	0.215	0.26	0.13	0.20	0.18	0.12	0.16
	11	0.26	0.30	0.09	0.21	0.24	0.10	0.21
	13	0.31	0.27	0.10	0.19	0.16	0.09	0.15
	17	0. 40	0.25	0.07	0.17	0.18	0.06	0.16
25 MIN	1	0.04	0.23	0.09	0.17	0.13	0.07	0.12
	3	0.06	0.15	0.09	0.12	0.09	0.09	0.08
	5	0.10	0.16	0.06	0.12	0.12	0.06	0.10
	7	0.14	0.18	0.06	0.13	0.15	0.03	0.12
	9	0.18	0.20	0.08	0.15	0.14	0.06	0.12
	11	0.22	0.17	0.06	0.13	0.11	0.05	0.10
	13	0.26	0.21	0.07	0.15	0.16	0.05	0.14
	17	0.34	0.15	0.04	0.10	0.14	0.03	0.11

Continuation .....

Table 4. Comfort Evaluation Studies of Males and Females. Comparison of Mean RMS 'g' and SD Results

Time	Freq. (Hz)	Reduced Comfort Boundary Levels from ISO (RMS 'g')	Mean RMS 'g' values of			SD Values of		
			Males	Females	Males + Females	Males	Females	Males + Females
1 HOUR	1	0.027	0.18	0.08	0.14	0.16	0.03	0.13
	3	0.04	0.14	0.08	0.11	0.11	0.07	0.09
	5	0.068	0.11	0.05	0.08	0.09	0.03	0.07
	7	0.094	0.16	0.07	0.12	0.15	0.03	0.12
	9	0.12	0.15	0.06	0.11	0.13	0.04	0.11
	11	0.15	0.14	0.07	0.11	0.13	0.04	0.10
	13	0.17	0.16	0.06	0.12	0.14	0.04	0.11
	17	0.23	0.12	0.04	0.09	0.10	0.03	0.08
2.5 HOURS	1	0.016	0.11	0.10	0.11	0.08	0.05	0.06
	3	0.024	0.09	0.06	0.08	0.07	0.06	0.06
	5	0.04	0.09	0.05	0 07	0.08	0.04	0.06
	7	0.057	0.12	0.06	0.09	0.13	0.04	0.10
	9	0.074	0.10	0.08	0.09	0.09	0.06	0.07
	11	0.09	0.09	0.07	0.08	0.10	0.05	0.08
	13	0.105	0.08	0.06	0.08	0.08	0.06	0.07
	17	0.14	0.07	0.04	0 .05	0.05	0.03	0.04

TABLE 5

Comfort Evaluation Studies of Males and Females. Comparison of Mean RMS 'g' and SD Results.

a_x axis excitation

Time	Freq. (Hz)	Reduced Comfort Boundary Levels from ISO (RMS 'g')	Mean RMS 'g' Values of Males	Mean RMS 'g' Values of Females	Mean RMS 'g' Values of Males + Females	SD Values of Males	SD Values of Females	SD Values of Males + Females
16 MIN	1	0.048	0.21	0.42	0.30	0.09	0.18	0.16
	3	0.071	0.16	0.27	0.21	0.06	0.17	0.12
	5	0.12	0.14	0.24	0.18	0.09	0.10	0.10
	7	0.165	0.15	0.24	0.19	0.07	0.08	0.08
	9	0.215	0.19	0.38	0.27	0.09	0.29	0.20
	11	0.26	0.18	0.28	0.23	0.05	0.20	0.13
	13	0.31	0.14	0.30	0.21	0.04	0.24	0.16
	17	0.40	0.13	0.20	0.16	0.04	0.15	0.10
25 MIN	1	0.04	0.33	0.37	0.35	0.26	0.10	0.20
	3	0.06	0.24	0.24	0.24	0.19	0.18	0.17
	5	0.10	0.24	0.23	0.23	0.25	0.17	0.20
	7	0.14	0.24	0.22	0.23	0.19	0.07	0.14
	9	0.18	0.21	0.24	0.22	0.15	0.13	0.13
	11	0.22	0.21	0.18	0.20	0.10	0.10	0.09
	13	0.26	0.24	0.17	0.21	0.14	0.10	0.12
	17	0.34	0.21	0.11	0.17	0.18	0.05	0.14

Continuation ..... Table 5. Comfort Evaluation Studies of Males and Females. Comparison of Mean RMS 'g' and SD Results.

Time	Freq. (Hz)	Reduced Comfort Boundary Levels from ISO (RMS 'g')	Mean RMS 'g' Values of			SD Values of		
			Males	Females	Males + Females	Males	Females	Males + Females
1 HOUR	1	0.027	0.18	0.21	0.20	0.06	0.08	0.06
	3	0.04	0.16	0.20	0.18	0.12	0.05	0.09
	5	0.068	0.15	0.15	0.15	0.10	0.05	0.08
	7	0.094	0.10	0.11	0.11	0.03	0.02	0.03
	9	0.12	0.12	0.17	0.14	0.03	0.13	0.08
	11	0.15	0.14	0.16	0.15	0.06	0.07	0.06
	13	0.17	0.10	0.16	0.13	0.03	0.08	0.06
	17	0.23	0.08	0.12	0.10	0.02	0.04	0.03
2.5 HOURS	1	0.016	0.08	0.37	0.20	0.03	0.14	0.17
	3	0.024	0.04	0.22	0.10	0.01	0.08	0.10
	5	0.04	0.06	0.13	0.09	0.02	0.06	0.05
	7	0.057	0.07	0.14	0.10	0.01	0.07	0.05
	9	0.074	0.06	0.14	0.09	0.02	0.08	0.06
	11	0.09	0.05	0.13	0.08	0.005	0.05	0.04
	13	0.105	0.06	0.14	0.09	0.01	0.06	0.05
	17	0.14	0.06	0.09	0.07	0.008	0.03	0.02

TABLE 6

a_x axis Excitation. Percentage Deviation in RMS 'g' Level Between 16-minute and 2.5-hour 'Reduced Comfort Boundary' Contours

Frequency, Hz	1	3	5	7	9	11	13	17
ISO 2631 (1974) (in %)	66	66	66	66	66	66	66	66
Current Findings (in %)	33	52	50	47	66	65	57	56

a_y axis Excitation. Percentage Deviation in RMS 'g' Level Between 16-minute and 2.5-hour 'Reduced Comfort Boundary' Contours

Frequency, Hz	1	3	5	7	9	11	13	17
ISO 2631 (1974) (in %)	66	66	66	66	66	66	66	66
Current Findings (in %)	54	62	63	52	55	62	57	70

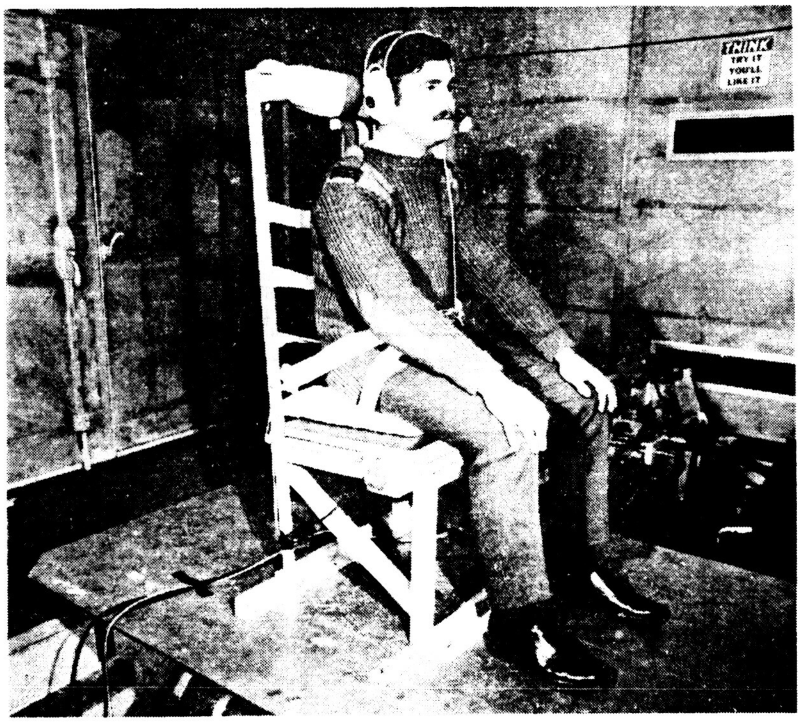


Plate 1.- Subject posture.

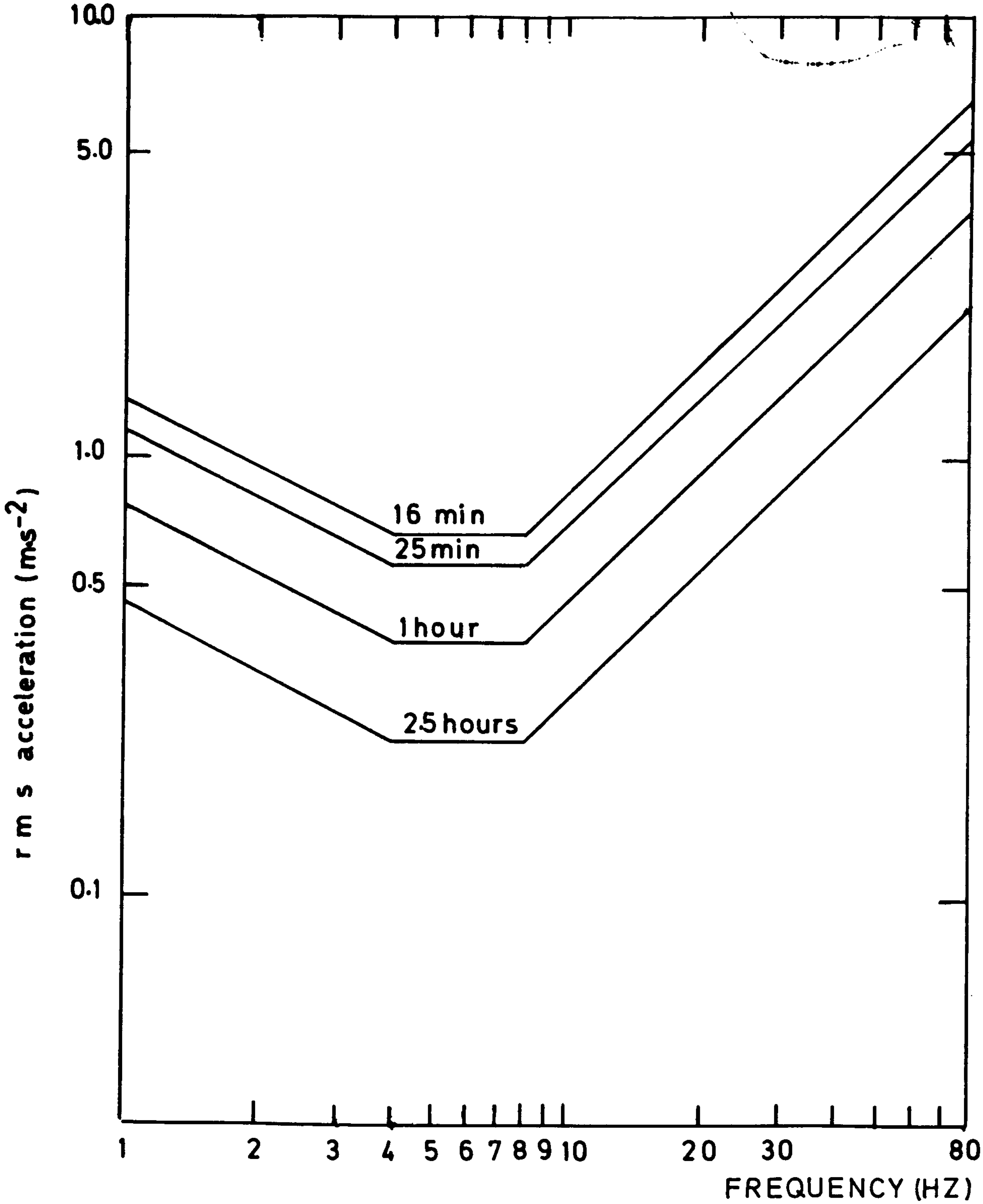


Figure 1.- Reduced comfort boundary (a_z).

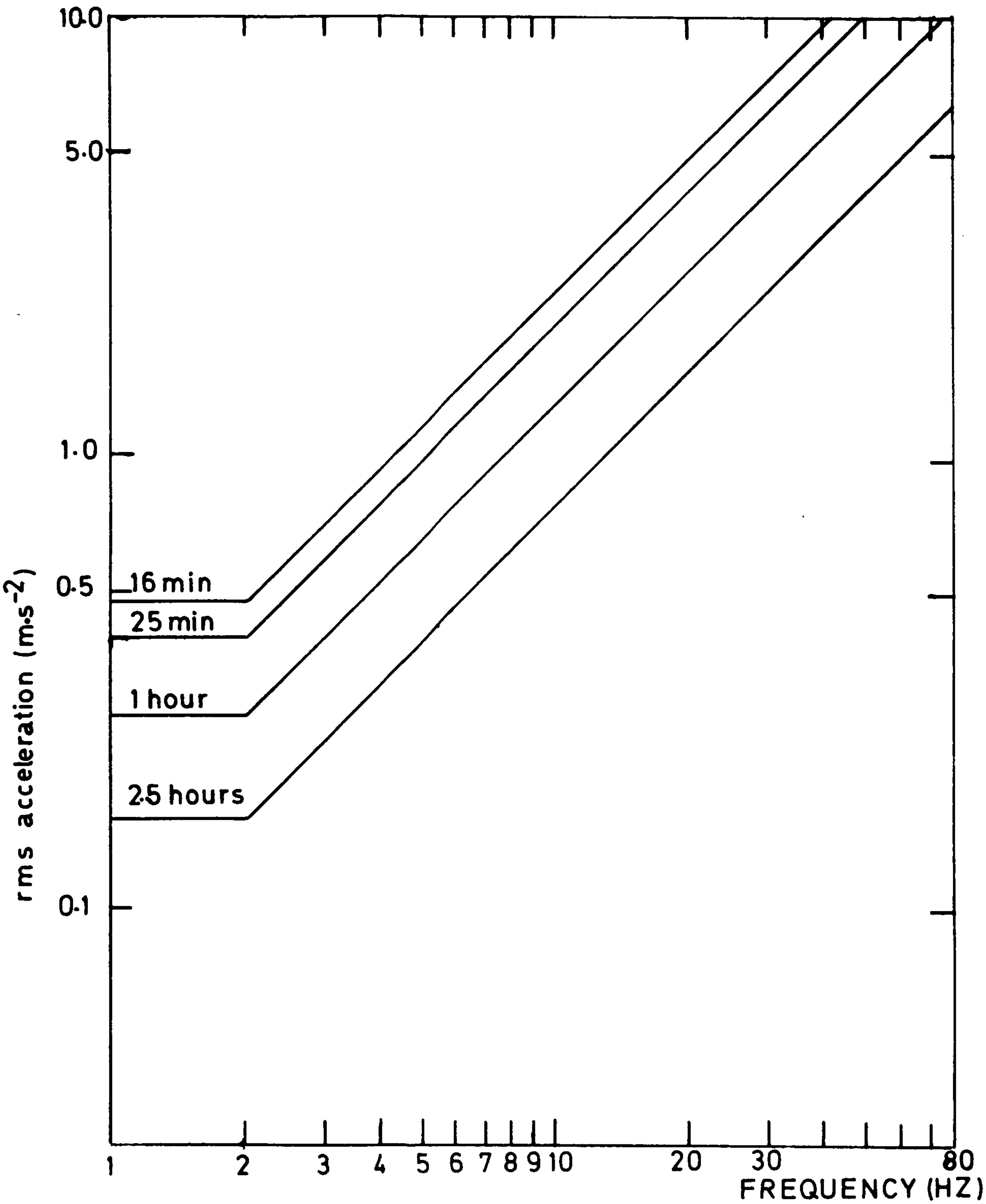
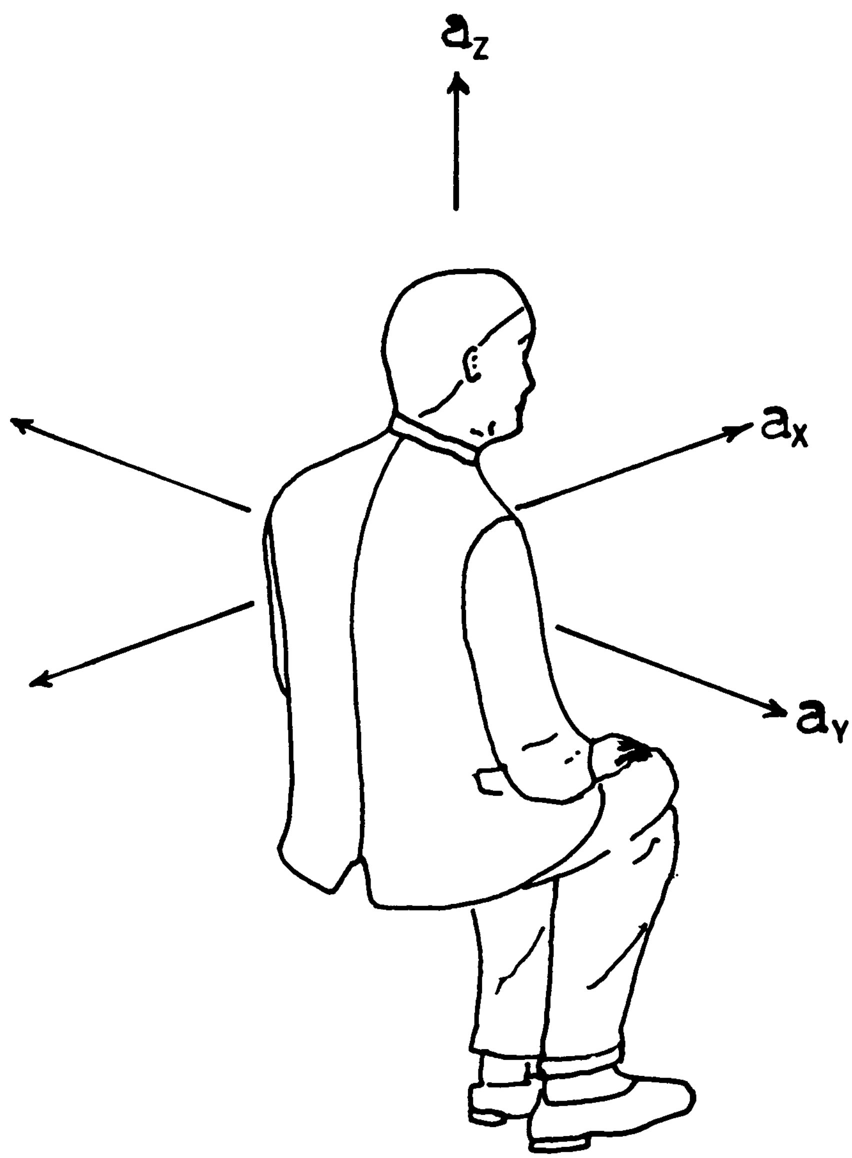
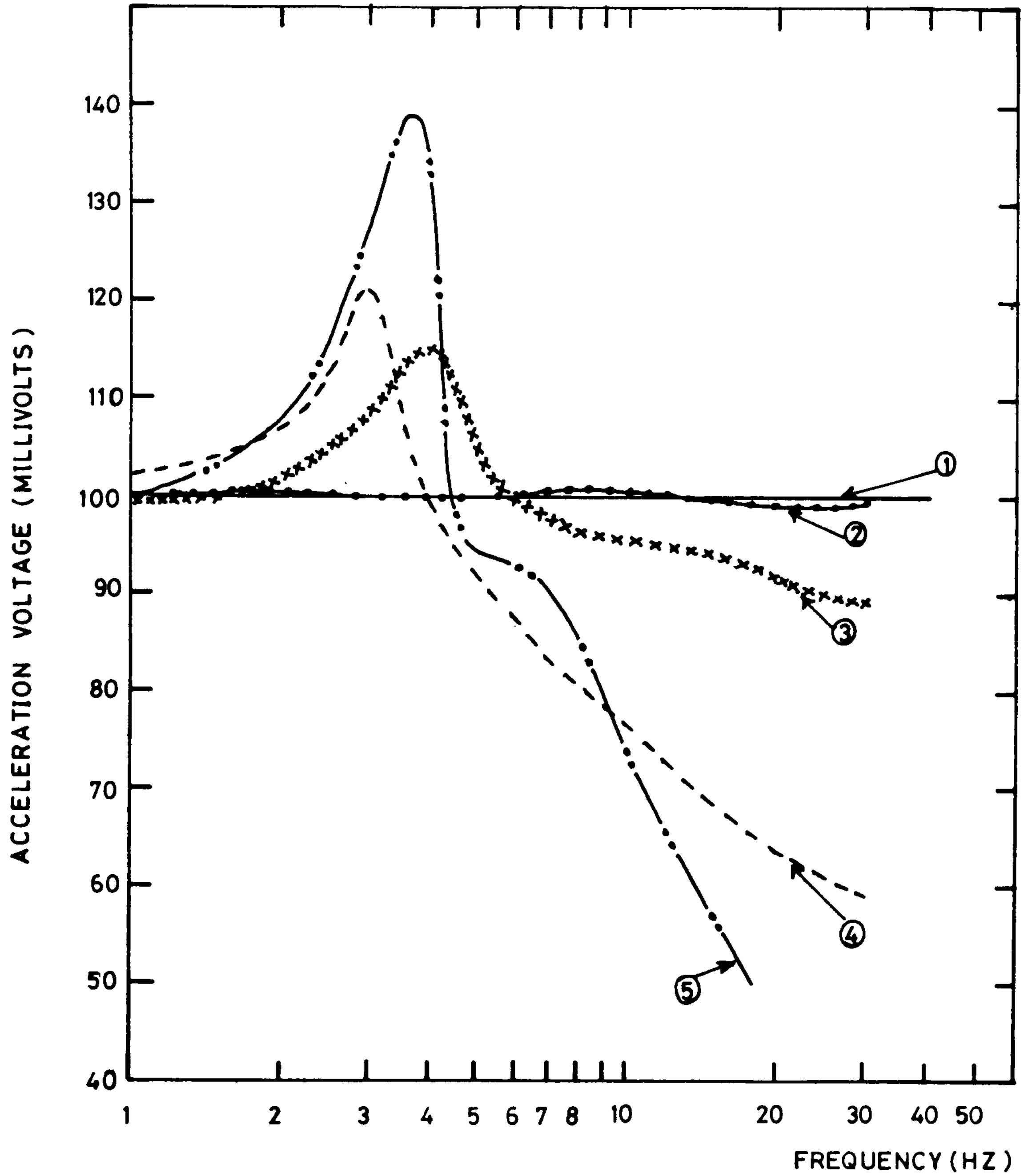


Figure 2.- Reduced comfort boundary (a_x and a_y).



a_x, a_y, a_z	= acceleration in the directions of the x,y,z axes
x axis	= back-to-chest
y axis	= right-to-left side
z axis	= foot-to-head

Figure 3.- Coordinate system for mechanical vibrations influencing humans.



① INPUT ACCELERATION ② BIRDSEED CUSHION RESPONSE

③ COMPOSITE (BIRDSEED ON 2.54CM SPONGE RUBBER) CUSHION RESPONSE

④ PLASTIC FOAM CUSHION RESPONSE ⑤ SPONGE RUBBER (5.1CM) CUSHION RESPONSE

Figure 4.- Dynamic response of cushions at subject-seat interface.

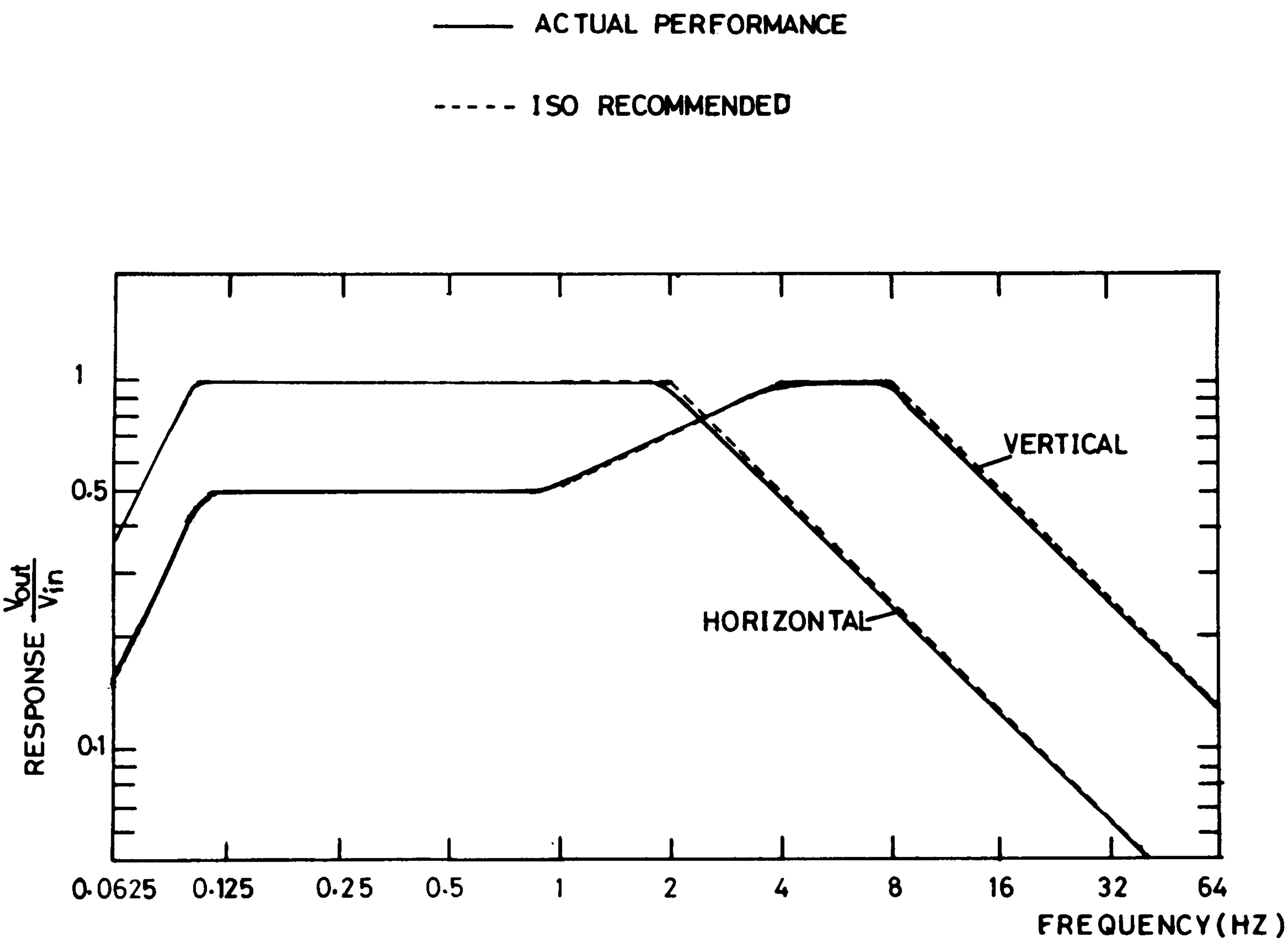


Figure 5.- Weighting filter characteristics.

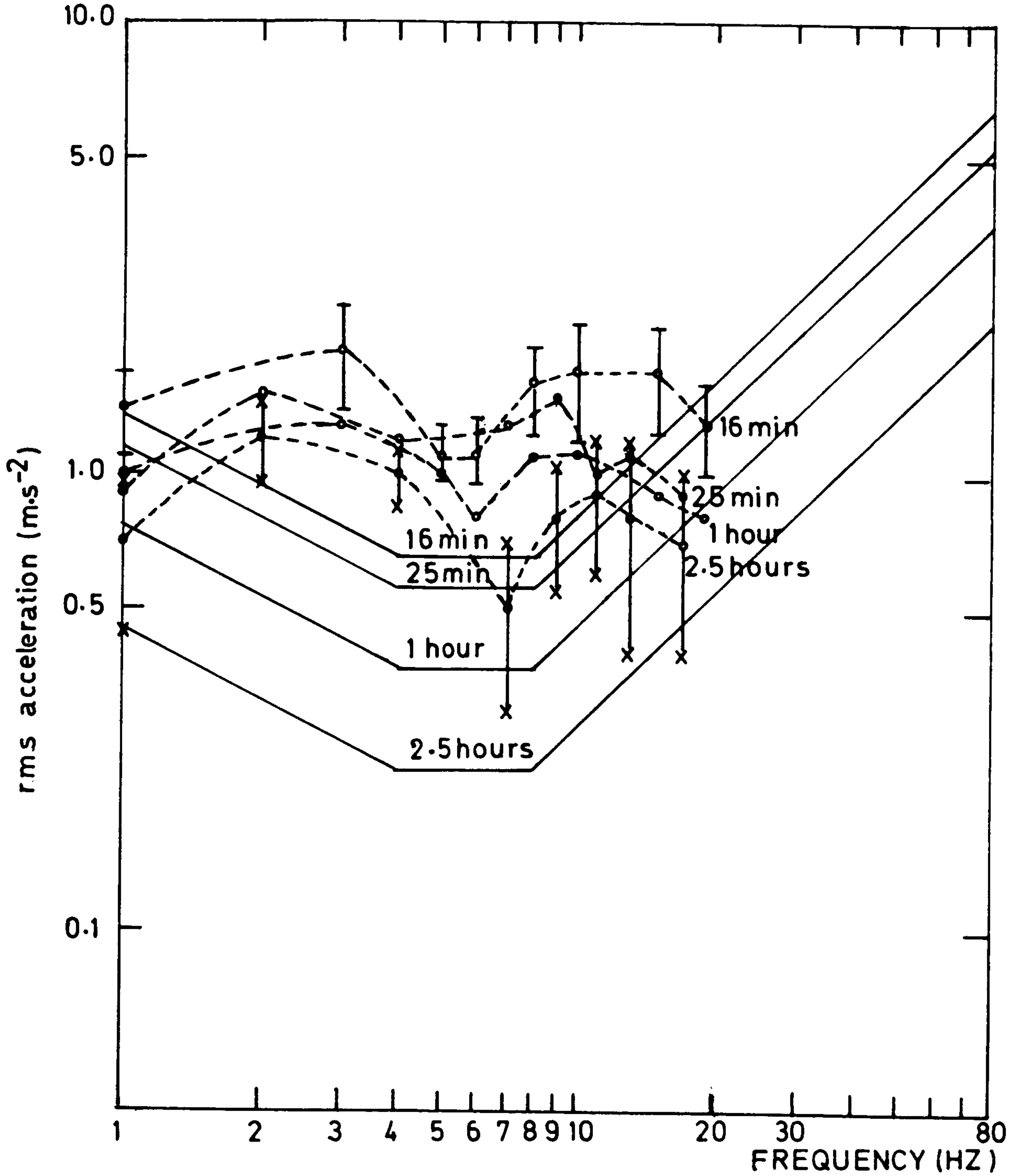


Figure 6.- Estimated comfort contours (a_z axis).

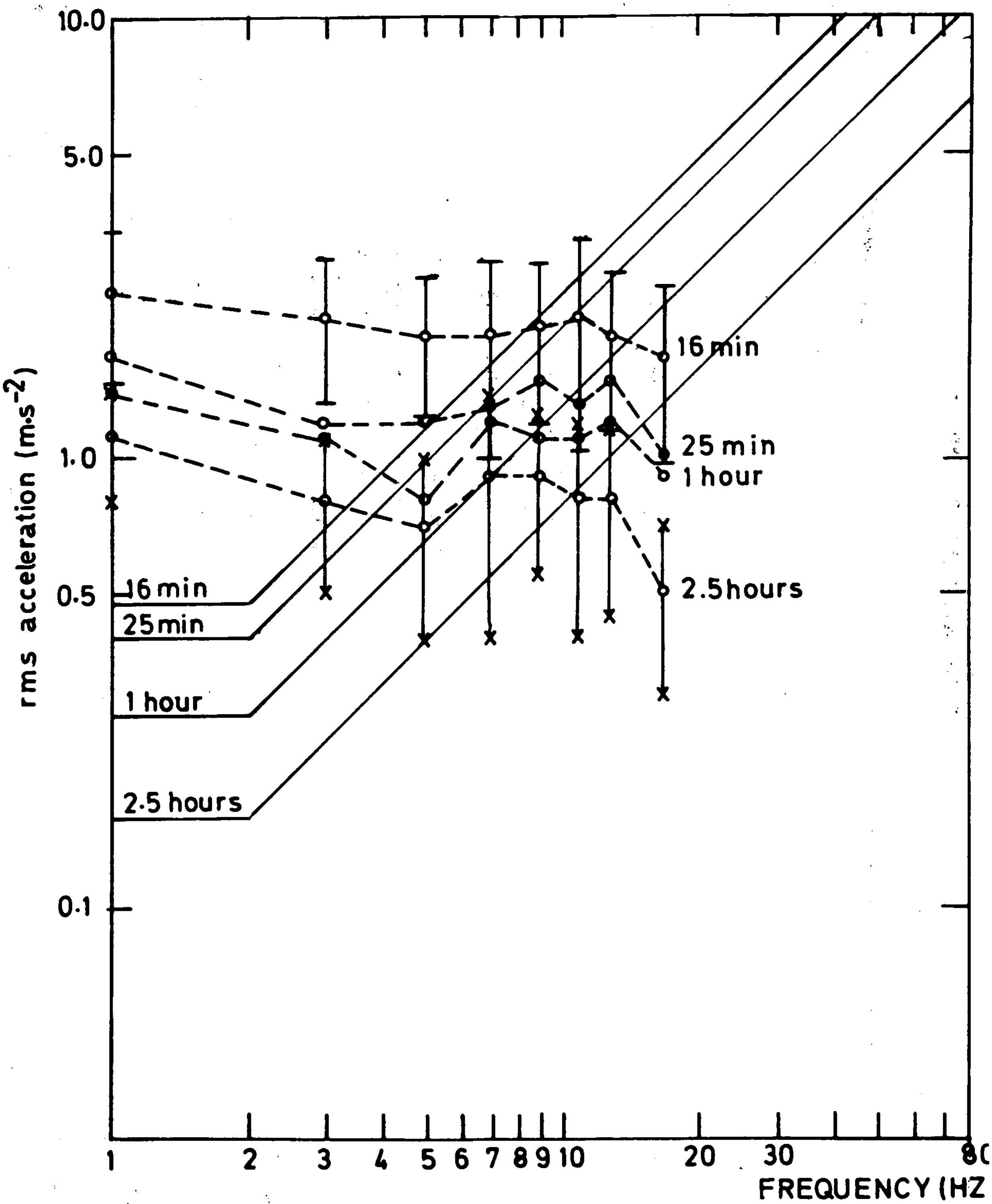


Figure 7.- Estimated comfort contours (a_y axis).

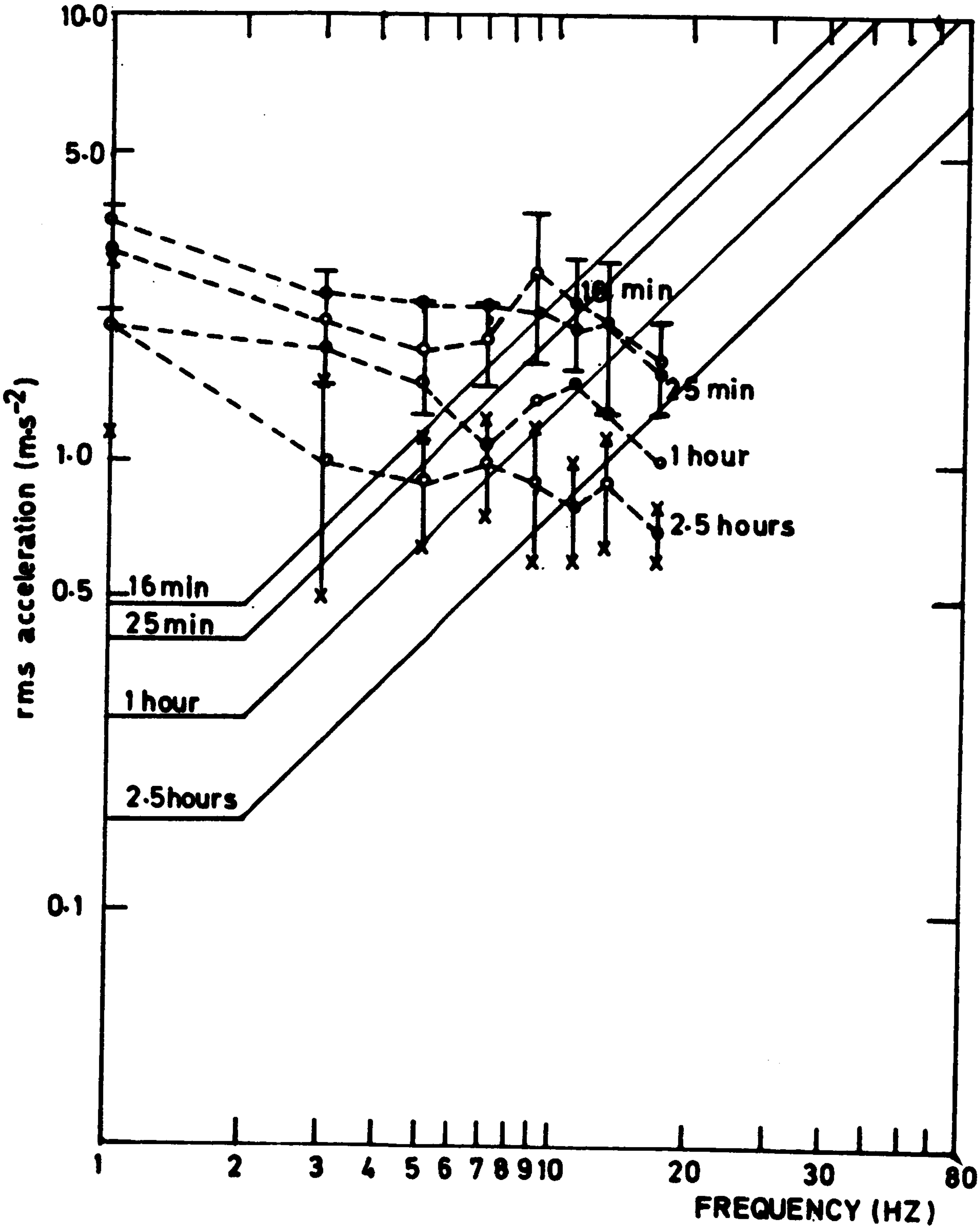


Figure 8.- Estimated comfort contours (a_x axis).

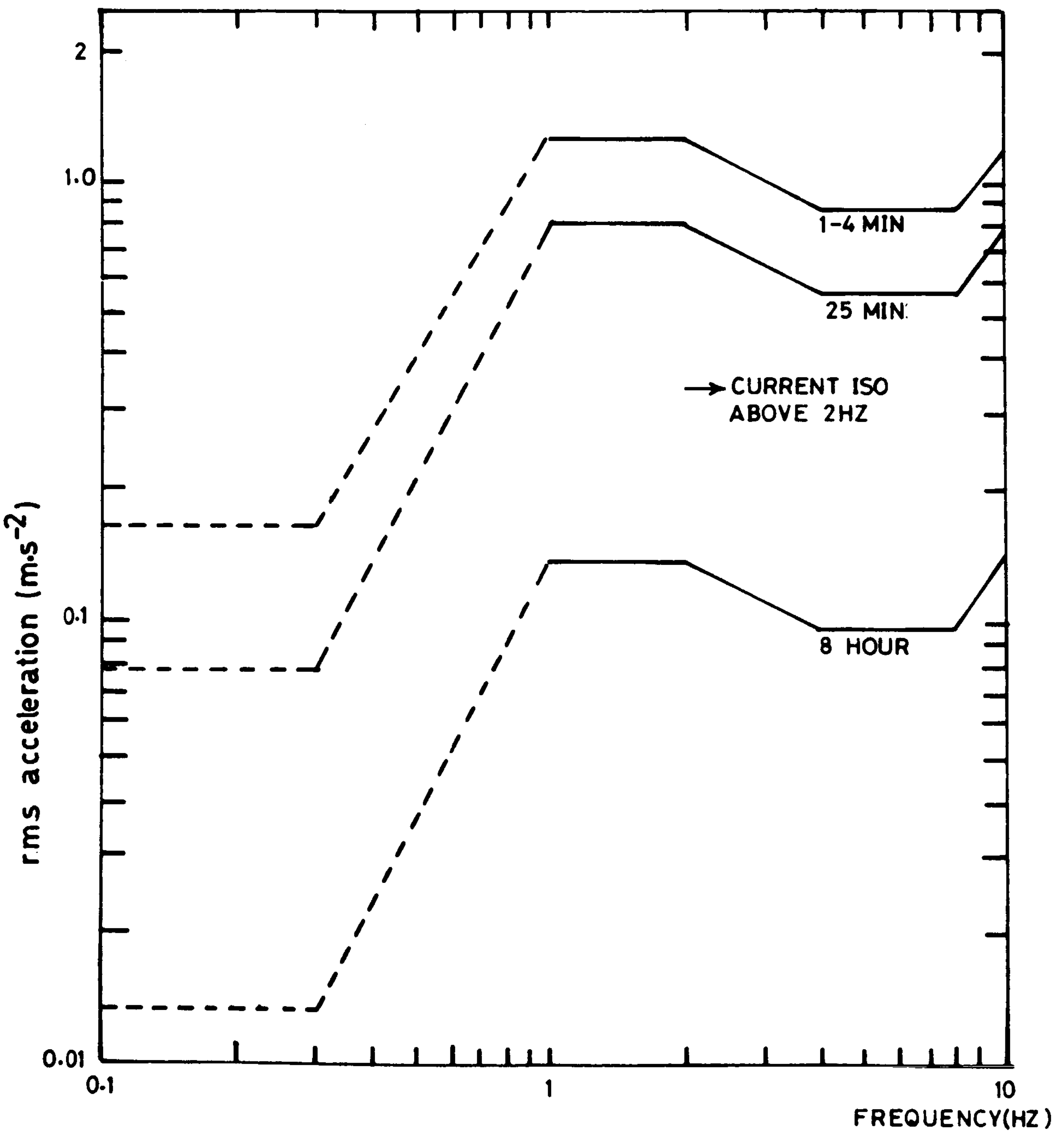


Figure 9.- Proposed vertical vibration reduced comfort boundary >0.1 Hz.

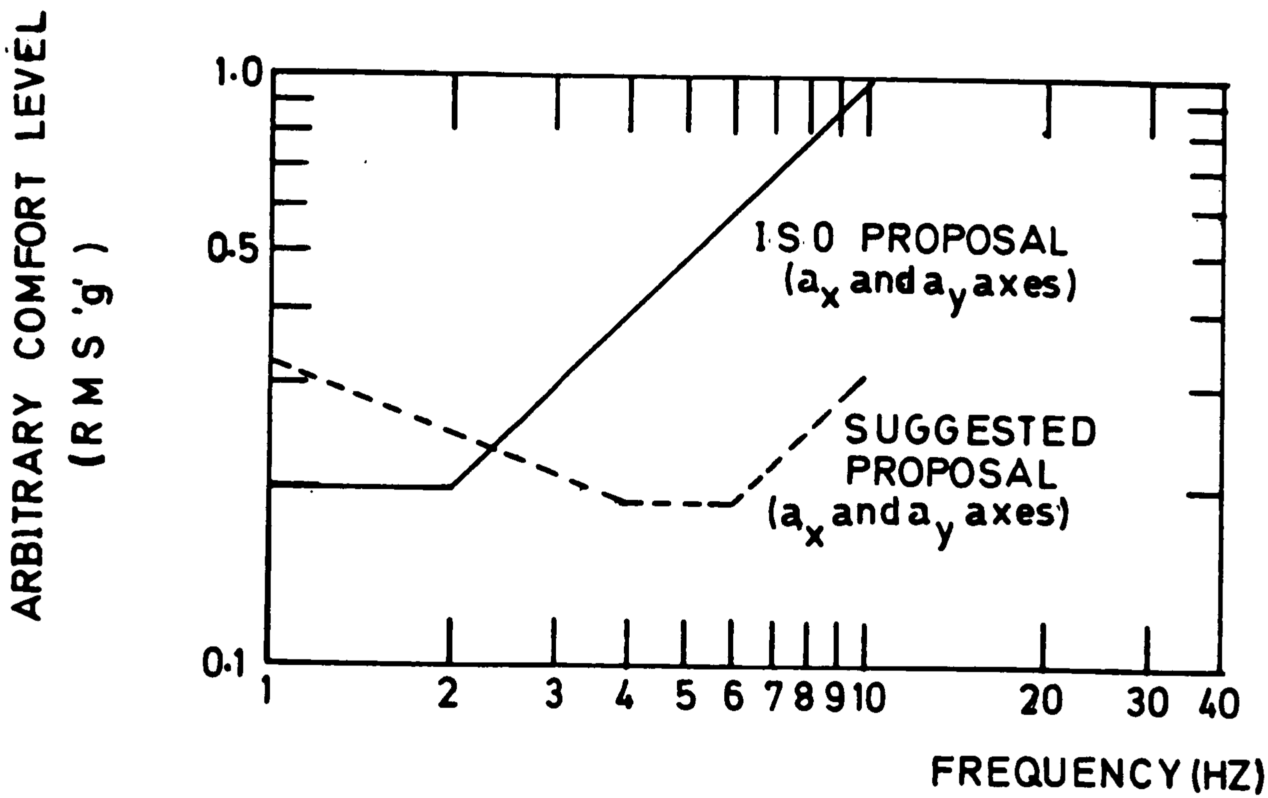


Figure 10.- A suggested contour shape for a_x, a_y axes response.

N76-16769

EFFECT OF VIBRATION IN COMBINED AXES ON SUBJECTIVE EVALUATION OF RIDE QUALITY

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INTRODUCTION

The first study of subjective evaluations of ride quality produced by simultaneous vibrations occurring in more than one axis was reported by Jacklin and Liddell (ref. 1). The results of that study showed that introduction of various combinations of amplitudes and frequencies in the horizontal axis lowered the thresholds for ratings of Disturbing and Uncomfortable in the vertical axis, for frequencies below 7 Hz. The experimental design of the study, however, did not permit detection of interactions between the effects of vertical and horizontal vibrations on subjective ratings.

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Holloway and Brumaghim (ref. 2) have studied the effects of narrow-band, random-frequency vibrations with center frequencies between 0.20 and 7 Hz applied simultaneously to the vertical and lateral axes. That study showed that increasing the amplitude of vibrations in the lateral axis led to lower levels of amplitude in the vertical axis being rated as Objectionable. As with the Jacklin and Liddell study, it was beyond the scope of the research to study possible interactions between the effects of vibrations in the two axes.

The studies herein reported investigated the effects of simultaneous sinusoidal vibration in the vertical and lateral axes on ratings of discomfort. The first experiment concentrated on the effects of variation of frequency in the two axes, and the second study concentrated on the effects of amplitude variation in the two axes.

EXPERIMENT I - VARIATION OF FREQUENCY

Subjects

The subjects for this research were 11 males and 13 females recruited from the undergraduate student body of Old Dominion University. The 24 subjects used were recruited from a larger list of volunteers who had been medically screened and approved by Langley Research Center. The mean age of the subjects was 23.7 years and the standard deviation of the ages was 8.2 years.

Apparatus

The apparatus used in this experiment was the Langley passenger ride quality apparatus (PRQA). This apparatus, designed as a simulated passenger aircraft, can present subjects with whole-body vibration of various frequencies, amplitudes, and waveforms in the vertical, lateral (side-to-side), longitudinal (fore-and-aft), pitch, and roll axes. For this experiment the PRQA was equipped with six tourist-class seats. Additional details about the PRQA can be obtained from Clevenson and Leatherwood (ref. 3) and Stephens and Clevenson (ref. 4).

Design

The experimental design used was treatments by treatments by sessions with subjects nested under sessions (Winer, ref. 5). The first treatment variable was the frequency of vibration input in the vertical axis; the 10 levels of vertical frequency employed were 0, 1, 2, 3, 4, 5, 8, 10, 15, and 20 Hz. The second treatment variable was frequency of vibration input in the lateral axis; the same 10 levels of frequency were used in the lateral axis as were used in the vertical. Groups of six subjects were tested simultaneously on the PRQA, and there were four groups, or sessions. For each group of subjects the apparatus was set at one level of vertical frequency, and all levels of lateral frequency were presented in random order with that vertical frequency. Then the next level of vertical frequency was presented. A different random order of lateral frequencies was used for each level of vertical frequency and a different random order of vertical frequencies was used for each of the four sessions. The amplitude of all stimuli was 0.15g (peak).

Rating Scale

The rating scale employed was a 9-point, unipolar scale. For each stimulus the subject was provided with a separate scale consisting of a line with 9 divisions, numbered from 0 to 8. Above the 0 was the anchor Comfortable or zero discomfort and above the 8 was Maximum discomfort. The subjects were instructed to use the scale as an equal-interval scale, rating stimuli between the numbered divisions as well as on them. The subjects were also instructed to rate the discomfort produced by the stimuli. Before beginning each new level of vertical frequency, the subjects were presented with two anchor stimuli. The first had no vertical input and a lateral input of 10 Hz and was described as "One that many people might give a low number rating". The second had a vertical input of 4 Hz and a lateral input of 5 Hz, and was described as "One that many people would probably assign a high number rating".

Procedure

The subjects were transported to the Langley Research Center from Old Dominion University, a distance of approximately 40 km (25 miles), in a late-model, nine-passenger station wagon. Upon arriving at the

Langley Research Center the subjects were taken to a conference room adjacent to the room housing the PRQA. Here the subjects were given their instructions regarding the experiment and appropriate safety procedures. The subjects were then seated in the PRQA and asked to fasten their seat belts.

Throughout the testing, two-way audio communication was maintained with the subjects and the subjects were also continually observed through a one-way mirror as part of the safety procedures.

Instructions regarding the anchor stimuli and the test stimuli were recorded on audiotape. At the beginning of each test stimulus the subjects were told "Begin" and at the end of the stimulus presentation the subjects were told "Rate". Each trial consisted of 5 seconds for the stimulus to reach the appropriate level, 15 seconds of stimulus, 5 seconds for the offset of the stimulus, and 10 seconds between trials. The subjects were given a 1-minute rest between each series of 10 stimuli and a 15-minute intermission halfway through the testing, i.e., after 50 stimuli.

Results

Table 1 shows the results of analysis of variance with repeated measures on two variables. Clearly, the most significant variable affecting the ratings of the subjects was the frequency of lateral vibrations. The effect of frequency in the vertical axis was also significant, as was the interaction between these two variables. The interaction appears to be due to each axis masking the effects of the other axis at frequencies rated as being of maximum discomfort, with the lateral axis masking the effects of the vertical more than in the reverse direction.

Figure 1 shows the mean ratings of the subjects as a function of the frequency of vertical input with frequency of lateral input as a parameter. Figure 2 shows the same data but with the ratings as a function of lateral frequency with vertical frequency as a parameter. The lateral axis appears to have a dominant effect at lower frequencies, whereas at higher frequencies the relative significance of the vertical axis is much greater than it is at lower frequencies. The significant interaction appears to be due to each axis masking the effects of the other axis at frequencies rated at maximum discomfort in the former axis, with the lateral axis masking the effects of the vertical more than in the reverse direction.

A multiple-regression analysis was subsequently computed using the physical measures of vertical and lateral frequency and various nonlinear transformations of these measures to predict the subjective responses of discomfort. The resulting predictive equation was used to generate the response surface presented in figure 3; it should be noted that the multiple correlation coefficient associated with the criterion variable and the predictor variables was 0.685, accounting for 47 percent of the variability in the individual subjective responses.

EXPERIMENT II - VARIATION OF AMPLITUDE AND FREQUENCY

Whereas the first experiment was primarily concerned with the effects of variation in frequency of vibrations simultaneously presented in the two axes, this experiment was concerned with the effects of variation of amplitude in the two axes on ratings of discomfort, and with interactions between the effects of amplitude and the effects of frequencies.

Subjects

The subjects for this research were 72 undergraduate students recruited from the student body of Old Dominion University in a manner similar to that used in recruiting subjects for Experiment I.

Apparatus

As in Experiment I the apparatus used was the Langley passenger ride quality apparatus (PRQA).

Design

The experimental design used was a 4 × 4 × 4 × 4 factorial design with 12 subjects nested in each of the vertical frequencies and with repeated measures over the vertical amplitudes, the lateral frequencies, and the lateral amplitudes. Thus, each subject was exposed to only one of the four vertical frequencies but experienced that frequency at each of its four amplitudes combined with 16 (or 4 × 4) lateral frequency and amplitude conditions. The four levels of vertical frequency were 2, 5, 9, and 15 Hz. The four levels of vertical amplitude planned were 0.05g, 0.10g, 0.15g, and 0.25g (peak). The four levels of lateral frequency were 2, 4, 8, and 16 Hz, and the four levels of lateral amplitudes planned were, like the vertical amplitudes, 0.05g, 0.10g, 0.15g, and 0.25g (peak). In addition, as a control condition, 12 other subjects experienced each of the vertical frequencies at each of the four amplitudes in the absence of lateral input. As a final control, another group of 12 subjects experienced each of the lateral frequencies at each of the four amplitudes in the absence of vertical input.

Groups of 6 subjects were tested on the PRQA simultaneously; 12 such groups were tested. For each of the 10 experimental groups plus 2 control groups that experienced lateral vibration, the apparatus was set at a level of lateral frequency and all combinations of vertical amplitude and lateral amplitude were presented with that level of lateral frequency before going on to another level of lateral frequency. For the control group that received only vertical input, the apparatus was set at a level of vertical frequency and all levels of vertical amplitude were presented with that before going on to another level of vertical frequency. To the extent possible, the order of presentation of levels of amplitude was counterbalanced.

Procedure

The rating scale and procedure used were the same as in Experiment I, except that the anchor stimuli and a 1-minute rest were given after each 8 trials rather than after each 10 trials.

RESULTS AND DISCUSSION

Before considering the analyses of the subjective ratings, a comparison was made between the amplitudes that were planned, the input amplitudes, and the amplitudes that were recorded from the PRQA during the testing, the output amplitudes. Although the magnitudes of the output amplitudes differed slightly from the input amplitudes, there appeared to be no major systematic variations between the planned inputs and the outputs across the experimental conditions. As noted above, the amplitudes that were planned were 0.05g, 0.10g, 0.15g, and 0.25g (peak); the means of the amplitude outputs were 0.06g, 0.10g, 0.15g, and 0.26g (peak).

The results of the analysis of variance of the ratings of discomfort, excluding the control conditions, are shown in table 2. All four main effects (vertical frequency, vertical amplitude, lateral frequency, and lateral amplitude) were significant, as were all six of the simple interactions between these four parameters of vibration. Two of the triple interactions were significant, as was the four-way interaction.

Figures 4 to 7 show the mean ratings of the subjects as a function of each of the parameters of vibration. These figures were obtained by averaging across all the remaining experimental conditions not shown in each figure. The first two of the figures, figures 4 and 5, show that the main effects found in Experiment I, regarding the effects of frequency on ratings of discomfort, were replicated in the second experiment. Figures 6 and 7 show that the effect of increasing amplitude of vibration in either axis is to increase ratings of discomfort, an expected finding.

The more interesting and important findings of the experiment are shown in figures 8 to 13, which show the simple interactions between the six pairs of vibration parameters. In each of these figures the discomfort ratings were averaged across both of the vibration parameters not shown in each figure, thus revealing the form of the interaction between the two variables that are shown. The interaction shown in figure 8, between vertical frequency and lateral frequency, is a replication of the interaction found in Experiment I, and shown in figure 1.

Figure 9 shows the interaction between the effects of the vertical amplitude and the lateral amplitude. It appears that the form of this interaction is terminative, since high amplitudes in either axis tend to mask the effects of variation in amplitude in the other axis.

The interactions between frequency and amplitude within each axis are shown in figure 10 for the vertical axis and figure 11 for the lateral axis. In both figures the effect of variation in amplitude is greatest at those frequencies rated as being of most discomfort while amplitude variation had less effect at frequencies rated as being of less discomfort.

The interactions between frequency in one axis and amplitude in the other are shown in figures 12 and 13. First, the interaction between vertical frequency and lateral amplitude is shown in figure 12; the other interaction, between lateral frequency and vertical amplitude, is shown in figure 13. In contrast to the form of the interaction shown in figures 10 and 11, these interactions are in the opposite direction, with amplitude variation having the greatest effect at frequencies rated as being of least discomfort. Perhaps a more appropriate conclusion, however, is that at frequencies rated as being of most discomfort, there is some masking of amplitude effects from the other axes while the effects of amplitude from the same axis are enhanced

Regarding the simple interactions, note should be taken that the three smallest interactions as reflected by the statistical values were found for interactions involving vertical frequency, suggesting that perhaps interactio with vertical frequency is the least important among those found. Regarding the other interactions, no pattern is apparent beyond that obvious from table 2. Although a significant four-way interaction was found, no explanation of it is readily apparent.

To summarize the results of Experiment II, it appears that the four major parameters of vibration not only affect ratings of discomfort, but they also interact with each other in their effects. Interactions between frequencies in the two axes and between amplitudes in the two axes were expected as was, to some extent, the interaction between frequency and amplitude withi one axis. However, the interaction between frequency in one axis and amplitu in the other was not expected.

Taken together, the results of these two experiments strongly suggest th there are effects on discomfort that occur when subjects are vibrated in several axes at once that cannot be assessed with research using vibration in on one axis. Although the interactions between the four parameters of vibration used in these experiments may be of less importance in accounting for discomf than are the main effects of these four major parameters, an understanding of these interactions may very well affect the precision with which standards ca be set to govern the acceptable limits for exposure of humans to vibration. I conclusion, these results also suggest the wisdom of further research on the effects of vibration in combined axes directed toward appropriate revision of the standard established by ISO in reference 6 regarding vibrations occurring in more than one axis simultaneously.

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TABLE 1. - THREE-WAY ANALYSIS OF VARIANCE WITH REPEATED MEASURES ON TWO VARIABLES

Source of variation	Sum of squares	Degrees of freedom	Mean square	F
Se	321.69	3	107.23	2.50
VF	1751.37	9	194.60	50.71**
LF	5680.88	9	631.21	327.56**
Se × S w. groups	858.31	20	42.92	
SE × V	200.27	27	7.42	1.93**
Se × L	146.89	27	5.44	2.82**
V × L	722.35	81	8.92	8.33**
V × S w. groups	690.80	180	3.84	
L × S w. groups	346.86	180	1.93	
Se × VF × LF	551.64	243	2.27	2.12**
VF × LF × S w. groups	1734.26	1620	1.07	

** $p < 0.01$

Notation:

F	mean-square ratio
LF	lateral frequency
p	probability
S	subjects
Se	sessions
VF	vertical frequency
w.	within

TABLE 2. - FOUR-WAY ANALYSIS OF VARIANCE WITH REPEATED MEASURES ON THREE VARIABLES

Source of variation	Sum of squares	Degrees of freedom	Mean square	F
VF	951.56	3	317.19	13.39**
LF	1178.82	3	392.94	105.73**
VA	1851.90	3	617.30	273.32**
LA	2160.80	3	720.21	260.80**
S w. VF	1042.38	44	23.69	
VF × LF	103.33	9	11.48	3.09**
VF × VA	173.37	9	19.26	8.53**
LF × VA	222.99	9	24.78	18.73**
VF × LA	103.65	9	11.52	4.17**
LF × LA	469.01	9	52.11	58.59**
VA × LA	249.03	9	27.67	39.92**
LF × S w. VF	490.58	132	3.72	
VA × S w. VF	298.13	132	2.26	
LA × S w. VF	364.52	132	2.76	
VF × LF × VA	39.26	27	1.45	1.10
VF × LF × LA	42.75	27	1.58	1.78*
VF × VA × LA	15.67	27	.58	0.84
LF × VA × LA	65.04	27	2.41	4.30**
LF × VA × S w. VF	523.86	396	1.32	
LF × LA × S w. VF	352.22	396	.89	
VA × LA × S w. VF	274.48	396	.69	
VF × LF × VA × LA	80.21	81	.99	1.77**
LF × VA × LA × S w. VF	665.45	1188	.56	

* $p < 0.05$
** $p < 0.01$

Notation:

F	mean-square ratio	S	subjects
LA	lateral amplitude	VA	vertical amplitude
LF	lateral frequency	VF	vertical frequency
p	probability	w.	within

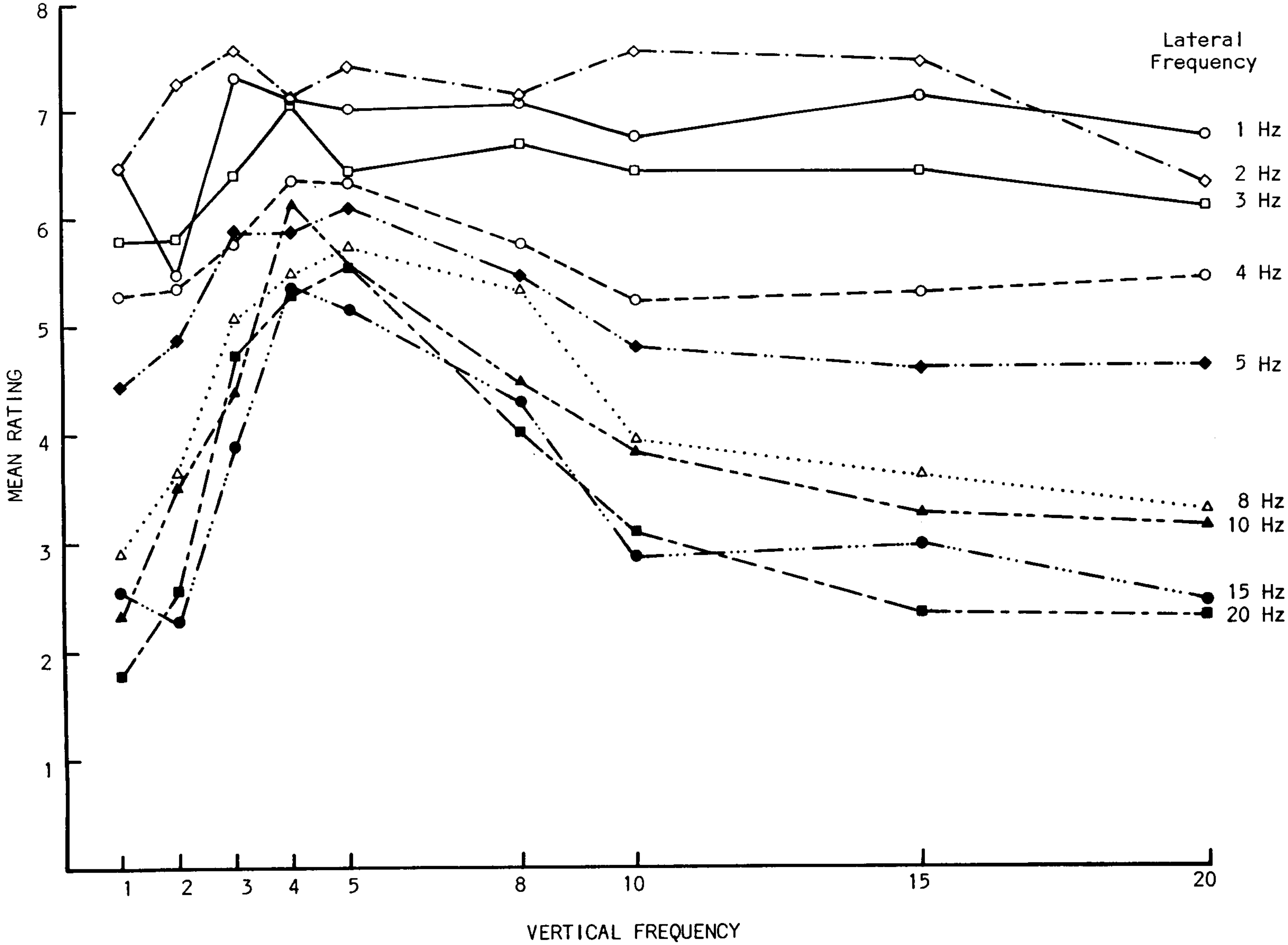

Figure 1.- Mean rating as a function of vertical frequency with lateral frequency as a parameter.

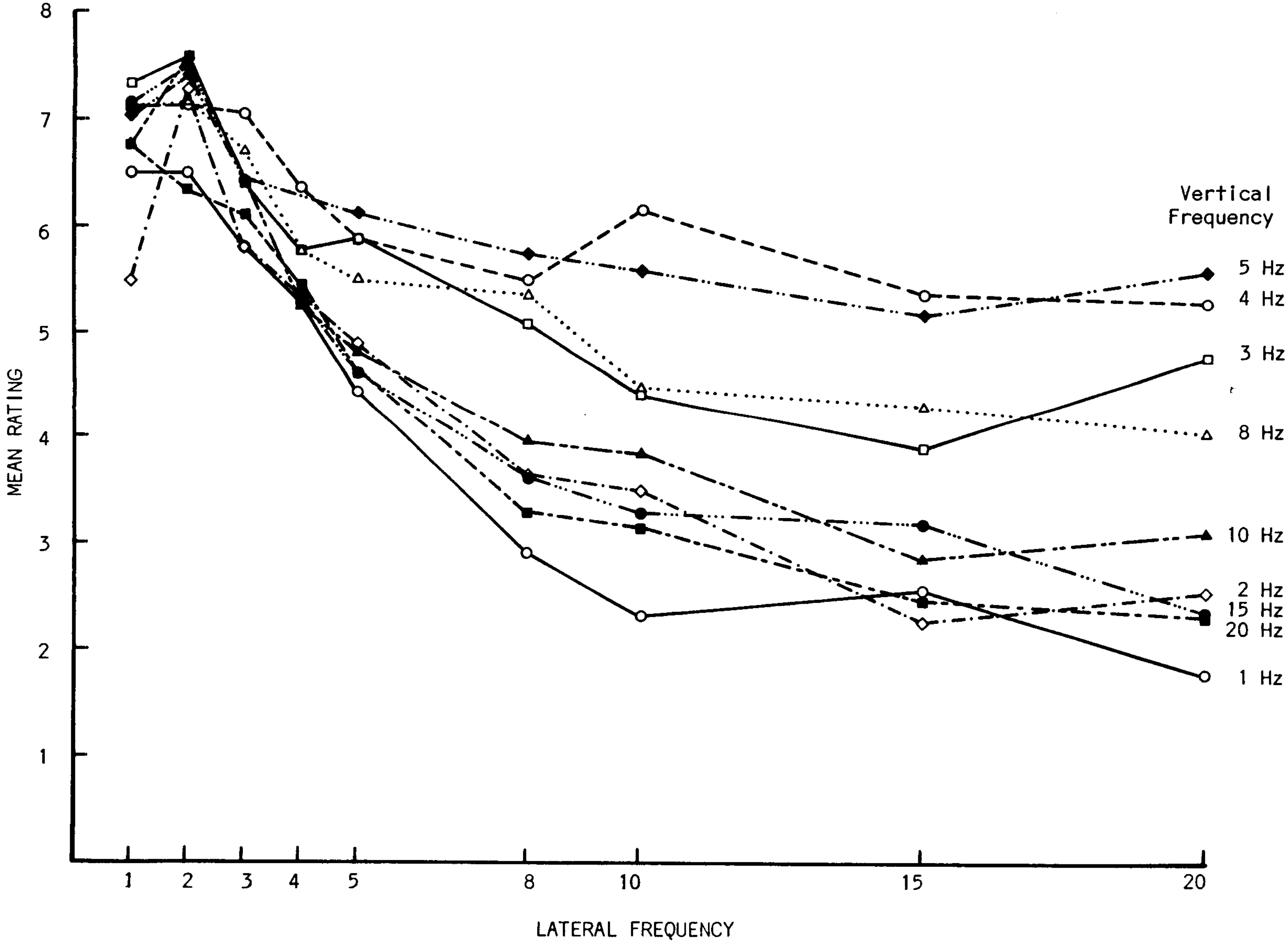

Figure 2.- Mean rating as a function of lateral frequency with vertical frequency as a parameter.

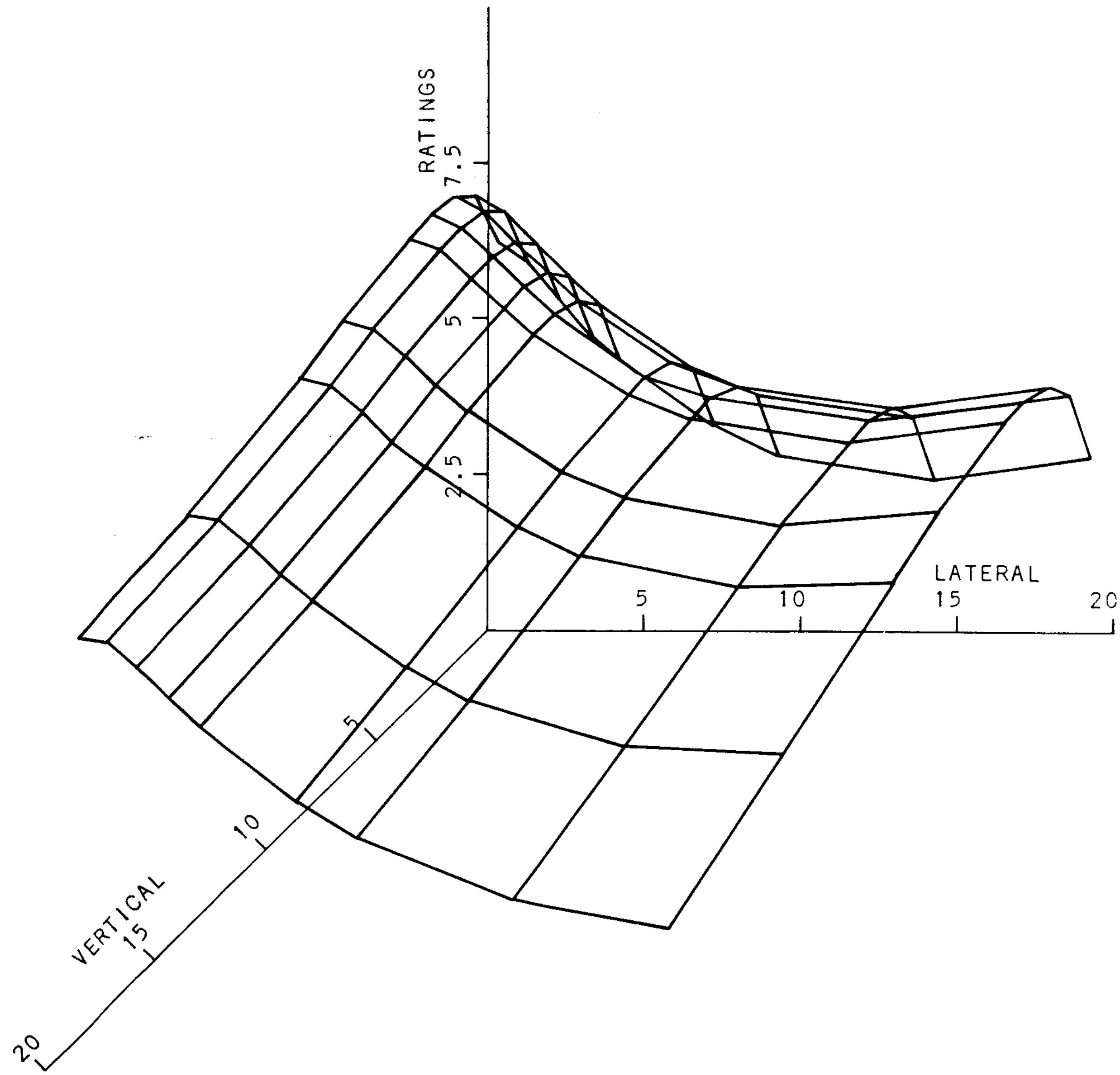

Figure 3.- Response surface predicted from vertical and lateral frequencies.

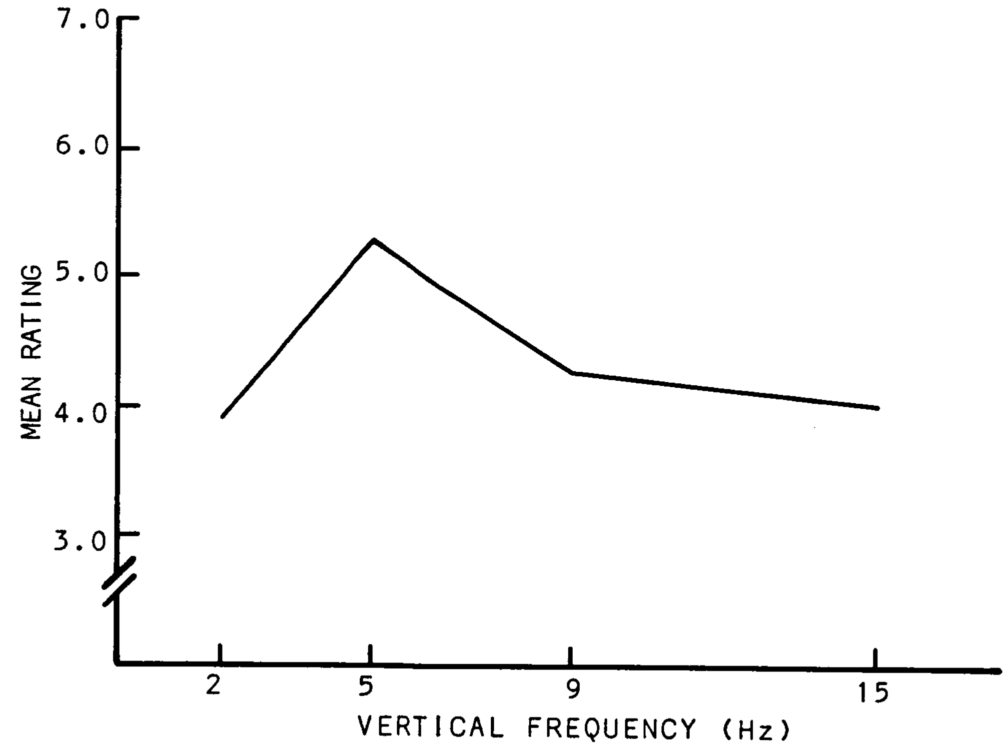

Figure 4.- Subjective rating as a function of vertical frequency.

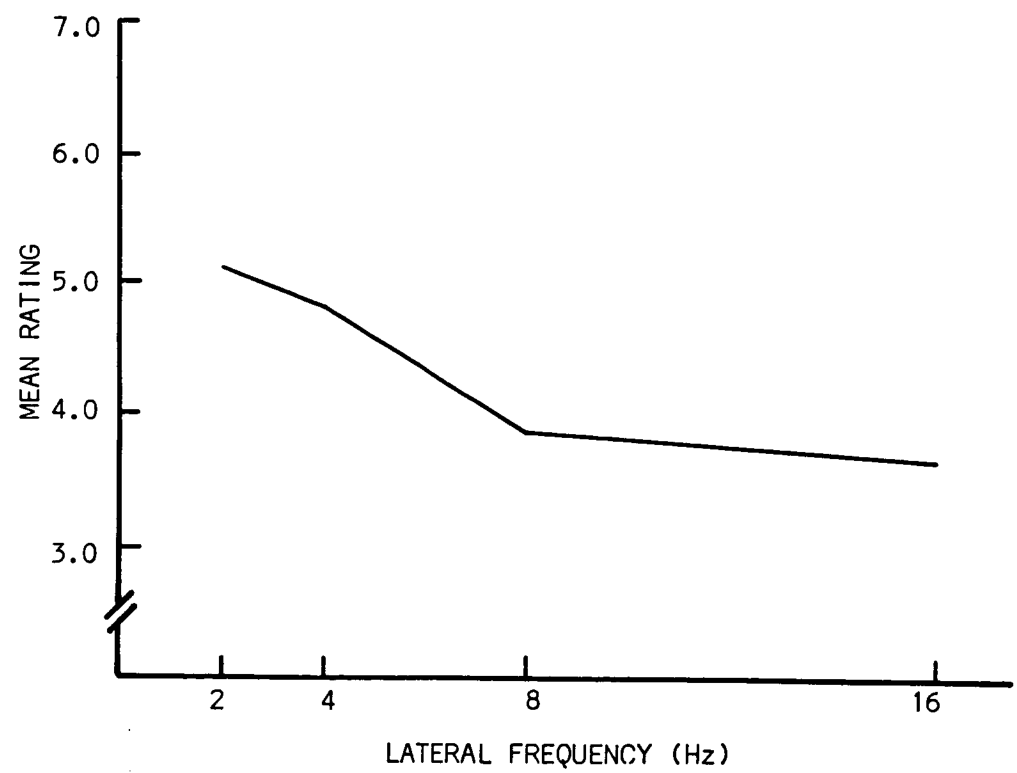

Figure 5.- Subjective rating as a function of lateral frequency.

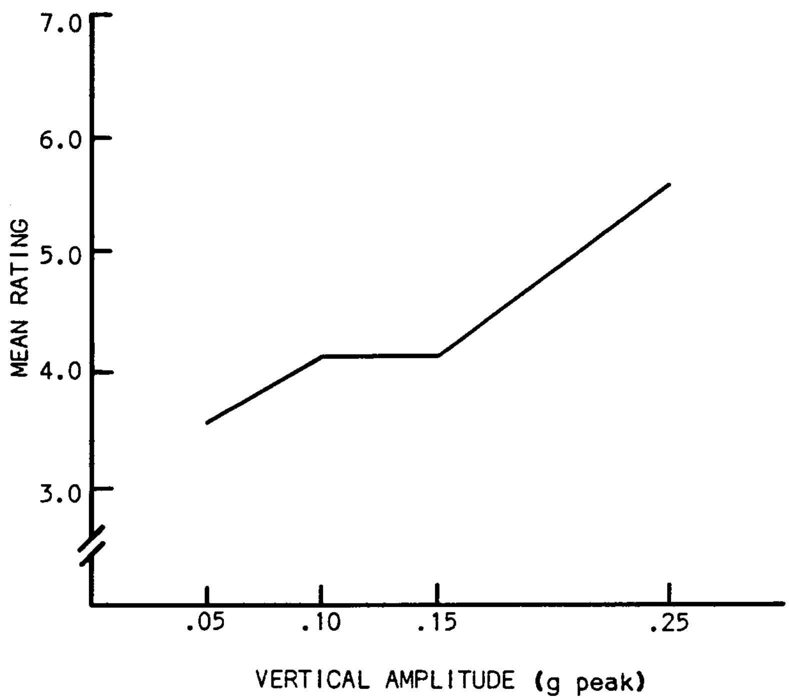

Figure 6.- Subjective rating as a function of vertical amplitude.

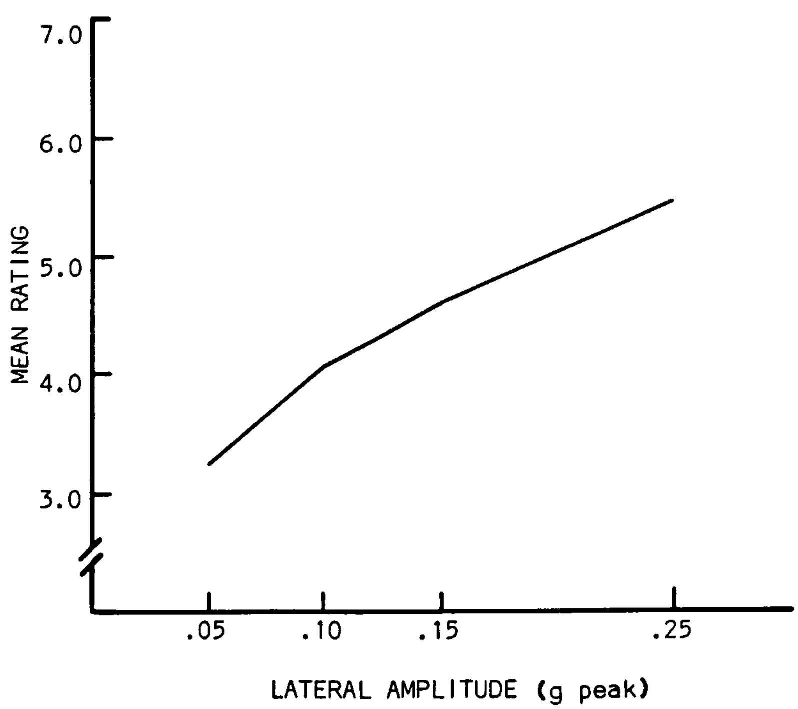

Figure 7.- Subjective rating as a function of lateral amplitude.

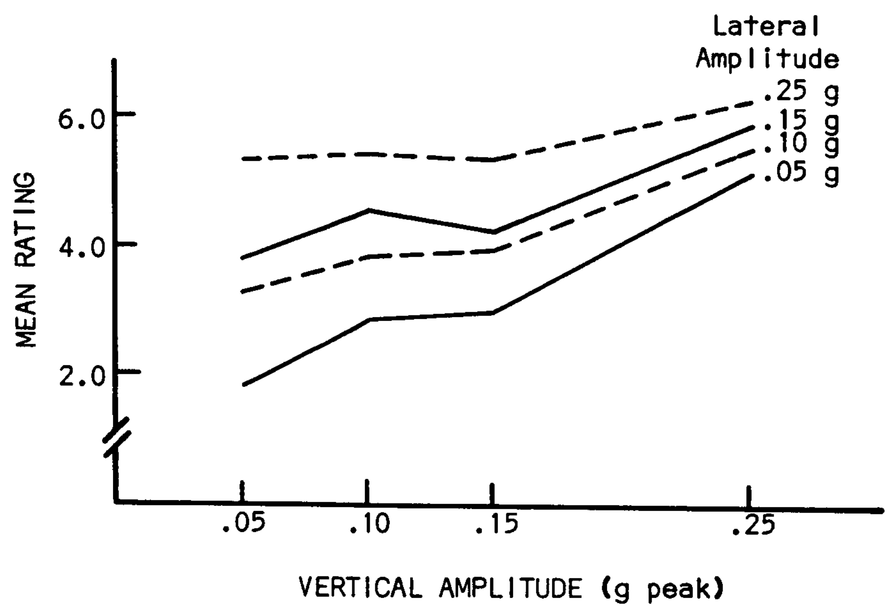

Figure 8.- Subjective rating as a function of the interaction between vertical frequency and lateral frequency.

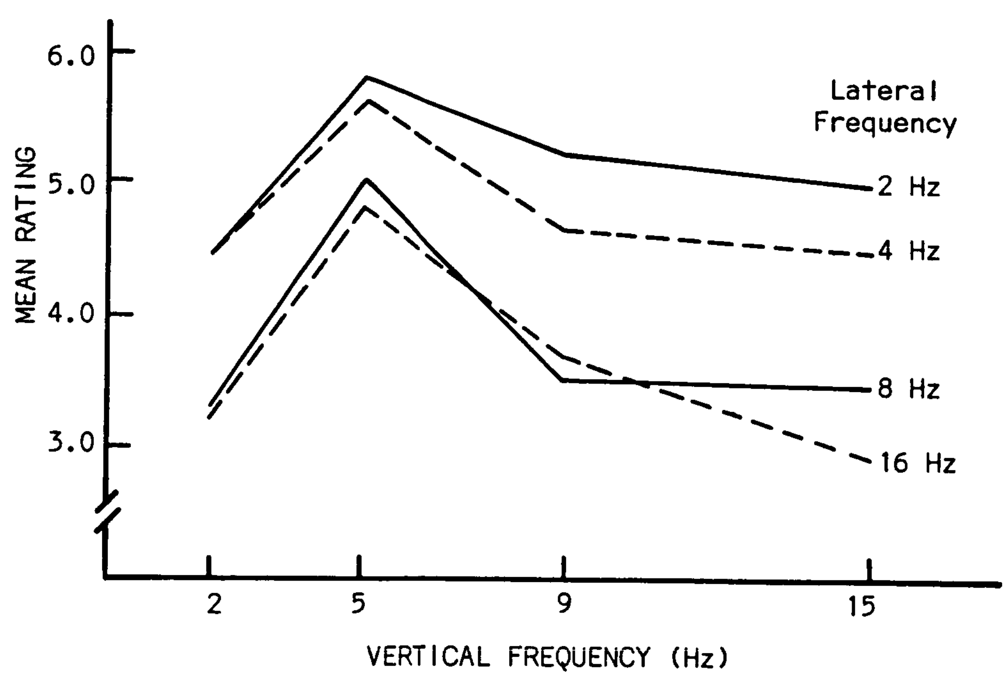

Figure 9.- Subjective rating as a function of the interaction between vertical amplitude and lateral amplitude.

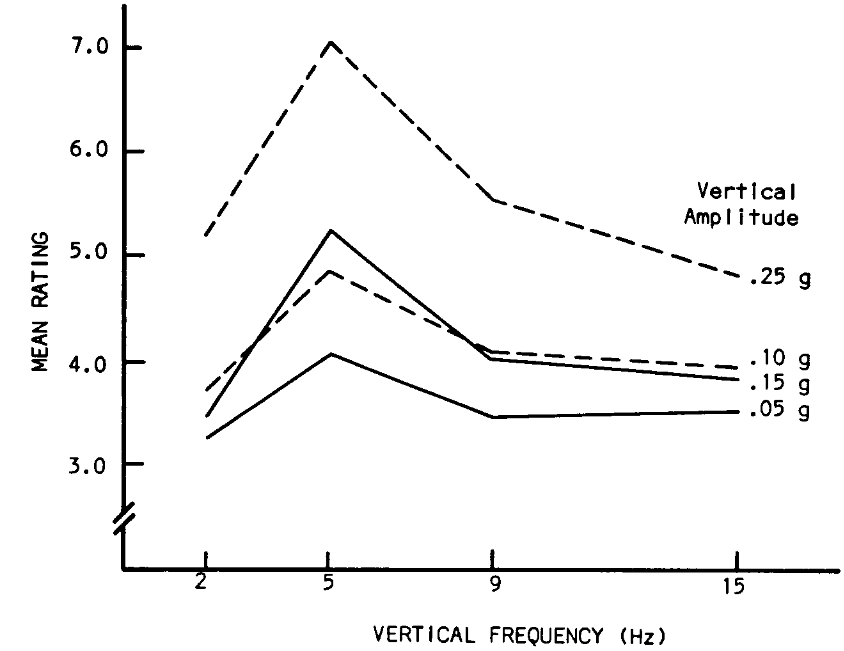

Figure 10.- Subjective rating as a function of the interaction between vertical frequency and vertical amplitude.

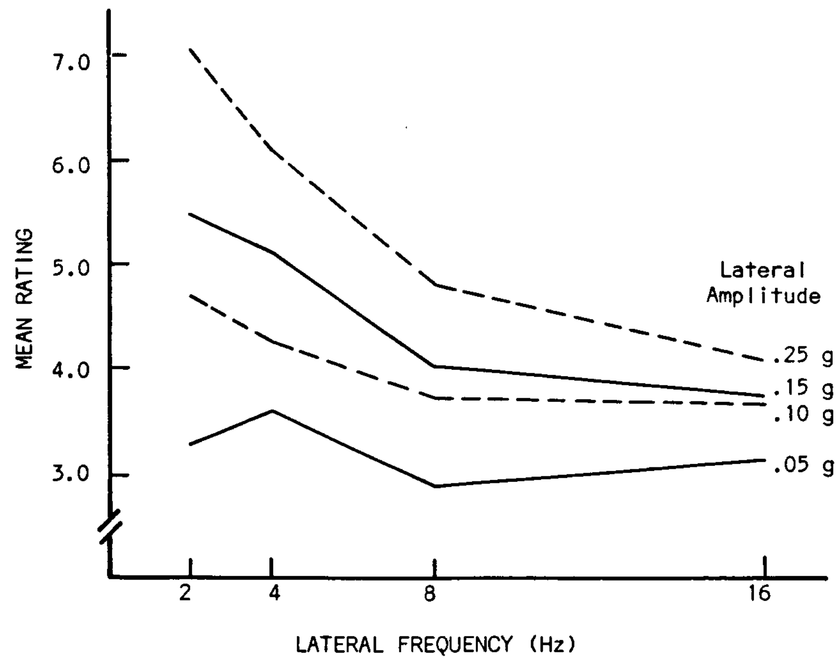

Figure 11.- Subjective rating as a function of the interaction between lateral frequency and lateral amplitude.

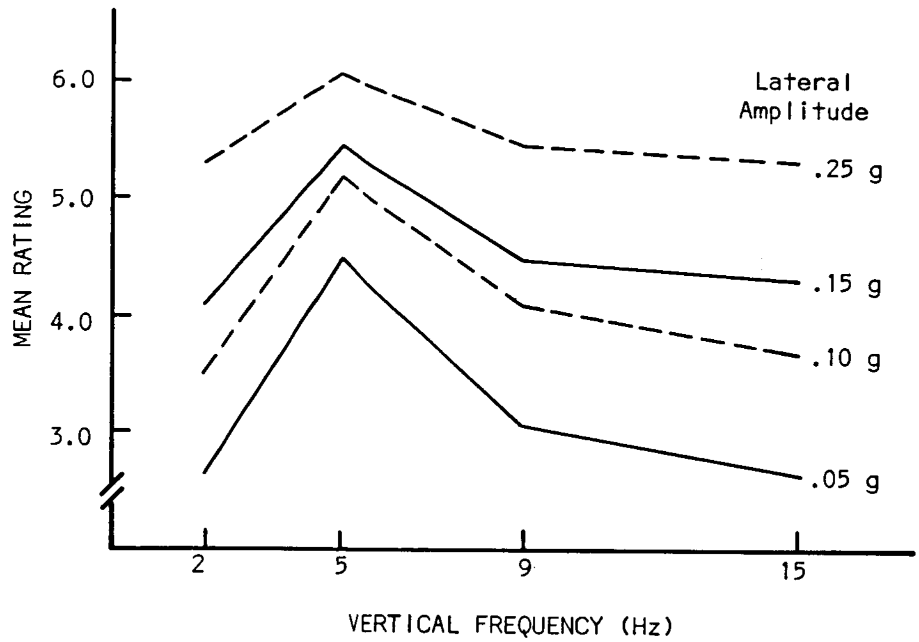

Figure 12.- Subjective rating as a function of the interaction between vertical frequency and lateral amplitude.

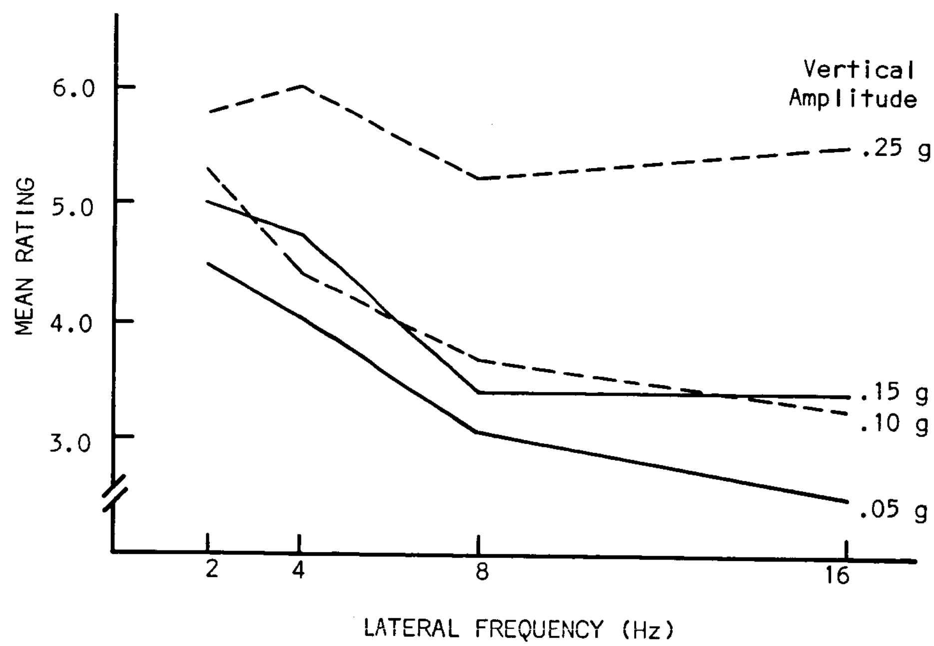

Figure 13.- Subjective rating as a function of the interaction between lateral frequency and vertical amplitude.

N76-16770

PASSENGER RIDE QUALITY RESPONSE TO AN AIRBORNE SIMULATOR ENVIRONMENT

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SUMMARY

Early studies of human response to motion were limited to the use of on-the-surface mechanical simulators. For air transportation, these ground-based simulators cannot reproduce the dynamic ranges of motion encountered aboard real aircraft. Some recent studies have been done aboard aircraft but the motion has been uncontrolled.

The present study was done aboard a special aircraft able to effect translations through the center of gravity with a minimum of pitch and roll. The aircraft was driven through controlled motions by an on-board analog computer. The input signal was selectively filtered gaussian noise whose power spectra approximated that of natural turbulence. This input, combined with the maneuvering capabilities of this aircraft, resulted in an extremely realistic simulation of turbulent flight. The test flights also included varying bank angles during turns.

Subjects were chosen from among NASA Flight Research Center personnel. They were all volunteers, were given physical examinations, and were queried about their attitudes toward flying before final selection. In profile, they were representative of the general flying public.

Data from this study include (1) a basis for comparison with previous commercial flights, that is, motion dominated by vertical acceleration, (2) extension to motion dominated by lateral acceleration, and (3) evaluation of various bank angles.

The significance of this study was its extension of the data base for the flight environment to areas previously not covered. These data should contribute to more effective modeling of subjective human response to an aircraft motion environment.

INTRODUCTION

Human response to motion has been studied for many years on many different types of vehicles (ref. 1). Early studies (refs. 2 and 3) were slanted towards crew performance using "shake" chairs. As time passed, the level of ground-based simulation became more sophisticated. The use of ground-base simulators is a very economical approach to researching human response to motion. However, this method lacks realism and is inherently limited in the type of motion that can be simulated. In an attempt to gain realism, some field studies (ref. 4) were done. In this case the experimentors were not able to control the motion environment so only a part of it could be studied.

One objective of studying human subjective response to motion research is to model the response as an aid in the design of future transportation systems Some work in this area has already been done (ref. 5). The success of such work will strongly depend on the completeness of the data base upon which it was formulated. An objective of this program was to provide human response data in areas beyond the capability of ground-based simulators. For this reason, an airborne simulator was used. A subject population was selected from a group of NASA Flight Research Center (FRC) volunteers and asked to evaluate their overall comfort from a passenger's viewpoint. A program of 5 flights was conducted on the NASA Flight Research Center's General Purpose Airborne Simulator (GPAS) aircraft. A flight-test engineer accompanied two subject passengers on each flight and controlled the experiment. For documentation purposes, cabin temperature and noise data were collected on selected flights. Unique on this aircraft were direct lift and side force generator control surfaces. Through these surfaces vertical and lateral accelerations can be produced on the aircraft with a minimum of pitch and roll. Single and combined axes tests were performed for subject passenger evaluation. Test flights also included a series of turns at various bank angles for evaluation. Typical test flights lasted slightly over one hour.

Data from this study include (1) a basis for comparison with data from previous commercial flights, i.e., motion dominated by vertical acceleration, (2) extension to motion dominated by lateral acceleration, and (3) evaluation of various bank angles.

TEST AIRCRAFT DESCRIPTION

The test aircraft used in this experiment was a modified JetStar Business Jet (figure 1). Modifications to the aircraft include the addition of direct lift control (DLC) and side-force generator (SFG) control surfaces. These unique features make this aircraft specially suited for the conduction of ride quality experiments. The DLC and SFG surfaces can be moved independently or in combination to provide vertical and/or lateral motion. However, due to the location of these surfaces on the aircraft, pure translational motion could not be produced but was always accompanied by a small amount of pitch and roll.

The aircraft cutaway (figure 2) shows the location of the DLC and SFG surfaces, the interior seating arrangement for subject passengers and the flight-test engineer, airborne analog computer, and data acquisition system. The seats used in this experiment were the aircraft manufacturer's original equipment as supplied with the standard aircraft.

From a tape recorder, located near the data acquisition system, a simulated turbulence signal was played into the airborne analog computer by the flight-test engineer. The airborne analog computer, in turn, generated signals to drive the DLC and SFG surfaces to induce aircraft vertical and lateral motion. The pilots monitored accelerations and made minor corrections as necessary to minimize pitch and roll.

INSTRUMENTATION AND DATA ACQUISITION

All aircraft motion parameters and the subject passenger responses were collected using a standard NASA Flight Research Center data acquisition system (DAS). The DAS samples the data 40 times per second, translates the data into pulse code modulated (PCM) format, and then records it on magnetic tape. Table 1 lists the parameters recorded on the DAS. For documentation purposes on selected flights, the passenger cabin temperature and noise level were recorded by the flight-test engineer using hand-held instruments.

The subjects were asked to rate their reactions to the flight motions according to a five-point scale ranging from very uncomfortable to very comfortable, as shown in table 2. The ratings were made by pushing one of five buttons on a box attached to the seat arm (figure 3). Although the subjects were instructed to change their ratings as they felt the need, it was necessary to obtain a considered rating during the last 15 seconds of each one minute test segment. To accomplish this, there was a command light on the rating box which was lit to request ratings. A reset button was also on the box and the subjects were requested to push this before any rating. This aided in removing signal ambiguity during data reduction.

An instrumentation package containing accelerometers and gyros was located in an area of the DAS near the aircraft center of gravity and was attached directly to the airframe.

TEST PROCEDURE

Since the results from this program would primarily impact the general flying public, it was desirable to obtain a subject population parallel to the makeup of that public sector. Consequently, volunteers were solicited from among Flight Research Center employees through the simple expedient of posting a notice on the bulletin boards requesting participation in a research flight program. No special incentives were offered, but employee interest in FRC activities was strong enough to result in 35 applications. For a variety of legal and medical considerations, only permanent full-time NASA employees in good health were accepted. Coupled with normal attrition over the life of the program, this finally reduced to a subject population of 16. The composition of this group compared favorably with the demographics of passengers on commuter aircraft in the northeast (ref. 6).

After selection, the subjects were asked to fill out a questionnaire to ascertain their general attitude towards flying for transportation. In general, the subjects enjoyed flying, even though most passengers flew on business trips. Prior to beginning the actual flights, the subjects were briefed on the nature of the program and their part in it. They were also given an explanation of the capabilities of the aircraft but were not told specifically what would occur during flight. When the flight program began, an assignment schedule was drawn up to permit each pair of subjects to report immediately before their flight for a final briefing.

Due to the onboard instrumentation, there was only room for two passengers at a time. Every effort was made to get all passengers enough flights in both seats to cover all the combinations available from the turbulence simulation system. Immediately following each flight, a quick debriefing was held to note any unusual occurrences. All passengers were also issued notebooks and encouraged to make whatever comments they wished. These comments are being studied for possible inclusion in a later report.

The simulated turbulence tape was generated by shaping a random signal on an analog computer and then recording it on magnetic tape. The random signal was obtained from a gaussian noise generator. The output of this source is a band of noise of uniform spectral density between 0 and 35 Hz. On the analog computer, this signal was shaped by using a second-order low pass filter with 0.7-Hz break frequency. These filter characteristics were chosen to generate an output signal whose power spectrum approximates that of natural turbulence (figures 4 and 5). The filter output was then scaled to be compatible with the maximum allowable input to the tape recorder. Three tape recorder channels were recorded with 1/3, 2/3, and maximum of full-scale signal amplitude. Another channel of the tape recorder was recorded with several test profiles containing 9 or 10 one minute segments. Each segment was manually adjusted to 0, 20, 40, 60, 80, or 100 percent of maximum amplitude. Segments of varying amplitudes were combined into test profiles either in a staircase or random fashion. A fifth recorder channel triggered the extra light on the subject's rating box to request a comfort rating during the last 15 seconds of each test segment. Subsequent subject comments indicated the need for an audible signal as well.

All test data were collected at an altitude of 6.1 km (20000 ft) and 250 knots indicated airspeed. The basis flight plan consisted of the following phases (with approximate times for each):

1. Take-off and climbout (20 minutes);
2. Simulated turbulence run 1 (10 minutes);
3. Turn number 1, 180° at 20° bank angle (5 minutes);
4. Simulated turbulence run 2 (10 minutes);
5. Turn number 2, 180° at 30° bank angle (4 minutes);
6. Turn number 3, 180° at 40° bank angle (4 minutes);
7. Descent and land (15 minutes).

During test profiles 1 and 2, the tape of simulated turbulence was played into the airborne analog computer which drove the DLC and SFG surfaces. By utilizing one or more tape recorder channels, single or combined axes tests were accomplished.

During single-axis testing a staircase or random profile was inputted to either the vertical or lateral axis while no input was made to the other axis. For combined-axes tests a staircase or random profile was inputted to one of the axes while a constant level of 1/3, 2/3, or maximum of full-scale signal amplitude was inputted to the other axis. All bank angle maneuvers were performed manually by the pilots and were to be made as well coordinated as possible. Both left and right bank maneuvers were evaluated with bank angles varying from 21° to 47°.

All data recorded on the DAS were reduced using the NASA Flight Research Center's Control Data Cyber 70 computer. Each one minute segment of each profile was partitioned into twelve 5 second parts. Mean and standard deviation values of vertical and lateral acceleration were computed for each part as well as for the entire one minute segment. The passenger subjective comfort rating was obtained by extracting the last rating found during the final 15 seconds of each one minute segment. These acceleration and rating data were then used to determine threshold values for comfort ratings of 2, 3, and 4 for each subject passenger on each flight. Ratings 1 and 5 were not included because of a general reluctance by the subjects to select these values. These threshold values were averaged over all subjects and all flights to generate the final comfort boundaries. The bank maneuver data were obtained by taking the passenger subjective comfort rating after a steady-state bank angle had been achieved. Lateral accelerations were examined to ensure the bank maneuver was well coordinated. The bank angles were averaged over all subjects for comfort ratings of 2, 3, and 4 to determine the final relationship. Computations were made using standard techniques. The cabin temperature and noise data were hand tabulated in a notebook. These data were averaged over all flights to obtain the final values of $71^{\circ} \pm 3^{\circ}F$ and 91 ± 2 dB.

RESULTS AND DISCUSSION

The fidelity of the motion simulation was evaluated by comparing the power spectral density (PSD) plots for the basic aircraft's typical response to natural turbulence, single-axis simulation, and combined-axes simulation. Figure 4 shows this comparison for vertical acceleration and figure 5 for lateral acceleration. These two plots indicate that the simulation profiles were representative of the natural turbulence case. Only acceleration variations about the mean value were used to generate these plots. Because the DLC and SFG surfaces are not located at the aircraft center of gravity, some pitching and rolling motion, respectively, is associated with the movement of these surfaces. These motions were kept to reasonably low levels by the pilot's use of manual controls. The pilots indicated the sharp "bucking" motion associated with natural turbulence was not as intense for the simulated turbulence.

Subjective data from single- and combined-axes tests are shown in figure 6. Only subject passenger ratings of 2, 3, and 4 were considered since there was a general reluctance on the part of the subject passengers to select the extreme ends of the rating scale. A wider choice of ratings should produce greater resolution. The solid curves are faired lines drawn through the average threshold data values. The straight line drawn at an angle through the origin shows the limit of previously collected commercial airline data (ref. 7). Data above this line duplicated and corroborated previous work. The data below the line represents an expansion of the data base as a result of this experiment to include motion which is dominated by lateral acceleration. This area of data was previously undefined in terms of riding quality for an aircraft environment. As indicated by earlier studies, the results show subject passengers are about twice as sensitive to lateral as compared to vertical accelerations. Also, subject passengers appear to be slightly more tolerant to vertical acceleration in the presence of a low level of lateral acceleration. Originally, the data were broken into groups according to sex, seat position, and flying experience. The results from these various data groupings did not indicate significant differences from the total group result. However, a volunteer group such as this is strongly disposed towards liking to fly.

Figure 7 shows the relationship of subject passenger comfort rating and bank angle. The normal acceleration during the bank was monitored to insure that the turn was coordinated. In general, any deviation on the part of the subject passenger from an upright posture increases the level of discomfort. For the aircraft used in this experiment, when a left bank was performed, the subject passenger could not see out of the left-hand windows because of the presence of the airborne simulation equipment and therefore lost sight of the horizon. However, no significant difference in the data was observed between left and right bank angles. Based on the results of this test, present commercial airline operating procedure limiting maneuvering bank angles to about 20° is acceptable from a passenger comfort standpoint, 30° being a maximum acceptable bank angle during a coordinated turn.

CONCLUDING REMARKS

A flight program of 55 flights was conducted to evaluate subjective human response to an aircraft motion environment. As a result of this program the data base has been expanded to include a motion environment dominated by lateral acceleration and to include bank angle information. The results reinforce the statement that subjects are about twice as sensitive to lateral as to vertical acceleration. The results also showed that current airline practice of limiting bank maneuvers to 20° provides minimal passenger discomfort.

A five-point rating scale was used throughout the program. Because of the reluctance of the subject to select either one or five, the rating scale collapsed to effectively a three-point scale. This represents a minimum number rating scale and it would be highly desirable to obtain more resolution.

During the tests a light on the rating box was lit to request a subject rating. It was felt that an audible signal would permit the subject to engage in more normal flight activities without having to continuously monitor the light.

As with all experiments of this type, the subject group is a continuous problem. It is desirable to have a larger and more representative group of the flying public to participate in such tests in the future.

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7. Jacobson, Ira D.; and Richards, Larry G.: Ride Quality Evaluation II: Modelling of Airplane Passenger Comfort. (Tech. Rep. 403217, Univ. of Virginia; NASA Grant NGR 47-005-181.) To be published in Ergonomics, vol. 18, 1975.

TABLE I.- DATA ACQUISITION SYSTEM PARAMETERS

ATTITUDE RATE	PITCH ANGULAR ACCELERATION
ALTITUDE	ROLL ANGULAR ACCELERATION
AIRSPEED	YAW ANGULAR ACCELERATION
PASSENGER RATING (2)	NORMAL ACCELERATION
PITCH ANGLE	LATERAL ACCELERATION
ROLL ANGLE	LONGITUDINAL ACCELERATION
PITCH ANGLE RATE	AIRCRAFT TURBULENCE INPUT SIGNAL (3)
ROLL ANGLE RATE	DIRECT LIFT FLAP ANGLE
YAW ANGLE RATE	SIDE FORCE GENERATOR ANGLE

TABLE II.- COMFORT RATING SCALE

RATING	DESCRIPTOR
1	VERY COMFORTABLE
2	COMFORTABLE
3	NEUTRAL
4	UNCOMFORTABLE
5	VERY UNCOMFORTABLE

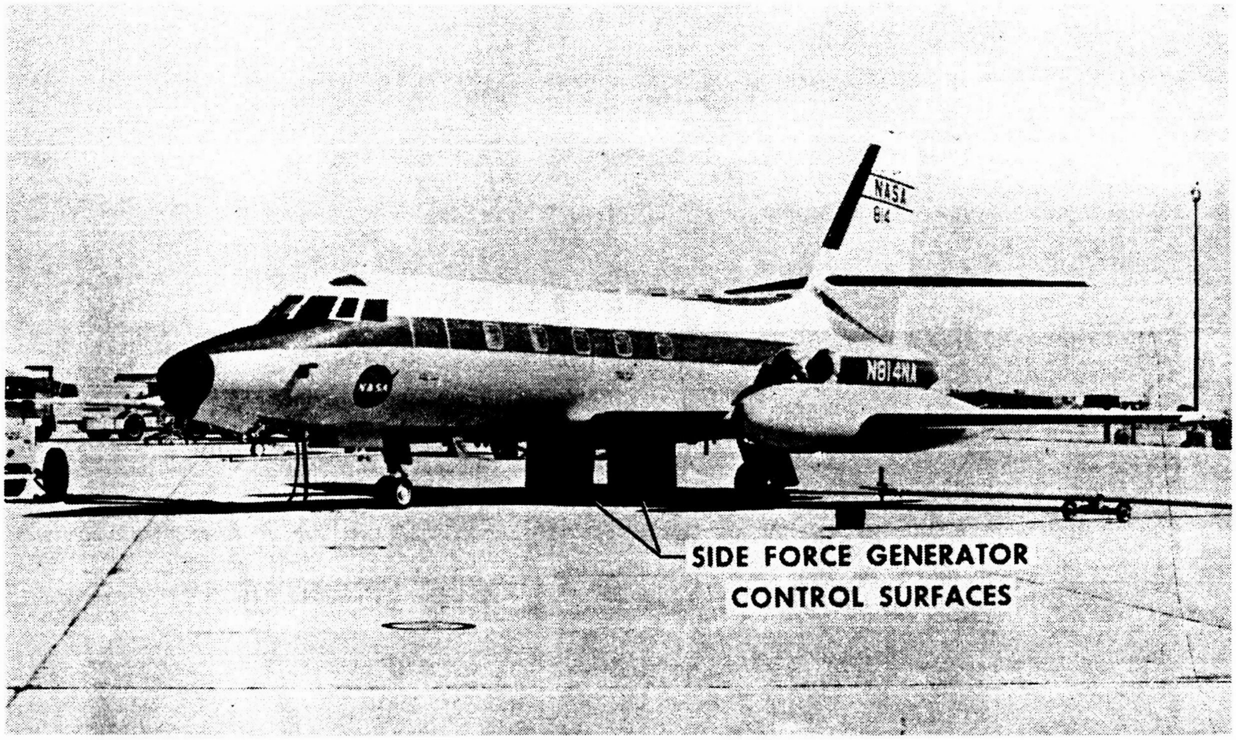

Figure 1.- NASA Jetstar.

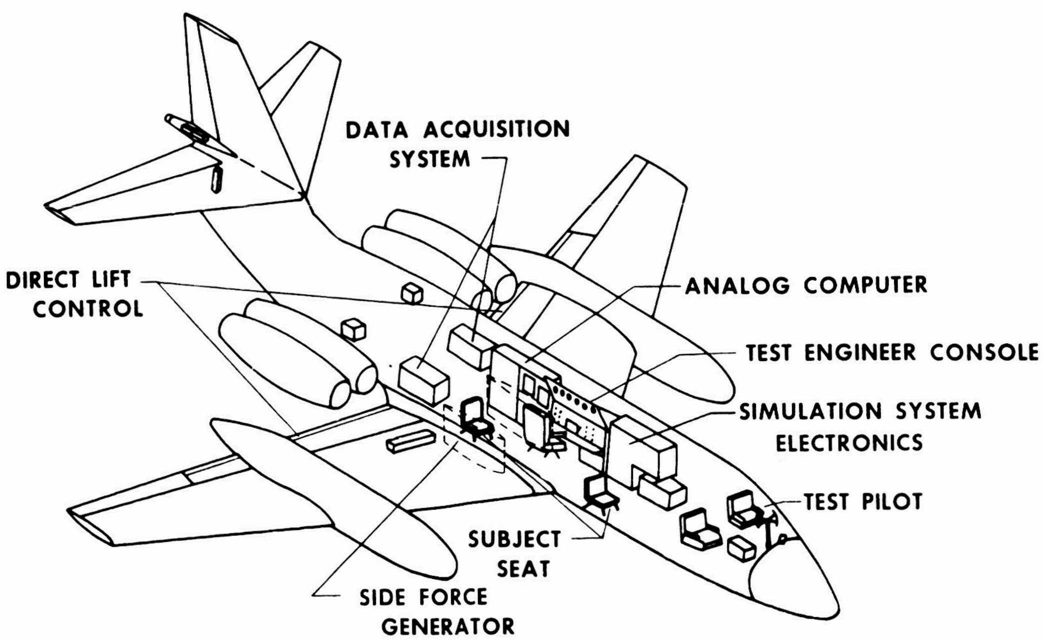

Figure 2.- NASA General Purpose Airborne Simulator (GPAS).

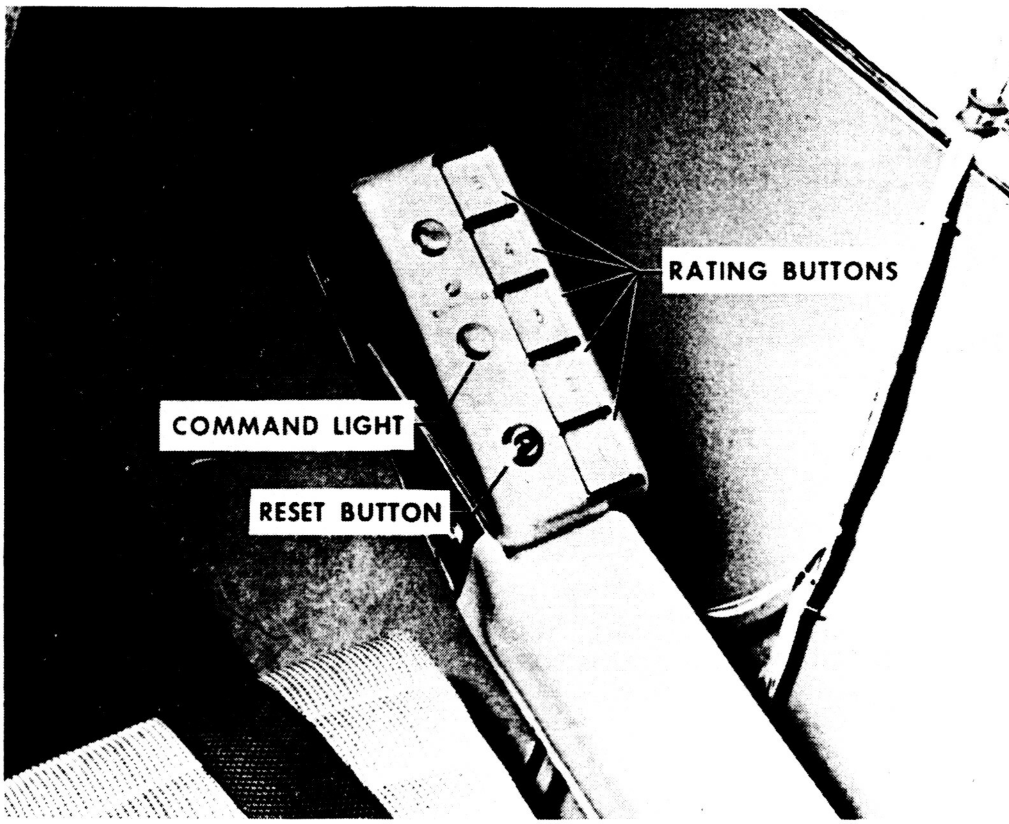

Figure 3.- Rating box installed on arm of Jetstar passenger seat.

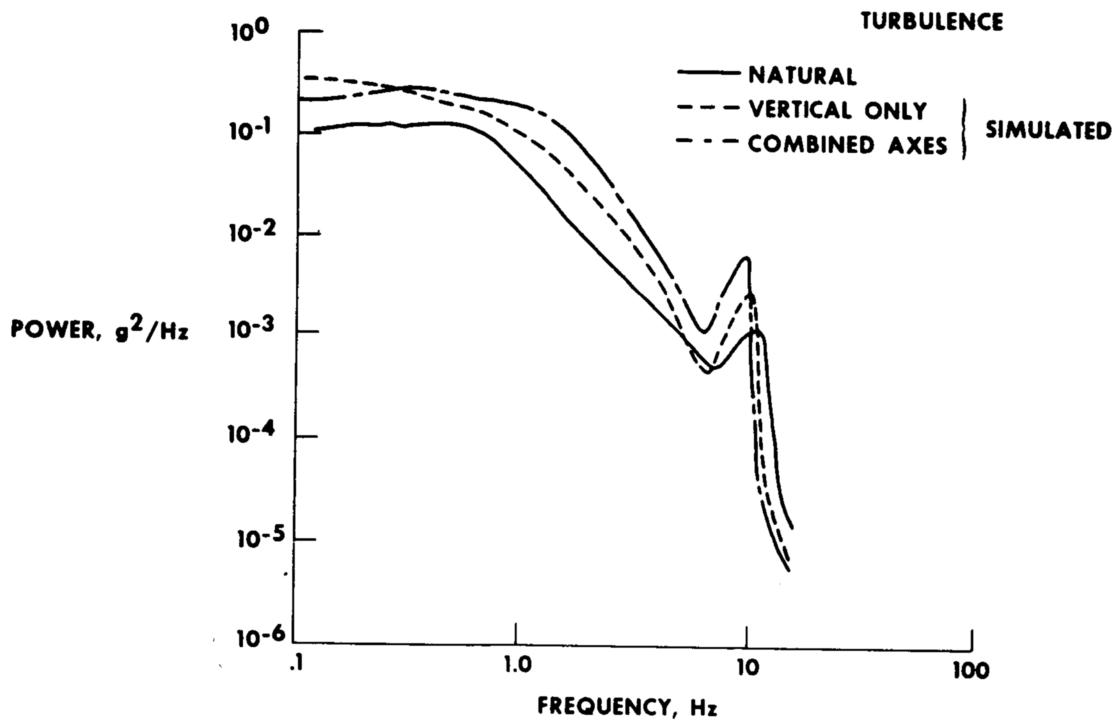

Figure 4.- Power spectral density of aircraft response to vertical acceleration.

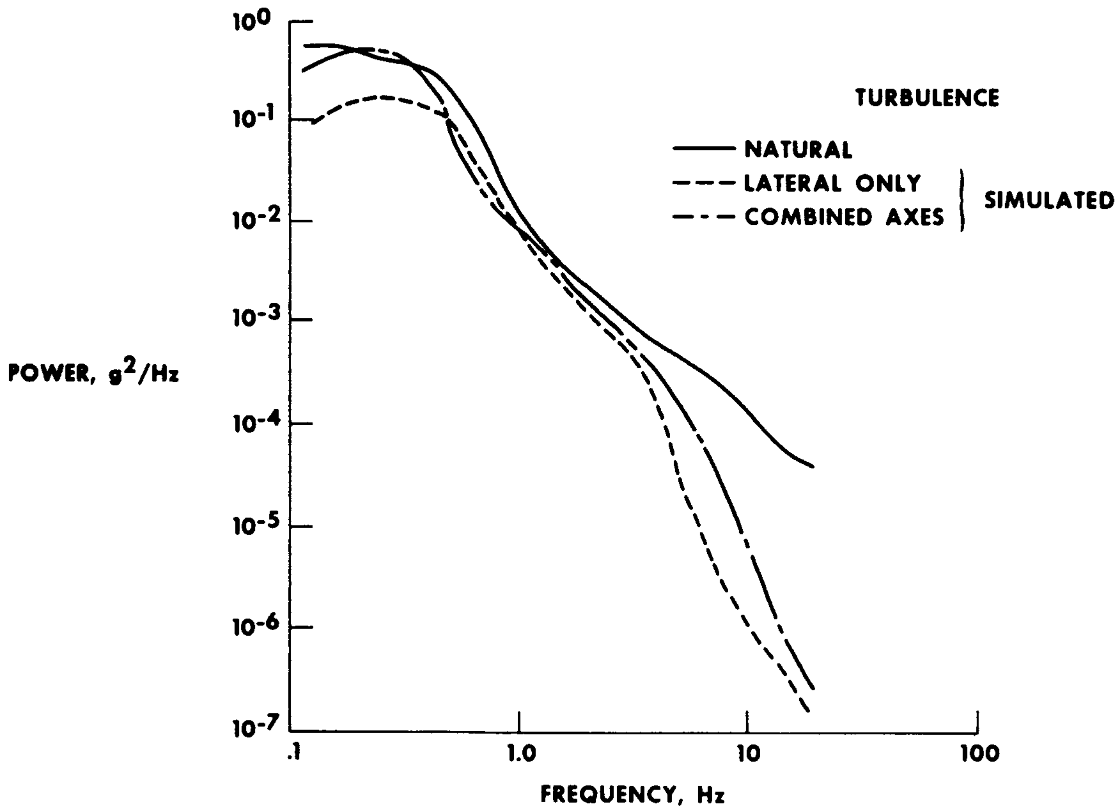

Figure 5.- Power spectral density of aircraft response to lateral acceleration.

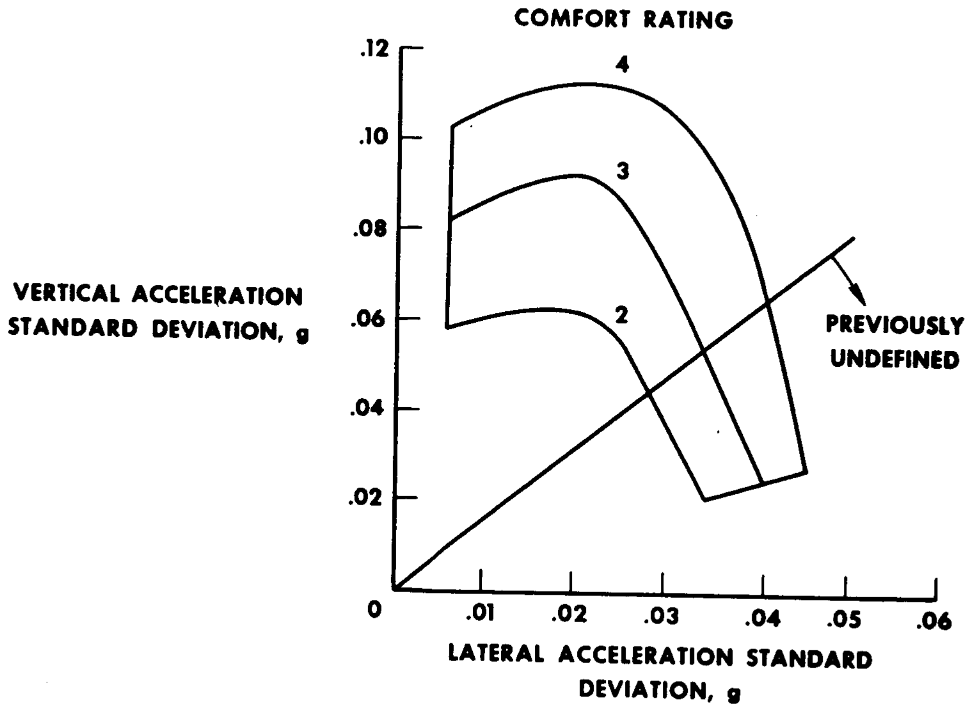

Figure 6.- Passenger response to aircraft accelerations.

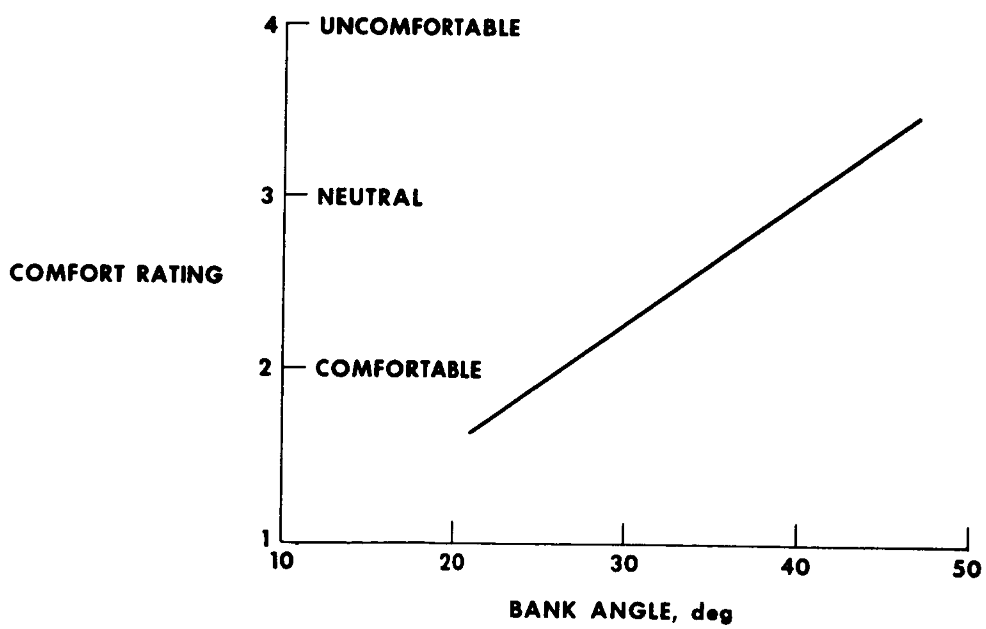

Figure 7.- Passenger response to bank angle.

N76-16771

RIDE QUALITY OF TERMINAL-AREA FLIGHT MANEUVERS

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SUMMARY

Complex terminal-area flight maneuvers being considered for airline operations may not be acceptable to passengers. To provide technology in this area, a series of flight experiments was conducted by NASA using the U. S. Air Force Total In-Flight Simulator (TIFS) aircraft to obtain subjective responses of a significant number of passenger test subjects to closely controlled and repeatable flight maneuvers. Regression analysis of the data produced a mathematical model which closely predicts mean passenger ride-comfort rating as a function of the rms six-degree-of-freedom aircraft motions during the maneuver. This ride-comfort model has been exercised to examine various synthesized flight maneuvers.

INTRODUCTION

Complex terminal-area flight maneuvers, used in conjunction with area-navigation and 4-D takeoff/approach techniques, are being considered to reduce fuel usage, noise pollution, and air-traffic congestion. Flight research to determine the feasibility of incorporating such unusual flight maneuvers into routine operations is part of NASA's Terminal-Configured-Vehicle Program (ref. 1). Such maneuvers, however, may not be acceptable to passengers since certain combinations of linear and angular motions are known to be upsetting to the human vestibular system. Several years ago, exploratory flight experiments concerning maneuver effects on ride quality conclusively indicated that criteria are needed which include more than just vertical and lateral motions (ref. 2). As ride comfort is a significant factor in determining acceptance and use of air transportation, a need exists for technology which will allow prediction of the degree of passenger comfort for terminal-area flight maneuvers.

Technology applicable to anticipated needs does not presently exist. Ride-comfort research has been conducted both in the field, aboard commercial and research vehicles, and in the laboratory using motion simulators. Laboratory simulators, however, lack motion capability sufficient to simulate flight maneuvers, whereas field tests aboard commercial vehicles do not allow precise control and repetition of a given maneuver. To provide the technology from which ride-quality predictive relations and criteria can be established for terminal-area maneuvers, a series of flight experiments was conducted by NASA using the U. S. Air Force Total In-Flight Simulator (TIFS) aircraft (fig. 1). The TIFS, piloted by a magnetic tape, was used to expose passenger test subjects to closely controlled and repeatable flight maneuvers. This paper describes these experiments, the regression analysis applied to the data to produce a ride-comfort model, and the results obtained when the model was exercised for various synthesized flight maneuvers.

TEST VEHICLE

The U. S. Air Force Total In-Flight Simulator (TIFS) is a C-131H transport (similar to a Convair 580 commercial transport) modified into a variable-stability research aircraft. Figure 2(a) illustrates the distinctive features of the aircraft. A simulation cockpit, mounted on the nose of the C-131H, is designed to place evaluation pilots in an environment configured to closely duplicate that of the cockpit of the aircraft being simulated. Special provision is made for independent control of the forces and moments about all three motion axes. Included are aerodynamic surfaces mounted vertically above and below each wing to provide side-force variation with very littl rolling or yawing moment, aileron-type flaps immediately outboard of the engine to provide direct lift control, and servo-operated throttles to provide longitudinal force variation. High-performance electrohydraulic actuators drive the existing ailerons, elevator, and rudder to produce rolling, pitching, and yawing moments, respectively. In the standard Convair cockpit, safety pilots monitor the simulation in progress and have the capability of disengaging the variable-stability system and resuming control of the aircraft at any time. The variable-stability system includes an analog computer and associated electronics located in the aft cabin. Inputs to the computer come from the evaluation pilot's controls and airplane motion sensors. A digital recording system capable of recording 58 individual variables, such as airplane motions and pilot control inputs, logs test results for engineering evaluation.

Figure 2(b) illustrates the TIFS modifications made for ride-quality testing. The standard TIFS simulation cockpit was replaced with a noise fairing. The aircraft cabin section between the cockpit and computer (fig. 3) was outfitted with wood paneling, curtains, and a carpet to create an airline-type atmosphere. Five standard Convair double seats were provided for the 10 test subjects. Each passenger seat was provided with a reading light, an adjustable outlet of conditioned air, a seat pocket with airsickness bag, and an emergency evacuation instruction card. A restroom, equipped with a marine-type toilet, was provided adjacent to the test-subject area. The TIFS hydraulic console area was soundproofed and trimmed with wood paneling to muffle the sound of continuous-duty hydraulic boost pumps. All but one pair of test-subject seats were adjacent to a window. An additional double seat for the flight-test director was provided immediately behind the test subjects, together with voice communications to the pilots and test engineer and a public address system for instructing the passenger subjects during flight. A closed-circuit television camera was mounted in the safety cockpit to record copilot head motions, and another camera was mounted behind a panel to record activity of a few of the test subjects. For the ride-quality experiments, the pilot-control inputs were replaced by magnetic tape command signals. These command signals were then combined with appropriate filtering and shaping to generate commands to the TIFS flight control surfaces necessary to produce the desired aircraft motions. A two-axis side stick controller gave the ccpilot the capability of maneuvering the aircraft with the variable-stability system engaged. Further details concerning the TIFS modifications for ride-quality research and test techniques employed are reported in reference 3.

FLIGHT TESTS

Flight Maneuvers

Maneuvers investigated individually consisted of one of three basic components (steady descent, steady turn, or longitudinal deceleration) of typical terminal-area flight maneuvers. A few combinations of two or three of these components were used to study subjective responses to more complex maneuvers (e.g., a turning decelerating descent, etc.). The range of each maneuver motion variable (e.g., approach angle, roll angle, etc.) somewhat exceeded the motion variable range normally encountered during terminal-area maneuvers of commercial passenger aircraft. Table I summarizes both the ranges of key motion variables and the number of unique variable combinations tested for each type of maneuver. Maneuvers, generally of 30-second duration, were sequenced at approximately 90-second intervals on 2 test tapes of 24 segments each.

The excellent repeatability of flight maneuvers provided by magnetic tape control of the aircraft is illustrated in figure 4. The figure presents time histories of four appropriate motion parameters measured during a particular maneuver flown on two different flights. The maneuver shown is a turning decelerating descent, which was probably the most complex and extreme maneuver tested and therefore was one of the most difficult to repeat. Differences in parameter values are relatively minor between flights and are essentially constant over the time duration of the maneuver for the three parameters (roll angle, pitch angle, and indicated airspeed) which were specifically controlled by the motion-command tape. Differences could be expected to be nearly constant because each of the three parameters was recorded on the drive tape in terms of parameter deviation from a reference flight condition. The slight shifts in parameter values between the two flights are associated with minor changes in reference flight conditions by the copilot to avoid weather, to stay within a certain test area, or to increase/decrease test altitude.

Passenger Subjects

Thirty-two passenger subjects of both sexes were chosen from among NASA employees, university students, and the general public to include a range of age and previous flight experience and to represent air travelers in general. Table II compares characteristics of the passenger subjects with those of air travelers in general. Approximately 1 hour prior to a given test flight, 10 of the test subjects were assembled and briefed on the purposes of the TIFS Ride-Quality Program in general and of the upcoming flight in particular. The subjects were informed of the types and degrees of motion to be tested and of the ability of any subject at any time to terminate the input motion by a simple hand signal (such termination, in fact, occurred more than once). After all questions were answered, each subject signed a statement of voluntary participation and boarded the aircraft.

Test Procedure

Once all passenger subjects were aboard and seated with seat belts secured the TIFS aircraft took off and during about 15 minutes climbed to the appropriate test area, altitude, and heading. The aircraft was then trimmed in straigh and level flight and the variable-stability system engaged. The motion-command tape recorder was started and the motion command signals were brought to full strength. For the next 30 to 40 minutes the aircraft was piloted by the tape recorder, with the exception of occasional pitch and roll trim changes by the copilot to keep the aircraft within safe test airspace. As the various test maneuvers were experienced in the aircraft, the beginning and end of each evaluation interval (typically 30 sec) were announced over the aircraft's public address system by the test director. At the end of each evaluation interval, each passenger subject recorded on a rating sheet his estimate of his own total comfort on a 7-point rating scale employing undefined descriptors ranging from "Very Comfortable" to "Very Uncomfortable" (see table III). In addition, each subject was asked to report in a "Comments" column any aspect of the passenger environment which he considered dominant in his assessment of personal comfort. Upon completion of the entire set of motion test segments, the motion command signals were attenuated, the tape recorder was stopped, the variable-stability system disengaged, and the aircraft returned to the Langley Research Center and landed. During the return trip, the passenger subjects completed summary questionnaires stating their assessments of the overall comfort (using the 7-point scale) of the test ride and of specific aspects of ride comfort (e.g., motion, noise, seat comfort, etc.). Upon landing, the passengers deplaned and, after a short debriefing, were dismissed.

RESULTS

The 2 test tapes of 24 segments each provided a total of 48 individual flight maneuvers to be repeated 4 times. With 10 subjects onboard each flight, the resulting 192 flight maneuvers provided a grand total of 1920 individual ride-comfort ratings. Space does not permit tabulation of individual ride-comfort ratings versus flight-condition variables. For each flight, the total number in each comfort rating is presented in table IV, however, to indicate that the entire ride-comfort rating scale was used and to provide a general idea of consistency between flights. It should be pointed out that between flights using the same maneuver tape there were sometimes differences in test altitude and air turbulence. The mean ride rating for all maneuvers was 3.63 and the corresponding standard deviation was 1.50.

During one 2-hour flight, subjects were exposed to two identical programed sequences to investigate possible changes in test subject's comfort ratings of identical segments spaced 1 hour apart. The results reported in reference 4 indicated no significant effect of time.

To illustrate detailed typical results, the time histories of all 13 recorded motion variables and the 40 individual ride-comfort ratings are presented in figure 5 for the 4 flights of a turning decelerating descent maneuver.

DATA ANALYSIS

Because of the great number of variables involved and the desire to develop a ride-quality model from the data, the regression analysis approach was used to analyze the data. Several analyses were employed to explore the suitability of various parameters and parameter combinations (e.g., peak value accelerations, rms velocities, etc.) to provide a meaningful model.

The simplest measure of each motion variable is the maximum deviation of that variable from zero during the maneuver interval. Initial correlation and regression analyses therefore used as input data maximum variable values which were read directly from time-history plots. However, this approach presented two major difficulties. It was frequently difficult to decide which value of a given variable in a given maneuver interval should be recorded. In addition, amplitude-duration effects were totally lost. Therefore, a further analysis used as independent variables the root-mean-square (rms) values of each motion variable. Table V presents the simple correlations existing among the various rms motion variables and the resulting individual ride-comfort ratings. On the basis of these correlations and to facilitate the comparison of results with those of vibratory-motion ride-comfort experiments, a linear regression analysis was used to obtain the following ride-comfort model based on rms linear accelerations and angular rates:

$$\text{Ride-comfort rating} = 1.65 + 8.32n_x + 15.1n_y + 21.5n_z + 0.183p - 1.20q - 0.238r$$

where: n_x = rms longitudinal acceleration; p = rms roll rate

n_y = rms transverse acceleration; q = rms pitch rate

n_z = rms normal acceleration; r = rms yaw rate

The multiple correlation coefficient for the model is 0.57 and the regression F statistic is 156. The relationship between the ride-comfort rating predicted by this equation for each test maneuver and the mean value of the corresponding 10 experimental passenger ratings for that maneuver is shown in figure 6. For the 192 maneuvers the rms difference between predicted ride-comfort rating and mean experimental rating is 0.55; the corresponding correlation coefficient is 0.85.

APPLICATION OF RIDE-COMFORT MODEL

Method of Application

The ride-comfort model has been used to predict the ride comfort of computer-synthesized simple turns, descents, and decelerations with pitchover. The rms value of each of the six motion variables over the maneuver duration wa calculated and then substituted into the regression equation to obtain a ride-comfort rating. A synthesized turn (fig. 7) is based on three assumptions: a sinusoidal roll-rate time history during turn entry and exit; the lift versus angle-of-attack characteristics of the aircraft (in this case the TIFS); and a level, fully coordinated turn. Roll angle is assumed to be the analytical integral of roll rate. The roll angle, in turn, specifies the normal acceleration, which together with the airspeed determines the pitch angle and hence the longitudinal acceleration. Euler transformations resolve the net aircraft rotation rate into both the assumed roll rate and the corresponding pitch and yaw rates. Parameters which can be varied are airspeed, maximum roll angle, maximum roll rate, and turn duration. The effects of aircraft motion response to atmospheric turbulence are approximated by superimposing on each of the six motion-variable time histories a random oscillatory signal having a zero mean and an appropriate standard deviation. These standard deviations (table VI) were selected by examining rms motion amplitude data obtained aboard commercial airline flights (ref. 5). Similar maneuver synthesis techniques were applied to steady descents and longitudinal decelerations. On an rms amplitude basis, agreement between corresponding variables is quite good. The ride-comfort rating predicted by the comfort model for the synthesized maneuver is 3.1; the mean experimental rating given the actual maneuver is 2.9.

Steady Turns

The variation of predicted ride-comfort rating with roll angle in a steady turn is shown in figure 8(a) for various turbulence levels. For zero and light turbulence, ride comfort is little affected by roll angles less than 20° but degrades rapidly and becomes "Uncomfortable" at about 50°. Turbulence intensity significantly degrades ride comfort for small roll angles but has a much smaller effect as roll angle increases. As the turbulence intensity increases, the roll angle above which ride comfort significantly degrades increases. For zero bank angle and for the various turbulence intensities, ride-comfort rating predicted by a two-degree-of-freedom regression model developed at the University of Virginia (ref. 6) are shown along the ride-comfort axis. Also shown in figure 8(a) are steady-turn data obtained by the University of Virginia during ride-quality flight experiments using the NASA Jetstar aircraft (ref. 7). Agreement is quite good.

The variation of predicted ride-comfort rating with roll angle for various airspeeds is shown in figure 8(b). The slight degradation of comfort at low roll angles with decreasing airspeed is due to increased longitudinal acceleration accompanying increase in pitch angle. For roll angles greater than 30°, decreasing airspeed improves the predicted comfort by increasing the aircraft yaw rate (because the yaw-rate regression coefficient is negative). For typical terminal-area airspeeds, the influence of airspeed on ride comfort in turns appears to be minor.

The variation of predicted ride-comfort rating with roll angle in steady turns for various maximum roll rates is shown in figure 8(c). Maximum roll rates typical of transport aircraft operations appear to have little influence on passenger comfort.

The effects of turn duration (time at maximum roll angle) on the variation of predicted ride-comfort rating as a function of roll angle are shown in figure 8(d). At roll angles less than 27°, increasing duration has a slightly beneficial effect. This effect occurs because of decreased rms roll rate and increased rms pitch and yaw rates (which have negative regression coefficients). For roll angles greater than 27°, the rapid increase in linear acceleration (particularly normal acceleration) with increasing roll angle reverses the situation, so that increased duration results in a degradation of comfort. However, in either case, the effects of turn duration appear to be minor except at high roll angles.

Steady Descents

The variation of predicted ride-comfort rating with steady descent pitch angle for various turbulence intensity levels is shown in figure 9. Because aircraft attitude is constant, ride comfort predicted by the regression model depends only on rms normal and longitudinal accelerations, which are symmetric about a zero pitch angle. This symmetry is due to the nature of the regression model employed and may not properly predict the ride comfort of large nose-up pitch angles. The degradation of ride comfort with increasing pitch angle is practically linear. The effects of turbulence intensity are relatively constant over the range of pitch angles shown. The ride-comfort ratings predicted by the University of Virginia regression model for a zero pitch angle are also shown on the figure.

Longitudinal Decelerations

The variation of predicted ride-comfort rating with average longitudinal deceleration is shown in figure 10(a) for various turbulence intensities. The airspeed is assumed to start at 200 knots and decrease over a 20-second interval (with a sinusoidal time history) at zero pitch angle. This deceleration is followed immediately by a 10-second pitchover to a final pitch angle of -5°. The effect on ride comfort of increasing average deceleration appears to be minor. The effects of turbulence are almost constant over the range of decelerations shown. Ride-comfort ratings predicted by the University of Virginia regression model at zero deceleration are also shown on the figure.

The effects of average deceleration on predicted ride comfort for various negative final pitch angles are presented in figure 10(b). The mean slope of this variation decreases as the magnitude of the final pitch angle increases and may actually become negative at pitch angles more negative than -10°. This is because the pitchover greatly increases the rms normal acceleration for higher airspeeds. Thus, increasing the average deceleration to decrease the airspeed at pitchover reduces the normal acceleration contribution to discomfort. Reduction of airspeed prior to initiating any substantial pitchover will therefore improve ride comfort. The results also suggest that the maximum negative pitch angle be limited to a value of 10°.

CONCLUDING REMARKS

A series of flight experiments has been conducted using a variable-stability research aircraft and a significant number of passenger subjects to investigate the ride quality of terminal-area flight maneuvers. The data obtained have been analyzed through multiple linear regression to produce a ride-comfort model. The model predicts the ride comfort of a flight maneuver as a function of the rms six-degree-of-freedom motions of the aircraft during the maneuver. Application of the model to computer-synthesized maneuver time histories indicates that:

(1) Roll angle during steady turns should be limited to a maximum of 30°

(2) The effects on ride comfort of roll rate, airspeed, and duration during steady turns are minor.

(3) Nose-down pitch angle during steady descents should be limited to a maximum of 10°.

(4) Ride comfort during longitudinal deceleration and pitchover is primarily dependent upon the change in pitch attitude and is only mildly affected by the average longitudinal deceleration.

(5) Reduction of airspeed prior to initiating any substantial pitchover will improve ride comfort.

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TABLE I. - TIFS FLIGHT TEST MANEUVERS

Maneuver type	Variables	Range	Combinations of unique variables
Descent	Pitch attitude	-13.5° to 6.5°	11
	Descent rate	-1.37 to 23.77m/sec	
	Initial altitude	1036 to 3231 m	
Turn	Roll attitude	±50°	23
	Roll rate	±20 deg/sec	
	Airspeed	135 to 205 knot	
	Altitude	427 to 3322 m	
Longitudinal deceleration	Longitudinal deceleration	0.06 to 0.18 g unit	10
	Descent acceleration	0 to 0.79 g unit	
	Final pitch attitude	-6.6 to 0.9°	
	Pitch rate	-5.2 to 0 deg/sec	
	Initial altitude	731 to 3292 m	
Combination	Longitudinal deceleration	0.06 to 0.18 g unit	
	Descent acceleration	0 to 0.6 g unit	
	Final pitch attitude	-5.2° to 3.3°	4
	Pitch rate	-4.9 to 0 deg/sec	
	Roll altitude	±42°	
	Roll rate	± 15 deg/sec	
	Initial airspeed	190 to 210 knot	
	Initial altitude	1006 to 3170 m	
Total number of maneuvers			48

TABLE II. - TIFS MANEUVER EXPERIMENT

PASSENGER SUBJECT CHARACTERISTICS

Characteristics	Air travelers in general, percent	Test subjects, percent
Age:		
20 yr and under	18	16
21 to 40 yr	45	53
41 to 60 yr	32	31
61 yr and over	5	0
Sex:		
Male	75	66
Female	25	34
Frequency of flying (number of times in last 2 yr):		
0	17	6
1 to 3		19
4 to 9	63	31
10 to 25		44
25 and over	20	

TABLE III. - RIDE-COMFORT RATING SCALE

Very comfortable	1
Comfortable	2
Somewhat comfortable . . .	3
Neutral	4
Somewhat uncomfortable . .	5
Uncomfortable	6
Very uncomfortable	7

Table IV. - RIDE-COMFORT RATING DISTRIBUTION

Maneuver tape	Flight test	Rating of -						
		1	2	3	4	5	6	7
II	1	16	43	44	38	58	31	10
I	2	28	53	46	29	50	26	8
II	3	32	48	37	28	68	26	1
II	4	23	61	43	43	51	17	2
I	5	5	47	52	52	68	16	0
I	6	5	41	43	55	81	11	4
II	7	13	70	62	38	46	7	4
I	8	9	43	63	46	55	20	4
Total		131	406	390	329	477	154	33

TABLE V. - CORRELATION MATRIX OF ALL EXPERIMENTAL FACTORS

	Indicated airspeed	Climb rate	Altitude	Flight path angle	Pitch angle	Roll angle	Yaw rate	Pitch rate	Roll rate	Normal acceleration	Transverse acceleration	Longitudinal acceleration
Comfort rating	.147	.232	.086	.210	.212	.199	.184	.378	.207	.493	.219	.305
Longitudinal acceleration	-.060	.415	-.163	.356	.820	.055	.104	.249	.004	.290	.156	1
Transverse acceleration	-.207	-.154	-.085	-.130	.085	.578	.616	.553	.616	.440	1	
Normal acceleration	.059	-.050	-.087	-.068	.157	.689	.642	.925	.585	1		
Roll rate	-.179	-.508	-.011	-.499	-.036	.909	.921	.764	1			
Pitch rate	-.163	-.288	-.111	-.277	.125	.865	.861	1				
Yaw rate	-.288	-.510	-.053	-.485	.033	.964	1					
Roll angle	-.174	-.505	-.034	-.502	-.010	1						
Pitch angle	-.154	.485	-.114	.426	1							
Flight path angle	.068	.980	-.110	1								
Altitude	.211	-.055	1									
Climb rate	.209	1										
Indicated airspeed	1											

TABLE VI. - MOTION-VARIABLE STANDARD DEVIATIONS ASSUMED FOR AIRCRAFT TURBULENCE RESPONSE

Variable	Turbulence Intensity			
	Zero	Light	Moderate	Heavy
Longitudinal acceleration, g unit	0	0.002	0.020	0.040
Transverse acceleration, g unit	0	.003	.030	.060
Normal acceleration, g unit	0	.010	.100	.200
Roll rate, deg/sec	0	.2	2.0	4.0
Pitch rate, deg/sec	0	.1	.6	1.1
Yaw rate, deg/sec	0	.1	.8	1.6

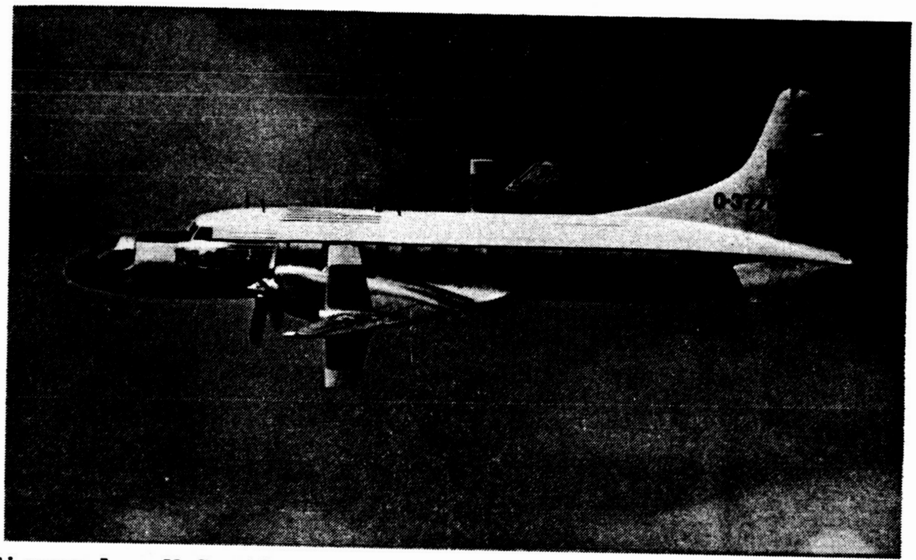


Figure 1.- U.S. Air Force Total In Flight Simulator (TIFS).

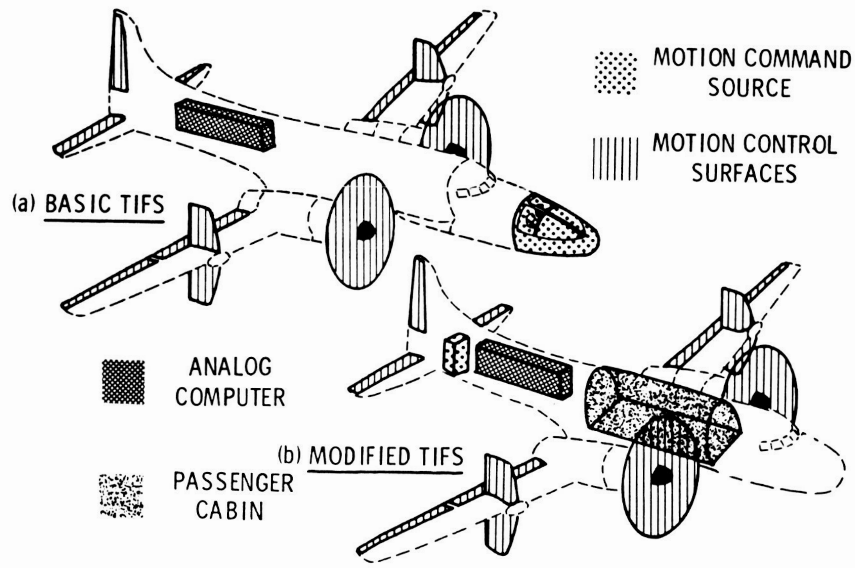


Figure 2.- TIFS modifications for ride-quality research.

Figure 3.- TIFS passenger cabin for ride-quality experiments.

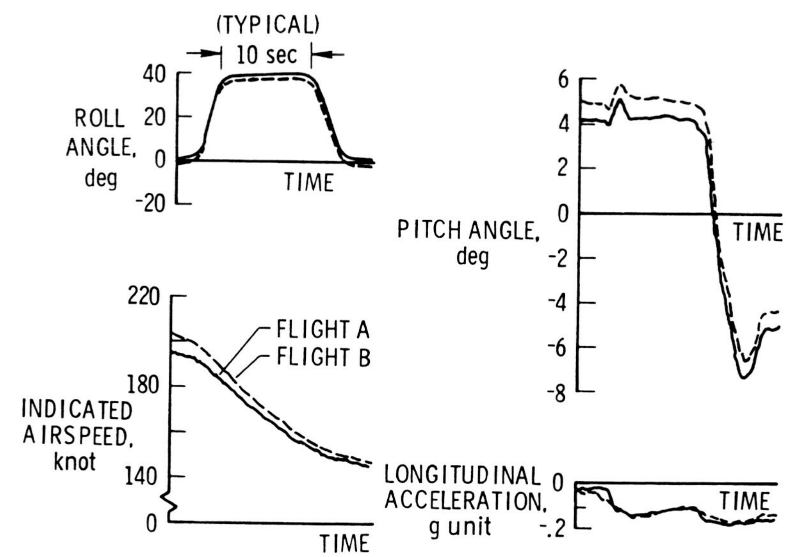
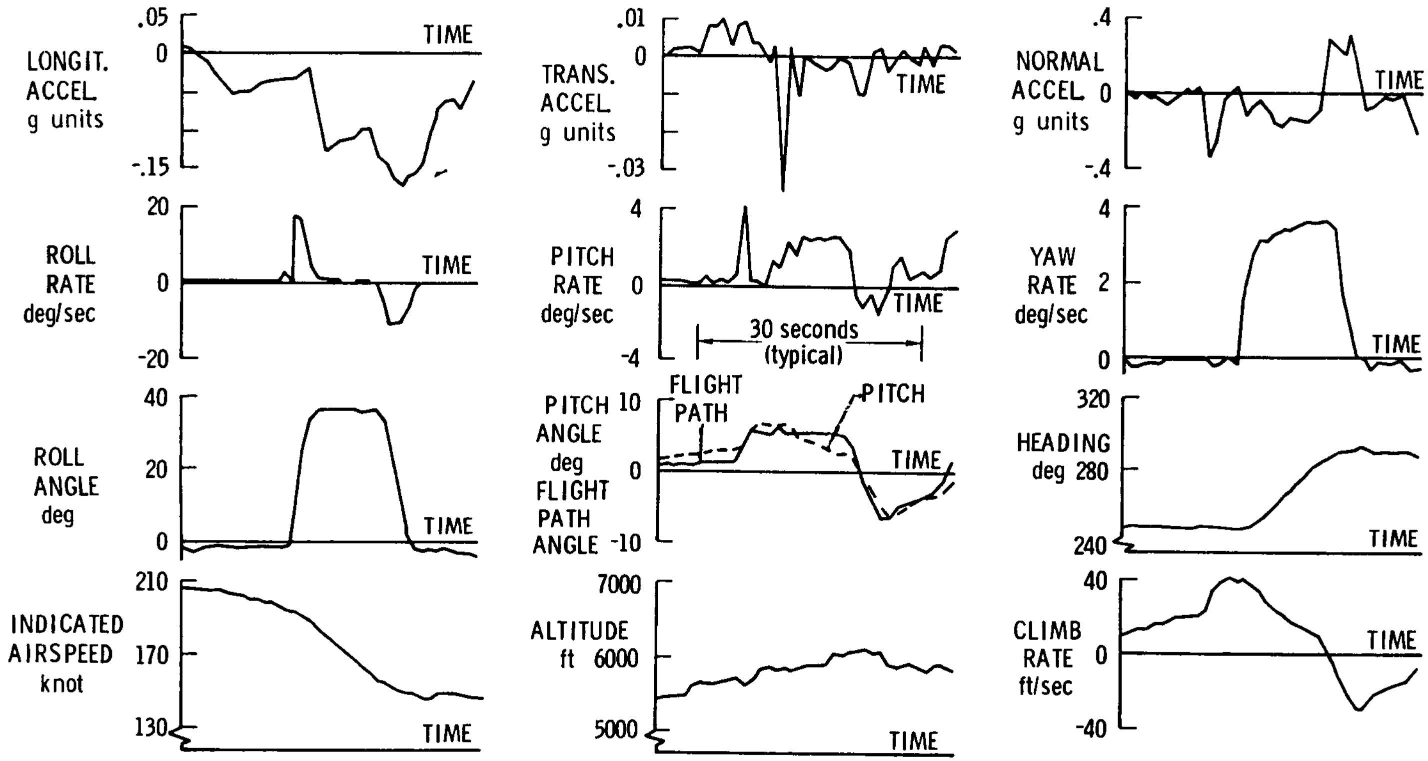


Figure 4.- Example of maneuver repeatability between flights for tape-controlled TIFS in turning decelerating descent.



(a) Time histories of recorded motion variables for turning decelerating descent.

FLIGHT	SEAT 1	2	3	4	5	6	7	8	9	10	MEAN RATING
	RATING										
2	2	7	6	2	4	5	4	2	6	7	4.5
5	5	5	6	5	5	5	4	5	2	3	4.5
6	5	4	6	4	3	3	5	5	5	5	4.5
8	4	3	5	3	5	5	5	4	4	5	4.3

(b) Individual ride-quality ratings obtained during maneuver repetition of four separate flights.

Figure 5.- Example objective and subjective data obtained during TIFS maneuver experiments.

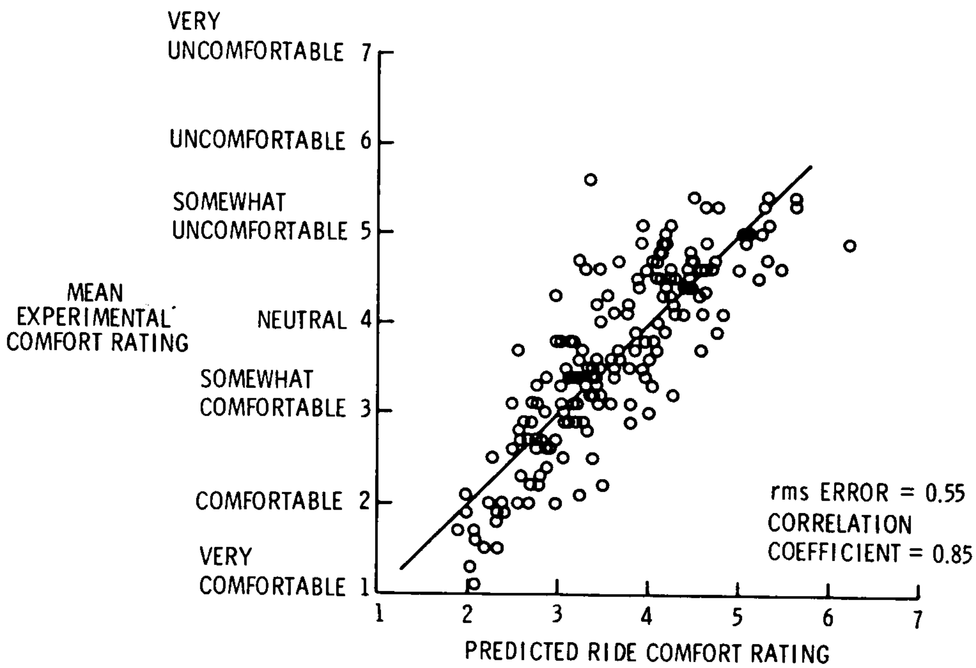


Figure 6.- Experimental rating as a function of model prediction of ride quality.

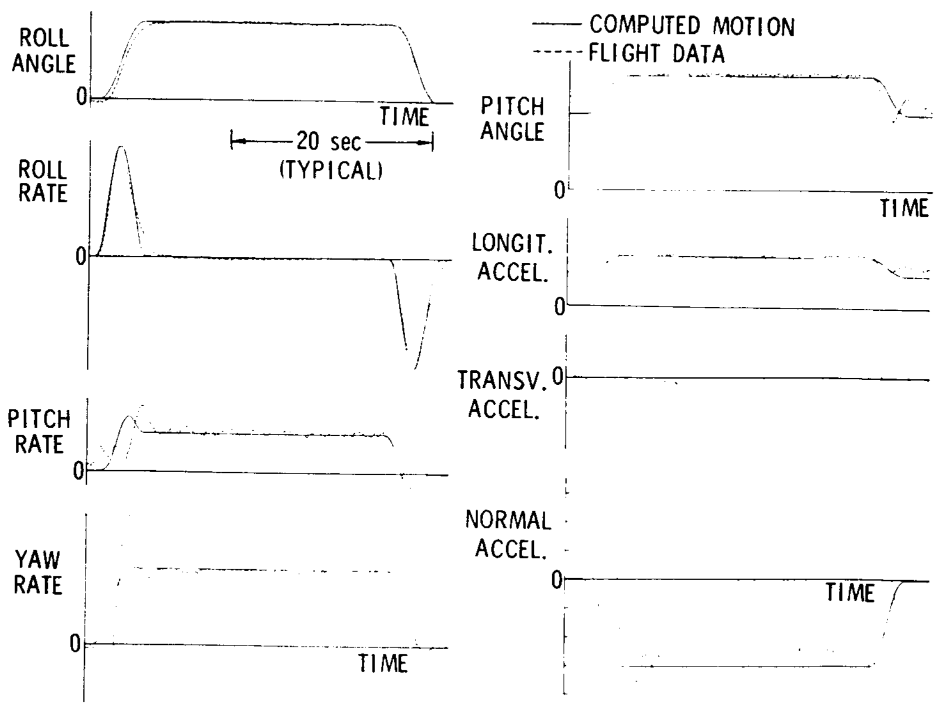
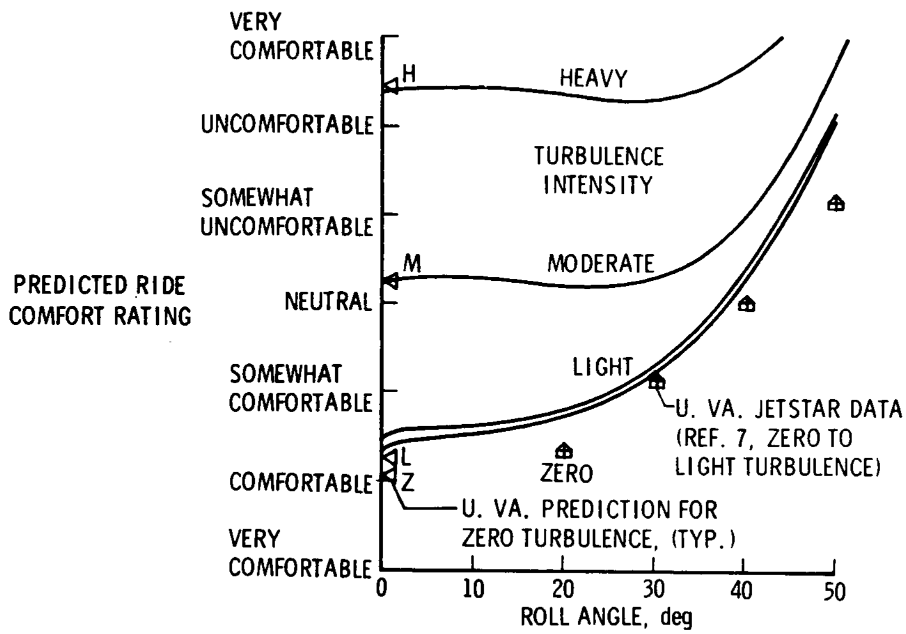
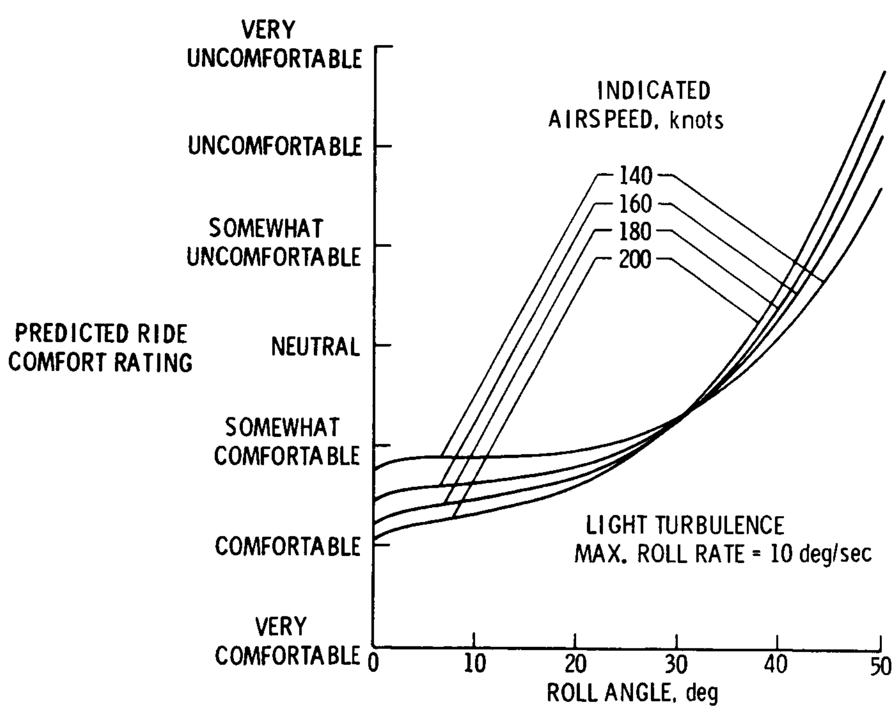


Figure 7.- Time history of example computer-synthesized maneuver (steady turn).

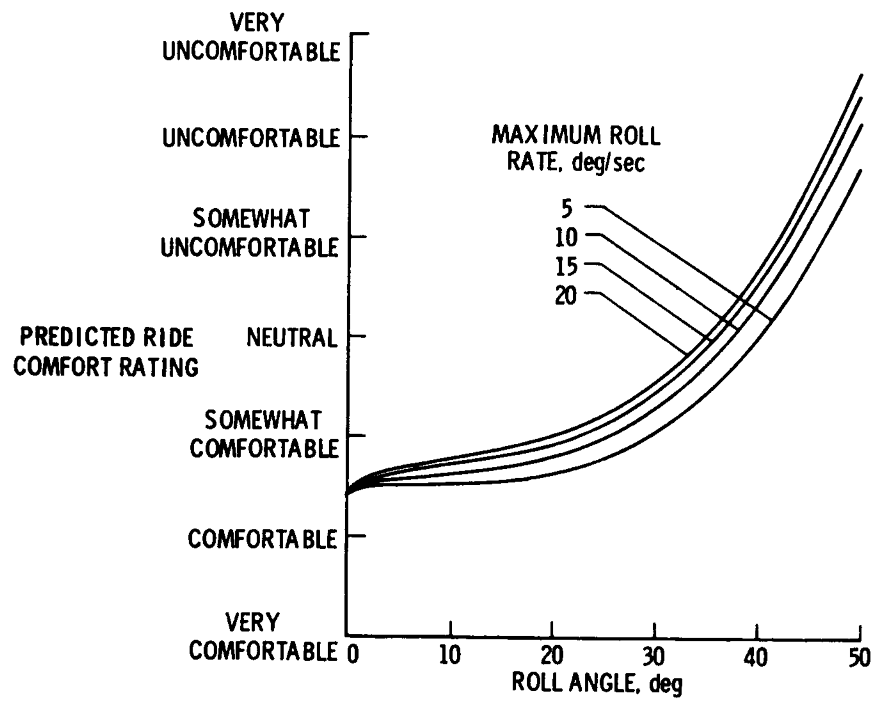


(a) For various turbulence intensities.

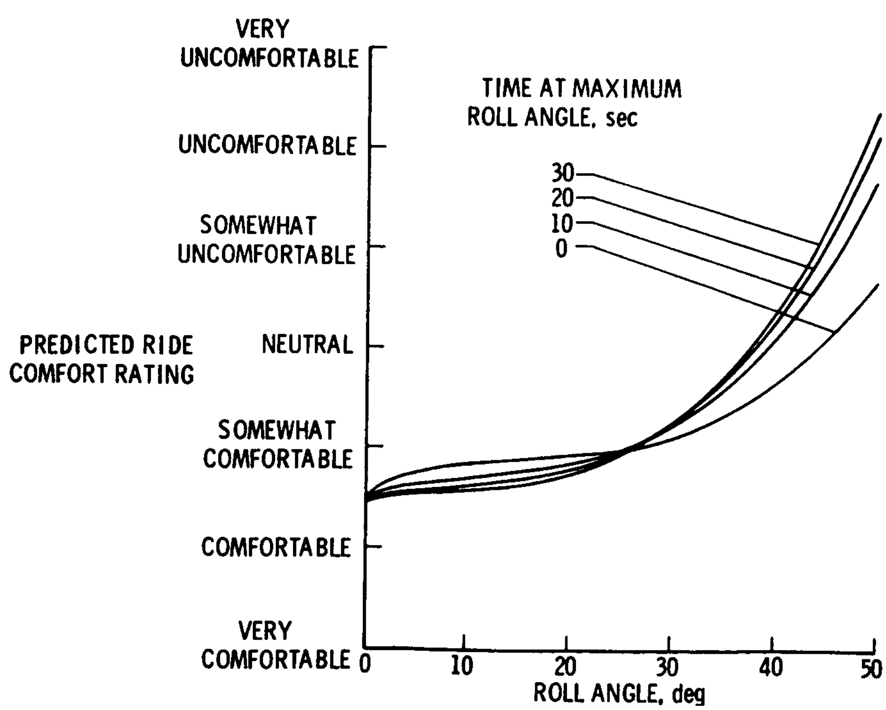


(b) For various airspeeds.

Figure 8.- Predicted comfort of steady turns.



(c) For various maximum accelerations.



(d) For varying durations.

Figure 8.- Concluded.

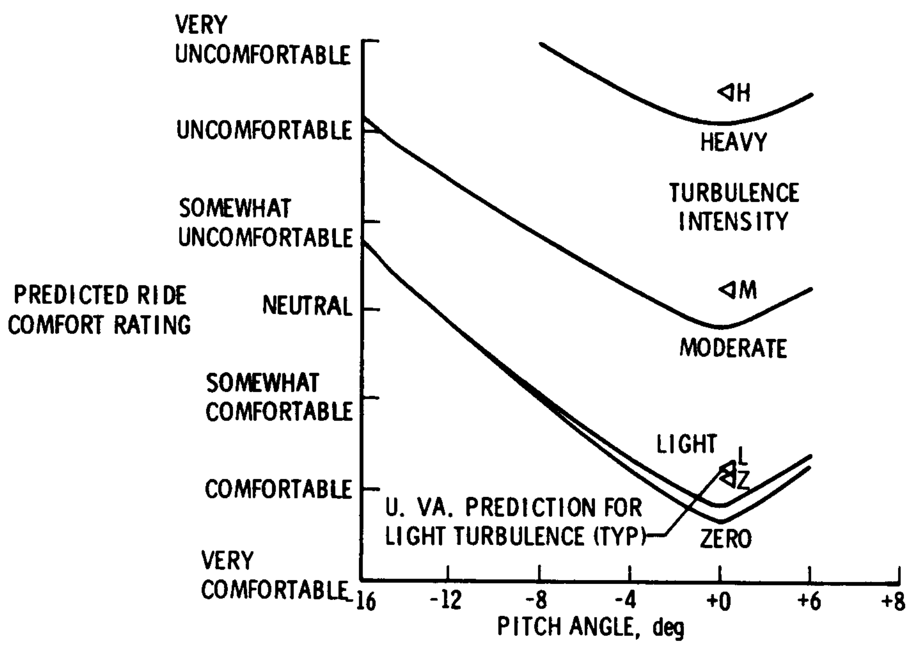
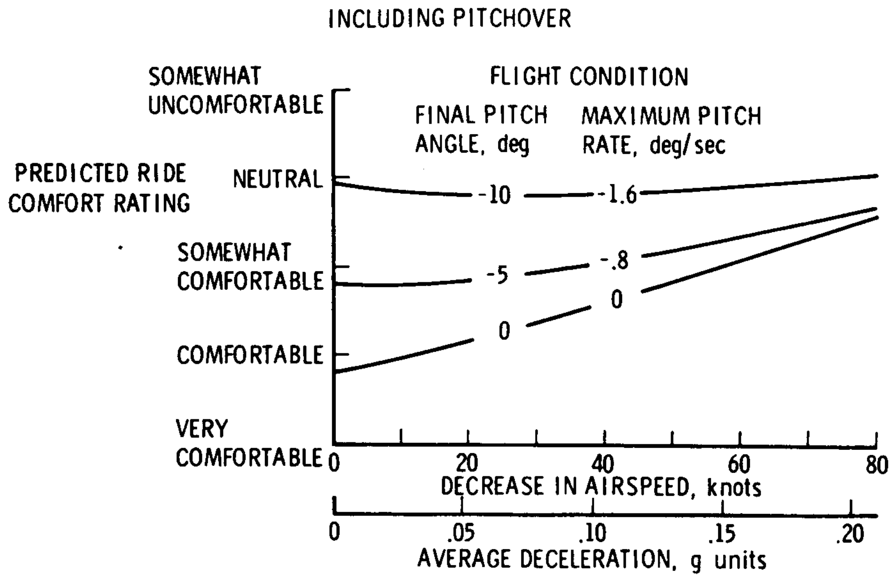
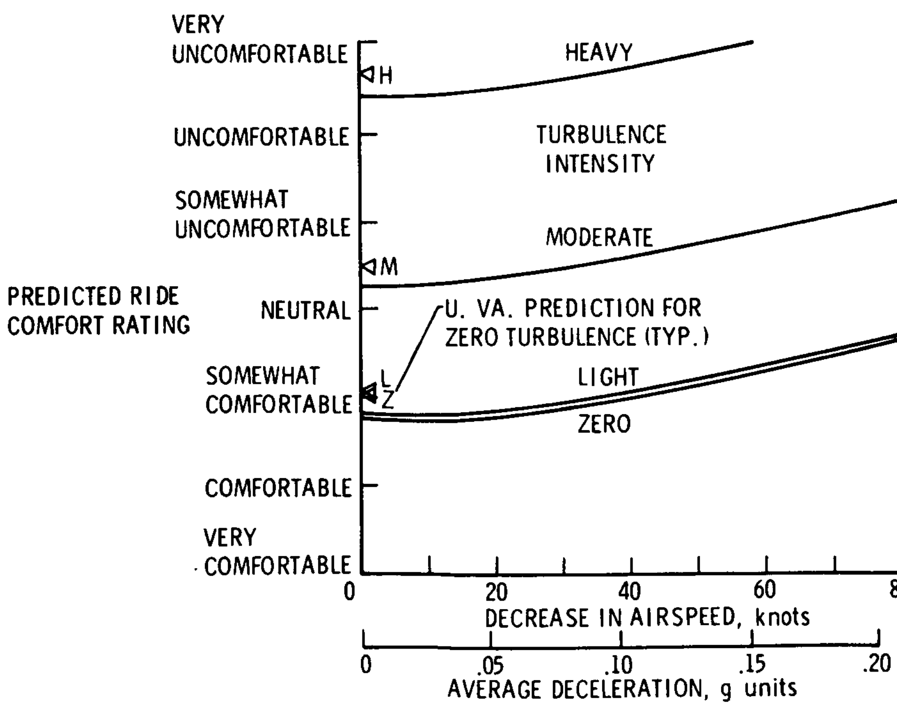


Figure 9.- Predicted comfort of steady descents for various turbulence intensities.



(a) For various turbulence intensities.



(b) For various final pitch angles.

Figure 10.- Predicted comfort of longitudinal decelerations.

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PASSENGER RIDE QUALITY DETERMINED FROM COMMERCIAL AIRLINE FLIGHTS

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SUMMARY

The University of Virginia ride-quality research program is reviewed. Data from two flight programs, involving seven types of aircraft, are considered in detail. An apparatus for measuring physical variations in the flight environment and recording the subjective reactions of test subjects is described. Models are presented for (1) predicting the comfort response of test subjects from the physical data, and (2) predicting the overall comfort reaction of test subjects from their moment by moment responses. The correspondence of mean passenger comfort judgments and test subject response is shown. Finally, the models of comfort response based on data from the 5-point and 7-point comfort scales are shown to correspond.

INTRODUCTION

The general goal of the research reported here is to determine the relation between passenger comfort and vehicle ride quality. This goal implies two problems: (1) characterize and measure vehicle ride quality--a physical problem involving analysis of the environment, and (2) characterize and assess passenger reactions to that environment--a psychological problem. Determining the relations between problems (1) and (2) is a psychophysical problem.

PROBLEM 1: ANALYSIS OF THE ENVIRONMENT

The University of Virginia ride quality program has been concerned mostly with aircraft. The flight environment for a passenger consists of (1) the seat in which he finds himself, (2) the surrounding space-both tactile and visual, and (3) the physical conditions acting on him, such as motion, noise, temperature, pressure, lighting, and so on.

An emphasis on ride quality implies primary interest in the motion variables and the seat. UVa has designed and built a portable ride quality measuring apparatus (see ref. 1). It permits continuous recording of a vehicle's motion characteristics in 6 degrees of freedom: 3 linear accelerations and 3 angular rates. Pressure, temperature, and noise are also recorded and separate channels permit voice entries and a numerically coded comfort response to be entered by a test subject. This instrument is carried aboard a vehicle, and after some processing, a trace like that in figure 1 results. Noise spectra are also processed--a typical output shown in figure 2.

Measurements and descriptions of the seat, surrounding space, overall noise level (dB(A)), and temperature are taken by hand. Thus, most of the problem of characterizing the environment has been solved.

PROBLEM 2. ANALYSIS OF PASSENGER REACTIONS

The problem of how to assess psychological reactions is more complex. First, one must decide which states or reactions are most relevant. Passenger comfort is clearly important; on the one hand, it seems to be the most direct psychological correlate of ride quality; and on the other, it would seem to be related to a passenger's satisfaction with a mode of travel, his willingness to use it again. If one is comfortable on this trip, all other things being equal, one will probably be willing to use this vehicle again. The point of ride quality research is to increase passenger acceptance of particular types of vehicles, so as to increase actual use of them.

Comfort is a state of feeling, an affective reaction. It is assumed to depend on inputs from the environment, especially motion and seat variables. The passenger receives various physical inputs continuously throughout a flight. In figure 3, aspects of the physical environment are assumed to map into sensations or perceptions. Conglomerate impressions may exist for motion and seat variables. These inputs all influence one's level of comfort.

A passenger's comfort level may also depend on his expectations, anxiety, state of health, and so on. Individual difference variables of interest include (1) attitudes, beliefs, fears, moods & anxiety-psychological factors, (2) age, sex, somatotype, tendency toward motion sickness & general health-physiological factors, and (3) previous flight experience, preflight experiences, socioeconomic status, demographic characteristics--situational factors.

Comfort level acts to determine satisfaction with a flight and is coded in memory for future decisions. Figure 3 outlines a theory of comfort--a set of hypothesized relations to be tested empirically. Consider further a passenger in an airplane; he has come into a situation for a purpose. The purpose is to travel, to get from one place to another, but the passenger might have any of several reasons for traveling. His being in this situation may be the result of a choice between competing modes of travel. Such a decision would be influenced by attitudes, beliefs, and expectations concerning, say, air travel, based on prior experience and communication. Finally, the passenger holds values--some specific to travel, others more general, and these values influence choice, decision, and evaluation concerning air travel.

All the considerations in this section influenced the development of two questionnaires. The questionnaires were designed for use on board regularly scheduled commercial flights involving fare-paying passengers. Both questionnaires asked for (1) demographic information, (2) attitudes about, purpose of, and frequency of flying, (3) the perceived importance of various physical factors in determining comfort, (4) a comfort rating, and (5) an evaluation in terms of willingness to fly again on this type of craft. A sample questionnaire is shown in Figure 4. Various other items will be discussed as the results are reported.

FLIGHT PROGRAMS

Two initial flight programs involving fare paying passengers were conducted. In the first, three planes were used: The Volpar Beech, Nord, and Twin Otter; in the second, three planes and one helicopter: Beech 99, Nord, Twin Otter, and the Sikorsky S-61 helicopter. All these aircraft are used for commuter service. One or two UVa test subjects were present on each flight. These subjects were specially trained and highly experienced. They operated the ride-quality apparatus on the various flights and provided ratings of their comfort levels throughout the flight. The goal was to obtain motion recordings and subject comfort ratings for about 10 two-minute intervals of a flight. In addition, a comfort rating for the total flight was also obtained from each test subject. A five-point rating scale was used in the first flight program, a 7-point one in the second.

All passengers on each flight were asked to complete a questionnaire. It was filled out near the end of the flight, about five minutes prior to landing. There were 758 passengers in the first flight program and 861 in the second. Figure 5 provides an overview of the data collected in the UVa ride quality program. Motion recordings, test subject ratings, and passenger questionnaire data were collected during the two flight programs. Test subjects were also used to gather data in simulators: in flight: TIFS (Total in Flight Simulator) and GPAS (General Purpose Airborne Simulator), and on ground: PRQA (Passenger Ride Quality Apparatus), VMS (Vision Motion Simulator), and RDS (Ride Dynamics Simulator). These simulators and the data from them will be discussed in detail by others at this conference: some simulator data will be reported briefly later in this paper.

QUESTIONNAIRE RESULTS

Characteristics of the samples of passengers in the two flight programs are shown in table 1 along with information from General Travel Surveys. Age, income, education, occupation and purpose of trip information is comparable for all three groups. The ratio of men to women deviates from that reported in general travel surveys. In flight program II, the proportion of women varied with plane type; a greater proportion of the passengers were women on the Twin Otter (32%) and the Beech (26%) than on the S-61 (14%) or the Nord (10%). In general, the proportion of women flying commuter service is quite small.

Both flight samples had a predominance of frequent travelers; 75% of the passengers in the first sample had flown 10 or more times in the prior two years, while only 2.3% had not flown before; in the second flight sample, 70% of the passengers had flown 4 or more times, but 16% were flying commuter service for the first time.

Attitudes toward flying were generally favorable. In the first sample, 45% of the passengers reported that they "loved to fly," 34% had "no strong feelings," and 21% "flew because they had to" and 0.7% said they disliked flying. This item was ambiguous: more than one response might be appropriate for a given passenger. In the second questionnaire, one item assessed attitude toward flying, while another asked whether one had to fly. The contingency table relating these two items is shown in table 2. Of those who have to fly, about ½ have no strong feelings about flying; of those who don't have to fly, 67% indicate that they like to do so.

Factors in satisfaction with air travel were rated similarly by both samples. The first sample rated safety and reliability of greatest importance, followed, in order, by time savings, convenience, comfort, and cost. In the second sample, time saving and on time arrival and departure were rated very important, with convenience and ride comfort rated moderately important.

Passengers report that thinking and looking out the window are the most frequently performed activities, with reading and talking also done with some frequency. Writing is rarely done and is rated difficult to perform on these types of planes. Ability to work (read or write) was however rated one of the least important factors in trip satisfaction.

COMFORT RESPONSE

Passengers were asked to report their level of comfort on their flight. A rating scale with adverb-adjective descriptors was used: a five-point scale for the first flight program, and a seven-point scale for the second. The distributions of comfort judgments taken over all passengers, flights, and plane types are shown in figure 6. The distribution on the left is from the first questionnaire with a five-point rating scale. Using the seven-point scale distributed the judgments in the middle range, fewer passengers found it necessary to use the neutral category. The percent of passengers using the extreme categories was about the same for both samples.

Comfort should be related to satisfaction. In terms of questionnaire items, the rated comfort level should correlate with willingness to fly again. Figure 7 shows the percent of passengers with no doubts about taking another flight plotted against comfort rating for the first sample. As rated comfort decreases so does the percent of passengers willing to fly again. For the second sample, the curve in figure 8 results. The same decline in percent of passengers satisfied occurs as comfort decreases. The adjectival labels for the two comfort scales were identical at the two extremes ("very comfortable," "very uncomfortable") and in the middle of the scale ("neutral"). For these three scale positions, the percent of subjects with no doubts about flying again are nearly identical, see the heavy dots in figure 8. Thus, subjects in both flight programs relate the comfort scale to satisfaction in the same way. Further, the curve drops in the predicted manner through scale points 2, 3, 5, and 6. Thus, not only does the relation between comfort and willingness to fly replicate, but the meaningfulness of the scale labels is supported by this replication.

COMFORT RATING AS A FUNCTION OF PLANE

Tables 3 and 4 show the distribution of comfort ratings according to the type of plane. For each program, these distributions differ as a function of the plane. For flight program I, the mean comfort ratings were 2.71 for the Nord, 2.97 for the Volpar Beech, and 3.02 for the Twin Otter. For flight program II, the order of aircraft by mean comfort response is S-61 ($\overline{X}$ = 2.71), Nord ($\overline{X}$ = 3.52), Twin Otter ($\overline{X}$ = 3.55), and Beech ($\overline{X}$ = 3.60). The Nord is rated more comfortable than the other two planes in both samples. However, the S-61 helicopter is rated the most comfortable vehicle in flight program II. It should be noted that it has the shortest average flight time (7-10 minutes).

Although women are relatively more prevalent on the Twin Otter and Beech, the least comfortable planes, the distribution of their comfort responses (see table 5) overrepresents the best comfort categories. Thus, these two aircraft may have higher mean comfort ratings than they would have given samples whose proportions more closely approximated those of the S-61 and Nord.

PHYSICAL FACTORS RELATED TO COMFORT

In the first flight program, passengers were asked to rank the importance of various physical factors in determining their level of comfort. Table 6 shows the results: seat comfort was seen as most important, followed by noise and temperature, then the motion factors. Figure 9 shows the mean rankings of the physical factors in comfort separately for men and women. Women rated seat comfort less important, and gave greater importance to the motion variables than did the men.

In the second questionnaire, passengers' perceptions of these various factors were assessed directly. This questionnaire contained rating scales for rather detailed aspects of the physical situation: motion, temperature, pressure, lighting, noise, workspace, ventilation, smoke, and odors. A separate set of items dealt with seat variables: firmness, width, adjustment, leg room, and shape. Passengers were asked to rate their discomfort on these various physical factors. Thus, passengers indicated what they thought influenced their comfort. These ratings of physical factors could then be related to overall comfort ratings to provide direct assessment of the perceived aspects of the flight environment presumed to be related to comfort.

Seven of the physical factors showed no significant relation to plane type. These were lighting, noise, odors, tobacco smoke, temperature, ventilation, general vibration, and turning. For five of these physical factors, 75% of the respondents indicated that they were "not uncomfortable" due to that factor. However, most respondents cited that they were at least "somewhat uncomfortable" due to noise and general vibration. Between 60% and 72% of the passengers experienced discomfort due to noise, and between 54% a 66% did so due to general vibration.

Significant relationships between plane type and response are evident for pressure, workspace, sudden jolts, up and down motion, backward and forward motion, side to side motion, and sudden descents. The strongest relation to plane type was found for up and down motion: Forty-eight percent of the passengers found the Twin Otter and Beech uncomfortable on this factor, while only 21% so rated the S-61 and only 12% of the Nord. Discomfo due to side to side motion is also significantly related to plane type. Over a third of the passengers on the Twin Otter and Beech reported discomfort, but only 17% of the S-61 passengers did, and only 10% of the Nord passengers. Similar patterns of differences emerge for sudden jolts, backward and forward motion, and sudden descents. In each case, the Beech and Twin Otter are associated with greater proportions of uncomfortable passenge However, on the last two physical factors, less than 25% of the passengers are uncomfortable. Pressure is also significantly related to plane type. The Beech is uncomfortable to 60% of the passengers, while the proportions for the other three plane types range from 26 to 38 percent. Workspace is rated uncomfortable by 81% of the Nord passengers, but by only 43% of the S-61 passengers. The Twin Otter and Beech are also rated poorly.

SEAT VARIABLES

Passenger reactions to five aspects of the seat were obtained. Passengers could "agree," "disagree," or "strongly disagree" with the statements: "The seat has enough leg room," "The firmness of the seat is satisfactory," "The seat is wide enough," "The shape of the seat is satisfactory," and "The seat can be adjusted to your satisfaction." Characteristics of the seats for the four aircraft are summarized in table 7. Seat firmness is generally satisfactory; 75% of the respondents agreed with this statement for the Nord, and even greater agreement was found for the other planes. All seats had foam cushions. Seat shape was rated poorly for the Nord, but not for the other three planes. The S-61 helicopter had the greatest percentage of passengers satisfied with both seat shape and firmness. Seat adjustment was uniformly poor, the highest percent agreement was for the S-61 and that was only 43%. Since none of the seats could be adjusted, it is assumed that some passengers were responding to the actual position of the seat rather than its potential for adjustment. Those passengers who agreed with the item on adjustment probably felt that the seats were already adjusted to their satisfaction.

Sixty-one percent of the passengers were satisfied with the seat width on the S-61, 57% on the Beech, but both the Twin Otter and Nord were rated unsatisfactory by most of the passengers. Only the S-61 satisfied a substantial proportion of the passengers with respect to leg room. These two seat variables are quantitative. Measurements of width and leg room are given in table 7. When the percentage of passengers satisfied is plotted against these measurements, figures 10 and 11 result. Seat width is related linearly to percent of passengers satisfied; further, the difference in width between a seat that satisfies 61% of the passengers (S-61) and one that satisfies 34% (Nord) is only 11.4 centimeters. Leg room is related to percent of passengers satisfied in a nonlinear fashion. There is a large increase in percent satisfied when leg room is increased from 24 to 27 centimeters. The S-61, which rates best on leg room, also rated best on workspace in the previous item.

PERCEIVED RELATIONS BETWEEN ENVIRONMENTAL FACTORS

Do passengers tend to respond as though certain environmental factors go together? Goodman and Kruskal's (ref. 2) gamma coefficient (γ) was chosen to index the degree of association between responses to different items. When all the environmental factors (physical and seat) are related to each other over all plane types, the γ's in table 8 result. Two major clusters are immediately apparent: one involving the motion factors and the other, the seat factors. The γ's within each cluster are quite large, while the γ's relating factors in the motion cluster to those in the seat cluster are small. Thus motion factors appear to be independent of seat factors. Workspace goes into the seat cluster and is strongly related to leg room and seat width. The motion factors are all highly interrelated, with general vibration associated with the motion cluster and with noise. Judgments of discomfort due to temperature and ventilation also tend to covary. It should be kept in mind that these results concern the structure implicit in response variation from the passengers and not necessarily the actual physical covariation present in the environment.

RELATION OF RATED SOURCES OF DISCOMFORT TO OVERALL COMFORT JUDGMENTS

Gamma coefficients were computed relating the degree of discomfort attributed to each of the environmental factors to the overall comfort rating and to the rated willingness to fly again. These values are shown in table 9. Ratings of noise, vibration, motion, and seat variables are significantly associated with comfort and evaluation judgments. Passengers perceive these factors as determinants of their comfort level, and their judgments covary in an appropriate way. These ratings of discomfort due to environmental factors are rather crude, but they suggest that our modelling efforts are concerned with the right variables.

MOTION VARIABLES AND TEST SUBJECT RESPONSES

The output from the ride quality measuring apparatus was processed by the NASA Langley time series analysis program and sent to Univ. of Va. as rms values and power spectra for the motion variables and appropriate digital representations of the other physical variables. Thus, a series of numerical values corresponding to each two minute flight segment was obtained, together with a rating of that segment by one or two test subjects.

Various models were examined for the data from the first flight program which had used a five-point comfort scale. In trying to predict comfort ratings from rms motion values, a simple linear model proved best. More complex models were tried but the small increase in the percent of variance they accounted for did not justify the added complexity. For the commercial airline data obtained using the five-point comfort scale, the model is given by

$$C = 2.0 + 7.6\,\bar{a}_{\text{TRANSVERSE}} + 11.9\,\bar{a}_{\text{VERTICAL}}$$

In all the tests done to date, vertical and transverse accelerations dominate the comfort responses. The constants are all significant at the 0.001 level or better, the Pearson correlation is 0.72, and the rms residual error is 0.59. The N for this model is 2976. This model is valid over the range of accelerations which were encountered in commercial operations, given by

$$\bar{a}_V \geq 1.6\bar{a}_T$$

For the range of accelerations $\bar{a}_V < 1.6\,\bar{a}_T$, the data from the flight-similator (Jetstar GPAS-see ref. 3) experiments were used giving an equation of the form

$$C = 2 + \bar{a}_V + 25\,\bar{a}_T$$

Again the constants are all significant at the 0.001 level or better. A composite of these two models is shown in figure 12 with isocontours of constant C indicating the transition region from comfortable to uncomfortable motions.

For each flight, test subjects provided an overall comfort rating and the mean of the passenger comfort ratings was computed. These quantities are plotted against each other in figure 13. There is some curvilinearity to the relation, but one can predict mean passenger response quite well using the overall rating from the test subjects.

The moment by moment test subject responses were related to their overall responses with a variety of types of models. The best fitting information integration model was a simple weighted average with the weight for each data point increasing as the time into the flight increased. The largest weights were given to the segments near the end of the flight. The best weighting function is shown in figure 14.

Data from the second flight program (7-point scale) were also used to model test subject reactions to the motion variables. The data reported here are only for the Nord and Twin Otter. Again, a simple linear model involving vertical and transverse accelerations was best. The equation for predicting comfort was

$$C = 2.1 + 17.1\bar{a}_T + 17.2\bar{a}_V$$

This equation yielded a multiple R of 0.75, thus accounting for 56% of the variance in comfort judgments. While the transverse component is significant, it should be noted that the correlation of vertical rms alone and comfort is 0.74. Further vertical and transverse accelerations are highly correlated ($r = 0.82$). Table 10 gives the summary statistics for the physical measures and test subject comfort responses, table 11 shows their intercorrelations.

Isocontours of constant C are plotted in figure 15 for equations based on both the 5-point model and the 7-point model. There are some discrepancies, but in general the models agree. If values of the model equations are solved in terms of a_V, and various a_V, values are inserted into both equations, the relations between the two models can be derived. Figure 16 shows the results of that process. Figure 16 may be used to convert 5-point comfort ratings to 7-point or vice versa. The two models produce predicted comfort values that are linearly related to each other.

FURTHER REMARKS

These two flight programs are part of a larger research effort dealing with ride quality. They were preceded by a flight program used to test the instrumentation (ref 1) and ground-based surveys (refs. 4, 5) used to develop and refine the questionnaires. The data from the flight programs are reported in greater detail in a series of papers appearing in the British journal Ergonomics (refs. 5 to 7). Additional commercial flight programs are now in progress.

In-flight flight similators are also being used to investigate ride quality. With test subjects and experimental aircraft, motion characteristics not normally seen on commercial flights can be realized. For example, a preliminary investigation of the effect of bank angle on comfort ratings was carried out on the Jetstar. The results are plotted in figure 17. Mean comfort responses do change with bank angle, and a 25° bank is probably a maximum for comfortable passenger operation (ref. 8). Ground-based simulator studies are also being pursued.

Other directions in which our research is going include (1) the effects of noise on perceived comfort, (2) the role of anxiety and mood in determining reaction to a flight, and (3) the extension of our research effort to other modes of transportation.

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Table 1. Characteristics of the flight samples

	General Travel Surveys	Previous In-flight Sample	Present In-flight Sample
N	3000+	758	861
Sex			
Male	75%	88%	80%
Female	25	12	20
Age			
20 & under	12	6	4
21-40	40	47	45
41-60	35	42	45
over 60	13	5	6
Education			
College	80	81	N.A.
Noncollege	20	19	N.A.
Occupation			
Executive Managerial Professional Technical	60	68	66
Other	40	32	34
Purpose of Trip			
Business	75	79	72
Other	25	21	28
Income			
Median	$22,000	$22,293	$24,069

Note: N.A. = not asked on this questionnaire.

Table 2. Feelings about flying versus "have to fly" responses for commuter flights

Have to fly	Feelings about flying			
	Positive	Neutral	Negative	N
Yes	40.4	48.8	10.8	498
No	66.8	30.0	3.2	280

Table 3. Distributions of passenger comfort ratings by type of aircraft*

	1	2	3	4	5	N
Nord	7.6	35.3	38.0	16.4	2.7	408
Volpar Beech	1.0	34.0	37.0	23.0	5.0	100
Twin Otter	5.1	24.8	38.9	25.2	6.0	234

* Table entries are percent of row total.

Table 4. Distributions of rated comfort by plane type*

	Comfort Rating							
	1	2	3	4	5	6	7	N
Beech	5.5	29.4	15.3	17.8	18.4	8.0	5.5	163
Nord	7.2	23.7	22.4	12.5	27.0	5.3	2.0	152
S-61	12.6	46.5	12.3	17.9	8.2	1.9	0.6	318
Twin Otter	6.2	28.7	16.9	14.9	22.1	7.7	3.6	195

* Table entries are percent of row totals.

Table 5. Distribution of comfort responses by sex *

	1	2	3	4	5	6	7	N
Male	7.1	32.8	16.9	17.8	17.1	5.7	2.6	662
Female	15.6	42.5	11.3	10.0	15.6	2.5	2.5	160

*Table entries are percent of row totals.

Table 6. Rank ordering of physical factors in comfort (first flight program)

	Total In-flight Sample	Sex		Ground-based Sample
		Male	Female	
Seat comfort	1	1	4	1
Noise	2	2	2	3
Temperature	3	3	3	2
Up & down motion	4	5	1	4
Pressure changes	5	4	6	7
Side-to-side motion	6	6	5	5
Work space	7	7	9	9
Lighting	8	8	7	6
Smoke	9	9	8	8

Table 7. Approximate seat dimensions and features

Aircraft	Width	Depth	Arm Rests	Leg Room*	Adjustment	Cushion Type
Twin Otter	41.3 cm	45.7 cm	No	24.1 cm	None	Foam
Nord 262	37.5 cm	44.5 cm	Yes	20.3 cm	None	Foam
Beech 99	44.5 cm	44.5 cm	No	20.3 cm	None	Foam
S-61	48.3 cm	45.7 cm	Yes	21.6-26.7cm	None	Foam

* Between seats (front of passengers seat to point of contact with the seat in front--in upright position).

Table 8 . Association (gammas) between ratings on environmental factors. Decimal points are omitted.

	Factors rated																		
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
1. Lighting																			
2. Pressure	50																		
3. Noise	42	67																	
4. Odors	57	35	57																
5. Tobacco Smoke	13	31	41	60															
6. Temperature	52	31	38	49	42														
7. Ventilation	46	33	42	53	56	90													
8. Workspace	46	27	47	30	26	47	46												
9. General Vibration	52	48	74	53	27	48	44	48											
10. Sudden Jolts	37	45	41	38	22	40	36	38	78										
11. Up & Down Motion	34	46	37	35	30	43	33	35	73	94									
12. Backward & Forward Motion	52	48	42	53	46	54	47	40	76	90	96								
13. Side to Side Motion	39	43	37	46	32	43	40	42	72	89	93	96							
14. Sudden Descents	43	50	42	42	26	50	51	34	71	89	89	87	87						
15. Turning	57	55	65	63	33	62	67	34	76	84	84	90	90	95					
16. Leg Room	24	27	31	02	28	26	29	66	23	26	32	26	29	23	07				
17. Seat Firmness	41	40	38	31	30	36	45	61	39	27	30	53	36	42	55	70			
18 Seat Width	25	13	33	14	22	22	29	61	37	29	34	29	34	30	15	66	74		
19. Seat Shape	22	26	34	23	26	26	23	48	32	20	24	31	29	38	45	63	87	75	
20. Seat Adjustment	26	14	35	25	26	19	30	54	31	29	29	27	33	32	19	66	68	70	74

Table 9 . Association (gammas - γ's) between rated sources of discomfort and overall comfort judgments and evaluations

Factor	Comfort Judgment	Willingness to Fly Again
Lighting	.27	.25
Pressure	.26	.28
Noise	.41	.38
Odors	.15	.29
Tobacco Smoke	.23	.15
Temperature	.27	.25
Ventilation	.31	.26
Workspace	.49	.46
General Vibration	.44	.39
Sudden jolts	.43	.47
Up and Down Motion	.46	.41
Backward and Forward Motion	.40	.49
Side to Side Motion	.48	.50
Sudden Descents	.35	.45
Turning	.28	.41
Leg Room	.54	.43
Seat Firmness	.54	.52
Seat Width	.52	.41
Seat Shape	.51	.48
Seat Adjustment	.47	.34

Table 10

Summary Statistics on Physical Measures and Comfort (rated by Test Subjects)

	Mean	Median	Mode	Std. Dev.	Maximum	Minimum	Skewness
Comfort	3.140	3.046	3.000	.935	6.000	2.000	.574
Yaw	.263	.119	.009	.372	3.646	.009	3.575
Roll	.961	.721	.455	.735	3.642	.112	.980
Pitch	.300	.211	.109	.252	2.227	.046	2.340
Longitudinal	.014	.013	.011	.009	.076	.001	1.826
Transverse	.014	.010	.001	.012	.080	.001	1.622
Vertical	.044	.034	.014	.031	.188	.008	1.529

Table 11

Intercorrelations of Physical Measures and Comfort (as rated by Test Subjects)

	Comfort	Yaw	Roll	Pitch	Longitudinal	Transverse
Yaw	.30					
Roll	.71	.50				
Pitch	.56	.66	.81			
Longitudinal	.30	.25	.41	.54		
Transverse	.68	.57	.86	.78	.40	
Vertical	.74	.40	.91	.78	.39	.82

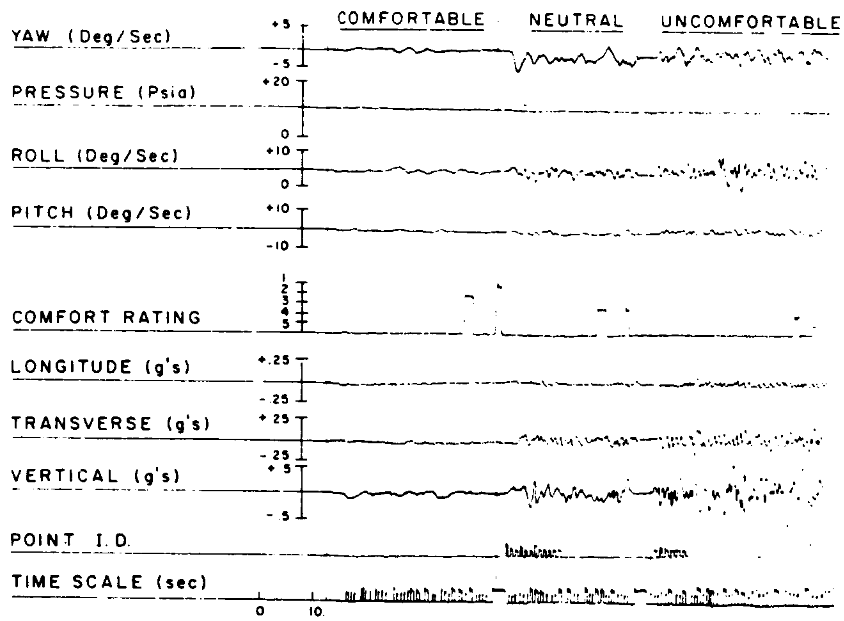

Figure 1.- Typical motion time histories.

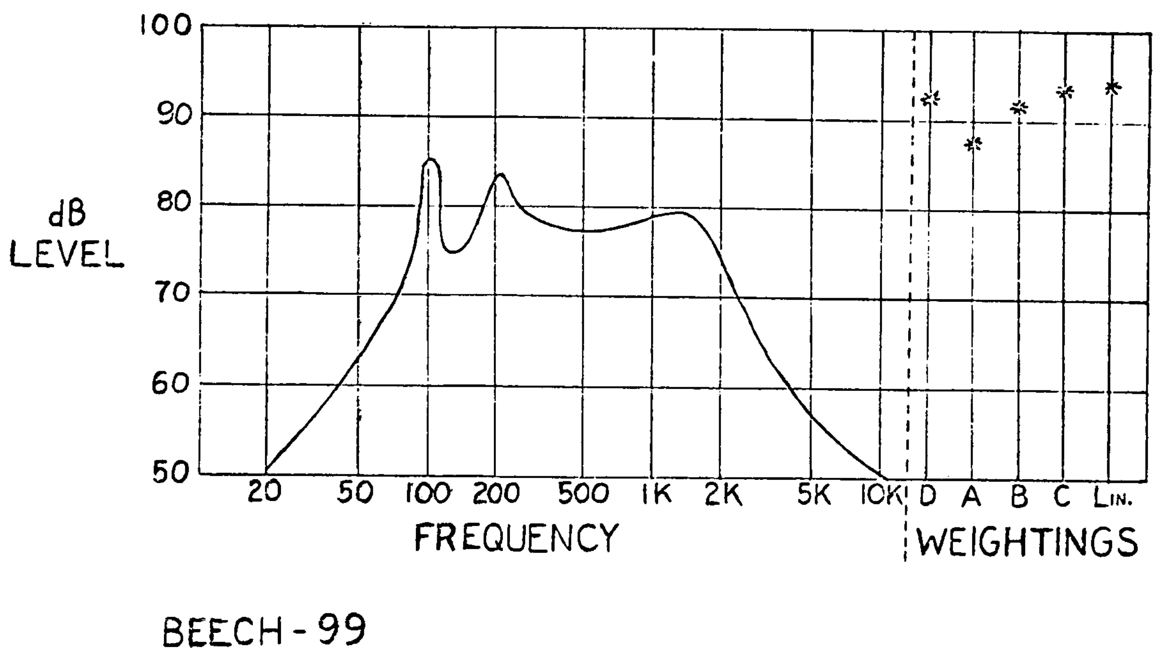

Figure 2.- Typical noise spectrum.

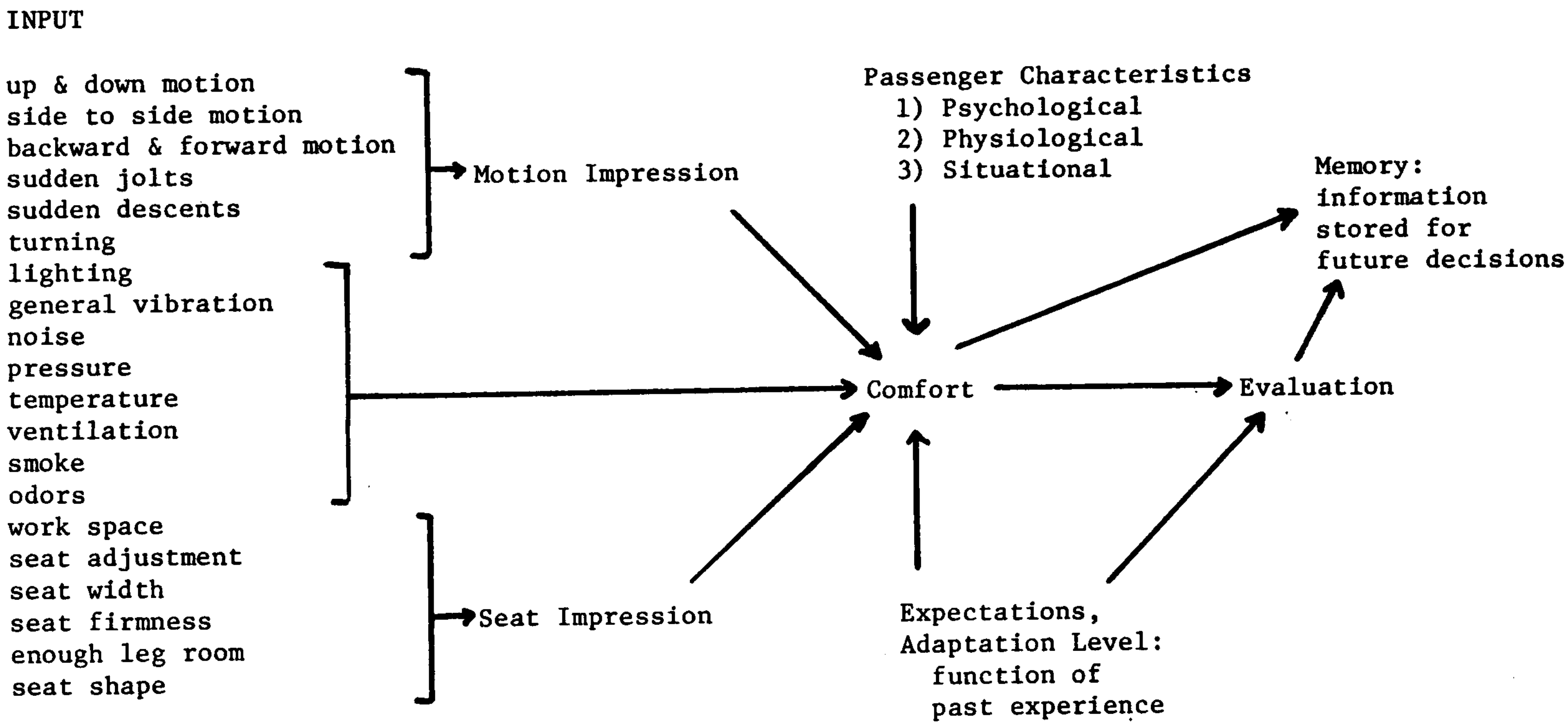

Figure 3.- Components of a theory of comfort.

ALLEGHENY COMMUTER
Operated by Atlantic City Airlines, Inc.

UNIVERSITY OF VIRGINIA

This questionnaire is part of an effort by Atlantic City Airlines, the National Aeronautics and Space Administration, and the University of Virginia to obtain from you, the flying public, information to be used in the improvement of transportation systems. The goal of the program is to identify the needs and desires of airline passengers, so that future systems may increase passenger satisfaction.

Your cooperation in completing this form will be most appreciated and can only be of benefit to you, the air traveler. Thank you, and enjoy your flight.

Maurice C. Young

Maurice C. Young
President, Atlantic City Airlines, Inc.

Please indicate only your first impression on each question. You need not answer any question that offends you.

1. Age ______ 2. Sex: ☐ M ☐ F

3. Education:
- ☐ High School not completed
- ☐ High School completed
- ☐ College

4. Occupation:
- ☐ Housewife
- ☐ Craftsman, Mechanic
- ☐ Professional, technical
- ☐ Professional, nontechnical
- ☐ Student
- ☐ Armed Forces
- ☐ Secretary, Clerk
- ☐ Salesman
- ☐ Manager, Official, Executive
- ☐ Other ____________

5. Industry of Employment ____________

6. Approximate Household Income (before taxes):
- ☐ Under $5,000
- ☐ $ 5,000-$ 9,999
- ☐ $10,000-$14,999
- ☐ $15,000-$19,999
- ☐ $20,000-$24,999
- ☐ $25,000-$29,999
- ☐ $30,000-$34,999
- ☐ $35,000 or more

7. What is the primary purpose of this trip?
☐ Business ☐ Personal ☐ Other

8. How do you feel about flying?
- ☐ I love flying
- ☐ I have no strong feelings about flying
- ☐ I dislike flying
- ☐ I fly because I have to

9. Approximately how many times have you flown in the past two years?
- ☐ None, this is my first flight
- ☐ 1-3
- ☐ 4-6
- ☐ 7-9
- ☐ 10 or more

10. How important is each of the following items in determining your feelings of comfort? Rank them using the numbers from 1 to 9, with 1 representing the **most important**, and 9 the least important. Please use each number only once.
- ______Pressure changes (ears pop)
- ______Noise
- ______Temperature
- ______Lighting
- ______Seat comfort
- ______Up and down motion (bouncing)
- ______Side to side motion (rolling)
- ______Work space and facilities
- ______Presence of smoke
- Other ____________

11. Place a check in the box which describes the importance of each of the following in determining your satisfaction with an airplane ride.

	Unimportant	Very Little Importance	Somewhat Important	Very Important	Greatest Importance
Comfort	☐	☐	☐	☐	☐
Convenience	☐	☐	☐	☐	☐
Cost	☐	☐	☐	☐	☐
Reliability	☐	☐	☐	☐	☐
Safety	☐	☐	☐	☐	☐
Time Savings	☐	☐	☐	☐	☐
Ability to Read	☐	☐	☐	☐	☐
Ability to Write	☐	☐	☐	☐	☐
Services on Board	☐	☐	☐	☐	☐
Surroundings	☐	☐	☐	☐	☐

12. Consider the motion you are experiencing. Indicate your reaction to this motion by checking the appropriate box:
- ☐ Very Comfortable
- ☐ Comfortable
- ☐ Neutral
- ☐ Uncomfortable
- ☐ Very Uncomfortable

(Please see last page)

13. How difficult does the motion of this flight make the following activities?

	Not at all Difficult	Slightly Difficult	Difficult	Extremely Difficult	Impossible
Concentration	☐	☐	☐	☐	☐
Reading	☐	☐	☐	☐	☐
Writing	☐	☐	☐	☐	☐
Sleeping	☐	☐	☐	☐	☐

14. After experiencing the motion of this flight, I would: (Check only one)
- ☐ be eager to take another flight
- ☐ take another flight (without any doubts)
- ☐ take another flight (but with some doubts)
- ☐ prefer not to take another flight
- ☐ not take another flight

15. Suppose a high-frequency shuttle service (8 or more round trips per day) were available at your local airport, scheduled to connect with flights of over 300 miles from a larger airport some distance away. Would you use the shuttle instead of ground transportation to the larger airport, if the cost were competitive?
☐ Yes ☐ No

16. Suppose a 25-passenger prop jet flew from an airport 15 minutes from your home or office to cities within 300 miles. Would you use this service rather than travel to a major airport an hour away?
☐ Yes ☐ No

THANK YOU FOR YOUR ASSISTANCE

Figure 4.- Sample of an in-flight questionnaire.

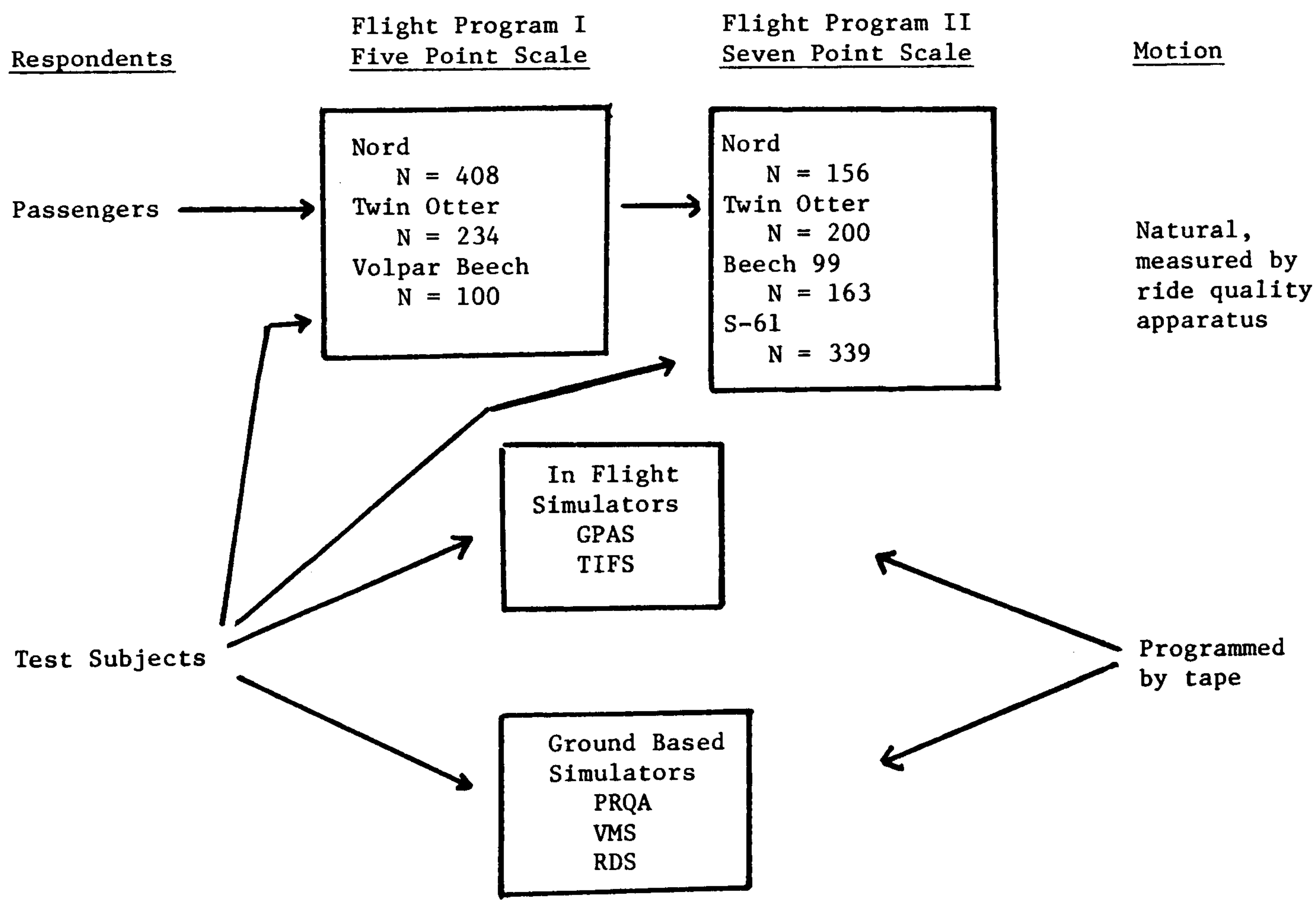

Figure 5.- Overview of the Univ. of Va. data base.

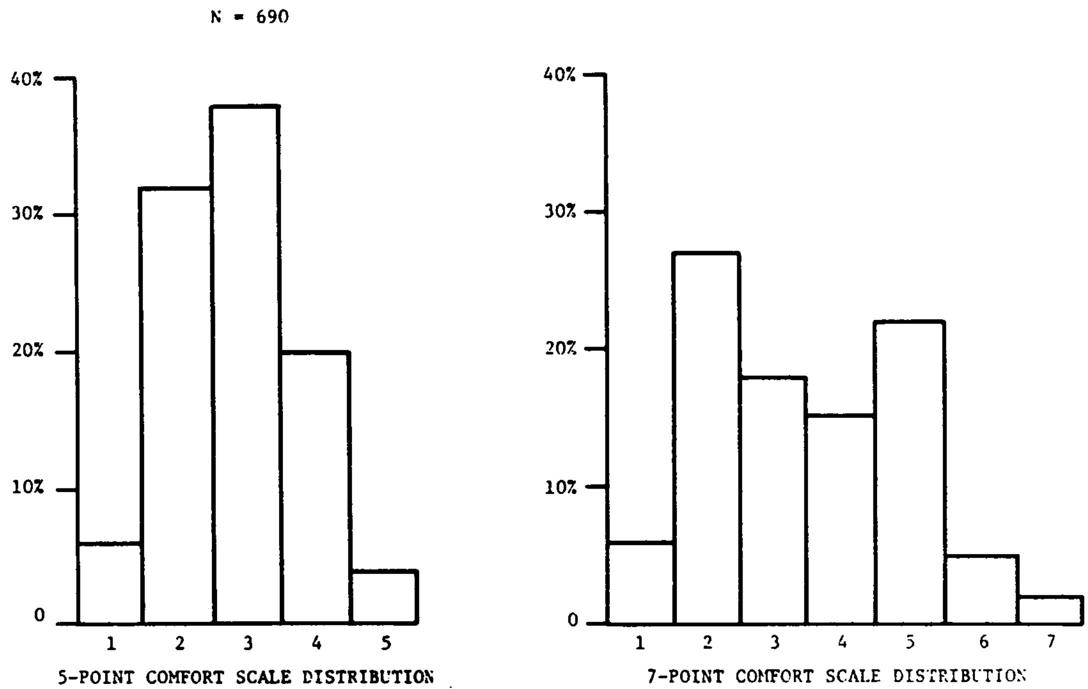

Figure 6.- Distribution of comfort responses.

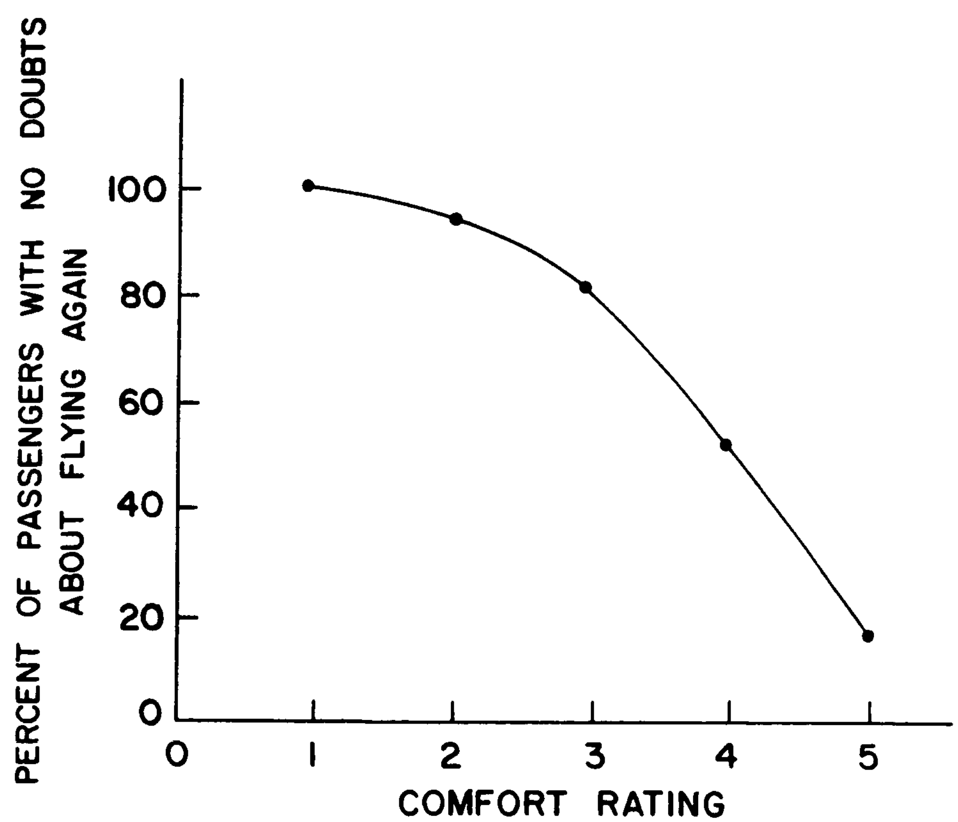

Figure 7.- Percent of passengers satisfied as a function of comfort level for the first flight program.

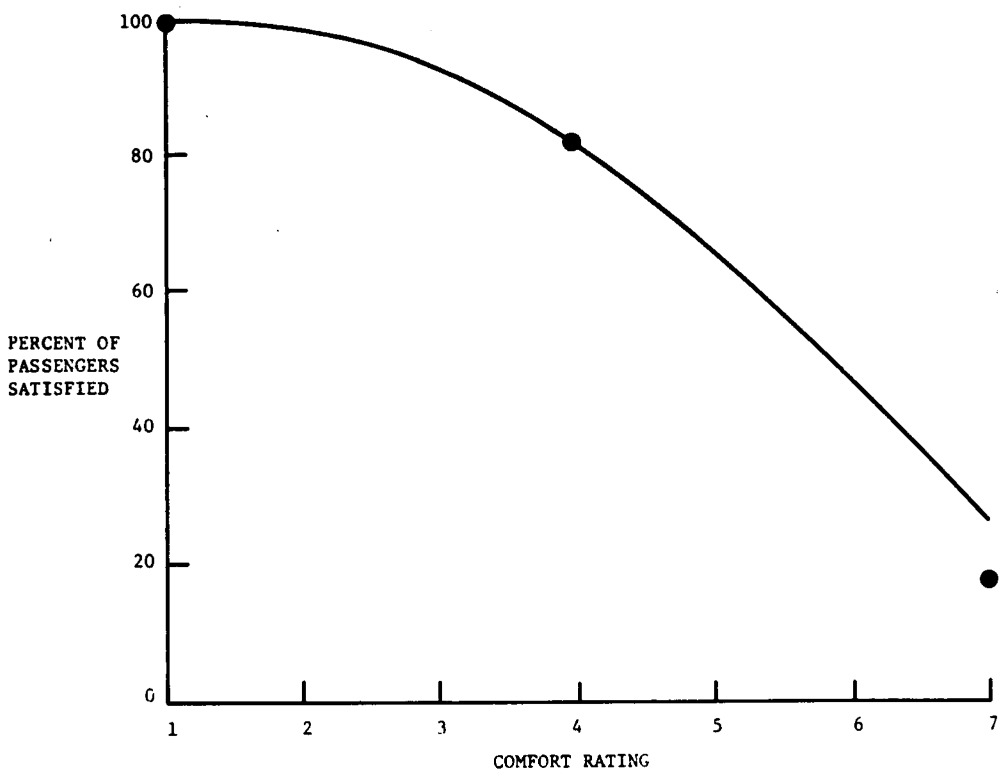

Figure 8.- Percent of passengers satisfied as a function of comfort level for the second flight program. (The dots represent results from the first flight program.)

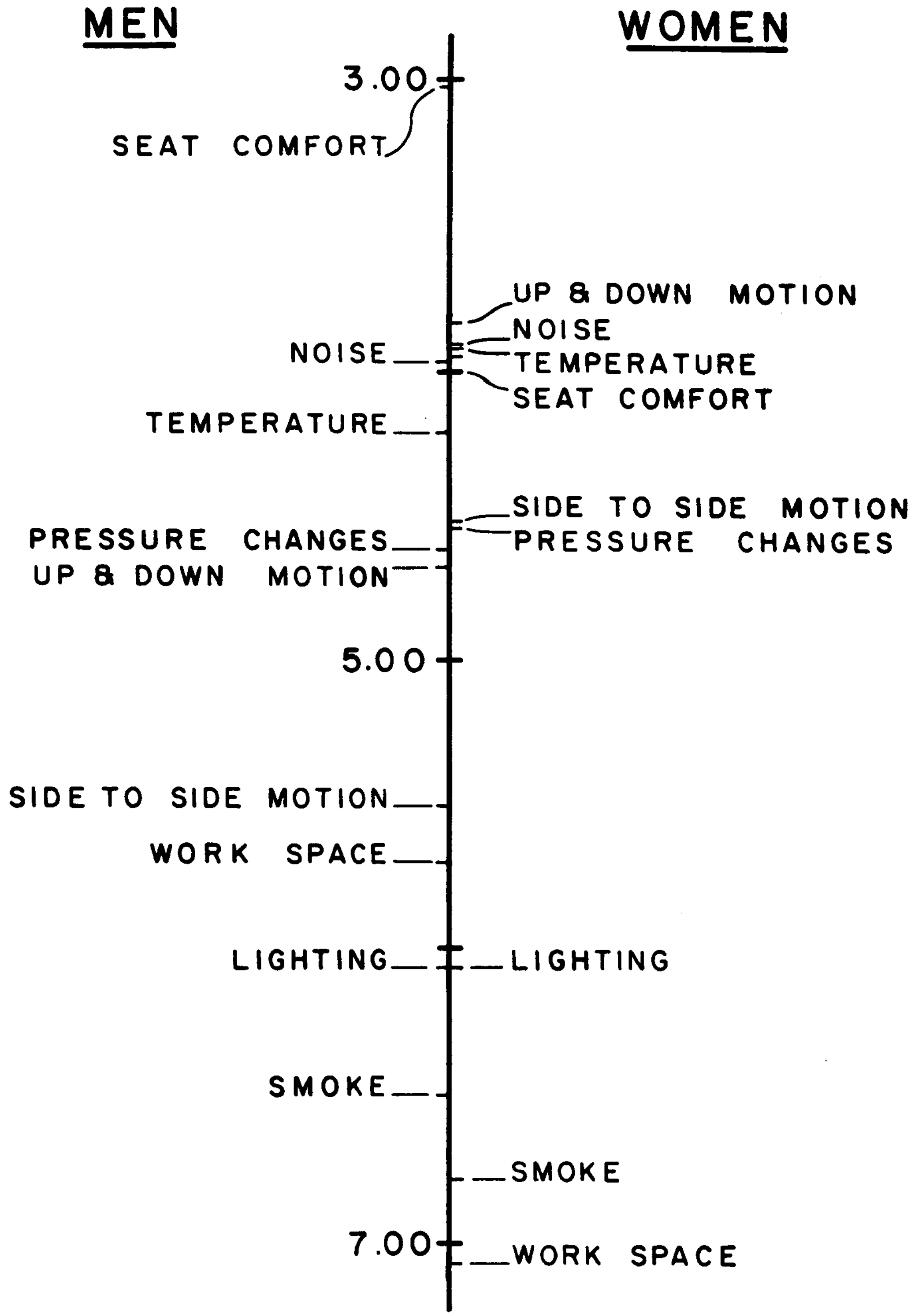

Figure 9.- Mean rankings of physical factors in comfort according to sex. (Low numbers indicate greater importance.)

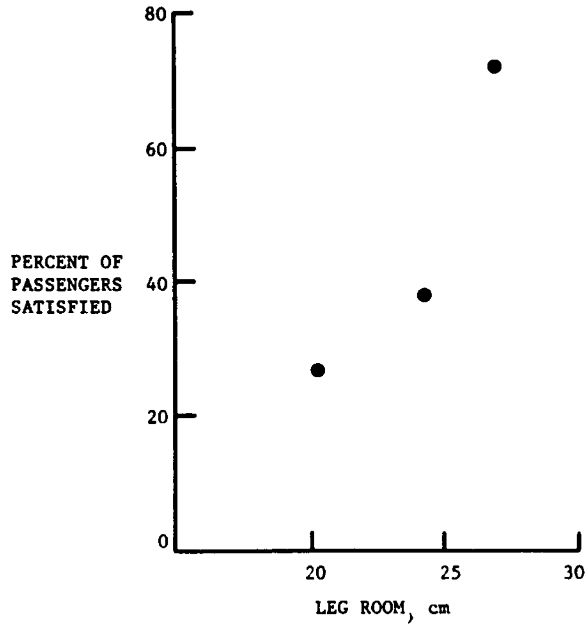

Figure 10.- Percent of passengers satisfied as a function of leg room.

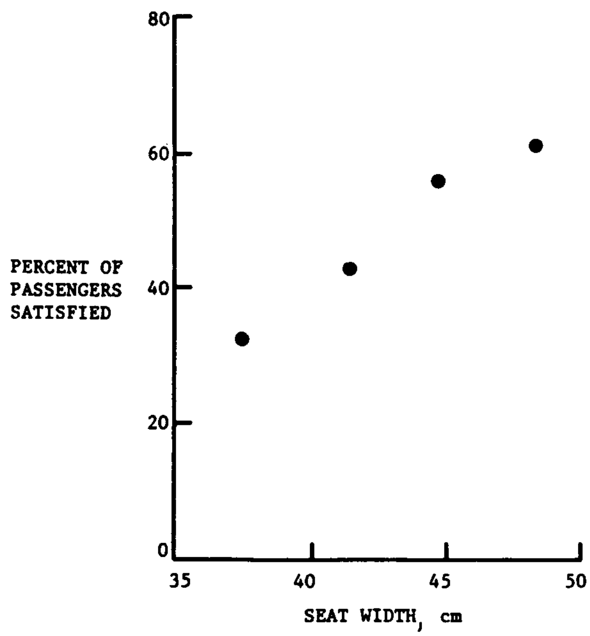

Figure 11.- Percent of passengers satisfied as a function of seat width.

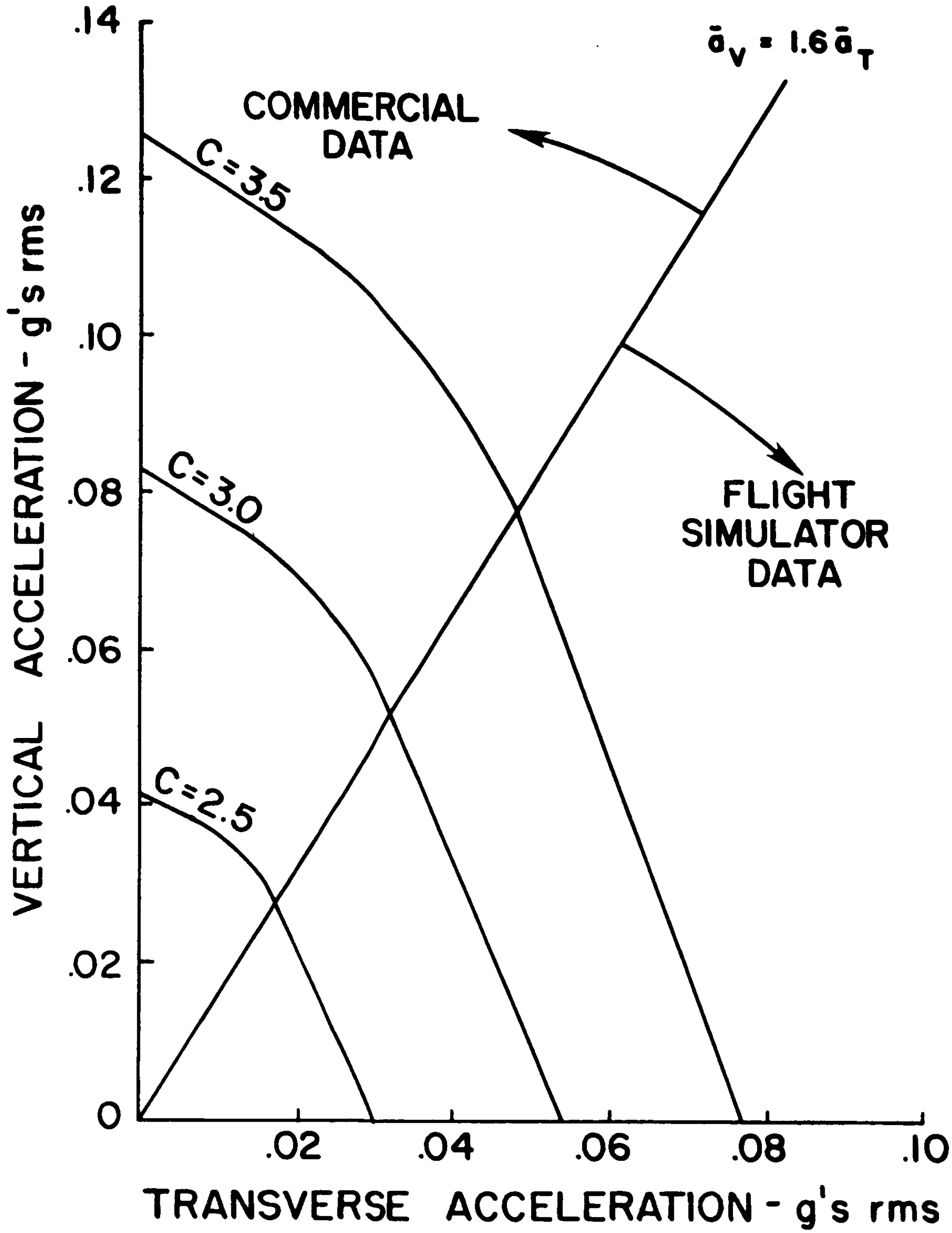

Figure 12.- Iso-contours for comfort responses based on 5-point scales.

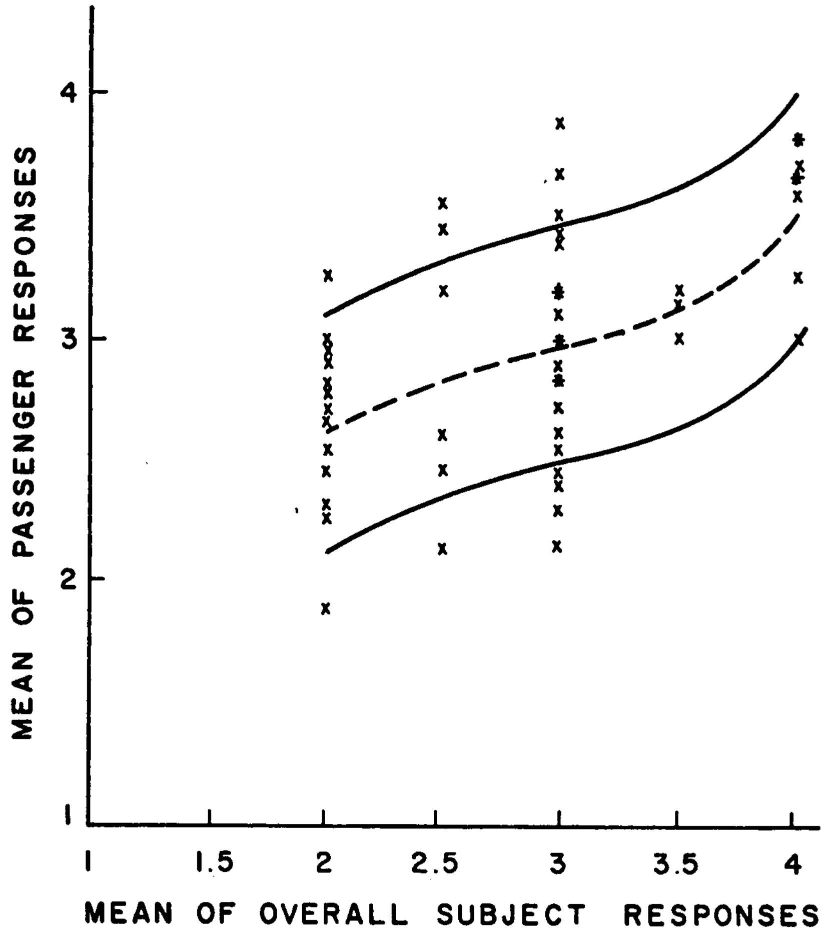

Figure 13.- Plot of mean passenger responses against mean subject responses.

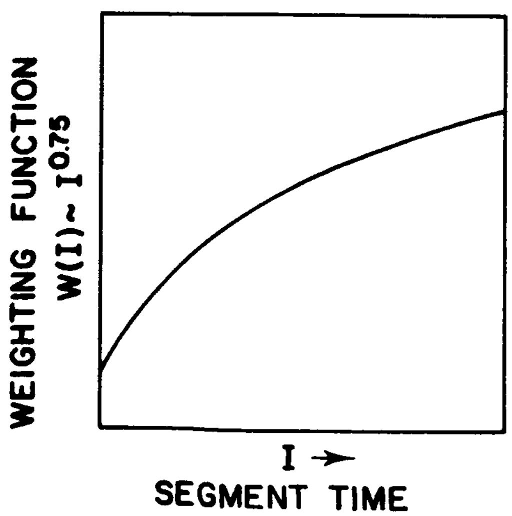

Figure 14.- Weighting function for integrating test subject responses.

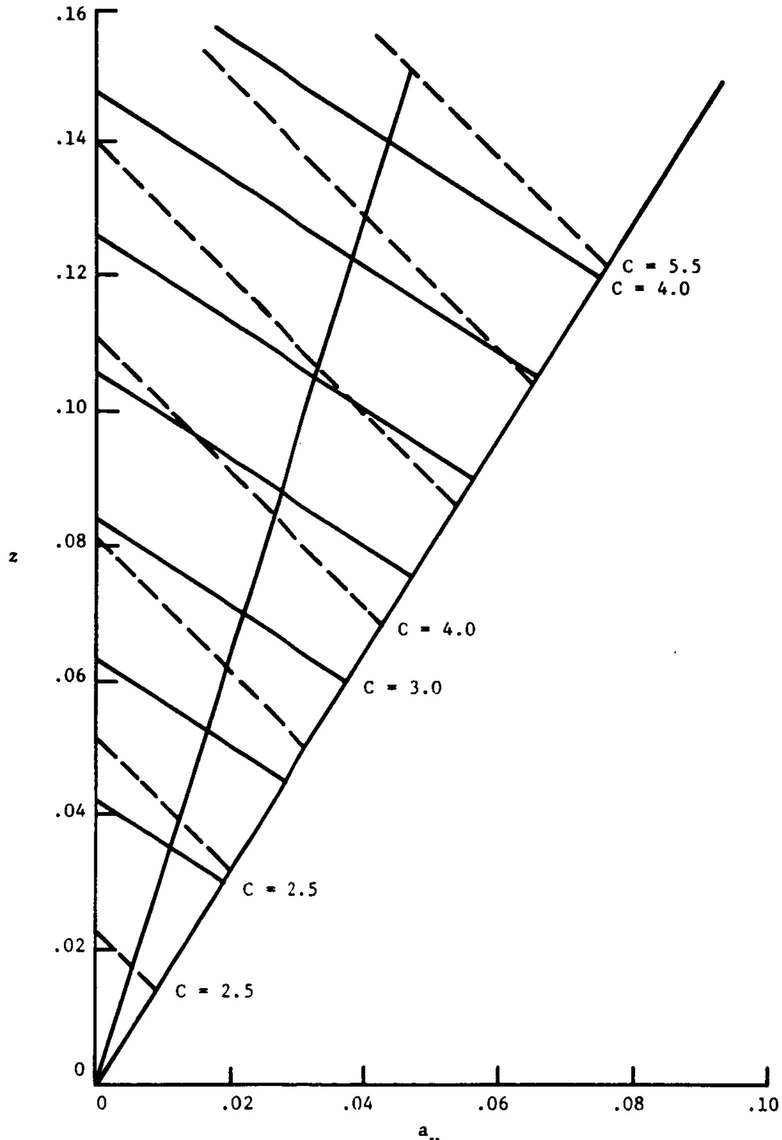

Figure 15.- Iso-contours from both comfort models. (Dotted lines are 7-point scale; solid are 5-point scale.)

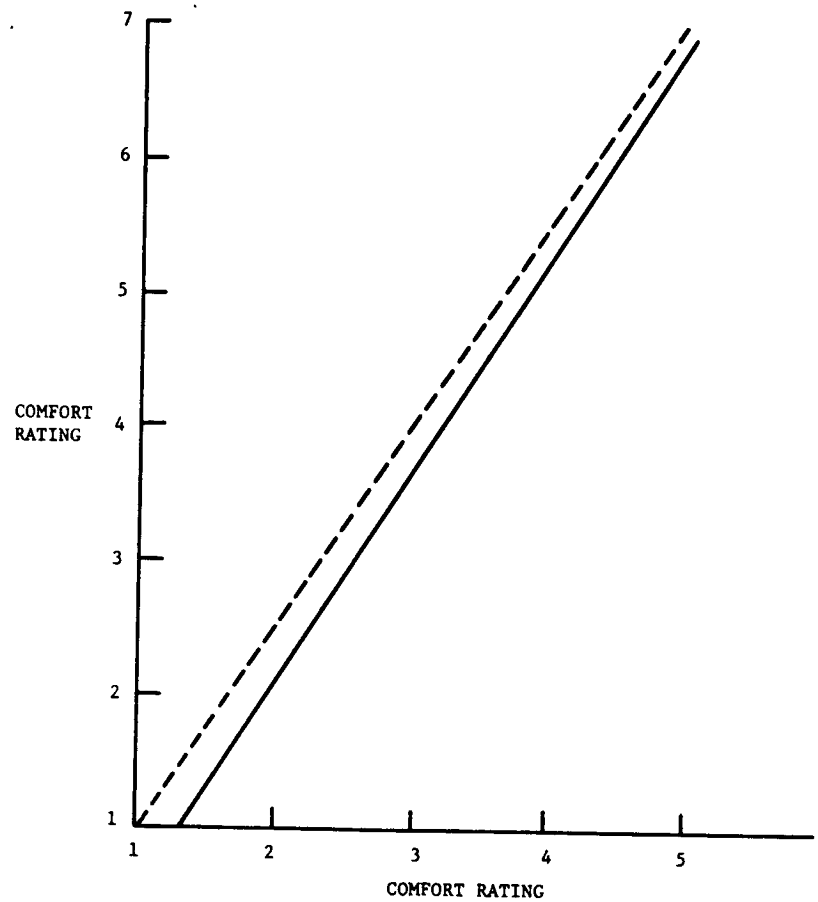

Figure 16.- Relation between comfort responses predicted from the two models (7 point and 5 point).

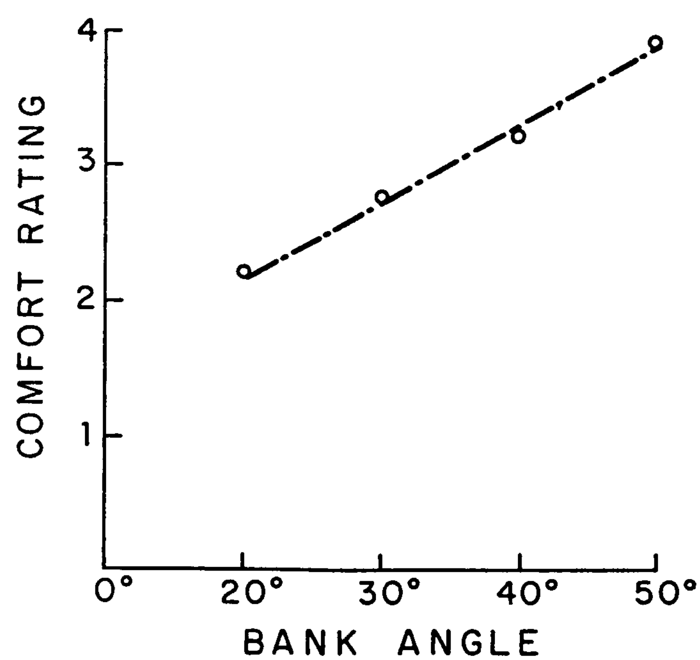

Figure 17.- Passenger responses to bank angles.

N76-16773

REACTION OF PASSENGERS TO PUBLIC SERVICE VEHICLE RIDE

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SUMMARY

The paper describes a series of questionnaire studies carried out on passengers in public service vehicles in the United Kingdom particularly cross-channel hovercraft, helicopter and train. It examines the effectiveness of the different rating techniques employed and demonstrates that useful and reliable information can be obtained on the effects of such physical parameters as vibration, vehicle motion and noise using rating methods which involve no external standards. It also presents some results obtained from analysis of the survey returns.

INTRODUCTION

In recent years problems caused by severe traffic congestion on many of the major road routes and in the hearts of most cities, assisted recently by the energy crisis, have given rise to pressures to increase the usage of public transport vehicles by travellers to whom alternative private transport is available. In addition, the developed social consciences of many legislators are insisting that even those to whom no alternatives are available are entitled to more enlightened treatment than they frequently receive at present.

Accordingly, pressure is on both designers and operators to ensure that new vehicles and new modes which use old vehicle designs in new ways shall provide travel which is cost effective, reliable, attains high block speeds and is comfortable.

The first three factors can be argued out and settled largely by designers and operators on the basis of existing information. The necessary data can be obtained without involving passengers in such systems directly, and effective decisions can often be made very early in the design process. The question of what constitutes a comfortable ride is, however, more difficult to settle and frequently involves the use of test subjects in prototype vehicles at a stage when major design changes are difficult and costly.

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The term "comfort" implies that some state of well-being exists within a person and it is this state of well-being which needs to be investigated. Such a subjective condition is generated by the combined effect of the many physical and psychological factors acting on the person, as well as by the physiological state of the man himself.

Generally speaking, the physical factors present in a transport environme fall into fairly well-defined groups. The psychological and physiological variables of the individual are, however, far more numerous and less definite. These may range from the passenger's attitude towards the particular vehicle and form of transport to his state of mind and state of health at the time. One of the notable characteristics of the psychological and physiological variables of individual passengers is the large variation which is possible within even a small group of travellers.

It is not surprising, therefore, that most of the previous work on the comfort of passengers in transport vehicles has been geared to discovering how the "passengers" react to the physical parameters of the environment. In the main such inquiries have been conducted in laboratory conditions, in an environment entirely divorced from the transport situation. There are only a few studies which were reported before about 1970 which referred to passengers in actual vehicles. Even now most investigators work almost entirely in laboratories.

It was with the aim of obtaining useful information from transport users themselves that a program of work was started at the University College of Swansea with the financial assistance of the Science Research Council of Great Britain, which included the use of questionnaire surveys carried out on different types of transport systems.

The basic objectives of the surveys designed and carried out by the Department of Mechanical Engineering and the Department of Psychology jointly are:

(1) Development of questionnaire approach.
(2) Identification of descriptors.
(3) Evaluation of semantic and numerical rating techniques.
(4) Correlation of ratings with measured motions.
(5) Determination of effects of sex, age and journey time on important environmental factors.

The paper discusses the items roughly in the order in which they occur in this list, although there are so many cross links that they will be exploited where possible, hopefully to clarify the approach and the results. The paper also draws fully on the information presented in the earlier paper in the symposium by the same authors (ref. 1).

STUDIES CARRIED OUT

During the period 1969-1973 a series of studies was performed on a variety of vehicle types. The principal surveys carried out are listed in table I. The first hovercraft survey using the SRN6 was a preliminary attempt carried out on a route between Southampton and Cowes (Isle of Wight). Unfortunately, the route carried more commuter traffic than had been expected so that after a few days all except one or two passengers per trip had been questioned previously. An attempt was made immediately after to sample a medium distance bus route between Swansea and Cardiff, but this was abandoned at an early stage because the buses were either so full as to make it impossible for passengers to complete the questionnaires or so empty as to make the returns per trip completely uneconomic.

The three SRN4 surveys were carried out on the cross-channel Dover to Boulogne/Calais route and formed a continuing programme of improvement, made possible by the extremely cooperative attitude of the staff of Seaspeed. Indeed, the cooperation received from all operators approached was very good. The helicopter survey was carried out on the British Airways (then British European Airways) route between Penzance and the Scilly Isles, and the train survey was done between Newport and Reading on the Swansea to Paddington (London) British Rail Intercity Service. Further questionnaires are hopefully planned for both helicopter and train but await the acquisition of financial support and final approval by the operators before they are carried out.

It will be noted that the time taken to analyze the questionnaires, which to some extent governed the interval between surveys, increased as time progressed. This was due to the increasing complexity of questionnaires which was made possible by the highly cooperative attitude of most of the passengers and by a gradually clearer questionnaire format.

On surveys V and VI recordings of the vehicle motion were made. These recordings were obtained by multiplexing six channels of acceleration information onto a UHER 4400 battery tape recorder via an encoding package, specially built by DYNATEL (also battery operated), which also provided the necessary signal conditioning for the half-bridge piezoresistive ENDEVCO accelerometers. The accelerometers were mounted in three boxes, one providing signals for the vertical, lateral and fore and aft directions, one for the vertical and lateral directions and one for the vertical direction only. These were mounted on the floor of the vehicle in suitable positions to give a reasonable indication of the overall vehicle movements, at any point in the vehicle, in three mutually perpendicular directions. In all cases the recordings were made during the whole of the period for which passengers were actually completing questionnaires. For survey V (train) recordings were taken for about 20 minutes in each of a succession of coaches, whilst for survey VI (hovercraft) this was done for the whole of the hovercraft flight.

QUESTIONNAIRE DESIGN AND RESPONSE RATES

Objectives

The basic objective of the questionnaire studies was to obtain quantitati subjective reactions of passengers to the motion and vibration present in movi vehicles in a form which could be correlated with objective measurements of th vehicle motion and vibration. A preliminary aim had to be, however, to develo a questionnaire format which would enable reasonably precise, repeatable, numerical information to be obtained from untrained fare-paying passengers about their reaction to environmental factors, particularly to factors which were not those about which they habitually thought or made comments.

As the surveys progressed it became increasingly apparent that passengers found great difficulty in extracting and considering just one or two physical parameters (particularly motion and vibration) from all the others present. In addition, the usefulness of any information obtained is diminished if other information concerning aspects of the passenger's reaction to the journey is not obtained at the same time.

As a result the questionnaire was enlarged both in its scope and its aims to include as many as possible of the physical and psychological factors though to be important in determining passenger comfort. Following this enlargement the problems of analysis and interpretation increased considerably. It should be emphasised, however, that the surveys carried out, some of which are to be discussed in some detail in this paper, were not pieces of unrelated work but were part of an on-going sequence in which the successes and failures of one were used to improve the design and operation of the next.

Layout

Figures 1 to 3 indicate how the design and layout of the questionnaires changed during the surveys. The questionnaires designed for the first two surveys were printed in horizontal format on small card since it was thought that card would provide more support for passengers to write on than would larger sheets of paper (fig. 1). The next survey, and half of the fourth were printed in larger type, but still using the horizontal format (fig. 2). A new vertical format was tried for the other half of survey IV and in slightly modified form for the final two surveys (fig. 3). The final column of table I shows the percentages of the questionnaires accepted by passengers which were returned fully completed. This demonstrates the improvement in overall returns obtained as the questionnaire design and the approach of the interviewers improved.

Use of Free Response Questions

During preliminary studies it very soon became apparent that non-technical and untrained people did not always understand very clearly what was intended when the words "vibration" and "motion" (defined to exclude the forward motion of the vehicle) were used. It was also found that problems arose in asking passengers to provide ratings of the intensity of vibration since there was no readily understood term which could describe this.

Accordingly, in addition to trying out different methods of obtaining subjective ratings of the environmental factors of interest, a considerable amount of effort was put into finding the words which could hopefully be used to describe vibration or motion intensity (as analogues of "loudness" for noise or "brightness" for lighting). In the course of this work it was found that many passengers described vibration and motion intensity in terms of situational phrases. Attempts were made to determine which phrases could be used realistically to describe end points on a scale of subjective intensity of vibration.

These efforts involved the use of unstructured questions (for example, the latter part of question 4 in fig. 1). Some of the changes in the early surveys were made in attempts to improve the response of passengers to these open-ended questions. The relative success of these changes can be inferred from the information given in table II.

During surveys IV, V, and VI the last page of the questionnaire was left blank with an invitation to the passenger to make whatever remarks he or she wished. Responses obtained referred to the whole range of services associated with the mode of transport being surveyed as well as providing comments on environmental factors within the vehicles and comments on the questionnaires. A great deal of useful information was gleaned from these remarks.

The next two sections of the paper describe the results of the attempts to obtain simple word descriptors and situational phrases and indicate the rating methods used to obtain passenger reactions. Inevitably there is a certain amount of cross linking between them.

IDENTIFICATION OF VIBRATION AND MOTION DESCRIPTORS

The questionnaires used in the early surveys contained either open-ended questions in which passengers were asked to record descriptions of the vibration felt, preferably using single words or very short phrases, or more structured questions to obtain words which could be used to describe the subjective intensity of vibration in the same way as the words loudness and brightness are used in connection with noise and lighting respectively.

Analysis of the early responses came up with very few words or phrases which could be usefully reduced to single-word descriptors, the majority of the responses being phrases which related the vibration or motion of the vehicle under investigation to another situation. Car and aircraft ride appeared ve frequently in these situational comparisons, particularly a bad flight to represent an extreme in the vehicle under survey.

Table III(a) shows an analysis of the 43 single-word responses, from a total of 295 questionnaires in survey I, obtained to a question asking for single words to represent the "least" and the "most" end of a vibration rati scale. It is readily seen that all of them have connotations other than simple vibration response - pleasantness, comfort, peace, and so on. Some o these words, and others gleaned from the comments supplied by passengers on "graffiti page" already referred to, were provided as a list on later survey (for example, survey III, question 14 in fig. 2) and passengers were asked t select the best description of their feelings. The results of this are show in table III(b) and are shown to be even more inconclusive since the words chosen to have a high priority really appear to relate to the quality of the vibration rather than its intensity.

The investigators were left to themselves to make a choice from inconcl data. After some laboratory studies (ref. 2) they decided to use the concept of smooth to rough to form the ends of a rating line in a later survey (surve VI), primarily on the grounds that roughness was a concept less tied (in thei opinion) to pecularities of the ride motions of particular vehicles than the other words selected. Additionally, from analysis of the free style respons produced by many passengers, some situational experiences, which were thought to be readily understandable by a majority of people questioned, were drafte for use as end points on rating scales. The use of these is described in som detail in the latter part of the next section. The situational scale ends devised for and used in survey III, IV and V are shown in table IV.

EVALUATION OF RATING TECHNIQUES

Limitations of Techniques Available for Field Studies

As has been stated earlier, the primary objective of the questionnaire surveys was to obtain numerical estimates of the severity of the relevant environmental factors from passengers in actual service vehicles. These coul then be compared with objective measures of the physical parameters deemed to be those most relevant to the factors under consideration.

There are two fundamental limitations which are inevitably imposed on an quantitative scaling method under field conditions. The first is that the method used will probably be of the pencil and paper variety. Theoretically it would be possible to use certain psychophysical techniques such as cross-modality matching of riding vibration by the use of noise signals. However, there are usually practical difficulties involved in using such techniques, either difficulties of application or difficulties of calibration and interpr tation.

The second difficulty is that during the course of the survey the stimul within the passengers' environment are generally at one predominant level wit only relatively short term excursions from that level. Train noise, for example, is generally of about the same order except when the train is crossing points or moving through a tunnel. Even for vehicle types such as the cross-channel hovercraft, major changes in weather do not usually occur within the space of a week or so, and one to two weeks was the time alloted for most of the surveys on financial grounds.

Effect of Scale Ends On Line Rating and Magnitude Estimation

There are three methods which can readily be used to obtain numerical ratings of subjective reaction to environmental parameters. These are listed briefly in table V, and typical forms of questions are shown in figure 4.

The methods categorized as "line rating" and "magnitude estimation" are obviously going to be severely affected by the choice of ends for the scales they are supposed to rate. Tentatively the authors have chosen to divide the scale ends into groups:

(1) Aesthetic (for want of a better word)
(2) Perception
(3) Tolerance
(4) Physical.

The Aesthetic group includes all pairs of scale ends which relate to subjective reactions which do not tend to make the passenger think specifically of one end of the scale or the other but are likely to attract reactions over the whole range. Perception and Tolerance groups tend to bias thoughts to one end or the other of a subjective scale and may also tend to include ideas related to the physical or physiological effects. The Physical group, as its name implies, refers pretty clearly to physical attributes of the environment without really asking for a relationship with a subjective feeling.

Figures 5 and 6 show the effects on ratings of a particular environment. (Each bar indicates the median and interquartile range of ratings for each scale end). The group classed as Aesthetic are centered with medians close to the rating of 5. The perception line shows a significantly higher rating, implying that the passengers were thinking about whether or not they could perceive vibration at all, whilst the tolerance line shows a significantly low rating with the implication that the passengers were considering whether or not they were being subjected to extreme physical effects.

The other factor which can affect the rating of environmental effects by the "line rating" method is the type of line used, particularly the way in which the line is divided into sections and whether or not the sections are labelled. A series of experiments was performed which convinced the authors that the differences in ratings caused by differences in line types were of negligible importance. These experiments have been fully reported in Oborne and Clarke (ref. 2).

Situational Scale Ends

Figures 7 and 8 show comparisons between ratings made along a 10 cm ratin line and those made by ascribing a number to the stimulus of the same vehicle using the same scale ends. Ratings were generally made at the same time by different people using parallel forms of questionnaires, the numbers being carefully matched so that equal numbers of each form were distributed on each journey. It can be seen that the relationship between the line and magnitude estimation ratings is very good for noise (fig. 7), but not so good for vibration (fig. 8).

It was the authors' intention in selecting the scale ends to try to find situations which could be clearly understood and accepted by as many people as possible. The hope was that they could also be used as a physical scale (with in reason) by using averages of physical measurements appropriate to the situations as the scale ends. Thus, standing next to a heavy lorry going uphi would usually result in a noise level of about 90 to 95 dB(A). Hence, there could also be hidden in the use of situational scale ends a method of providin passengers with a pseudocalibration on a physically recoverable basis. This has not been investigated yet in view of the fact that the scale ends of interest in riding investigations need to be refined to get better agreement between answers obtained by different rating methods.

Graphic Rating

The third type of rating referred to in table V is the graphic rating in which guiding phrases are placed along the line. In the earlier surveys considerable attention was given to the possibility of using such a rating technique in a similar way to Shackel and others (ref. 3) who had used it for the study of seat comfort. However, some testing, which is fully reported in Oborne and Clarke (ref. 4), convinced the investigators that it was not a particularly good method because of possible confusion as to the meanings of the steps on the scale.

Figure 9 shows a five-point comfort scale which has been used, both in a defined and in an undefined form, in both laboratory studies and field studies The laboratory studies are discussed in Clarke and Oborne (ref. 1) and in more detail in Oborne and Clarke (ref. 5). The relevant point to be raised here is that the laboratory studies showed that the scatter between individual respons of subjects to vertical sinusoidal excitation could be reduced by providing definitions of the points on the five-point rating scale.

CORRELATION OF RATINGS WITH MEASURED MOTIONS

Survey VI (see table I) was conducted during September 1973 on the Seasp route operated between Dover (England) and Boulogne (France) using SRN4 hovercraft. During the course of this survey recordings were made of six components of acceleration at the following three places on the floor of the passenger spaces:

(1) Rear port side cabin; vertical, lateral and fore-and-aft acceleration
(2) Front port side cabin; vertical and lateral acceleration
(3) Front starboard side cabin; vertical acceleration

The positions of the accelerometer boxes were selected so as to enable a reasonable estimate of vibrations in three directions experienced by small groups of passengers to be made. All that has been done so far has been to assume the levels of vertical acceleration to be roughly constant over the rear of the rear cabin and over the front cabin, with a very simple assessment of root-mean-square vertical acceleration being made for the time segments of each journey in which the majority of passengers completed questionnaires. Programs to enable more sophisticated analysis of the tape recordings to be carried out on a PDP11/10 computer with Micro Consultants A/D convertor, which has been recently bought by the Mechanical Engineering Department, are still being prepared.

Survey VI combined questions asking for ratings of overall comfort, motion comfort (motion being defined in the questionnaire as "motion of the craft due to the waves") and vibration comfort, using the five-point scale shown in figure 9. It also contained questions asking for ratings of overall comfort on a 10 cm rating line with scale ends "Very Comfortable" and "Very Uncomfortable" and for ratings of motion comfort and vibration comfort on a similar line with scale ends "Smooth" and "Rough".

Relationship Between Category Ratings and Line Ratings

The first exercise was to relate the category ratings to appropriate line ratings. For example, the mean and standard deviation were calculated for the ratings on the comfort line of all passengers who checked the overall comfort of the vehicle as "Just Comfortable". The values of the mean plus or minus the standard deviation were taken as being rough boundaries of the "Just Comfortable" region in the rating line. This was repeated for the other four categories of overall comfort. The line was then sectioned by taking the boundaries thus obtained and halving the overlaps and underlaps of the rough ranges. The final result is shown in figure 10. Also shown are the results for the vibration comfort and the motion comfort rating lines.

From these results figure 11 is produced by matching the boundaries of the categories for overall comfort against the same categories for motion comfort and vibration comfort. The curves indicate, for example, that someone rating overall comfort at 6 is likely to have rated motion comfort at about 5.6 and vibration comfort at 6.5.

Relationship Between Motion/Vibration Ratings and Vibration Measurements

The next stage is to obtain some relationship between mean motion rating and measured accèlerations. As a first attempt it was assumed that the passengers were identifying the motion due to the waves and the vibration as separate effects and were able to rate the two effects separately without trouble. Some uncalibrated power spectra have been obtained for the hovercra ride recordings. Figure 12 gives the general pattern of these, indicating a high value of spectral density in the lower frequency range, between 0 and 3 Hz say and a high narrow peak, generally occurring between 10 and 12 Hz. Accordingly it was tentatively decided to identify the motion effect with low frequency effects and for convenience to filter the recordings in the frequen range 0 to 4 Hz. The vibration was tentatively identified with the peak at about 12 Hz on the grounds that for frequencies up to about 50 Hz this peak generally stood out well above the noise floor, and for convenience the reco were filtered to pass the octave band 8 to 16 Hz.

Segments of the tapes were identified which covered time intervals during which a sufficient number of ratings were made. For each of these segments the mean motion and mean vibration rating were obtained and compared with the root-mean-square acceleration of the record filtered between 0 and 4 Hz and between 8 and 16 Hz respectively. The results are the regression lines shown in figures 13 and 14 respectively. It can be seer that the straight line relationship is the best simple fit that could be achieved between either of the pairs of variables and that despite the fact that the correlation coefficients achieved are not particularly high there is a good indication that more sophisticated analysis of the recordings could be expected to achieve better fits.

Effect on Overall Comfort of Vibration in Different Frequency Ranges

The final stage is to make use of figure 11 and produce figure 15 which shows how the overall comfort rating varies with the root-mean-square accele: tion in the two bandwidths from 0 to 4 Hz and from 8 to 16 Hz. At this stage all that can really be obtained from figure 15 are some general deductions about the relative equivalences between motion effects and vibration effects.

As a check of sorts on the rather tortuous argument which has produced figure 15 an attempt was made to reconcile overall comfort ratings and vibration and motion ratings directly. Some results from the survey carried out on the train service from Newport to Reading were added for good measure. Figures 16 and 17 show the plots of mean values of overall comfort rating against mean values of vibration rating and mean values of motion rating respectively. It can be seen from figure 16 that the plots for vibration rating from train surveys and hovercraft surveys are similar, and that for all those plots a simple straight line regression is likely to be a good fit.

Figure 18 shows the regression line obtained from making such a fit. It is not suggested, incidentally, that the close match between train and hovercraft vibration lines is other than coincidence. However, examination of the points on figures 13 and 14 indicates that the root-mean-square acceleration values which actually occur in the hovercraft lie within the range 0.2 to 1.5 m/sec^2 for both frequency bands examined, and it seems reasonable to suggest that extremes of passenger ratings on the journeys sampled would coincide approximately with extremes of physical values.

On this basis the agreement between the hovercraft lines of figure 18 and the lines of figure 15 is quite good. The only obvious discrepancy is that the relative positions of the motion and vibration lines are interchanged between the two figures.

The fact that reasonable looking curves can be obtained from the sort of arguments which have been used in this section is encouraging. Agreement of a sort between two different uses of the data is fairly good and indicates that there is good reason to believe that passengers can be induced to provide information about vibration and motion effects. The pursuit of more elaborate techniques for analysis of the data is therefore worthwhile, and in due course, when equipment and programmes are working properly, this will be done.

EFFECTS OF AGE, SEX AND JOURNEY TIME ON IMPORTANT ENVIRONMENTAL FACTORS

This section will discuss some aspects of the passenger and his journey and their effect on the passenger's overall assessment of his journey comfort. The overall intention was to evaluate more clearly the importance to passengers of the various factors which make up the total comfort effect and how these change with time.

Relative Importance of Environmental Factors

In the train study it was decided to try to discover whether passengers were confusing two types of questions. The first type asks how important a particular environmental factor is to a passenger's feeling of comfort. The second type asks for a specific subjective rating of the level of that factor in a particular vehicle. There had been some doubt from reading passenger comments in previous studies as to whether passengers were actually providing ratings of intensity levels when asked to do so, or whether they were really indicating importance of a parameter relative to some undefined datum.

To do this, a separate single-sheet question set was issued to the passengers on the train in addition to the questionnaire. The relevant portion of the separate sheet is shown in figure 19, and the histograms of the responses are shown in figure 20. It can be seen that the results from the five environmental factors listed fall distinctly into two groups. A surprising result is that seat comfort is listed with temperature and ventila tion in view of the strong effect which it is thought seat comfort (or seat design at any rate) will have on reactions to motion. The interesting point is that passengers feel that seat comfort, ventilation and temperature are more important than vibration and noise as environmental factors, whilst at the same time they feel that suitable criteria have been set for the first three but not for the last two. The different shapes of the "importance" and "level" responses indicate quite clearly that passengers are able to rate the levels of environmental factors as a separate issue from the expression of feelings about relative importance of the same factors. The high response which indicated that train vibration and noise are too high can also be taken as adequate justification for continuing with investigations such as this.

Figure 21 shows information on the relative importance of different environmental factors which was gleaned from the comments made by passengers during the course of survey IV. Here four out of the first seven factors listed as having upset passengers in connection with a total service are vehicle environmental factors, with vibration and motion being near the head of the list.

Effect of Age of Passengers

The effect of age on comfort rating of the hovercraft is shown in figure 22 and 23. Figure 22 indicates a very slight decrease in sensitivity to overall ride at ages 50 and above (high comfort ratings mean less comfortable ride). Figure 23, on the other hand, indicates a greater age response to motion and to vibration. Both factors appear to show a general trend in which sensitivity decreases with increasing age. The overall effect is one showing a sharp decrease from a high number of objections at ages below 10 with the effect then flattening out for overall craft motion but continuing to decrease for vibration. The effects at extreme ages may be coloured by relatively low proportions of passengers in these age groups. Generally speaking, however, there is an indication that the very young find both the overall craft motion and the vibration unpleasant.

Effect of Sex of Passengers

The effects of sex on reactions were more difficult to establish overall since the population sampled was very unbalanced. Considerably more men than women were questioned over the whole range of surveys. However, there are indications that whereas men and women appear to have much the same reactions to vibration and motion, women tended to react more favourably to the overall comfort level provided than men.

Effect of Journey Time

Finally, some information about the effects of time of exposure can be deduced. On questionnaires distributed during surveys V and VI passengers were asked to write the time at which they completed a certain part of the task. By subtracting the journey start time from this some guidance as to length of exposure to that particular journey could be obtained. The exposure time varied from 0 to 20 minutes for the hovercraft trip, and from 0 to 150 minutes for the train ride. Correlation of the estimated trip duration with ratings of vibration effect, overall motion effect and with overall comfort indicated no discernable change for a period up to 150 minutes. This is in complete contradiction to the predictions in ISO 2361 (ref. 6) which indicates a change from 100 percent acceptance at 4 minutes to about 30 percent acceptance at 150 minutes, thus indicating, in the view of the authors, the falsity of the ISO time dependence predictions for reduced comfort at these levels of vibration.

There are, however, indicated changes over long journey times in the relative importance of different environmental factors on comfort. Figures 24 and 25 indicate cumulative plots of the quoted dominant factors for the hovercraft (up to 20 minutes) and for the train (up to 140 minutes). Allowing for variations due to small group sizes, there is no significant change over the indicated 16 to 17 minutes, but there is an indication that there may be an increase in the importance of seat comfort towards the end of the journey, with vibration and motion decreasing in importance accordingly. For the longer train journeys the marked change is the increase, as time goes on, of the rating of temperatures at the expense of noise and vibration.

CONCLUDING REMARKS

The work described set out to determine whether or not questionnaire studies of ordinary fare-paying passengers in public service vehicles could be used to provide repeatable and reliable information about individual environmental parameters.

The results have exceeded expectations. They show that, provided due care is taken in the design of the questionnaires, high response rates can be obtained. The use of a format in which the same question is asked in different ways, or the use of parallel forms in which different groups of people are asked the same question in different ways, enables cross checks on numerical ratings to be carried out in a way which enhances their value and meaning.

The surveys have also provided an appreciable amount of information about the effects of different physical, demographic and personal factors in ride comfort, much of this being in an understandable numerical form which can be used directly in further analyses.

Finally, the surveys have resulted in the collection of a large number of passenger comments on all aspects of the service provided and of the vehicle design. Much of this information is still waiting to be extracted and used.

ACKNOWLEDGEMENTS

Thanks are due to the Science Research Council of Great Britain for financial support, to Mrs. Susan Oborne and others for much hard work in gathering and preparing data for processing and to Professor C. E. M. Hansel and Professor F. T. Barwell of the University College of Swansea for advice and support.

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TABLE I. - SURVEYS CARRIED OUT BETWEEN SEPTEMBER 1969 AND OCTOBER 1973

Survey	Vehicle	Date	Number of questionnaires completed	Percent of questionnaires issued and returned completed
I	Hovercraft SRN6	Dec. 1969	295	74
II	Hovercraft SRN4	Apr. 1970	519	71
III	Helicopter S-61	Aug. 1970	483	81
[a]IV	Hovercraft SRN4	Feb. 1971	1066	78
V	Train	Feb. 1971	1602	97
VI	Hovercraft SRN4	Sept. 1973	691	80

[a]Survey IV was in two parts of nearly identical size. Survey IV(a) was a repeat, apart from one or two modifications to the wording, of survey III but applied to the SRN4 hovercraft, thus giving a questionnaire which had been applied to two different vehicles. Survey IV(b) was a new design run parallel with survey IV(a) to give comparisons between two layouts. Survey IV(b) led directly to surveys V and VI.

Table II. - RESPONSE RATE TO QUESTIONS ASKING FOR DESCRIPTION

Survey	Type of information required	Response rate, percent	Mean response rate to other questions, percent
I	Motion	50	77
	Provide scale ends	53	
	Scale ends	45	
II	Motion	45	91
	Provide scale ends	35	
III	Motion	27	93
	Noise	60	
	Fuselage vibration	16	
IV(a)	Motion	26	84
	Noise	49	
IV(b)	Motion	75	93
V	Vibration	94	98

Table III. - STUDY OF MOTION AND VIBRATION DESCRIPTORS

(a) Single Word Descriptors Offered as Vibration Scale Ends

[Forty-three passenger responses from survey I]

"Least" scale end		"Most" scale end	
Descriptor	Number of responses	Descriptor	Number of responses
Smooth	24	Rough	14
Pleasant	6	Bumpy	12
Comfortable	5	Uncomfortable	4
Gliding	2	Shake	4
Relaxing	2	Jarring	3
Peaceful	2	Nauseating	2
Enjoyable	1	Bounce	2
		Unpleasant	1
		Lurching	1

(b) Ordering of Descriptors by Passengers

Descriptor	Survey III, helicopter		Survey IV(a), hovercraft	
	Place	Percent of passengers	Place	Percent of passengers
Bumpy	5	4	1	28
Shaky	1	41	5	12
Bouncy	3	5	2	14
Judder	2	39	3	14
Jolty	4	4.5	4	14
Rough	6	3	6	7
Lurch	7	2	8	3
Plunge	8	1	7	7
Heave	9	0.5	9	11

TABLE IV. - SITUATIONAL PHRASES SELECTED AS SCALE ENDS

Scale	"Least" scale end	"Most" scale end
Noise	Sitting in a soundproof room	Standing next to a heavy lorry going uphill
Vibration	Complete rest	Travelling in an old car over an unmade road

TABLE V. - RATING TECHNIQUES

Rating method:
- Magnitude estimation
- Rating on line:
  - Unsectioned line
  - Sectioned line
- Graphic Rating

Scale ends:
- Descriptors:
  - Aesthetic type
  - Perception type
  - Tolerance type
  - Physical type
- Situational phrases

3. Are you aware of any sideways or up-and-down movement in this Hovercraft (other than the forward movement)?

Definitely Possibly Not sure Probably not Definitely not

If you have answered "definitely not" to Question 3 then end the questionnaire here; otherwise continue.

4. Would you describe the motion (other than forward) that you are experiencing on this Hovercraft as:-

a) A type of vibration Yes No

OR

b) Another type of motion Yes No

If you answer yes to 4b then please describe:-

Figure 1. - Format of questionnaire for survey I.
(Questions printed on both sides of card.)

Please do not attempt the next question till nearer the end of your journey.

13. When you have considered all the factors that might affect your reaction to the journey, could you please rate this particular journey on the scale opposite, by putting a cross on the vertical line.

IT WAS FRIGHTENING
IT WAS UNPLEASANT
IT WAS UNCOMFORTABLE
IT WAS TOLERABLE
IT WAS PLEASANT
IT WAS RELAXING
IT WAS VERY SMOOTH

14. The amount, or intensity, of sound is commonly described in terms of its 'loudness'. Similarly the intensity of light can be described in terms of its 'brightness'. The following is a list of words which could be used to describe the intensity of an up-and-down motion or 'vibration'. Please tick those which you consider to be relevant and then ring the one word which you consider to be most applicable.

Bouncyness	Jolty	Plunge	Heave
Shake	Bumpyness	Lurch	Judder
Roughness	Any others (Please specify).		

Figure 2. - Format of questionnaire for survey III.
(Questions printed on one side of paper only.)

8. Please rate the levels of (a) MOTION OF THE CRAFT DUE TO THE WAVES and (b) CABIN VIBRATION that you are at present experiencing, by putting a cross on the line corresponding to your judgement. 15

N.B. You do not have to keep to the sectioning on the line.

(a) MOTION DUE TO THE WAVES

SMOOTH ROUGH 17

(b) CABIN VIBRATION

SMOOTH ROUGH 16

9. In terms of the following scale, where do you think the COMFORT level of this particular journey would fall?

Please tick the appropriate box.

Very comfortable ☐

Comfortable ☐

Just comfortable ☐

Uncomfortable ☐ 18

Very uncomfortable ☐

10. In terms of the following scale, where do you think the TEMPERATURE of this hovercraft would fall? 19

HOT COLD 22

Figure 3. - Format of questionnaire for survey VI.
(Questions printed on one side of paper only.)

a) Line rating

Please rate the level of noise that you are at present experiencing in this compartment, by putting a cross on the line corresponding to your judgement.

Loud Quiet

b) Magnitude estimation rating

Please try to imagine the levels of vibration which would be experienced:

a) At complete rest, and
b) Whilst travelling in an old car over an unmade road.

If the former (complete rest) was valued at 0, and the latter (travelling in an old car) was valued at 100, what value would you give to the present level of vibration in this compartment?

c) Graphic rating

In terms of the following scale, where do you think the noise level in the compartment would fall?

Please tick the appropriate box.

Very quiet Fairly quiet Moderate Fairly loud Very loud

Figure 4. - Rating methods used in questionnaire studies.

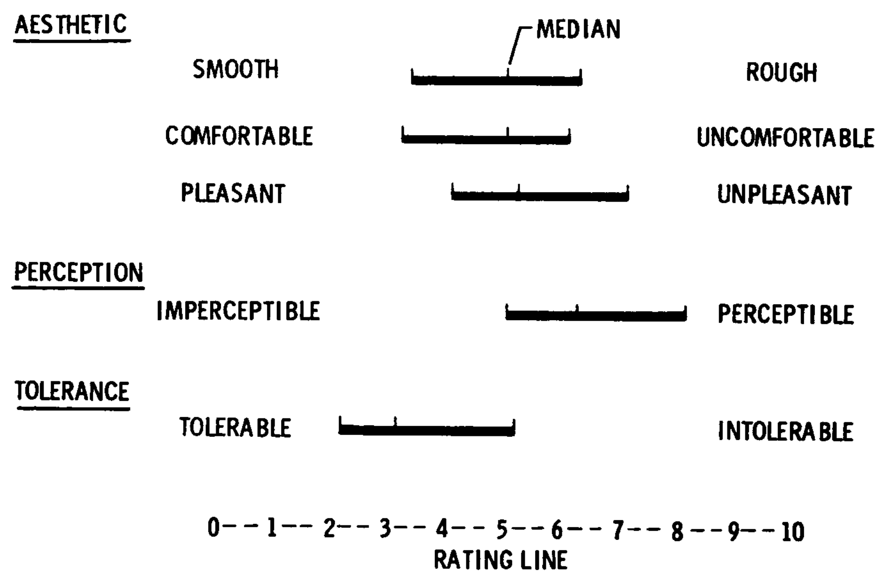

Figure 5. - Effect of scale ends on rating line responses to ride vibration of SRN6 hovercraft.

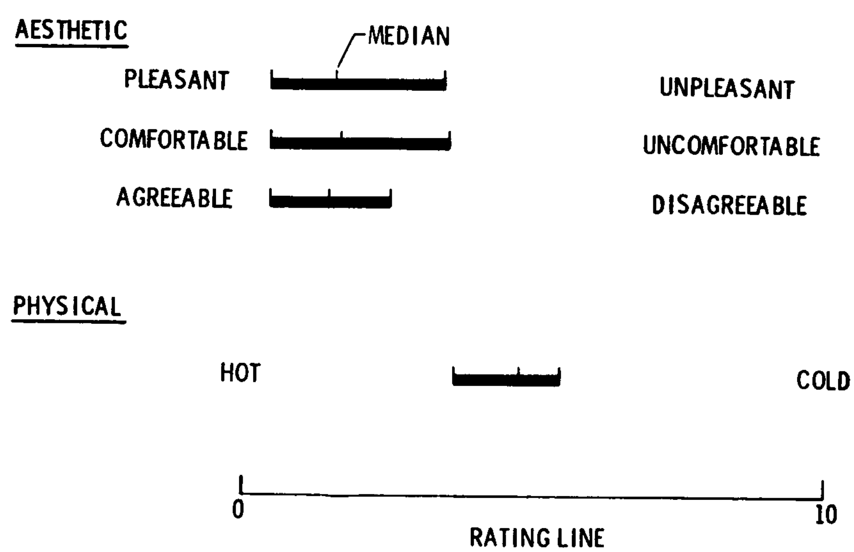

Figure 6. - Effect of scale ends on rating responses to temperature of SRN4 hovercraft.

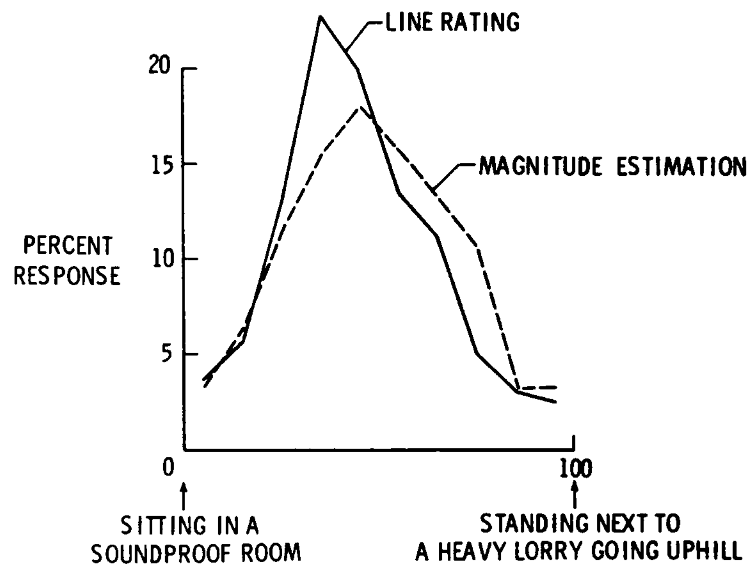

Figure 7. - Comparison of ratings of hovercraft noise obtained by means of line rating and magnitude estimation using situational end points for scales.

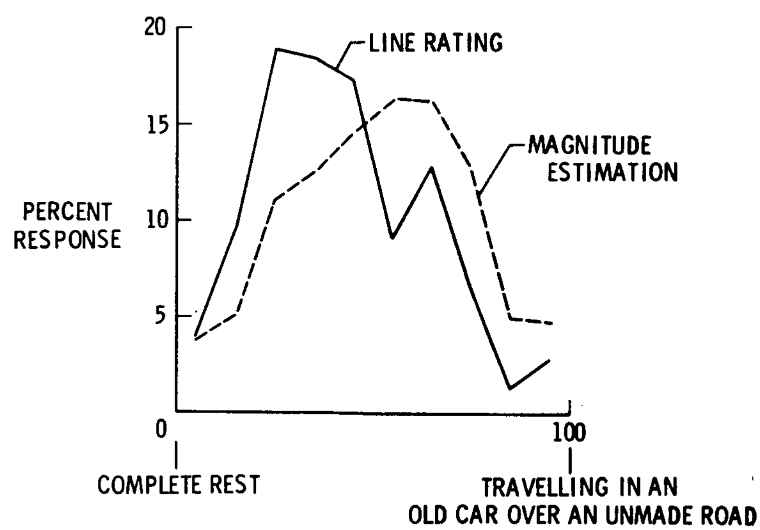

Figure 8. - Comparison of ratings of hovercraft vibration obtained by means of line rating and magnitude estimation using situational end points for scales.

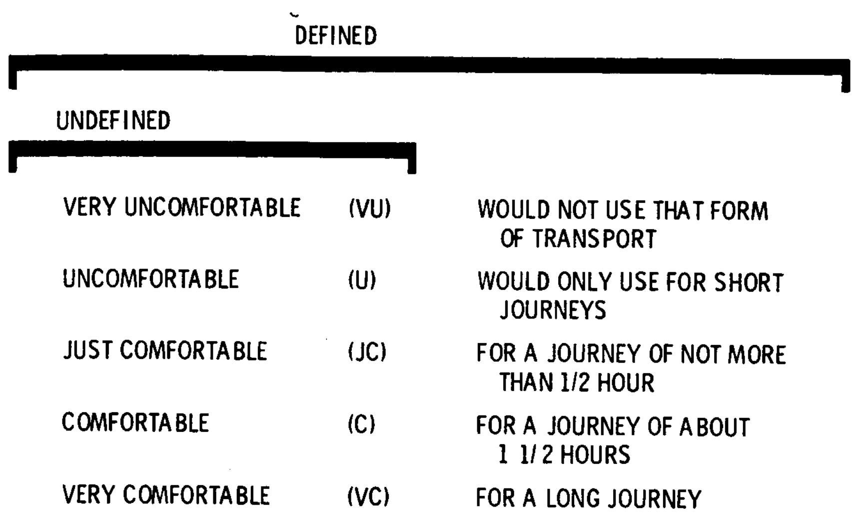

Figure 9. - Defined and undefined semantic rating scales.

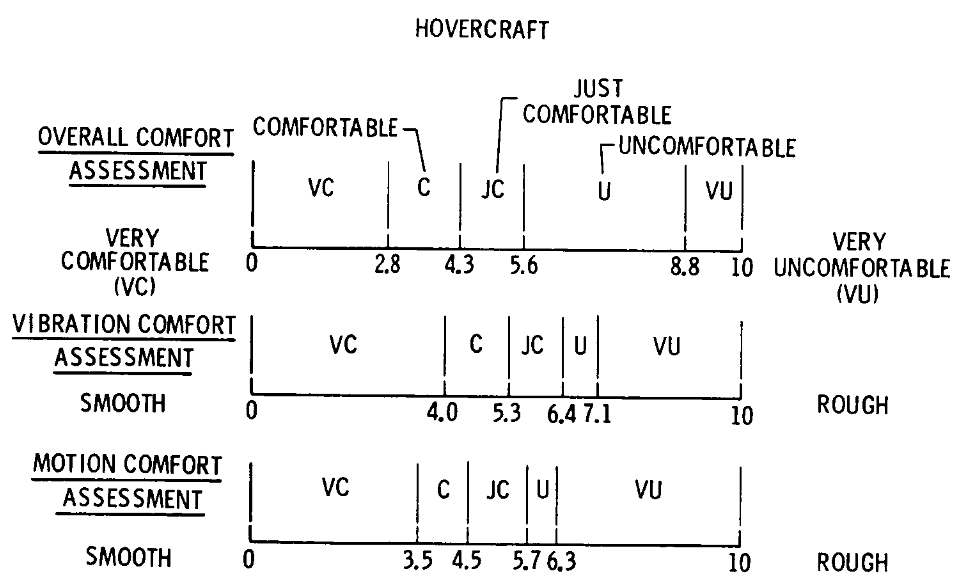

Figure 10. - Linking of semantic assessments with line ratings.

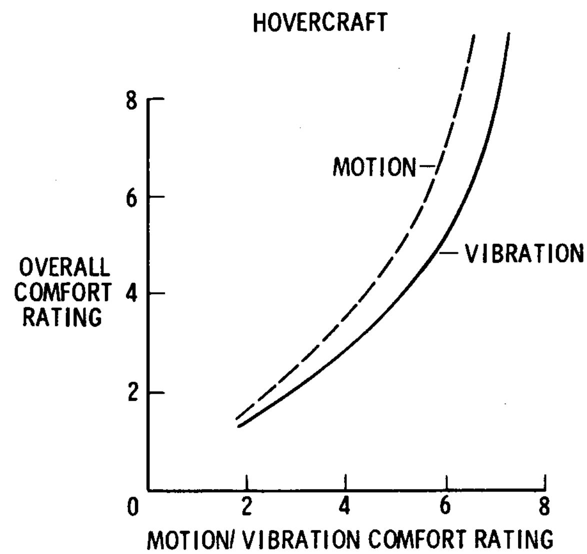

Figure 11. - Relationship between overall comfort rating and motion or vibration comfort rating.

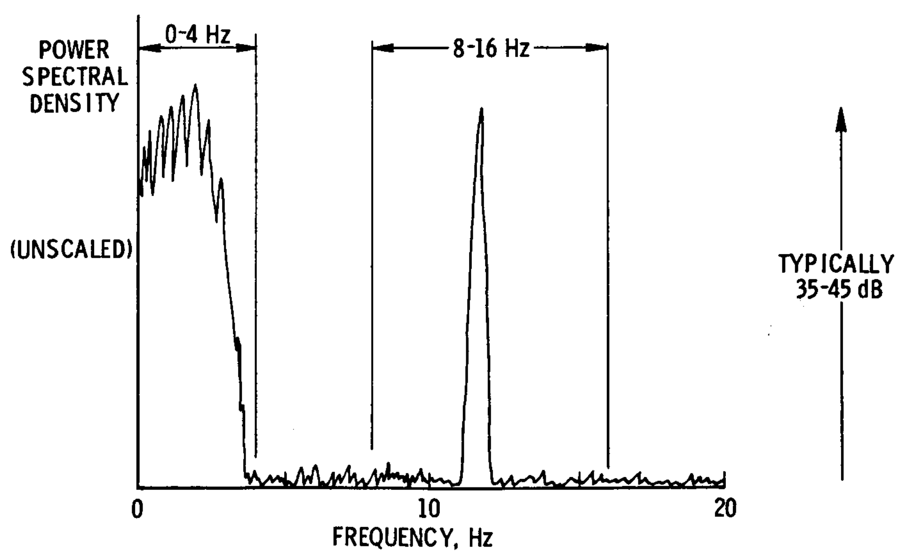

Figure 12. - Typical spectral density shape for vertical hovercraft motion.

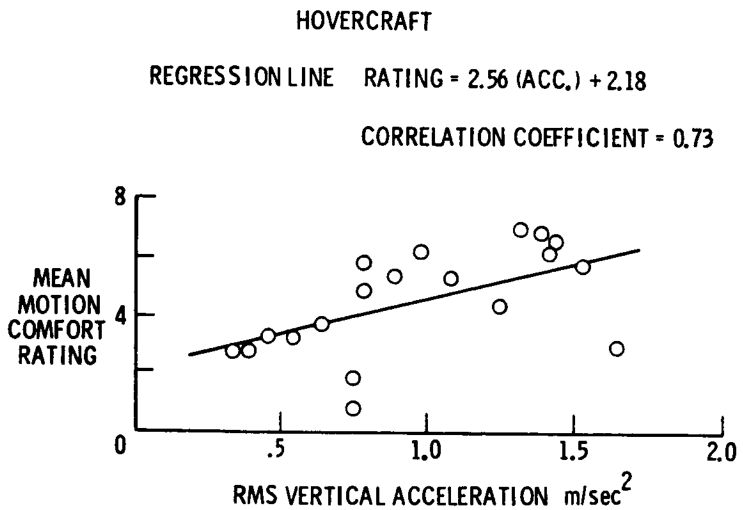

Figure 13. - Relationship between mean motion comfort rating and vertical vibration in the 0 to 4 Hz band.

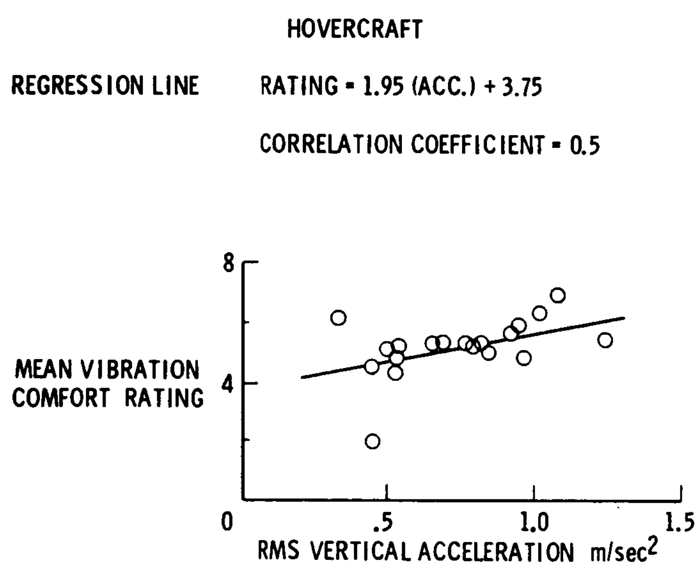

Figure 14. - Relationship between mean vibration comfort rating and vertical vibration in the 8 to 16 Hz band.

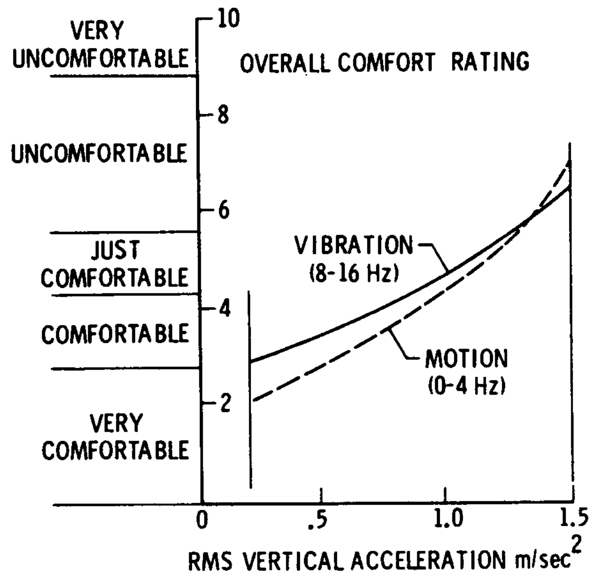

Figure 15. - Relationship between overall comfort rating and vertical vibration/motion for SRN4 hovercraft.

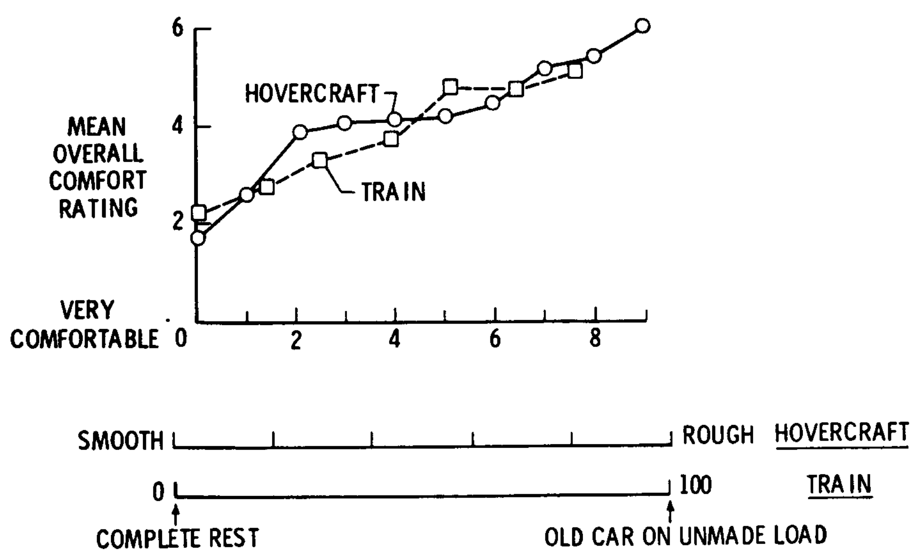

Figure 16. - Relationship between overall comfort rating and vibration rating (SRN4 hovercraft and train).

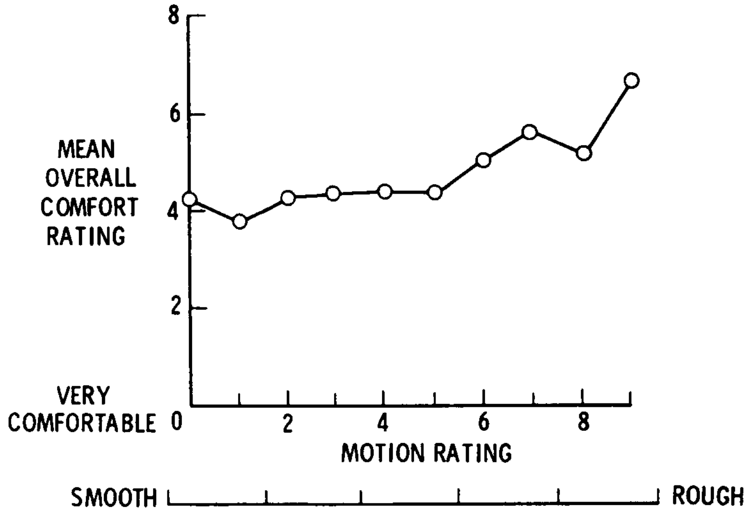

Figure 17. - Relationship between overall comfort rating and motion rating (SRN4 hovercraft).

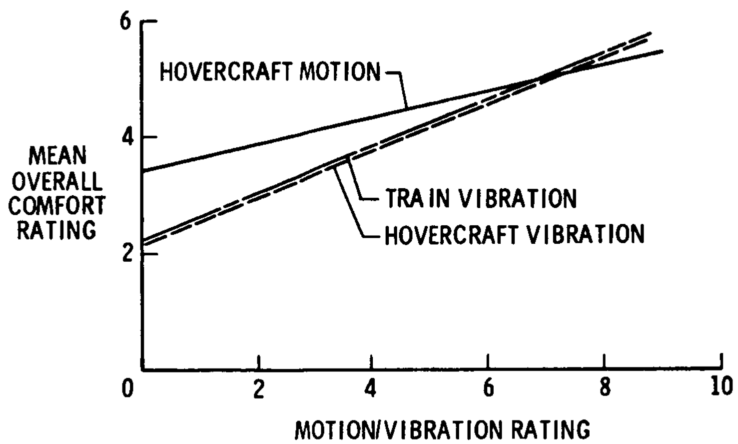

Figure 18. - Relationship between overall comfort rating and motion/vibration ratings (regression lines for data in figures 16 and 17).

Could you now rate the following factors in terms of how important you consider each factor to be in determining the comfort of THIS journey. Would you also ring, by the side of each factor, whether its level is 'too high' (1); 'high'(2); 'just right' (3); 'low' (4); or 'too low' (5) in this compartment.

(a) Temperature 1 2 3 4 5

Not important — Very important

(b) Vibration 1 2 3 4 5
(i.e. any movement of the train other than forward)

Not important — Very important

(c) Noise 1 2 3 4 5

Not important — Very important

(d) Seat Comfort 1 2 3 4 5

Not important — Very important

(e) Ventilation 1 2 3 4 5

Not important — Very important

Figure 19. - Extra question sheet issued on train survey (survey V).

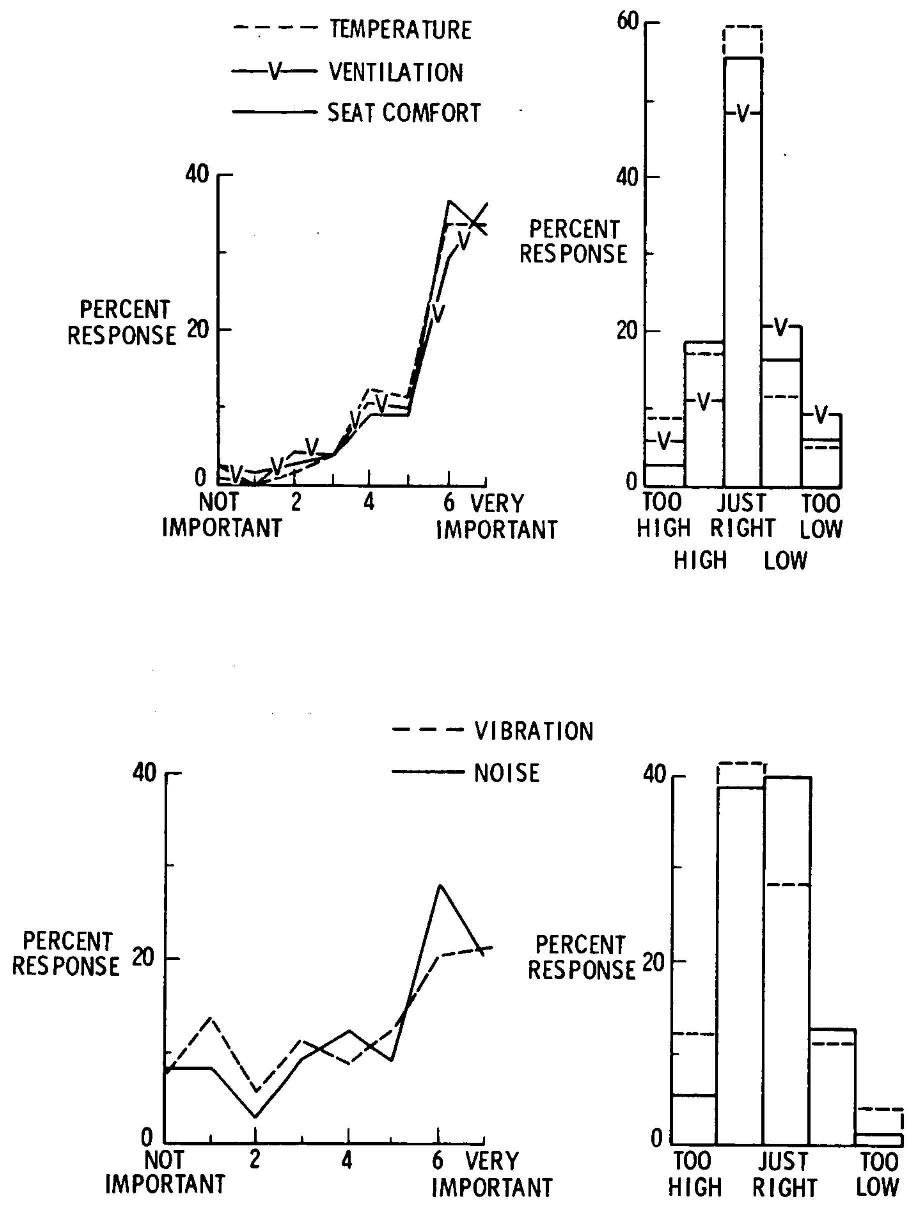

Figure 20. - Comparison of assessed importance of train environmental factors with ratings.

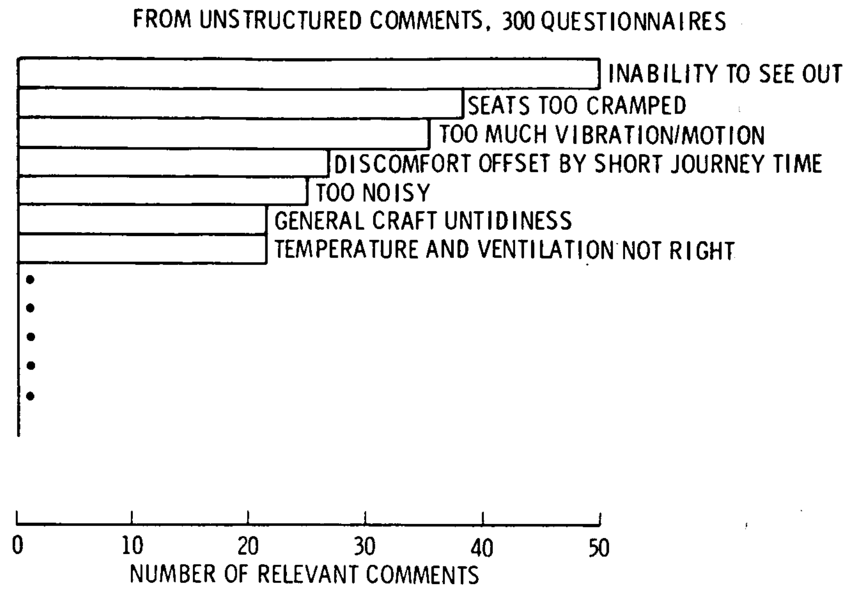

Figure 21. - Factors assessed as being important by hovercraft passengers.

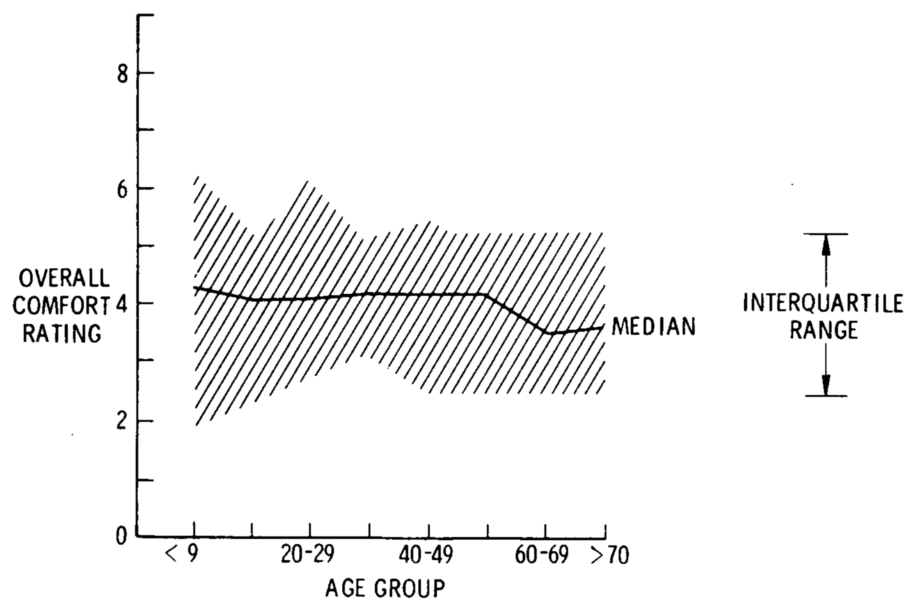

Figure 22. - Effect of passenger age group on overall comfort rating (line represents median; hatched area represents interquartile range) from hovercraft survey IV.

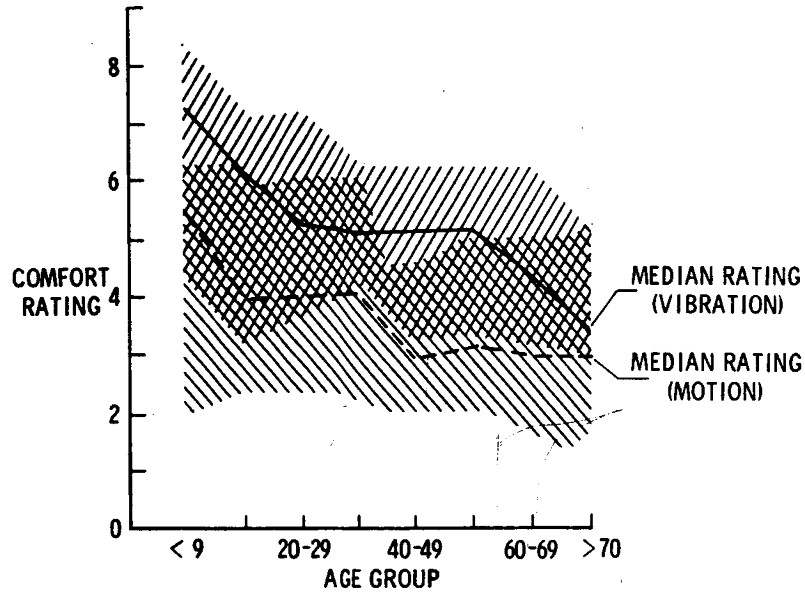

Figure 23. - Effect of passenger age group on vibration and motion comfort ratings (lines represent medians; hatched areas represent interquartile ranges) from hovercraft survey IV.

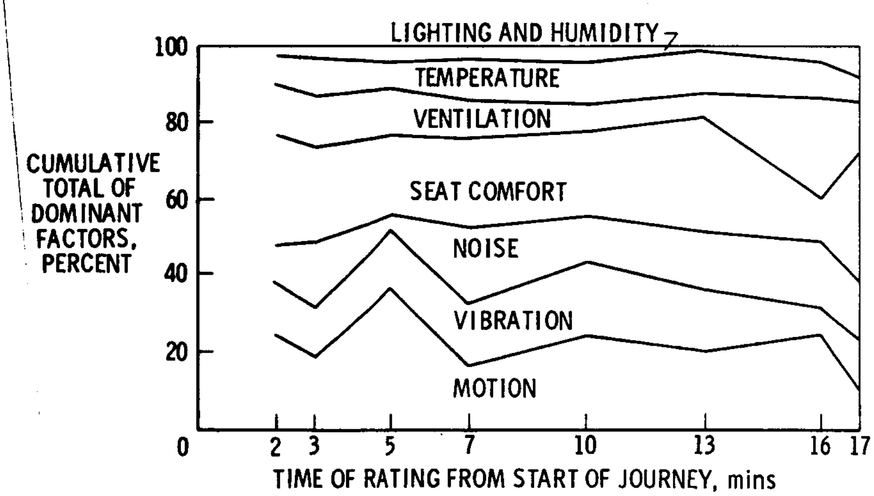

Figure 24. - Variation with time of exposure of proportion of passengers rating environmental parameters as dominant (from hovercraft survey IV).

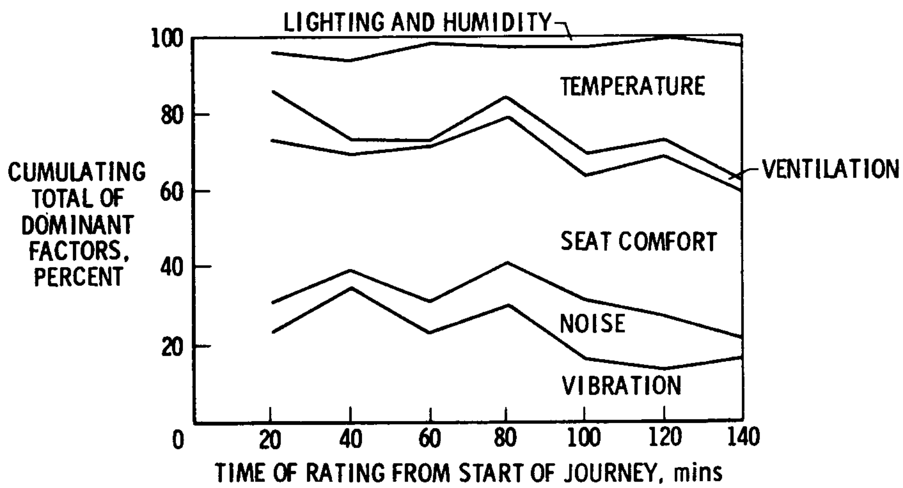

Figure 25. - Variation with time of exposure of proportion of passengers rating environmental parameters as dominant (from train survey V).

N76-16774

A REVIEW OF RIDE COMFORT STUDIES IN THE UNITED KINGDOM

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SUMMARY

United Kingdom research which is relevant to the assessment of vehicle ride comfort has been reviewed. The findings reported in approximately 80 research papers are outlined and an index to the areas of application of these studies is provided. The data obtained by different research groups are compared and it is concluded that, while there are some areas of general agreement, the findings obtained from previous United Kingdom research are insufficient to define a general purpose ride comfort evaluation procedure. The degree to which United Kingdom research supports the vibration evaluation procedure defined in the current International Standard on the evaluation of human exposure to whole-body vibration is discussed.

INTRODUCTION

This paper provides an outline of United Kingdom research into areas of subjective response to vibration that are relevant to the assessment of vehicle ride quality. The desire to report on the relevant United Kingdom effort in one paper has necessitated some degree of selection. Studies have been included in the review wherever it is considered that they help to provide an overall picture of the evolution of research. Thus, while some experiments may have failed to provide any useful findings for designers, they have been included if it is considered that they may provide a foundation which will enable others to increase the practical value of their work.

Figures 1 and 2 provide a guide to the numbers of research papers on all aspects of human response to vibration that have been published in the previous years of this century. It will be seen that United Kingdom researchers produced fewer than half the number of papers published by workers in the United States of America. Of the 350 papers produced from the United Kingdom about one in four concerns ride comfort and approximately 80 of these form the subject of the present review. Almost two-thirds of these studies have been conducted in relation to ride in some specific vehicle while the remainder are concerned with human response in the laboratory with no particular vehicular application.

In 1973 the present author published the findings of a questionnaire survey of human response to vibration research in the United Kingdom (ref. 1).

Fifty-seven groups known to be interested in human response to vibration were asked to describe their experimental facilities and outline their past, present, and future research work. It was found that, between 1965 and 1972, the majority of laboratory experiments were conducted at Universities but less than half resulted in a publication. The helicopter and hovercraft environments were the principal concern of those conducting field research but again fewer than half of the studies resulted in a publication. Thirteen University theses and fifteen review papers were referenced and, of forty-nine other papers describing experimental work since 1965 most were departmental reports and memoranda. At least sixteen groups in the United Kingdom were found to employ one or more persons in some capacity to study human response to vibration. Five groups estimated that during the year beginning October 1971 they spent more than one man year on such research and the survey suggests that during that year a total effort of about 20 man years was spent directly on the study of human response to vibration. Approximately half of this effort was spent in Universities.

The specification of vibration limits for transport systems does not divide itself neatly into three separate problems concerned with human comfort, performance, and health. For example, no study of the discomfort produced by whole-body vibration can reasonably ignore the potentially large effects that can be produced by changes in body posture and seating (e.g., refs. 2, 3, and 4). However, while the physical movement of the body undoubtedly determines subjective reaction to vibration and some of the many studies of biodynamic response to vibration are highly relevant to the assessment of ride quality, they are not included in this review. Similarly, while the effect of vibration on manual control or vision (refs. 5, 6, and 7) may affect a persons' rating of a ride, findings concerning changes in performance are generally excluded. Such effects are included when in an experiment designed to measure subjects' opinions, it appears that effects other than discomfort have dominated their reactions.

In the following sections the United Kingdom research relevant to the study of ride comfort is outlined in an approximately chronological order. An index to the research conducted on the different aspects of the subject is presented in table 1.

SUMMARY OF RESEARCH

Possibly the first published scientific paper to consider human annoyance due to whole-body vibration was a 1902 report by Mallock (ref. 8). He conducted a study for a Committee of the Board of Trade who were appointed to investigate complaints about vibration by persons living in houses near the Central London Railway. The committee concluded that the high "unspring-borne" load of locomotives was the cause of the problem and were unequivocal in "recommending the adoption of a type of locomotive or motor in which the load not carried on springs is reduced as far as possible." Few studies of human response to vibration have produced such a clear and practical conclusion. In his report, Mallock states that "a variation of less than 1 percent in the effective force of gravity is noticeable, and that if the variation is as great as 4 or 5 percent the result is distinctly unpleasant." It seems probable that this conclusion mainly relates to vertical motions with a frequency of about 15 Hz in buildings.

In 1911 Digby and Sankey (ref. 9) presented a paper to the British Association in which they stated "... It has long been known that different persons are affected in different manners by the same conditions of vibration. So far as the authors are aware this subject has not yet been the subject of any very definite study and investigation ..." They proceeded to report on their own findings of large individual differences in sensitivity to vibration of the hand and an apparent decrease in sensitivity after 30 or 40 minutes of the test. They point out the possible importance of whether age, sex, state of health, over-work or railway travelling, and occupation or class affect sensitivity to vibration. They indicated their desire to study response to motions containing third and fourth harmonics and to motions containing recurrent intermittent vibration. When presenting their paper they apparently invited members of the British Association to visit their laboratory on Mondays or Thursdays and "form the subjects of experiment." Digby and Sankey presumably found their task too great (or members of the British Association were uncooperative) for no further account of these authors' studies was published. Digby and Sankey appear to have assumed that response to vibration was dependent on the vibration velocity. In 1923 Eason (ref. 10) reviewed knowledge of human susceptibility to vibration and concluded that acceleration was the unit best suited to describe human response when there were a range of frequencies present.

The determination of the manner in which response varies with frequency appears to have been one of the objectives of a study conducted by Constant (ref. 11) at the Royal Aircraft Establishment, Farnborough, in the early 1930's. He employed a wooden beam hinged at one end and mounted at the other on an eccentric driven by an electric motor. The subject was seated at some position along the beam corresponding to a chosen amplitude of vibration. The vibration frequency was increased and the subject stated "when the amount of unpleasantness reached an arbitrary level fixed by himself." In a paper to the Royal Aeronautical Society, Constant reported that it was extremely difficult to obtain consistent results from such an unreliable measurement of unpleasantness. He also found that the maximum permissible amplitude of vibration at a given frequency varied considerably according to whether the subject was sitting or standing and on the particular attitude adopted in each posture. It also depended on the susceptibility of the individual. However, the variation of the maximum permissible amplitude with frequency was always the same and he concluded that the results gave the best estimate which could be obtained at that time. The single curve covers a frequency range from about 12 Hz to 80 Hz and is shown as curve (e) in figure 3. (This curve assumes that Constant's data relate to peak to peak displacement.) It is not a coincidence that current vibration criteria for United Kingdom aircraft bear a remarkable resemblance to the 1931 curve produced by Constant.

A paper by Postlethwaite (ref. 12) in 1944 gave detailed consideration to the similarity between curves of constant vibration sensation ("trems") and the phon curves of equivalent sound sensation. His analysis of the previous research did not, unfortunately, lead to experimental studies. In 1956

Steffens of the Building Research Station (13) reviewed the application of previous research to the assessment of building vibration. He presented some measurements of building vibration due to road and rail traffic, pile driving, blasting, and machinery and concluded that the levels proposed in Germany by Reiher and Meister were the most useful for assessing this type of problem. (In 1963 the same author provided a more extensive review of a number of alternative evaluation methods (ref. 14).)

In 1956 Willis (ref. 15) considered the possibility of providing sprung seats to alleviate aircrew discomfort during high speed low level flight. He reports that some low level test flights in turbulence suggested that the predominant bumps occurred at frequencies between 1.3 and 7 Hz with levels normally less than ± 1.5 g but occasionally up to ± 3 g. The conclusion of the study conducted at the Royal Aircraft Establishment, Farnborough, was that a seat suspension having a travel of about ± 15.2 cm could be useful but that further study of seat suspensions, aircraft motion and human response was required.

Much of the research relevant to ride comfort conducted during the 1940's and 1950's took place in Germany and the United States of America. In the United Kingdom during 1958 Loach (ref. 16) presented a paper to the Institution of Locomotive Engineers in which he proposed a new method for assessing ride quality in railway carriages. The method was based on the work of Mauzin (of the Societé Nationale de Chemin de Fer) and Dr. Ing. Sperling (of the Deutsche Bundesbahn). The analysis technique detailed by Loach involved the manual determination of the distribution of peaks and the average frequency in an acceleration time history. These data were then modified by frequency weightings, that originated from the work of both Mauzin and Sperling, into a Comfort Riding Factor expressed in hours. Loach states that this is the amount of time before which an average passenger in a coach will experience a sense of fatigue and he says that a carriage regarded as adequate by an average passenger corresponds to a six hour riding factor. However, he cautions against "too literal an interpretation of what the units really mean ... that a value is numerical means that it can be compared with values similarly obtained on other tests." A curve of "equal comfort" corresponding to a three hour riding factor for vertical sinusoidal vibration is shown as curve (c) in figure 4. The corresponding curve for a six hour riding factor occurs at about half the acceleration levels of the three hour contour.

A very similar procedure for railway carriage ride assessment was described by Batchelor in 1962 (ref. 17). A graphical vibration time history of at least ten seconds of vibration is obtained. By visual inspection the low frequency component is assessed and drawn over the complex waveform. The amplitude distribution of peaks of the low frequency component is then counted (ignoring signs) and the mean level of the peaks is calculated and called the "mean acceleration" and is associated with a "predominant frequency." The deviations of the high frequency peaks from the drawn-in low frequency components are then assessed to determine their amplitude distributions. A ride index is then determined for each component by consulting graphs showing (for vertical and lateral directions) ride indices corresponding to each frequency-amplitude combination. The ride index of the complex motion is calculated from the tenth root of the sum of the tenth powers of the ride indices of each frequency component. The frequency contours for a ride index of 3.5 (just satisfactory) are shown as curves (a) and (b) in figure 4. It appears that the methods reported by both Loach and Batchelor are intended for measurements made on the floors of carriages rather than at the passenger-seat interface.

The comparison of objective and subjective measurements of vehicle riding comfort was the basis of a study conducted at the Motor Industry Research Association Laboratories by Aspinall in 1960 (ref. 18). Using the method of paired comparison with 12 subjects he compared the subjects' rankings of the ride in seven cars with the objective data obtained by ride evaluation procedures based on recommendations published by Dieckmann, Janeway, and Loach. Subjects appeared to be confident as to which vehicle they would prefer to travel in and which gave the least vertical motion, but they had difficulty in judging the roll and pitch of the vehicles. The author concluded that the objective methods were satisfactory for detecting the wide differences in vehicle riding comfort but that they were likely to differ from a subjective assessment when fine differences are involved. In a subsequent report, Aspinall and Oliver (ref. 19) published the findings of a similar study in which groups of subjects were exposed to motions in three vehicles. The rides of the vehicles were modified by altering tyre type and pressure, spring rates, dampers, seat flexibility and types of road surface. A good correlation was reported between subjective assessment of the low frequency ride of a vehicle and the average vertical acceleration recorded between a passenger and his seat and passed by a 0.75 to 6.0 Hz filter. The average floor acceleration passed by a 7 to 75 Hz filter also showed a good correlation with subjective assessments. After further studies of car ride (ref. 20) the development of a ride meter was described by Oliver in 1968 (ref. 21). This meter had selectable integration times (30 seconds to 6 minutes) and a plug-in filter such that it could provide a measurement of the average acceleration in the 0.2 to 50 Hz band or the 7.0 to 50 Hz band.

In an experiment reported in 1961, Jones and Drazin (ref. 22) of the Institute of Aviation Medicine required pilots to control a two seater aircraft at various levels and frequencies of roll and pitch oscillation while the head motion of a subject in the rear seat was recorded. This subject was required to rate each motion condition on a four point scale of subjective tolerance. For frequencies of roll from 0.2 to 3 Hz they concluded that when the maximum linear acceleration of the head was less than 0.1 g the conditions were entirely acceptable. When this acceleration was greater than 0.2 g the condition was entirely unacceptable. With the pitch motions, all conditions (0.25 to 1 Hz in the range 3 to 6 degrees per second) produced severe and persistent nausea. Most of the vibration experiments conducted at the Institute of Aviation Medicine have been concerned with the physiological effects of vibration. However, in some cases the experimenter has taken the opportunity to obtain the subjects' opinions of the motions to which they have been exposed. Guignard (ref. 23), for example, investigated the value of an anti-g suit as an anti-vibration device by exposing eight seated subjects to vertical sinusoidal vibration at frequencies from 4.8 to 9.5 Hz at vibration levels of ± 0.5 and ± 1 g. He reported that inflation of the anti-g suit did not alter the increase in pulmonary ventilation or decreases in visual acuity and reaction time that occurred during the vibration exposure. Subjects' ratings of the vibration on a 100 mm line with ends labelled "absolutely delightful" and "tolerable" were also unaffected by inflation of the anti-g suit. However there was subjective evidence that subjects might have been prepared to tolerate the experimental situation longer with the suit inflated. On the basis of the subjects' comments the author therefore concluded that the inflated anti-g suit is of some potential value to men exposed to severe low-frequency vibration in flight.

A series of studies of human perception to low-frequency motion were conducted by E. G. Walsh at the University of Edinburgh in the early 1960's. In a paper published in 1964 (ref. 24) he reported on a study to determine perception thresholds to sinusoidal motion at 0.33 and 0.11 Hz. By studying a subject with bilateral vestibular failure he concluded that sensations in the inner ear were the principal means of first perceiving the motions. His results with normal subjects led him to conclude that peak acceleration did not adequately indicate whether the motion would be perceived and he suggested that his results implied that the appropriate receptors may respond to rate of change of acceleration (jolt or jerk).

The ride comfort of tractor operators has been the subject of a series of studies by J. Matthews and colleagues of the National Institute of Agricultural Engineering (refs. 25 to 31). After an extensive review of previous relevant research (ref. 25) the vertical, longitudinal, lateral, roll, and pitch motions of two pneumatic tyred tractors were recorded while driving on an unmetalled track, rough pasture, and newly ploughed land with deep furrows. Vertical acceleration of the tractors was concentrated in the 2 to 5 Hz frequency range and peak levels greater than 1 g were reported in all axes. The author concluded that longitudinal and transverse components were significant and possibly more important than vertical vibrations under some conditions. The construction of two tracks simulating gently undulating surfaces (with obstacles up to 2.54 cm in height) and severely uneven ground (continuous undulations of 15.2 cm or more) was recommended. The theoretical derivation of the design and construction of experimental lengths of these test tracks is presented in a later paper together with some results obtained on the tracks (ref. 27). Some measurements of vibration on different seats obtained with tractors on farm surfaces is compared with vibration spectra predicted from a knowledge of the seat transmissibilities (determined in the laboratory) and the vibrations on the tractor bodies. Agreement between the two sets of data was fairly good but the author concluded that laboratory measurements alone could not be used to assess the ride quality of seats. A more recent paper from the National Institute of Agricultural Engineering (ref. 29) discusses the design of a ride meter built to conform with the frequency weightings defined in ISO 2631-1974. Papers by Stayner and Beam in 1971 (ref. 30) and Stayner in 1972 (ref. 31) report the findings of studies in which this ride meter was used to determine the effects of driver weight, type of tractor, tyre and ground surface, and the age of the seat on the vibration attenuation performance of tractor suspension seats.

Limits for helicopter vibration were considered in a 1965 paper by Jones (ref. 32). After reviewing some of the previous research in the frequency range up to 30 Hz he concludes that "vibration levels greater than about 0.1 g are objectionable over most of this frequency range." He concludes that a vibration standard with some chance of success would be a limit of 0.1 g up to about 20 Hz and thereafter the curve obtained by Constant (ref. 11) should be followed. This is shown as curve (d) in figure 3. Jones recommends that vibration in all three linear axes should be recorded "close to his (the pilot's) head but on the seat structure."

In 1965 W. D. Bryce (ref. 33) conducted an experiment at the National Gas Turbine Establishment to determine maximum levels of lateral seat vibration for passenger comfort in a proposed rotor-borne aircraft. One hundred and twenty-one subjects took part in an experiment and were mainly exposed to a slowly increasing vibration amplitude (at each of ten frequencies) until the individual reported any particular disturbance. Three-quarters of the total subject comments concerned blurring of the visual field but many subjects reported no adverse effects up to the maximum level of vibration possible with the apparatus. The author draws the tentative conclusion that in the lateral axis levels below a peak acceleration limit of 0.2 g from 3 Hz to 8 Hz and a constant velocity limit from 8 to 40 Hz will be acceptable to 95% of the population for a short period. The limit proposed by Bryce is shown as curve (a) in figure 3.

In 1966 D. R. Leonard (ref. 34) of the Transport and Road Research Laboratory reviewed the problem of determining acceptable limits for bridge movement. He reports on some measurements of the vibration of bridges and describes some experimental work with pedestrians walking and standing on a bridge forced into vibration in the laboratory. Two new tolerance limits were then proposed for walking and standing subjects. (The limits for standing subjects are shown as curve (a) in figure 4.) This work was extended to buildings when Whiffin and Leonard (ref. 35) later published a survey of traffic-induced vibrations. This paper includes a consideration of the mechanism of vibration generation by vehicles and some vibration measurements. They conclude that the most satisfactory way to minimise the effect of traffic-induced vibration is by maintaining road surfaces to a good standard. The problem has been reviewed again in the context of the general adverse effects of road vehicles on the environment by Burt (ref. 36). He states that new roads in Britain are among the smoothest in the world (no irregularities exceeding 10 mm in a 3 m length) and it is doubtful whether there is a case for higher standards to reduce the generation of vibration. In conclusion it is suggested that a systematic survey is desirable to establish the scale of the nuisance and help estimate the financial benefits of improved standards of maintenance. A very different approach to road smoothness was adopted in 1973 by another worker at the Transport and Road Research Laboratories (ref. 37). He investigated the suitability and effectiveness of humps for alerting drivers and controlling vehicle speeds. Humps 3.66 m (12 ft) long and 0.10 m (4 in) high showed some promise for controlling vehicle speeds but the author concludes that their use should be undertaken with caution where vehicle speeds are high.

A. G. Woods (ref. 38) reported in 1967 on a combined study of the effects of low-frequency sinusoidal and random vibration on comfort and performance. For vertical motion at three levels of acceleration with frequencies from 1 to 10 Hz and lateral vibration with frequencies up to 7 Hz three or four subjects made ratings on a six point scale. While the data for vertical motion showed a very definite increase in unpleasant effects around 5 Hz, reaction to lateral vibration indicated a slight and gradual decrease in the effects as the frequency increased at constant acceleration. (Contours that correspond to the comment "some unpleasant effects cannot be ignored" are shown as curv (b) and (c) in figure 3.) There was somewhat more tolerance to lateral than vertical vibration in the 3 to 7 Hz frequency range and there was slightly greater tolerance to the random vibration spectra employed in the experiment than the corresponding sinusoidal motion.

Many measurements of vibration in aircraft have been obtained by worker in the Structures Department of the Royal Aircraft Establishment, Farnboroug (e.g., ref. 39). The analysis method has mainly consisted of an analysis of the distribution of peak accelerations recorded at some position in the aircraft and is oriented towards an understanding of aircraft response rather than human reaction. Some data obtained by this method of analysis is presented by Silverleaf and Cook (ref. 40) in a 1969 review of ride comfort high speed marine craft. They say that the ready availability and ease of operation of equipment to count peaks outweighed the possibility that the da so obtained might be of limited value in assessing ride comfort. The author interpret some previous research as implying that a reasonable acceleration limit for journeys of one hour or more should be between 0.1 g and 0.15 g at low frequencies. They state that foilcraft with submerged foils and autopilot systems have achieved this performance but that it had not been achieve by hovercraft of reasonable commercial size. Silverleaf and Cook concluded that the standard of ride comfort that can be achieved may be a crucial factc in the commercial use of high speed marine craft in open-water routes. In a 1969 review of passenger comfort in hydrofoils Shurmer (ref. 41) of the British Aircraft Corporation advocated further research to develop equipment to give an overall ride index and, in the following year, Lovesey (ref. 42) c the Royal Aircraft Establishment produced a general review of the hovercraft environment.

In 1970 Ashley reported the first use in the United Kingdom (ref. 43) of the method of intensity matching to determine the effect of vibration frequency on subjective response to whole-body vibration. He employed a method somewhat similar to that previously used with whole-body vibration by Miwa in Japan and employed in psychoacoustics research for many decades. In the firs part of the study standing male subjects were required to move from a vibrato adjusted to produce a given level of vertical sinusoidal motion at 6 Hz to a vibrator producing a random vertical vibration. For each of four levels of sinusoidal motion (corresponding to the 1, 2.5, 4, and 8 hour fatigue decreased proficiency limits in ISO 2631-1974 (ref. 44)) the level of the random motion was varied by the experimenter until the subject considered tha it was equally annoying to the sinusoidal motion. The mean levels of the random vibration determined from 27 subjects were then used as fixed levels against which six subjects compared sinusoidal motions from 0.7 to 20 Hz. By adjusting the level of the sinusoidal motions to produce 'equal annoyance' Ashley was able to determine four mean constant annoyance contours. One such contour is shown as curve (a) in figure 5. He concludes that his results are in excellent agreement with the (then proposed) ISO frequency contours.

E. J. Lovesey of the Royal Aircraft Establishment published an evaluati of the effects of bead-filled cushions upon comfort during vibration in 1971 (ref. 45). By increasing the vibration level until subjects considered the motion slightly uncomfortable he concluded that bead-filled cushions were slightly more comfortable than sponge-rubber-filled cushions with most lateral vibrations and during 2 and 4 Hz vertical vibration. The sponge cushions gave a more comfortable ride with vertical vibration at 8 Hz and 20 Hz. All cushions were preferable to a bare seat and, similar to Woods (ref. 38), Lovesey found that at 2 Hz, the maximum amplitude of the heave acceleration that was acceptable was approximately twice that of the lateral vibration. At higher frequencies the relative importance of the two axes without a cushion was reversed--the maximum level of lateral vibration was about double that of vertical vibration at 8 Hz and about treble at 20 Hz.

Human perception of whole-body vibration was the subject of an extensive study reported by McKay from the University of Southampton in 1971 (ref. 46). He determined a median threshold of perception of about ± 0.003 g in a group of forty-eight subjects over the frequency range 1.5 to 100 Hz. However, the effect of vibration frequency on the acceleration threshold was significant as were the differences between standing and sitting and male and female subjects. The median threshold determined by McKay is shown as curve (f) in figure 4. He was particularly interested in determining why the threshold of perception curves reported from previous research differed over an intensity range of 40 dB. In later work (ref. 47) he therefore conducted experiments to determine reasons for this variance and found that the background vibration frequency, acoustic noise, footrest, subject versus experimenter presentation of the stimulus, and the vision, footweat, sex, posture, and attention of the subjects all significantly influence the perception of vibration.

A study to compare response to sinusoidal and random vibration was reported to the United Kingdom Informal Group on Human Response to Vibration by Ashley and Eames-Jones in 1971 (ref. 48). A number of standing subjects adjusted the level of three different spectra of random vibration "to be equal in disturbance sensation" to a given level of a 6 Hz sinusoid. For all three spectra (which covered the frequency range of either 0.5 or 3 Hz to 20 Hz) the authors found that their subjects would accept about 50% more acceleration from random than sinusoidal vibration. At the same meeting, G. Rowlands (ref. 49) of the Royal Aircraft Establishment reported on an experimental demonstration of some International Organization for Standardization vibration levels to subcommittee and panel members of the British Standards Institution. The subjects were required to read, write, talk, and indicate their reactions while exposed to conditions of vertical and lateral vibration corresponding to the 4 and 25 minute ISO fatigue decreased proficiency times. It was reported that most subjects found the levels extremely disturbing and all stated that they would not accept or tolerate these levels in most forms of transport.

A survey of the vibration and ride comfort problems of various transport organisations was compiled by Allen (ref. 50) of the Royal Aircraft Establishment, Farnborough, in 1971. The survey, conducted to assist the Science Research Council in considering research grant applications from Universities, includes the opinion of about twenty different organisations. The author concluded that there was an urgent need for further research which should be equally divided between the study of the effects of vibration on crew and driver efficiency and passenger comfort. Particular areas of research considered to require attention were response to multiaxis, random, long duration, and low-frequency motions. Study of the interactions between seat design and vibration effects, vibration, and other environmental stresses as well as the application of laboratory research to real life environments were also considered to require attention.

A further 1971 report by a worker at the Royal Aircraft Establishment (ref. 51) provides data obtained from three axis vibration measurements made on the floor of fourteen commercial and military vehicles. The data show tha motion was not restricted to the vertical axis and the author therefore suggested that future laboratory studies should include the study of response to fore and aft and lateral vibration.

Three papers (refs. 52 to 54) describing the Ph.D. research conducted by Jones at the University of Salford were published during 1972 and 1973. In his first experiment sixty seated subjects (thirty men and thirty women) were alternately exposed to two vertical sinusoidal motions for eight seconds. On of the motions was a reference of 20 Hz and the other was set by the experimenter to one of thirteen frequencies in the range 4 to 80 Hz. For each of six levels of the reference (± 0.1 to ± 0.6 g) the subject was required to vary the level of the other motion until he considered it to be "equal in sensation on a comfort basis to the reference vibration." The authors report some significant differences between the response of men and women. Compared to their sensitivity at 20 Hz the females were more sensitive than the males to 60 and 80 Hz and to the lower two levels of 4 and 6 Hz. Jones and Saunders suggest that their results are in fairly good agreement with the shape of the curve given in ISO 2631-1974 (ref. 44). This experiment is also presented in a later paper (ref. 53) together with some results obtained with the same experimental method using ten standing male subjects and when employing a 10 Hz reference vibration with sitting male subjects. Compared to their sensitivity to 20 Hz the standing subjects were less sensitive to 4, 5, and 6 Hz than seated subjects. (Curve (c) in figure 5 shows a contour obtained with standing male subjects.) Jones and Saunders report no change in the shape of the curve due to the change of reference frequency. The third paper from these authors (ref. 54) reports on the use of the method of magnitude estimation with sixty seated subjects (thirty men and thirty women) and ten standing males. They determined 'growth functions' from Stevens' Psychophysical Law and concluded that, because the effects of vibration frequency, subject sex and subject posture were small, a value of 0.93 for the exponent in this law will give an adequate overall approximation. By analogy with the phon curves and sone scale in acoustics they proposed units of vibration intensity VICS (Vibration Contours) and units on a subjective ratio scale VIMS (Vibration Magnitude Scale).

At the University of Salford Hempstock and Saunders (ref. 55) were also concerned with Stevens' Psychophysical Law. They exposed subjects to noise and vibration sequentially and required them to alter the level of the dependent variable (noise or vibration) until it produced a sensation equivalent to a fixed value of the independent variable (vibration or noise). Assuming Stevens' Psychophysical Law for both noise and vibration with an exponent value of 0.6 for noise, they proceeded to use the results of their experiment to calculate exponents for vibration. They found that the vibration exponent was two or three times greater when noise was used as the independent variable than when vibration was the independent variable. Thus, for example, the mean values suggest that while subjects would adjust 65 dB(A) noise to be equivalent to 1.0 m/s^2 rms of vibration, they would adjust 1.0 m/s^2 rms of vibration to be equivalent to 80 dB(A) noise! The authors conclude that for whole-body vibration there exists no single value of the exponent in Stevens' Psychophysical Law.

Another study of combined noise and vibration is reported by Fleming and Griffin (ref. 56) from Southampton University. They conducted an experiment to determine the subjective equivalence of 1000 Hz pure tone noise and 10 Hz sinusoidal whole-body vertical vibration. Each of 20 male subjects was exposed to all 64 possible combinations of 8 levels of noise (65 dB to 100 dB SPL) and 8 levels of vibration (0.20 m/s^2 rms to 1.2 m/s^2 rms). Both stimuli were presented simultaneously for a period of 10 seconds and subjects were required to indicate whether, if they were to be presented with the combination again, they would prefer that the noise or the vibration should be reduced. The conditions for equivalence ranged from 0.2 m/s^2 rms at 69 dB to 1.2 m/s^2 rms at 94 dB. The authors present their results in a form that enables an estimate to be made of the percentage of subjects who prefer noise or vibration at any of the given combinations of the two stimuli. It is claimed that the results could be employed as a practical guide to reducing either the noise or the vibration in some environments. A study of subjective responses in a combined noise and vibration environment is also reported by Innocent and Sandover (ref. 57) of Loughborough University. They conclude that "noise and vibration acting together give rise to a discomfort level which is equivalent to the summated discomfort levels of the stresses acting separately."

Pilot reaction to helicopter vibration has been studied in recent years by Griffin (refs. 58 to 61) and workers at Westland Helicopters Ltd. (refs. 62 to 65). Griffin conducted three inflight experiments in Army Scout AH Mk 1 helicopters (ref. 58). A subsidiary finding from the experiments was that pilots often failed to detect changes of up to four to one in the level of vibration that occurred when the aircraft were flown in different flight conditions. There was good evidence that pilots based their judgements of the amounts of vibration on their anticipation of what happens in the various flight conditions rather than the physical levels of the motions they experienced during the particular experimental flights. Since the acceptability of the vibration in aircraft is often based on the judgement of a pilot, it was recommended that further consideration should be given to the benefits of supplementing this method with objective measurement systems. The study also provided some detailed data on the vibration encountered in the helicopter, and the degree to which it was transmitted to the pilot.

The studies of pilot vibration conducted by Westland Helicopters Ltd. have also produced large amounts of data on the vibration in some helicopters. Attempts to correlate the objective measurements with pilot assessments of the motion have shown that the mean vibration levels corresponding to the points on a 10 point rating scale tend to increase as the rating increases. However, there are many vibration conditions that deviate from this trend. Jackson and

Grimster (ref. 65) report that measurements made on some rigid structure within production helicopters show that when the peak level of any vibration component in any axis exceeds ± 1.8 cm/sec the crew consider the aircraft "rough" and unserviceable.

New limits for helicopter vibration have recently been proposed by Griffin (ref. 66) in collaboration with the Royal Aircraft Establishment, the Institute of Aviation Medicine and Westland Helicopters Ltd. Two alternative evaluation methods allow for the specification of limits for the whole-body vibration of aircrew in terms of vibration measured on either the structure of the aircraft or at the crew-seat interface. In summary, the 'normal' limits correspond to 0.4 m/s^2 rms in the vertical (a_z) axis (for frequencies from 4 to 8 Hz) and 0.3 m/s^2 rms in the fore and aft (a_x) and lateral (a_y) direction (for frequencies from 1 to 2 Hz). The frequency weightings defined in ISO 2631-1974 (ref. 44) are used to determine the effect of other frequencies. Data taken from previously published studies have been analysed and it is claimed that they show that these new limits (largely based on laboratory studies) are reasonable. The proposed limit for vertical vibration is shown as curve (f) in figure 3.

Ashley and Rao (ref. 67) of the University of Birmingham reported on an experiment in which subjects, seated in the laboratory on a car seat, were subjected to separate sources of whole-body vertical vibration and vertical foot vibration. In the first experiment five subjects were exposed to random foot vibration and required to ask the experimenter to adjust the level of sinusoidal seat motion until it gave an "equal sensation effect." This was repeated for frequencies in the range 2 to 20 Hz to give an equal sensation contour for sinusoidal vibration. In the second experiment a random seat vibration was adjusted to give equal sensation to a random foot motion. In the third experiment various frequencies of sinusoidal foot vibration were adjusted to be equivalent to a random vibration and so give a contour of equivalent sensation for foot vibration. In the fourth experiment various frequencies of sinusoidal seat vibration were adjusted to equivalence with sinusoidal foot vibration and so yield a second sensation contour for seat vibration. The authors state that the two equivalent sensation contours for seat vibration differ by less than 25% and conclude that this is a good justification for the use of the equal sensation technique.

Human response to vibration research at Swansea University has been concentrated on the study of subjective response to vibration and is mainly described in four papers (refs. 68 to 71). In 1973 Oborne and Clarke (ref. 68) presented an account of the not insignificant practical problems that have to be surmounted when conducting a survey of passenger comfort. McCullough and Clarke (ref. 69) discussed the problems inherent in the semantic scales employed by many previous researchers. They state that such scales have only ordinal properties and that there will be inaccuracies when they are translated from one language to another. Further, they point out that words can be understood to mean different things to different people at different times and they claim that this effect is undoubtedly responsible for a large proportion of the variance in the previous data. McCullough and Clarke then suggested that by using Stevens' Psychophysical Law, it may be possible to construct a family of equal sensation contours based upon a single vibration threshold contour. The authors present a brief outline of two experiments on response to cutaneous and whole-body vibration and conclude that "attention should be directed away from experiments in which semantic labels are used to classify the intensity of vibration and towards experiments which are designed to develop a ratio scale relating subjective and physical magnitudes."

In 1974 Oborne and Clarke (ref. 70), describing a study in which they determined semantic 'comfort labels' for the intervals between frequency contours, rejected both a semantic category selection method and the method of intensity matching for the determination of the frequency contours. Subjects were required to rate various levels of 11 frequencies of vibration on a 10 cm line with ends labelled 'smooth' and 'rough.' Four equal sensation contours were then constructed for ratings of 2, 4, 6, and 8 cm along the rating line. Further, subjects were presented with the vibration stimuli again and asked to rate the motions on a six point semantic scale. The experimenter placed five appropriate phrases between and above the four equal sensation contours. (The contour dividing 'neutral' and 'uncomfortable' for standing subjects is shown as curve (b) in figure 5.) The authors add a note of caution on the use of rating lines. They say that there is evidence that passengers make ratings not only in terms of the scale ends but also in terms of what levels of vibration they expect to experience in the vehicle. In their most recent paper Oborne and Clarke (ref. 71) report on a laboratory experiment in which standing subjects were each required to rate ten different vibration stimuli on thirty different 10 cm rating lines (five different sectionings of the lines combined with six different semantic ends). Finding that all thirty different lines produced generally similar results the authors concluded that the fears of other authors over the confusion generated by the use of different sectioning and semantics is unfounded.

A somewhat similar experiment included in a series of studies conducted at the University of Southampton was reported in 1972 by Fothergill (ref. 72). This investigation involved three experiments designed to determine whether subjects differentiated between various adjective scales, whether results obtained by category selection methods differ from those obtained by category production, and whether background acoustic noise affects a subject's rating of vibration. The first experiment tested the hypothesis that subjects disregard the adjectives on which they are asked to scale their sensations and substitute some personal psychological scale. A group of 20 subjects were divided into two subgroups such that one group rated a small number of motions on an open ended 10 cm scale with ends marked "not unpleasant" and "very unpleasant." The second group rated the same stimuli presented in the same order on a similar scale marked "not annoying" and "very annoying." There was good evidence to conclude that the difference in adjective did initially result in different ratings but that after a small number of judgements other variables associated with the scale and range of stimuli became more dominant sources of variance. In a second experiment with a five point semantic scale it was found that the levels corresponding to the extremities of the scale were higher when determined by a category production method than when determined by category selection. The reverse occurred for the three central descriptors of the scale. In a third experiment it was found that when a background white noise at 85 dB(A) was presented the subjects considered that the lowest point on a five point semantic scale generally corresponded to a slightly higher vibration level than when a lower noise level of 54 dB(A) was present. The other four points on the scale, particularly the highest point, corresponded to slightly lower vibration levels in the presence of the higher noise level.

More recent experiments at the University of Southampton have employed an intensity matching technique in which the subject adjusts the level of one stimuli to produce the same degree of discomfort as some other stimuli. Fothergill and Griffin (ref. 73) first studied the method and investigated th between and within subject variability and the effect of varying the frequenc of the standard vibration against which other frequencies are matched. Although subjects had only a low confidence in their matches, the within subject variability was low and very much smaller than between subject variability. As the frequency separation of the two vibrations to be matched became greater, the subject variability also increased. Although only small differences were found between the results obtained with different frequency standards, it was concluded that a 10 Hz sinusoid was the best choice for th future research.

Fothergill and Griffin have conducted three experiments to study the discomfort of multiple frequency whole-body vertical vibration (ref. 74). Subjects were required to adjust the level of a 10 Hz sinusoidal vibration such that it produced a degree of discomfort equivalent to that caused by a variety of multiple frequency stimuli including motions containing predominant beats and up to four sinusoidal components. The levels of the 10 Hz vibration equivalent to the complex motions were always well predicted by the root mean square of the levels of 10 Hz equivalent to the individual sinusoidal components in the complex motion. The authors point out that the equivalent discomfort of the multiple frequency motions could therefore be determined by weighting the vibration spectrum with an electronic network having a frequency response given by the manner in which the discomfort due to vibration varies with vibration frequency. They considered the possibility of inhibition occurring in the response to multiple frequency motions but concluded that the complexity inherent in methods based on models of inhibition was unnecessary. They also compared the results of the study with the recommendations published in the International Standard ISO 2631-1974 (ref. 44). Some more recent research by Griffin (ref. 75) shows that for practical purposes the above method for assessing the discomfort of multiple frequency motions can also be employed to evaluate some random motions, including motions with crest factors greater than three.

Fothergill and Griffin have also conducted a detailed study of the determination of the subjective magnitude of 10 Hz sinusoidal vertical vibration by both magnitude estimation and magnitude production methods (ref. 76). In brief, it was found that for all fourteen subjects participating in the experiment the rate of increase of subjective reaction with increasing vibration level was greater when determined by magnitude production than when determined by a magnitude estimation method. The mean exponents of Stevens' Psychophysical Law were 1.7 (magnitude production) and 1.1 (magnitude estimation). This compares with a mean value of about 1.0 determined by Fleming and Griffin (ref. 56) in the same laboratory at Southampton University from the combined noise and vibration experiment described earlier.

In a recent paper presented to the Institute of Acoustics in 1975 Griffin reported on vibration measurements made in cars, trucks, and buses driven over four different roads (ref. 77). The roads varied in roughness from 'good' to 'poor.' Fore-and-aft, lateral, and vertical vibration were recorded at the subject-seat interface of a person sitting in a passenger seat and, simultaneously, on the vehicle floor beneath this seat. The recorded data were analysed to determine the frequency, amplitude, and axis distribution of the motions at the two measuring locations. The seat vibration data were weighted by the frequency weightings defined by the International Organization for Standardization and the seat transmissibilities were determined. The author reported that the ISO weighting procedure for vibration evaluation indicates that vertical vibration was the predominant motion. Frequencies below about 10 Hz contributed most to the weighted value in the vertical axis and the frequency associated with the peak weighted acceleration level was found to depend on the vehicle type. The weighted vibration levels varied according to the type of road and type of vehicle. On the 'good' road the weighted vertical levels were 0.2 m/s^2 rms and greater, while on the 'poor' road the levels were 0.5 m/s^2 rms or more. In trucks and buses weighted levels higher than the one minute reduced comfort boundary were recorded on the 'good' road and well in excess of the 1 hour fatigue decreased proficiency level on the 'poor' road. In the vertical direction crest factors at the seat were normally in excess of three. The vertical transmissibility of the seats varied but all showed an amplification at some frequencies below 10 Hz and attenuation at higher frequencies.

OTHER PUBLICATIONS

The United Kingdom was one of the two countries to vote against accepting the proposals that became International Standard ISO 2631-1974(E), Guide for the evaluation of human exposure to whole-body vibration. However, before copies of the International Standard became available in 1974 the British Standards Institution published a Draft for Development, Guide to the evaluation of human exposure to whole-body vibration (ref. 78). The Draft for Development is very similar to the International Standard and there is no conflict between the vibration evaluation methods given in the two documents. The reasons for the issue of a BSI Draft for Development as opposed to a British Standard (or approval of the International Standard) was that it was felt that the proposals were only "of a provisional nature because much of the available information relating to the effects of vibration on humans is in fact of a provisional or even contradictory nature."

An earlier publication from the British Standards Institution provides a "Guide to the safety aspects of human vibration experiments" (ref. 79). This document discusses some of the ethical and safety measures that experimenters should consider and it proposes that experiments should be classified into four schedules according to the levels of the vibration and the fitness of the subjects. These schedules range from experiments with levels below the ISO 'fatigue decreased proficiency limits' for which with fit subjects no medical certification or supervision is required, to experiments with levels exceeding the ISO 'exposure limits' when subjects should be required to have medical certification and a medical officer should be present during the experiment. The document also provides a list of medical conditions which would generally render a person unfit to be a subject in a vibration experiment.

Many other aspects of human response to vibration are currently under consideration by sub-committee and panel members of the British Standards Institution (e.g., response to building vibration, multiple frequency vibration, hand-arm vibration, and impacts). One study of great importance and having a wide interest concerns the specification of limits for human exposure to low frequency vibration. Suitable simulation facilities have not been available in the United Kingdom to conduct relevant experimental work but some limits for vibration in the frequency range 0.1 to 1 Hz have been formulated on the basis of previously published research (ref. 80). G. R. Allen of the Royal Aircraft Establishment at Farnborough has undertaken the task of evolving the limits which, at present, comprise "Severe Discomfort Boundaries" and a "Reduced Comfort Boundary." The Severe Discomfort Boundaries are based on motion sickness data and, for a 20 minute exposure, take the form of a constant acceleration limit of 1.0 m/s^2 rms from 0.1 to 0.3 Hz rising to 3 m/s^2 rms at 0.6 Hz and tentatively extrapolated to 6.7 m/s^2 rms at 1 Hz. For longer periods of exposure the acceleration limits decrease in inverse proportion to the square root of the exposure duration. The reduced comfort boundary is based on laboratory studies of discomfort due to factors other than motion sickness during vibration. At present it is described by a contour which increases by a factor of five in acceleration as the frequency is increased from 0.1 to 1.0 Hz.

CONCLUSIONS

The findings of about eighty studies conducted in the United Kingdom to investigate the effect of vibration on human discomfort have been summarised. The laboratory studies of the effects of frequency of sinusoidal vertical vibration on comfort have produced some agreement on the shape of the curves (see figure 5) with the mean sensitivity of subjects showing a maximum around 5 Hz. Although there are also data to show how to assess some non-sinusoidal motions the available results fall far short of that which is required to provide a complete general procedure for assessing the complex multiaxis motions, that characterise most vehicle rides. There are some data on the relative differences in the sensitivities of individual subjects to different frequencies but, above threshold, little understanding of the absolute differences in individual sensitivity to any vibration condition.

There have been no satisfactory studies which suggest how comfort limits should change with the duration of exposure to vibration or how to assess motions whose level varies greatly during an exposure.

Studies conducted in relation to specific transport systems (aircraft as in figure 3 or the railways as in figure 4) show a high degree of agreement. (The curves (a) to (d) in figure 4 could be raised or lowered to allow for different ride indices or exposure times but those shown seem reasonable in the light of the context in which the limits are reported.) In these figures two curves could be identical but, being associated with different evaluation procedures (e.g., the method of assessing non-sinusoidal motion), could correspond to widely different limits. Evaluation methods have not always been adequately defined by those proposing limits and it is often not clear where the vibration levels are to be measured. Where there are such differences between two procedures, their importance is dependent on the motions being assessed. Since it is not possible to evaluate this in the present paper, the following comparison of the curves in figures 3 and 4 assumes that their proposers would expect them to apply to sinusoidal motion at a passenger-seat interface. It may be observed that for vertical vibration around 5 Hz all authors (Batchelor, Loach, Woods, Jones, Griffin, Jackson, and Grimster) quote limiting levels in the range 0.4 to 0.7 m/s^2 rms. They all advocate the same or higher levels at higher frequencies and, with the exception of Jackson and Grimster (ref. 65), they advocate the same or higher levels at lower frequencies. Although some authors of the above limits quote measurements of transport vibration to support their proposals there has been relatively little systematic investigation of their validity. In view of the differing applications of the limits and the limited attempts at verification it is surprising to find such a high degree of agreement.

One of the objectives of research in this area is to define a ride evaluation procedure which will not only give a numerical indication of vehicle ride but also provisionally indicate how the ride changes as the many physical variables change. The United Kingdom research outlined in this paper comes close to providing the most simple procedure for stationary vertical vibration with only two variables: level and frequency. There is very little information originating from the United Kingdom on how these variables interact with motion in other axes, on the importance of vibration duration or variations in vibration level, frequency or axis with time. There are some data on the relative importance of noise and vibration but reports of the significance of other physical variables that may affect human response without changing the vibration exposure are largely apocryphal.

An hypothesis as to how human response to vibration depends on four physical variables (vibration level, frequency, axis, and duration) was published as International Standard ISO 2631-1974(E) (ref. 44). From research conducted in the United Kingdom, the United States of America and many other countries this document defined vibration limits for the preservation of comfort, the preservation of working efficiency, and the preservation of health and safety. The data presented in figure 5 show a broad similarity to the shape of the ISO contour for vertical vibration although some curves depart from the shape by up to a factor of two in acceleration level at some frequencies. The vibration limits shown in figure 3 and figure 4 approximately correspond to the ISO 25 minute reduced comfort boundary and, in view of the many other potential sources of variation in analysing a ride motion, this may seem to be reasonable agreement. Debate over the contention in the ISO standard that levels three times greater are required before there is a significant risk of impaired working efficiency and that, for 25 minutes, it would be unsafe to exceed levels six times greater does not come within the scope of this paper.

It appears therefore that both the vertical frequency weighting and some of the limits for human comfort defined in the ISO Standard can be considered to be in harmony with some United Kingdom research. However such agreement is not generally sufficient for design purposes. There is discord between United Kingdom Research (Fothergill and Griffin (ref. 74)) and the ISO preferred method of assessing complex vibration. There are data which lead t the conclusion that the suggestion in ISO 2631-1974 that the limits may not apply to motions having crest factors greater than three is a very severe practical limitation. However there are also some United Kingdom data to suggest that, while the crest factor may not be the most appropriate unit to describe the 'peaky' nature of a motion, the tentative limit of three given i ISO 2631-1974 could possibly be increased to 5 or 6 for some motions. Finall there are no United Kingdom data to support the time dependency defined in IS 2631-1974 and at present there are insufficient published data to draw conclu sions regarding the validity of the data for non-vertical vibration.

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TABLE I

AUTHOR	DATE	REFERENCE	PAGE	AIR	RAIL	ROAD	SEA	AGRICULTURAL	BUILDING	LABORATORY
Allen	1971	50	479	X	X	X	X	X		
Allen	1974	80	486	X	X	X	X	X		X
Ashley	1970	43	478							X
Ashley and Eames Jones	1971	48	479							X
Ashley and Rao	1972	67	482							X
Aspinall	1960	18	475			X				
Aspinall and Oliver	1964	19	475			X				
Batchelor	1962	17	474		X					
Bryce	1965	33	477	X						X
B.S.I.	1973	79	485							X
B.S.I.	1974	78	485	X	X	X	X	X	X	X
Burt	1972	36	477			X			X	
Constant	1932	11	473	X						X
Digby and Sankey	1911	9	473							X
Eason	1923	10	473						X	
Fleming and Griffin	1975	56	481							X
Fothergill	1972	72	483							X
Fothergill and Griffin	1975	73	484							X
Fothergill and Griffin	1975	74	484							X
Fothergill and Griffin	1975	76	484							X
Griffin	1970	61	481	X						
Griffin	1972	58	481	X						
Griffin	1973	1	471	X	X	X	X	X	X	X
Griffin	1974	59	481	X						X
Griffin	1975	3	472							X
Griffin	1975	5	472							X
Griffin	1975	77	485			X				
Griffin	1975	66	482	X						X
Griffin	1975	75	484							X
Grimster	1974	63	481	X						
Grimster et al	1974	62	481	X						
Guignard	1959	4	472	X						X
Guignard	1959	6	472							X
Guignard	1964	23	475	X						X
Hawkins and Griffin	1972	60	481	X						
Hempstock and Saunders	1973	55	480							X
Hilton	1970	29	476					X		
Holliday	1974	64	481	X						
Innocent and Sandover	1972	57	481							X
I.S.O.	1974	44	478	X	X	X	X	X	X	X
Jackson and Grimster	1972	65	481	X						
Jones, A. J. and Saunders	1972	52	480							X
Jones, A. J. and Saunders	1972	53	480							X
Jones, A. J. and Saunders	1974	54	480							X
Jones, G. M. and Drazin	1961	22	475	X						
Jones, J. P.	1965	32	476	X						
Leonard	1966	34	477						X	
Loach	1958	16	474		X					
Lovesey	1970	42	478				X			
Lovesey	1971	45	479							X
Lovesey	1971	51	480	X	X	X	X			X
Mallock	1902	8	472		X				X	
Matthews	1964	25	476					X		
Matthews	1964	26	476					X		
Matthews	1966	28	476					X		
Matthews and Talamo	1965	27	476					X		X
McCullough and Clarke	1974	69	482							X
McKay	1971	46	479							X
McKay	1972	47	479							X
Mitchell	1969	39	478	X						
Oborne and Clarke	1973	68	482	X			X			
Oborne and Clarke	1974	70	482							X
Oborne and Clarke	1975	71	482							X
O'Hanlon and Griffin	1971	7	472							X
Oliver	1968	21	475			X				
Oliver and Whitehead	1966	20	475			X				
Postlethwaite	1944	12	473							X
Rowlands	1971	49	479							X
Rowlands and Maslen	1973	2	472							X
Shurmer	1969	41	478				X			
Silverleaf and Cook	1969	40	478				X			
Stayner	1972	31	476					X		
Stayner and Bean	1971	30	476					X		
Steffens	1952	13	474						X	
Steffens	1963	14	474						X	
Walsh	1964	24	476							X
Watts	1973	37	477			X				X
Whiffin and Leonard	1971	35	477						X	
Willis	1956	15	474	X						
Woods	1967	38	477	X						

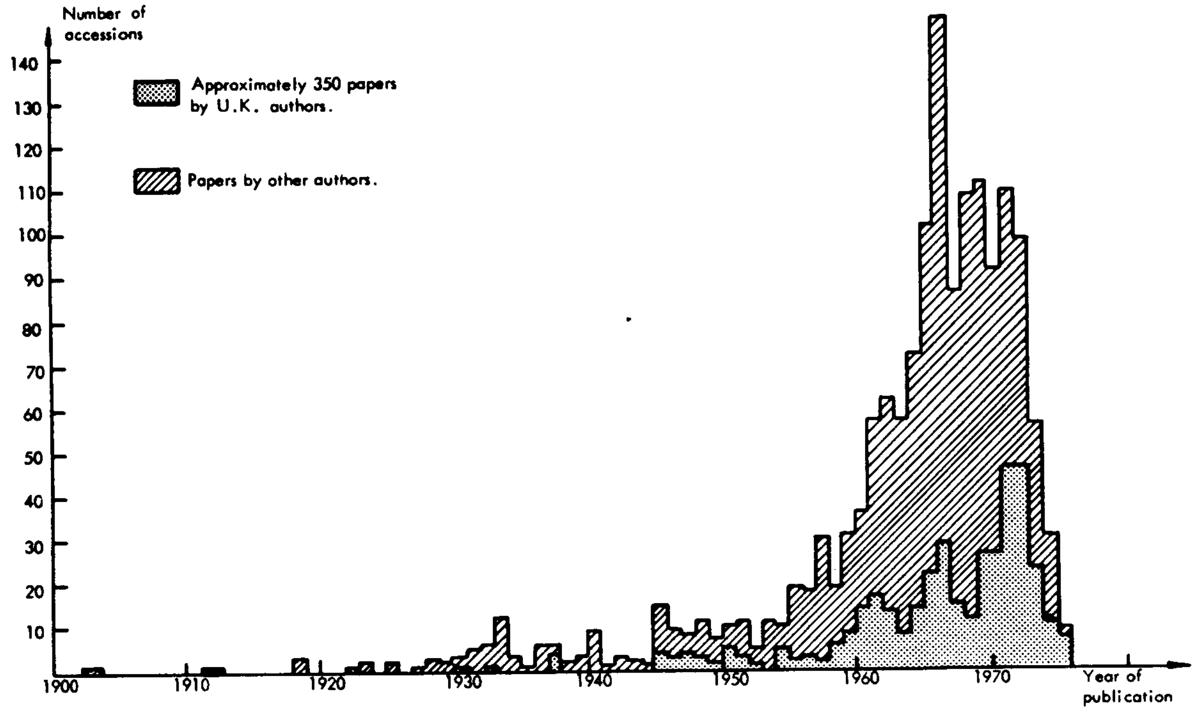

Figure 1.- Histogram showing the year of publication of accessions to the human response to vibration literature collection at the Institute of Sound and Vibration Research (June 1975).

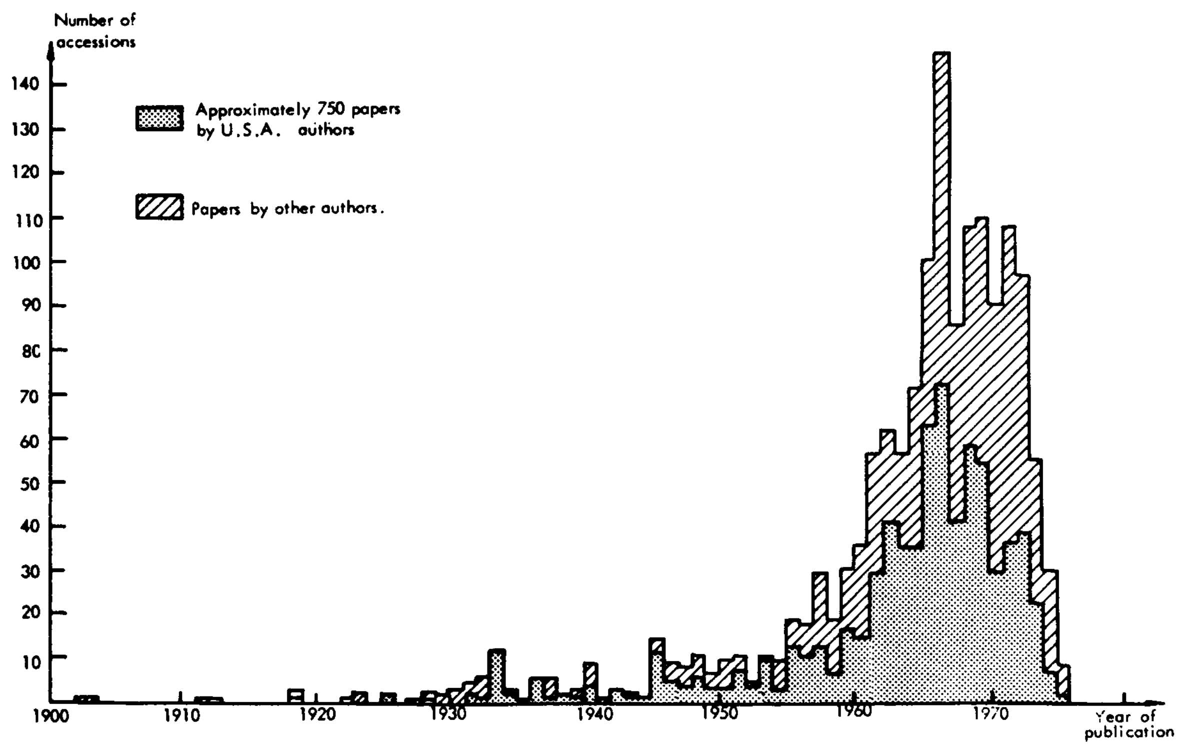

Figure 2.- Histogram showing the year of publication of accessions to the human response to vibration literature collection at the Institute of Sound and Vibration Research (June 1975).

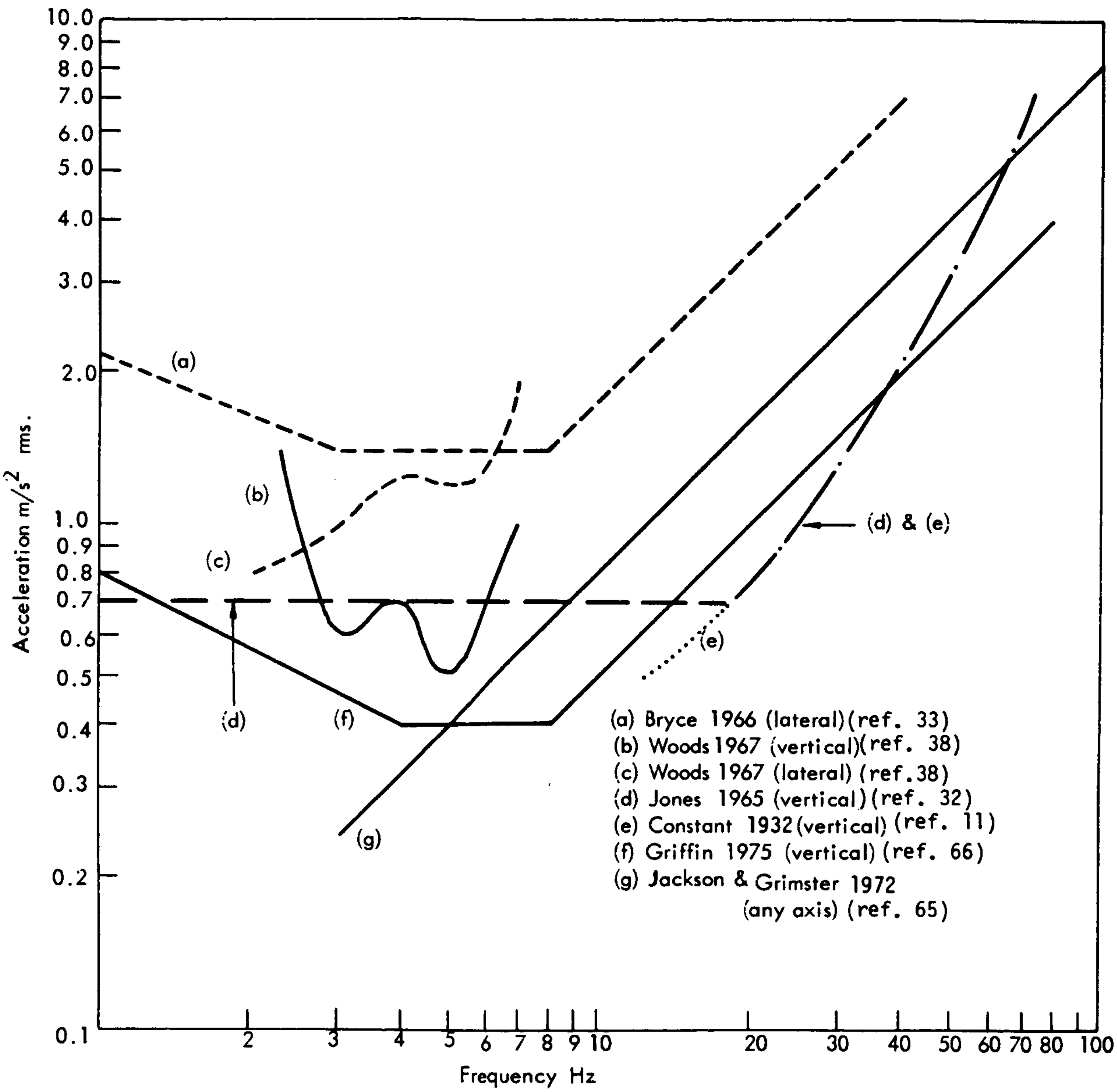

Figure 3.- Aircraft vibration limits proposed in the United Kingdom.

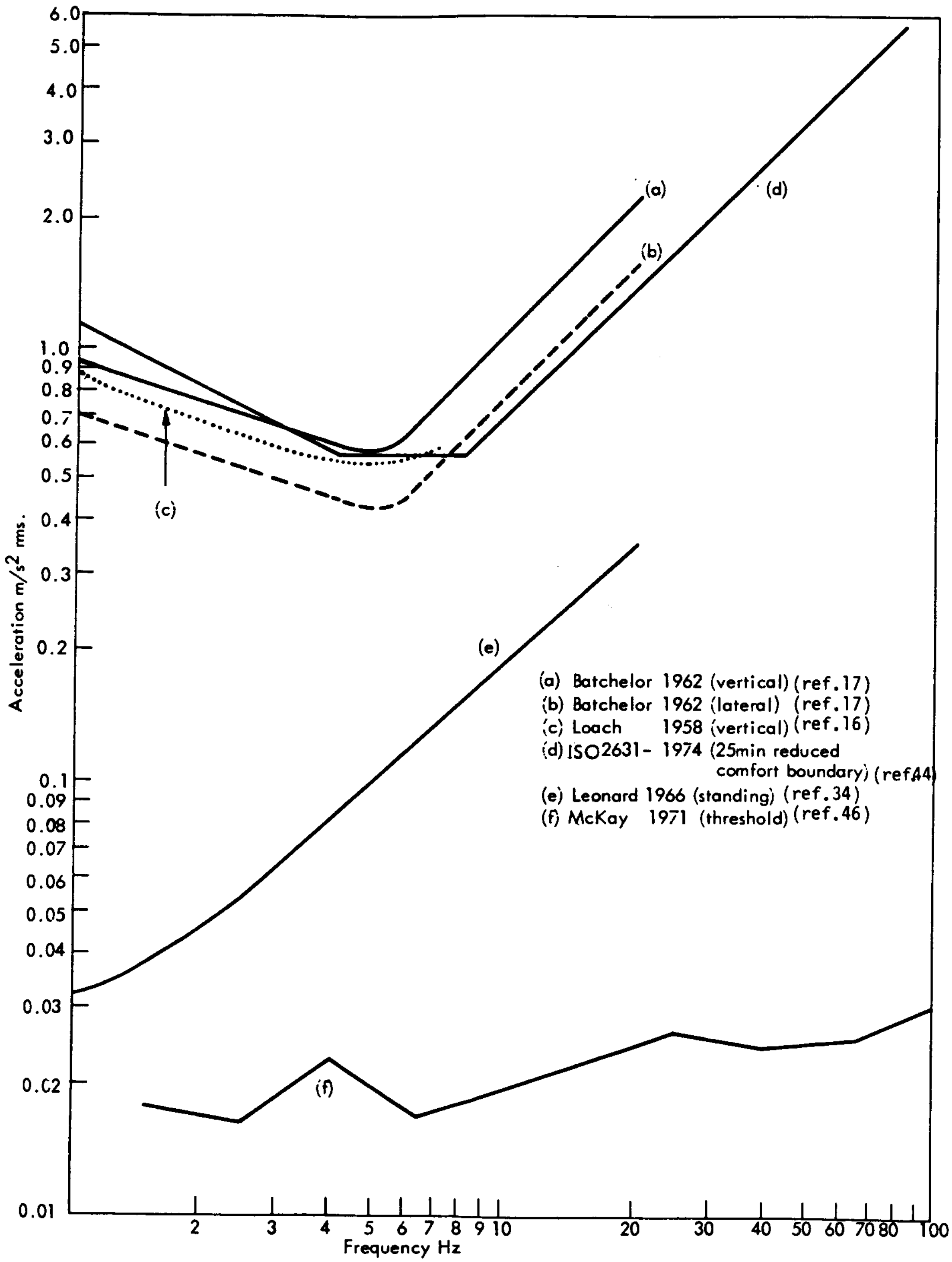

Figure 4.- Frequency contours for railway vibration, bridge vibration, and the threshold of perception of whole-body sinusoidal vibration as reported by some United Kingdom workers.

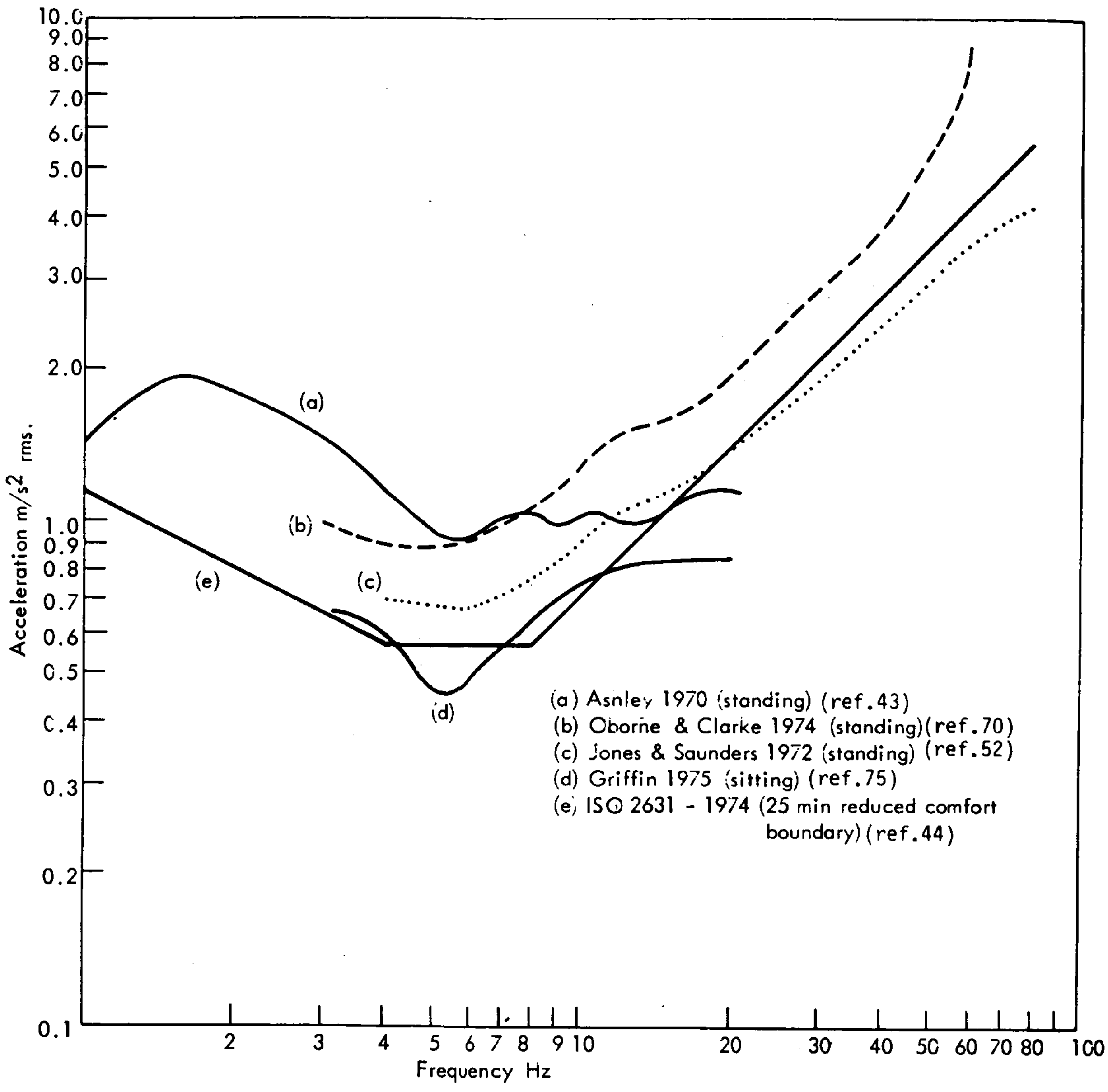

Figure 5.- A comparison of contours of equivalent sensation to whole body sinusoidal vertical vibration determined in the United Kingdom.

N76-16775

RIDE QUALITY AND INTERNATIONAL STANDARD ISO 2631 ("GUIDE FOR THE EVALUATION OF HUMAN EXPOSURE TO WHOLE-BODY VIBRATION")

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SUMMARY

The evolution of the Standard, which is aimed at promoting research and production of more data, and providing some design guidance, is outlined and its contents summarised. Some of the assumptions and information on which it is based are analysed. Certain problem areas which the author considers need particular attention are briefly discussed.

Its application to vehicle ride quality is considered in the context of the safety, efficiency and comfort of crew and passengers. The importance of establishing the precise criteria against which vibration limits are required is underlined, particularly the difficulties of first defining comfort and then postulating appropriate levels.

Some current and future work related to improving the Standard is outlined and additional suggestions offered.

INTRODUCTION

Problems of ride quality have been with us since transport began, from the ancient coracles and chariots to the more recent aircushion and spacecraft. Vibration is an important, sometimes a dominating feature of the ride environment, and causes undesirable effects ranging from back troubles and other pathological problems (ref.1), contributing to fatal air crashes by impairing pilot efficiency (ref.2), to 'simple' discomfort. Consequently it has been a topic for considerable research, and numerous 'standards' for acceptable vibration levels have been postulated. None of these is universally applicable and none has received widespread acceptance until the recent issue of ISO Standard 2631 (ref.3).

The objects of this paper are, in the context of vehicle vibration requirements, to review briefly the evolution of ISO 2631, to outline its contents and their foundation, and to analyse them. The application of the Standard to vehicle ride quality is discussed. Finally the work proceeding or planned to improve and supplement the Standard is reviewed and suggestions made to fill other important gaps.

This Symposium is concerned primarily with ride quality requirements related to passenger comfort and acceptance. However, the safety and performance of the crew and vehicle are also influenced by ride characteristics, so that all these aspects are covered in this paper.

Apart from certain factual information related to ISO 2631, the paper presents my personal views which are not necessarily those of the ISO Sub-Committee (ISO/TC108/SC4) involved.

EVOLUTION OF ISO 2631 AND GENERAL REMARKS

Work on the Standard officially commenced in June 1964 at the first meeting of Working Group 7 of ISO Technical Committee 108 at Aix-les-Bains, where the first draft proposal,"Classification of the Influence of Mechanical Vibration of Man", was tabled. This copied almost entirely a German specification, VDI 2057 of October 1963 (ref.4) which in turn evolved from the work of Dieckman on 'K Values' (ref.5). It was aimed primarily at defining levels for various strengths of perception of vibration, that is, the response of the body as a load-measuring device. Only tentative examples were given, in an Appendix of the relevance of these curves to subjective tolerance, which was acknowledged to be influenced by important variables other than vibration *per se*. A graph summarizing the proposals, which applied equally to vertical and horizontal vibration, is given in fig.1, with the implied time-dependency of tolerance in fig.2. The final document (ref.3) was published in July 1974, after approval by 19 countries with the UK and USSR expressing disapproval on technical grounds. For comparison, extracts from it are included in figs.1 and 2.

In the metamorphosis of the Standard there were considerable changes, not only in the shape and levels of the 'limits' but also in its coverage and fundamental purpose. Perhaps the most important change has been in the emphasis in the final document, which is absent from the original, that its first purpose is "... to facilitate the evaluation and comparison of data gained from continuing research in this field" and only second "... to give provisional guidance as to acceptable human exposure to whole-body vibration". Another change in philosophy particularly important in relation to this Symposium concerns the original declared scope of the work of the ISO Working Group which was "... with a goal to ensure safety and performance capability of man". 'Comfort' considerations soon began to be discussed and included, but for myself, the Standard still has the flavour of a document aimed mainly at industrial working life exposure, with the "maintenance of proficiency" as the focal point. Levels for the preservation of health and comfort are factored above and below the "fatigue-decreased proficiency boundary" (F-DP). As discussed in more detail subsequently, in my opinion its recommendations should only be used with considerable discretion for design standards, particularly those related to passenger comfort or acceptance.

It appears that the evolution has had to depend on considerable assumptions necessary at the time because of lack of information. The final document may at first sight have the appearance of considerable precision and coverage, particularly if designers turn to the graphs and tables without carefully reading the all-important qualifications in the text. In fact in an authoritative paper (ref.6) on the subject, it is contended that the Standard "relates various human responses to the dynamic motions and exposure time experienced. ... [but] makes no judgment on the permissibility or advisability of the occurrence of these responses in specific situations (e.g., vehicles). It recognizes that to a considerable extent human responses, primarily behavioral and performance effects, depend upon the attitude, motivation, age, experience, and many other biodynamic and psychological factors which characterize the exposure situation...". This important reservation on the applicability of the Standard to vibration requirements is not, however, included in the document itself. Rather, it is an expert interpretation which may well not be applied by the normal, less well-informed user of the Standard.

In fact, as discussed subsequently, human response to vibration is such a complex problem that personally I consider it most unlikely that it will ever be possible to produce clear cut standards covering all situations. Designers, operators, etc., should use 2631 for general guidance. They should explore the many variables in their particular situation and if necessary adjust the proposed levels. This difficulty has been appreciated by the ISO Sub-Committee involved, which is working to fill some of the gaps and to produce addenda for specific applications such as vibration in buildings and in ships.

PRECIS AND ANALYSIS OF ISO 2631

The main contents of the document including the recommended limits and the important supporting text are summarised below. The paragraph numbers in brackets refer to the appropriate paragraphs in ISO 2631. Information in quotes is taken verbatim from the Standard. Important information which may be overlooked in scanning the full Standard is printed in italics. The précis (P) is slightly indented to distinguish it from the analysis (A).

My limited analysis of the Standard is based to some extent on official records and presented some difficulties in preparation. The Standard itself only gives limited information on the logic and evidence on which it is based, and the references included are not specifically cited in the text. The background to the Standard has however already been covered in some depth in a paper (ref.6) at a previous Symposium on vehicle ride quality, and to some extent in an AGARD paper (ref.7). For the sake of completeness my analysis reiterates some of the contents of these earlier papers.

P. (0) "INTRODUCTION"

PURPOSE "*First*, to facilitate evaluation and comparison of data gained from continuing research in this field and *second* to give *provisional* guidance as to acceptable human exposure to whole-body vibration."

OVERRIDING QUALIFICATION "These limits are defined explicitly in numerical terms to avoid ambiguity and encourage precise measurement. *However when using these criteria and limits, it is important to bear in mind the restrictions placed upon their application*."

(1) "SCOPE AND APPLICATION"

("Addenda ... providing modified guide lines for particular applications may be issued from time to time".)

Primarily whole-body vibration applied to standing or seated *man*. *Provisionally* applies to recumbent or reclining man, *not* to local vibration to limbs or head.

1-80Hz, periodic and random or non-periodic vibration.

Criteria	Nomenclature of corresponding limit	Application
"Preserving comfort"	"Reduced comfort boundary"	Passenger (transport) accommodation
"Preserving working efficiency"	"Fatigue-decreased proficiency" (F-DP) (previously entitled "Fatigue time-limit of decreased proficiency")	Vehicle driver or machine operator
"Preserving safety or health"	"Exposure limit"	(Not declared, assumed to apply to any situation.)

Population cover "... people in normal health: that is persons who are considered fit to carry out normal living routines *including travel* and to undergo the stress of a typical working day or shift."

A. The criteria are simplified generalisations and have important subdivisions which considerably influence the appropriate limits, for example the nature of the task and physical, psychological or 'activity' discomfort.

Reaction to vibration varies widely between individuals and individual groups. A more specific definition of population cover and limits for particular populations is ultimately needed. (See subsequent proposals for vibration below 1Hz.)

Average reaction of a group (and most of the evidence for the limits seems to be based on average results on fit young men) may be less relevant than reaction of particular individuals. The proposals do not necessarily apply accurately to women and certainly not to children or old people.

P. (3) "CHARACTERISATION OF VIBRATION EXPOSURE"

DIRECTION: Linear vibration only, using an orthogonal system related to major body axes:-

a_z , foot-to-head (longitudinal, popularly 'vertical' for standing or seated man)

a_x , chest-to-back (fore and aft)

a_y , side-to-side (lateral)

"Angular vibrations ... are frequently an important part of a vibration environment. For example ... the pitching or rolling motions of the seat may be more disturbing than the rectilinear vibrations. However little information on the effects of angular (or the rotational) vibration is yet available." In practice, the centre of rotation can often be assumed to be far enough from the body for the resulting motion to be represented by linear vibration alone. The Standard requests that, wherever practical, data on angular vibration should be recorded to increase knowledge.

The limits given in the Standard "... should be regarded as very tentative in the case of vibrations having high crest factors ...".

INTENSITY: *Primary quantity shall be acceleration* in m/s^2 rms $\left(\text{or } g = \frac{m/s^2}{9.81}\right)$ *measured at entry into body itself.*

If peak values are measured, convert to rms. For random vibration *crest factor* (peak/rms) must be determined. *Limits given can only be applied very tentatively if crest factor exceeds 3.*

A. Relatively little field or laboratory work has been carried out on angular vibration. Limits will be particularly difficult to define because of measurement problems and the fact that reaction will be critical to the position of the centre of rotation in relation to the body.

To define the input completely, strictly speaking another linked parameter such as force or impedance is needed. As an extreme example two surfaces may vibrate (accelerate) together with little or no force or interaction between them. This may be important with regard to the effects of posture, arm and foot rests, harnesses, etc. In many situations however, acceleration alone is probably an adequate descriptor of the vibration input, particularly in view of the many other variables involved.

'Crest factor' is not precisely defined, particularly the duration over which the peak/rms ratio should be measured. Ride comfort in certain situations will depend on reaction to 'jolts and bumps' with crest factors exceeding 3. There is little or no guidance for such situations either in this Standard or elsewhere. (ISO/TC108/SC4 has recognised and is endeavouring to fill this important gap. Also, a draft Standard is in preparation for desirable limits for large single shocks (covering accidents, etc.).)

P. (4) "VIBRATION EVALUATION GUIDE"

"FATIGUE-DECREASED PROFICIENCY BOUNDARY" (F-DP) Beyond this boundary "vibration can be regarded as ... carrying a significant risk of impaired working efficiency in many kinds of tasks, particularly those in which time-dependent effects ('fatigue') are known to worsen performance, as for example in vehicle driving".

The limits are expressed as rms acceleration versus frequency for exposure times from 1 minute to 24 hours and are summarised in figs.3 and 4 for longitudinal and transverse (fore and aft and lateral) vibration respectively. The recommended proportional reduction in permissible vibration with time is shown as the curve in fig.5.

The limits are for general guidance only and the value applying t a particular situation"... depends on many factors including individual factors as well as the nature and difficulty of the task ... a more stringent limit may have to be applied when the task is of a particular demanding perceptual nature or calls for the exercise of a fine manual skill. By contrast some relaxation of the limit might be possible when ... the performance of the task (for example, heavy manual work) i relatively insensitive to vibration ... tentative data ... suggests tha a *range of correction of +3dB to -12dB (that is ... 1.4 to 0.25 times t rms acceleration specified by the boundary)* may be envisaged." A graph illustrating this range is given in fig.6.

"EXPOSURE LIMIT" This limit, summarised in figs.3 and 4, is of the sam shape as the F-DP boundary but set at twice the level (6dB). The limit is stated to be set at approximately half the level of the threshold of pain, or limit of voluntary tolerance obtained from laboratory studies (*men*.

"REDUCED COMFORT BOUNDARY" This boundary, summarised in figs.3 and 4, : *assumed* to follow the same shape as the F-DP boundary but at approximate one third of its level (-10dB). *"In the transport situation the reducec comfort boundary is related to the difficulties of carrying out such operations as eating, reading and writing."*

The boundary is qualified by the following statement, the significance of which is illustrated graphically in fig.7. "It is anticipated that additional tables will be developed ... for a finer differentiation of comfort in various situations, such as in offices, in various types o private residence, *on ships*, etc. *The range of such correction factors might extend from +3dB (1.4 ×) to -30dB (1/30) (the approximate threshol of perception).*"

"NOTE ... it should not be taken as implying that there exists in all circumstances a simple hierarchical relationship between the intensities of vibration likely to impair health, working efficiency or comfort."

A. The shape of the acceleration/frequency curves is the same for all three criteria and is based on the assumption that the overriding influence on human response to vibration is due to the biomechanical response of the body and bod parts. This contention is supported by empirical evidence from laboratory and field research, mainly on young men.

More specifically, for the a_z (longitudinal) direction the trough in the acceleration/frequency limits assumes a minimum in driving point impedance between 4 and 8Hz, that is a major resonance of the human body in this frequency region. The corresponding minimum between 1 and 2Hz in the a_x and a_y (transverse) directions assumes minimal impedance and a major (shoulder girdle) resonance in this region. The increasing slope at 45° on the log/log scale of the a/f curves above 8 and 2Hz respectively, implies that if the body behaves as a linear mass/spring/velocity-damped system, then the response is directly related to the total force acting on the mass, that is, to the total input force.

The decreasing slope in the a_z/f curves between 1 and 4Hz is proportional to $a/\sqrt{f}$ ($\frac{1}{2}$ on log/log scale), which is a compromise between the original VDI proposals (ref.4) of a constant a_z and an alternative suggestion that it should be proportional to $1/f$ (1:1 on the log/log scale).

The shape is a simplified generalisation which may be a reasonable approximation in certain specific cases. It is open to considerable adjustment, firstly because the body, and the associated response to vibration, frequently do not behave as a single order mass-spring-system. There are several important sub-systems, head-on-shoulders, spine, etc., usually with resonant frequencies higher than the dominant ones (*circa* 5Hz and 2Hz for a_z and a_x, a_y directions respectively) and for certain applications these may modify the shape of the curves particularly above 8Hz (a_z) and 2Hz (a_x and a_y). Also the a_z shape between 1 and 4Hz fails to reflect the likely peak in tolerance around 1½ to 2Hz, attributable perhaps to evolutionary acclimatization to walking frequency (ref.8). The criteria of acceptability may be founded on one or more of the several possible response characteristics of the body system. These could range from absolute displacement, relative displacement, applied force, absorbed power (heat) in the total body system or a sub-system, to force in a particular body sub-system. It seems likely that the basic responses controlling safety, efficiency and comfort will differ even for a particular situation. For example safety (preservation of health) is likely to be dominated by the load or force in particular body parts whereas performance may be dependent on relative and/or absolute displacement. This emphasises the importance of the qualification in the Standard concerning the simple hierarchical relation between exposure, F-DP and reduced comfort limits.

The shape and acceptable levels are affected by many variables not at present covered in this Standard. Some of these are briefly considered subsequently. Apart from possible variations due to different biodynamic criteria (above) it is conceivable that the appropriate shape may differ between a short exposure, a long casual exposure and a repeated working life exposure.

With regard to the actual levels specified, the Standard informs us that:-

(i) The F-DP boundary is based on data "mainly from studies on aircraft pilots and drivers".

(ii) The exposure limit is "set at approximately half the level considered to be the threshold of pain (or limit of voluntary tolerance) for healthy human subjects ..."

(iii) The reduced comfort boundary "is derived from various studies conducted for the transport industries".

Apart from the time-dependency, no specific variations in the suggested levels for the three criteria are suggested. The possible wide variation needed to cover specific situations is acknowledged in the tentative correction factors of +3 to -12dB for F-DP and +3 to -30dB for reduced comfort. The much wider range for reduced comfort is presumably because this reaction is more susceptible to psychological influences than is reduced proficiency.

It appears (ref.9) that the a_z acceleration levels and the +6 and -10dB hierarchical relationships between "exposure limit" and "F-DP" and "reduced comfort" and "F-DP" are, for durations between 1 and 100 minutes, based largely on a survey by Notess (ref.10). My own plot, in fig.8, of the Notess data supports the indications from a previous survey (ref.11) that for short exposures the F-DP and reduced comfort levels are set on the high side, certainly for random vibration. This may explain the negative bias in the tentative correction factors (-12 and -30dB as against +3 and +3dB for F-DP an reduced comfort respectively).

The difficult problem of variation in levels with duration of exposure i discussed below and that of population effects has already been mentioned.

P. "EVALUATION OF FREQUENCY SPECTRUM" The preferred method of evaluation i to compare the acceleration level for single or multiple (discrete) frequencies or for 1/3rd octave bands, separately against the recommende level at each frequency or 1/3rd octave band centre frequency. *This procedure assumes "that in respect of human tolerance no significant interactions occur between the vibration effects of different frequencie but states that there is no published evidence to decide between the accuracy of this preferred method and the suggested simplified alternati weighting and summation procedure (below).*

Under "NOTES" an alternative method of evaluation is described "to allow the characterisation of a vibration environment ... by a single quantity and to simplify measurements for situations in which spectrum analysis is difficult or is inconvenient". The overall vibration signal between 1 and 80Hz is weighted by an electronic network which adjusts ea 1/3rd octave band level to the equivalent of the 4-8Hz level for longitudinal (a_z) and the 1-2Hz level for transverse (a_x, a_y) vibration. The *by implication* these weighted 1/3rd octave levels are summed to give one overall rms level (analogous to the dBA overall weighted level for noise This level is then compared with the permissible value in the 4-8Hz band for a_z and 1-2Hz band for a_x, a_y . The Standard declares (in my opinion not necessarily correctly for all applications) that this method "results in an over-conservative assessment of the effects of vibration ... for a vibration spectrum closely following the shape of th limits the summated level is 13dB (4½ ×) higher than for the preferred worst single frequency or 1/3rd octave band method". It does not clearl point out that the summation method may be the more accurate if, as discussed subsequently, for a multi-frequency input the conditions are such that human reaction is caused by an integrated effect rather than b response to one particular frequency or 1/3rd octave band.

MULTIAXIS VIBRATION For vibration occurring in more than one axis simultaneously it is recommended that "the corresponding limits apply separately to each vectorial component in the three axes" (therefore it i assumed that there is no interaction between the axes).

A. For complex single axis vibration, the Standard implies that reaction is dominated by the vibration at one frequency or in a single 1/3rd octave band, that is there is little or no interaction between different frequencies. An alternative method of weighting and integrating the component parts to give one characteristic number is suggested but it is implied that this is mainly to simplify data measurement and analysis and stated that it "results in an over-conservative assessment of the effects of vibration".

For multiaxis vibration it is recommended that each axis should be evaluated separately.

Since, as discussed previously, the acceleration/frequency contours are based on biodynamic response, then some integrated effect in reaction is to be expected for a complex vibration. Therefore, the levels obtained by the second method may be the more accurate and not 'over-conservative'.

Surprisingly, little clear cut evidence exists to elucidate this important point, for performance, comfort or safety criteria. Work at ISVR, Southampton University indicates that for short duration sensation at least, the integrating method is the more accurate, although the shape of the contours is by no means correct for all individuals.

For multiaxis vibration also, biodynamic considerations supported by some laboratory work (ref.12), suggest that some interactions will occur and that a summation method may therefore be more appropriate.

P. DURATION OF VIBRATION The tolerable acceleration level is assumed to decrease with increasing exposure time from 1 minute up to 24 hours, that is a daily permissible dose, as illustrated in fig.5. This relationship applies "... *when the exposure is repeated daily over many years, for example for an industrial worker ... or for a transport driver. For exposure which is much less frequently experienced, for example by the casual traveller the acceptable exposure ... may well be higher*".

This time relationship applies to continuous exposure or (without mitigation for recovery) to intermittent exposures. A fractionating method of summing for exposure times at different amplitudes or frequencies is given.

A. This is perhaps the most important yet least substantiated part of the Standard. The limited supporting evidence used (refs.6, 9, 10, 13) dates back to 1956. It covers frequencies of about 1Hz only and some of it apparently consisted of people's *estimates* based on short exposures, rather than on actual *experience* of prolonged vibration. More recent investigations (refs.14 and 15) indicate that for casual exposures at least, any performance decrement due to vibration does not get worse with time, at least for exposures up to 3 hours.

In fact, two different time-dependency relationships may apply. The first (which should perhaps be asymptotic to the horizontal at 24 hours to cover continuous exposure in ships, etc.) would safeguard health against repeated exposures over many years, that is provide a cumulative working life "exposure limit". In this connection the work in hand by NIOSH (refs.16 and 17) could produce information concerning the validity of the present curve. The second shape, for F-DP and reduced comfort boundaries, and applying to both working life and casual exposures may well be much flatter than the present curve.

Lastly, the Standard does not give the precise method of evaluating the acceptability of a complex long duration exposure, where the level is varying continuously. The whole question of defining, measuring and calculating the vibration dose would be considerably simplified if an energy relationship (dose $\propto a^2t$) were adopted, as suggested in fig.5. This would enable a vibration dose meter to be used, analogous to a noise dose meter. This problem is discussed further in the next section.

Here, there are two basic considerations. The first concerns the reaction of the vehicle occupants to vibration *per se*, the second the significanc of this reaction in the overall acceptability of a particular means of transport. This paper concentrates mainly on the first aspect, although the secon which has already been considered fundamentally in a paper at a previous ride quality symposium (ref.18) and elsewhere (ref.19), is particularly relevant to vibration requirements for the comfort of passengers.

As discussed in more detail elsewhere (ref.11), human reaction to whole-body vibration (HRV) is very complex, but can be represented by the following qualitative equation:-

$$HRV = f(V,G,E,Ph,Ps,Ad,Ac)$$

where V	= Vibration input	physical, largely extrinsic factors
G	= Geometry of seat and other interfaces	
E	= other Environmental inputs (noise, etc.)	
Ph	= Physiological influences (health, biorhythms, etc.)	physiological factors
Ps	= Psychological influences (mental state, motivation, experience, expectation, etc.)	
Ad	= Adaptibility (posture etc.)	behavioural factors
Ac	= Activity (driving, etc. for crew; speaking, talking, drinking, eating, etc. for passengers)	
f	= 'function of'.	

Unfortunately but not unexpectedly, ISO 2631 and in fact all the availat laboratory and field data cover and quantify only a small part of this equati(In my view if 'HRV' is likely to be critical in any particular transport situation, it is necessary to explore this situation in depth against the bacl ground of the ISO *guide* and other information, and perhaps to conduct laboratory and field studies, before realistic limits, particularly regarding proficiency and comfort, can be postulated.

The significance of 'HRV' and any limits associated with it will vary considerably with the different sectors of the transported population and the criteria employed to judge acceptability. For this purpose the population car conveniently be divided into four main groups:-

(i) Drivers, pilots, seamen, etc. and other crew directly or indirectly responsible by their actions or their health for the safety of the vehicle anc its occupants. For people in this group it is essential to ensure that vibration does not significantly impair their performance or by its immediate or cumulative effects, their health. The (safe) exposure limit and the F-DP boundary are particularly relevant, as are the physical and physiological factors in the equation above. Psychological influences are likely to have little effect on these limits.

(ii) Other vehicle crew such as cabin crew. For these, safety considerations will be less important but long-term health effects should be considered and vibration interference with activities such as serving food and drink. Again exposure limit and F-DP are the prime considerations.

(iii) Regular passengers such as commuters who may travel day after day, year after year. Here, long-term health (exposure limit) and comfort considerations (reduced comfort boundary) are important. Psychological influences, expectancy, experience, etc. may well predominate in the latter, but the purely vibration aspects of these will probably be subordinate to the broader question of overall acceptability.

(iv) Passengers making occasional business, social or pleasure trips. For these, comfort considerations (reduced comfort boundary) will predominate, with psychological influences playing a major part.

The preceding remarks have demonstrated the complexity of deriving realistic vibration ride quality limits. In this paper it is only possible to suggest a philosophy, the basic principles involved in such derivations. A simplified hypothetical example for aircraft ride is chosen to demonstrate this, using the recommendations in ISO 2631 with certain indicated adjustments which are considered justifiable.

Example

An aircraft is to be designed for use on routine flights from A to B and return, of 2 hours each way. It will carry a flight crew, cabin crew and passengers comprising regular commuters and casual travellers. What are the desirable maximum vibration levels for the occupants?

Solution

Maximum vibration levels must be checked against the three criteria of:-

(i) Health of crew and regular travellers.
(ii) Efficiency of operators and hence safety of occupants.
(iii) Comfort of passengers and crew.

In order to postulate limits against these criteria it is assumed that:-

(i) The operators will work an 8 hour day for a number of years. The ISO 8 hour exposure limit should therefore be applied to prevent any possibility of cumulative effects of vibration on health.

(ii) Based on published evidence to date (refs.14 and 15) the pronounced time-dependency in ISO 2631 is unlikely to apply directly to proficiency. The 2 hour F-DP level is assumed to be more appropriate than the much lower 8 hour F-DP level which should theoretically be applied.

(iii) The reduced comfort limit is also considered to be less dependent on exposure time, and the 2 hour reduced comfort level is assumed to apply even to travellers who make a return journey on the same day. The comfort limit is probably the most difficult to quantify. As indicated previously in this paper and discussed elsewhere (e.g. ref.18) it is affected by many variables other than by vibration *per se* and should be considered as only part of a wider *journey* acceptance criteria, that is, door-to-door satisfaction. An arbitrary level has been selected to facilitate comparison with the other limits, and in real life may well need considerable adjustment either way.

These three limits, including the alternative and more conservative 8 hour F-DP level are plotted in fig.9 as 'acceptable' vibration in rms m/s^2 against daily exposure time. The acceptable level can either be expressed as the 'worst third octave' normalised to 4 to 8Hz or, probably more accurately, as one weighted, summed value. The levels which 'must' not be exceeded on safety or health grounds together with those which should preferably not be exceeded to ensure passenger comfort are indicated on the graph.

This is a very simplified treatment of a particular problem in which a steady vibration level has been assumed for each and every daily journey. In practice vibration level and frequency will vary considerably and probably in somewhat random fashion. ISO 2631 is not explicit on how to sum a long duration complex waveform but it is implied that, assuming the ISO time-dependency curves do apply, the following procedure will be necessary. This is given for the simplified weighted summation method and will be further complicated if the preferred 'worst third octave' method is to be applied:-

From the taped or calculated record of weighted summed acceleration level versus time a histogram is constructed relating the various times t_1, t_2, etc. spent in various narrow bands of acceleration levels with centre amplitudes A_1, A_2, etc. having corresponding permissible exposure times of T_1, T_2, etc. The vibration is acceptable if

$$\sum \frac{t_1}{T_1} \leqslant 1 \quad .$$

As previously reasoned and assumed in the above example, the ISO time-dependency relationship may not apply to all three criteria. However to maintain driver proficiency a given maximum level must not be exceeded at any critical period. This raises the practical question of the likely short duration increases above the normal desired maximum, for example vibration on a rough runway or during severe turbulence. For this, each case must be considered on its own merits with a guiding principle that if an increase in vibration however short-lived, occurs simultaneously with a vital vehicle control activity, then it must not cause decrement in performance sufficient to impair safety or cause gross discomfort to the occupants.

The practical significance of temporal variations in vibration with reference to the desired comfort level is probably even more complex and is well beyond the scope of this paper. Also, the ride requirements to prevent severe discomfort and injury to passengers and crew due to sudden encounter with severe turbulence have not been considered. Fortunately such encounters are rare, but still enough to cause a significant number of severe injuries every year. The general problem of the ride requirements concerning repeated shocks is briefly considered in the next section.

It is appropriate to underline here some of the problems of relating the ISO reduced comfort levels to the vibration requirements for passenger comfort, problems evidenced by the qualifications concerning the possible wide variation in the levels which ISO 2631 has wisely included. Briefly as illustrated diagrammatically in fig.10 there are at least three different kinds of discomfort reaction. Each of these may be provoked by a widely different level, which in turn will vary with the type of transport and population covered. The first reaction is that due to direct physical or physiological disturbances and will usually only be provoked by relatively high vibration levels. The second is that due to psychological or mental disquiet caused by vibration rising significantly above the level normally to be expected, thus engendering feelings of apprehension or alarm. The corresponding acceptable vibration level may well be very low in some transport such as cable cars, in fact little above perception! The third and perhaps the most common type of discomfort is that caused by interference with activity. Work by Brumaghim (ref.20), suggests that sensitive activities such as reading will tend to lower the discomfort reaction level.

Unfortunately, much vibration 'comfort' research has been concerned with very short duration, 'sensation' effects with no subject activity and the results cannot therefore be applied directly to real life situations. Surprisingly, little work seems to have been conducted on the effect of vibration on passengers' ordinary activities such as reading, writing, eating, drinking, thinking and sleeping, and even less on the effects of prolonged exposure. In an investigation (ref.14) at RAE eight subjects were subjected to four 3 hour sessions of vertical vibration at 5Hz and $1.2 m/s^2$ rms, that is, the one hour F-DP or the 3 hour exposure limit, whilst carrying out various tasks including writing. Although, on average, performance was immediately degraded by the vibration, there was no evidence of significant worsening with time. Also whereas subjects initially considered the vibration to be extremely uncomfortable they seemed to adapt to it and several spontaneously remarked that as time went on they "almost forgot about it".

POSSIBLE FUTURE IMPROVEMENTS RELATED TO ISO 2631

In the Standard itself some of the gaps are acknowledged and it is stated that "addenda ... may be issued from time to time". Some of the shortcomings have already been discussed in this paper. Planned or desirable modifications or additions are briefly outlined below. For completeness, modifications to 2631 and work on associated standards which may be only indirectly related to ride quality, are considered. Some of the improvements can be formulated from existing data, but most will need more information from laboratory and field experiments. The list may therefore serve as a useful guide for those evolving future research programmes.

Agreed necessary by ISO/TC108/SC4, proposals already drafted

(1) An addendum to 2631 covering exposure to vibration below 1Hz has been drafted. This will fill an important gap, since in several forms of transport there is much energy in this region which causes a most undesirable reaction, motion sickness. The proposals, based on a critical literature survey (refs.21 and 22) and aimed at preventing motion sickness in 90% of adult casual travellers are summarised in fig.11. This shows the very sharp frequency dependence of reaction and indicates the particular need to minimize vibration in the 0.1 to 0.3Hz range. Tentative "reduced comfort" limits are shown in fig.12. These are for exposures up to 4 minutes only, there being an almost complete lack of data for longer periods. At this lower level, 'comfort' reaction, at least for short exposures, is much less frequency dependent than the motion sickness reaction.

(2) A "Guide to the evaluation of human exposure to mechanical shock" is aimed at defining acceptable limits for a sudden and violent (accident type) shock and will eventually be issued as a separate standard.

(3) A "Guide to the evaluation of hand-transmitted vibration" is well advanced and should be issued as a separate standard in the next year or so. Although aimed primarily at minimisation of the occupational risk ('white fingers') of continued exposure to vibration from chain saws, vibrating tools, etc., it is relevant to vibration from steering wheels, handlebars, etc. in transport.

(4) An addendum to 2631: "Vibration and shock limits for occupants in buildings" is well advanced. This gives various weighting (reduction) factors to the acceleration levels in 2631 for different types of buildings, and is relevant to traffic-induced vibration.

Agreed necessary by ISO, some work commenced

(1) An addendum to 2631 defining acceptable vibration levels in ships is planned. Data are being collected mainly by Japan.

(2) Information is being collected, aimed ultimately at providing recommended limits for human exposure to repeated ('low level') shocks and vibration with crest factors greater than 3. This will cover an important gap in ride quality (rough ride) requirements for many forms of transport. A possible approach to this is shown in fig.13.

(3) Information is being collected on the transmission of vibration through the body with particular reference to the effects of posture, seat and harness design. Weighting factors may eventually be introduced into 2631 to cover these effects. For example harness tends to attenuate main body (low frequency resonances but to amplify higher frequency (head, shoulder) vibration.

(4) Basic information on body impedance, analogues, etc. is being prepared.

(5) The tentative nature of the time-dependency curve in 2631 has been recognised and it will be reviewed when more data are available.

(6) Work has commenced on the problem, for complex spectra, of 'worst frequency or 1/3rd octave level' versus 'summed, weighted assessment'. As previously discussed some recent evidence suggests that the latter method may be more accurate.

Other suggested improvements

The following list is based on experience as a research worker on aircraft and other vibration effects. It is by no means comprehensive and reiterates some of the points made in the above analysis of 2631.

(1) More specific definitions of *criteria*, the first essential for progress on better *limits*, particularly for ride quality, passenger comfort considerations.

(2) A better definition of population cover and/or limits for specific populations. This needs more laboratory and field information on individual rather than average response of men, women and children.

(3) Adjustment to the shape of the acceleration/frequency curves. Different shapes may be needed for different criteria (safety, proficiency and comfort) and for sub-divisions of these.

(4) The gap in angular vibrations in the present specification requires filling as soon as better information is available.

(5) A better definition of 'crest factor' is needed, together with guidance in human response to vibrations with high crest factors (linked with (2) in previous list).

(6) Adjustments (weighting factors) are needed to the F-DP and reduced comfort boundaries, against more specific criteria, and hence an elaboration of the present suggested +3 to -12 and +3 to -30dB variations in recommended levels.

(7) The appropriate methods of evaluation of complex single or multiaxis vibration need to be defined for various applications.

(8) The whole question of the time-dependency of acceptable acceleration levels requires to be reviewed. As an interim measure, a constant energy relationship in place of the present shape, would seem more plausible and would considerably simplify analysis.

CONCLUDING REMARKS

ISO Standard 2631 should make an important contribution to our understanding and alleviation of the unwanted effects of vibration on man, particularly if its many qualifications are heeded. Firstly it provides a common basis for the gathering, analysis and comparison of field and laboratory information, and secondly it provides some design guidance which will become more and more useful as the Standard is improved. It has important implications for legislators, operators and research workers alike: this should stimulate work to explore further some of its controversial proposals.

The production of the Standard has been hampered by the lack of suitable data and the long time involved in its preparation and approval. Thus it is already in need of some updating and refinement. However in view of all the considerable technical, administrative and other difficulties which had to be surmounted, the final document, which has been approved by the great majority of the countries involved, represents a considerable achievement.

It is hoped that this paper will be useful to those involved in vehicle ride quality, firstly by helping them to understand, interpret and apply what is inevitably a complex standard. Secondly it may encourage research which will assist current efforts to improve the Standard. Specific topics which it is considered need especial attention have been discussed and most are in some way applicable to ride quality. The comments concerning the need for more realistic comfort tests, including individual not just average response, are particularly relevant.

ACKNOWLEDGMENTS

I should like to acknowledge the considerable help received from my colleagues at RAE and elsewhere, particularly Dr. G. Bobbert, in preparing thi paper. Also ISO 2631, and hence this paper, would probably never have emergec but for the stimulating efforts of Dr. H. von Gierke.

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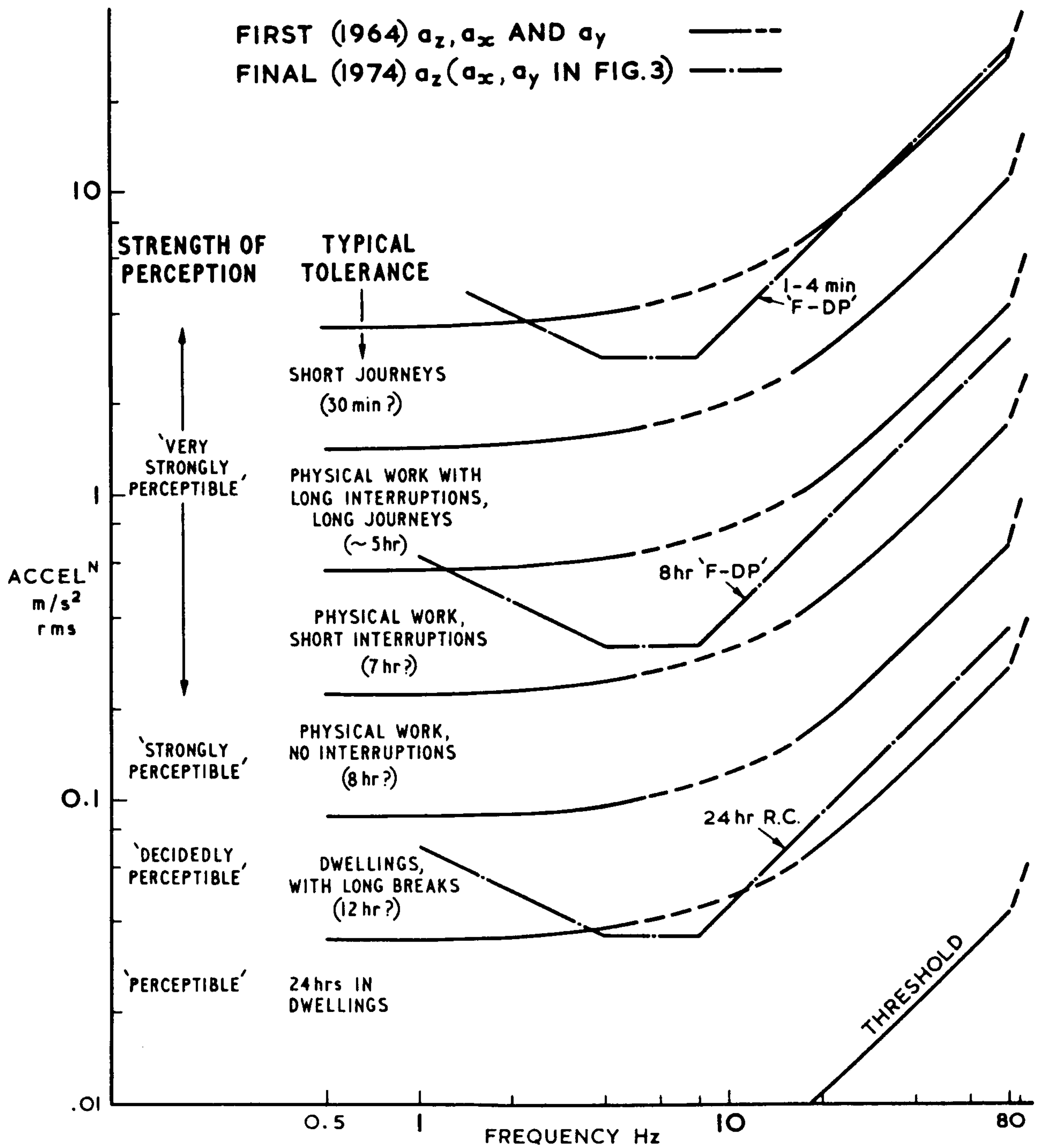

Figure 1.- Evolution of ISO 2631: Acceleration/frequency curves.

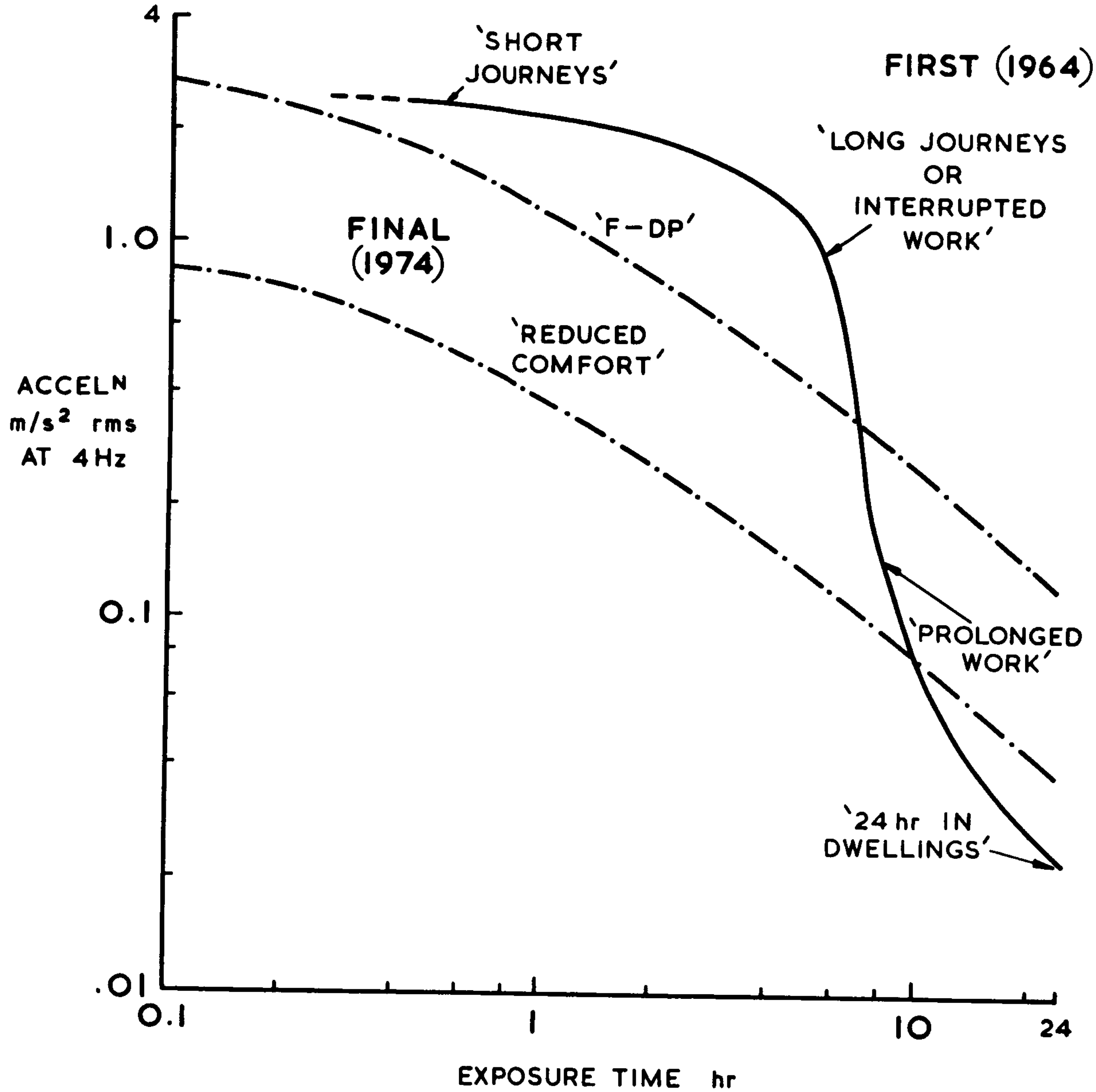

Figure 2.- Evolution of ISO 2631: Time-dependency.

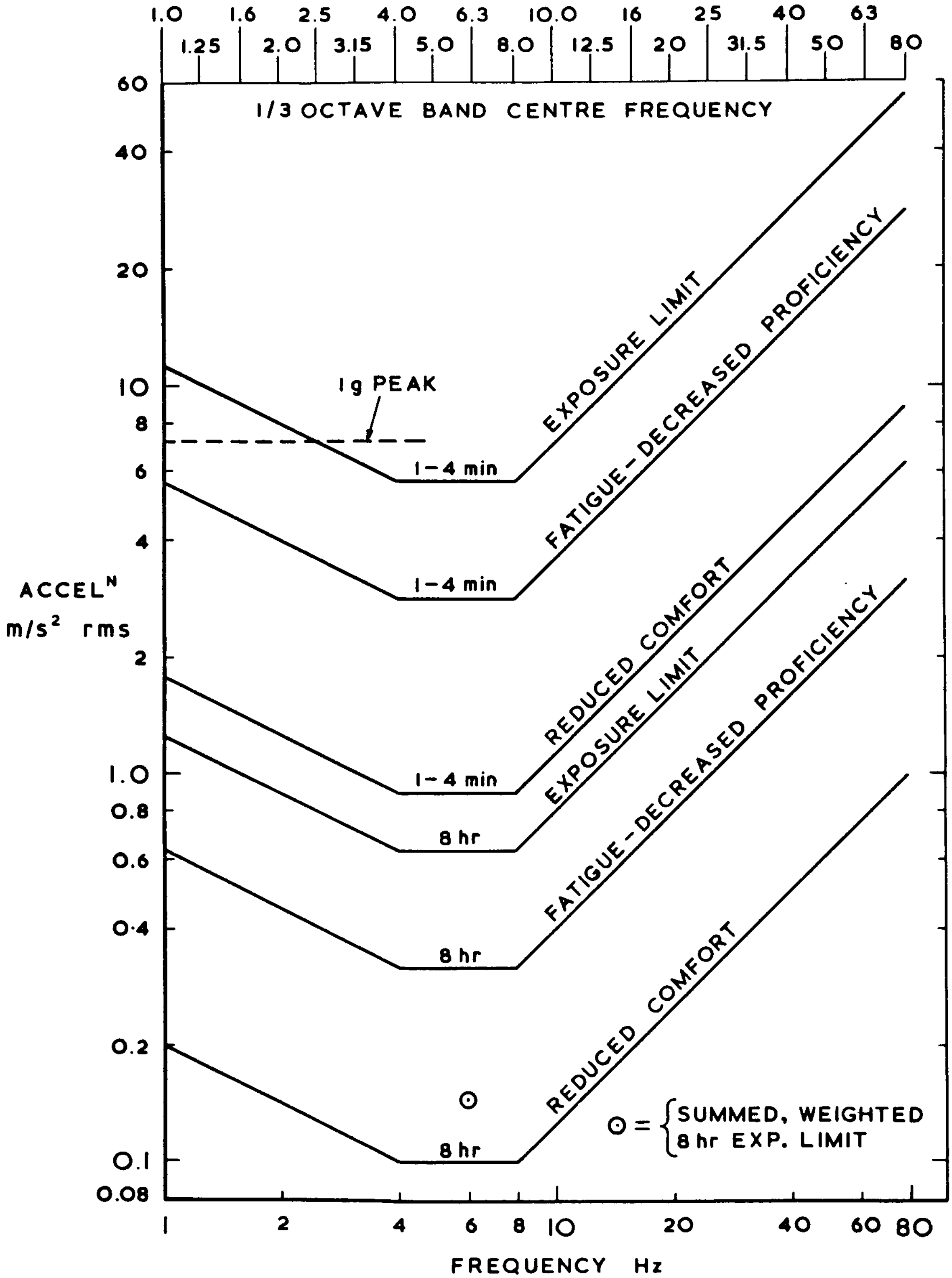

Figure 3.- ISO 2631: Acceleration/frequency curves for longitudinal, a_z, axis.

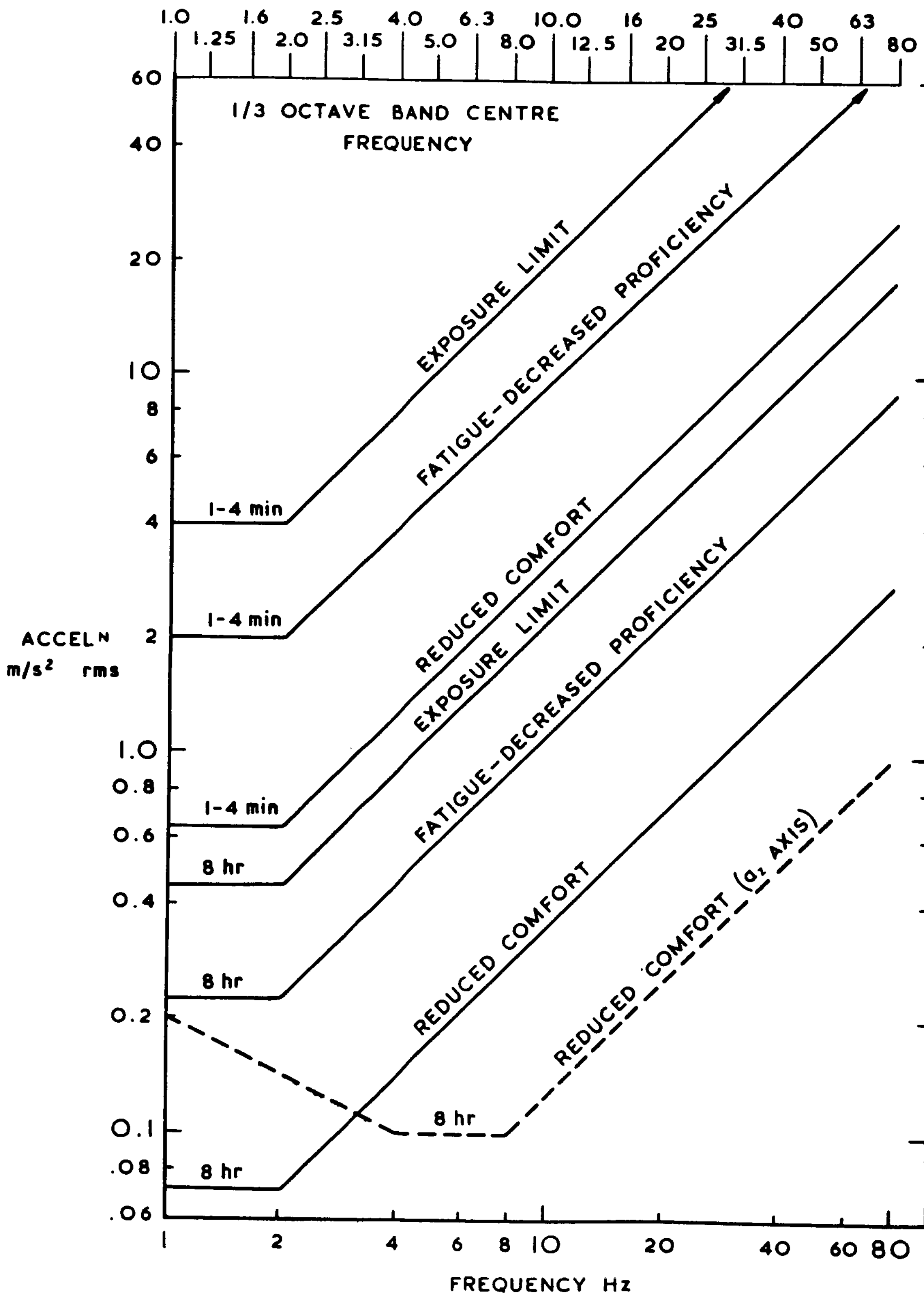

Figure 4.- ISO 2631: Acceleration/frequency curves for transverse, a_x, a_y, axes.

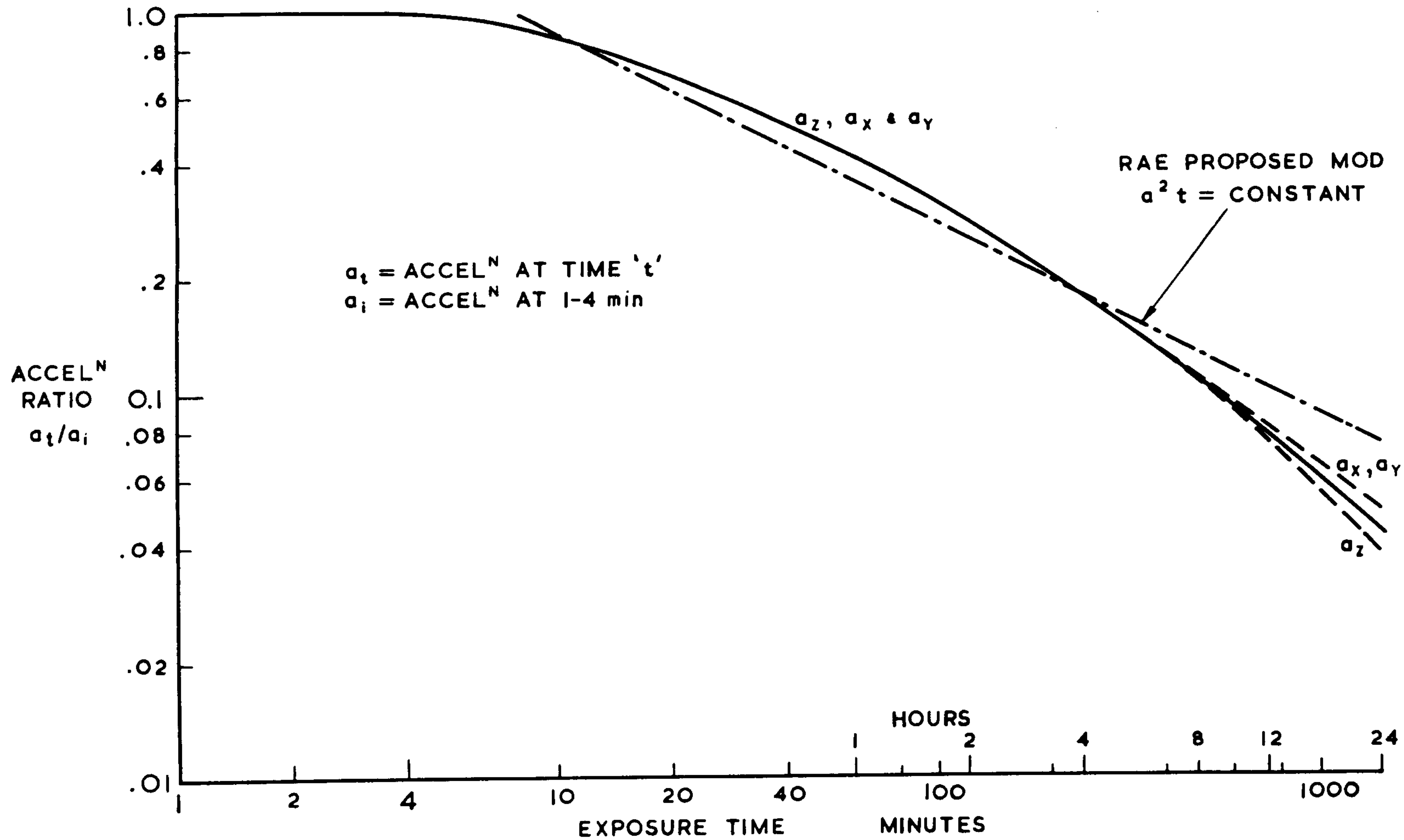

Figure 5.- ISO 2631: Acceleration time-dependency.

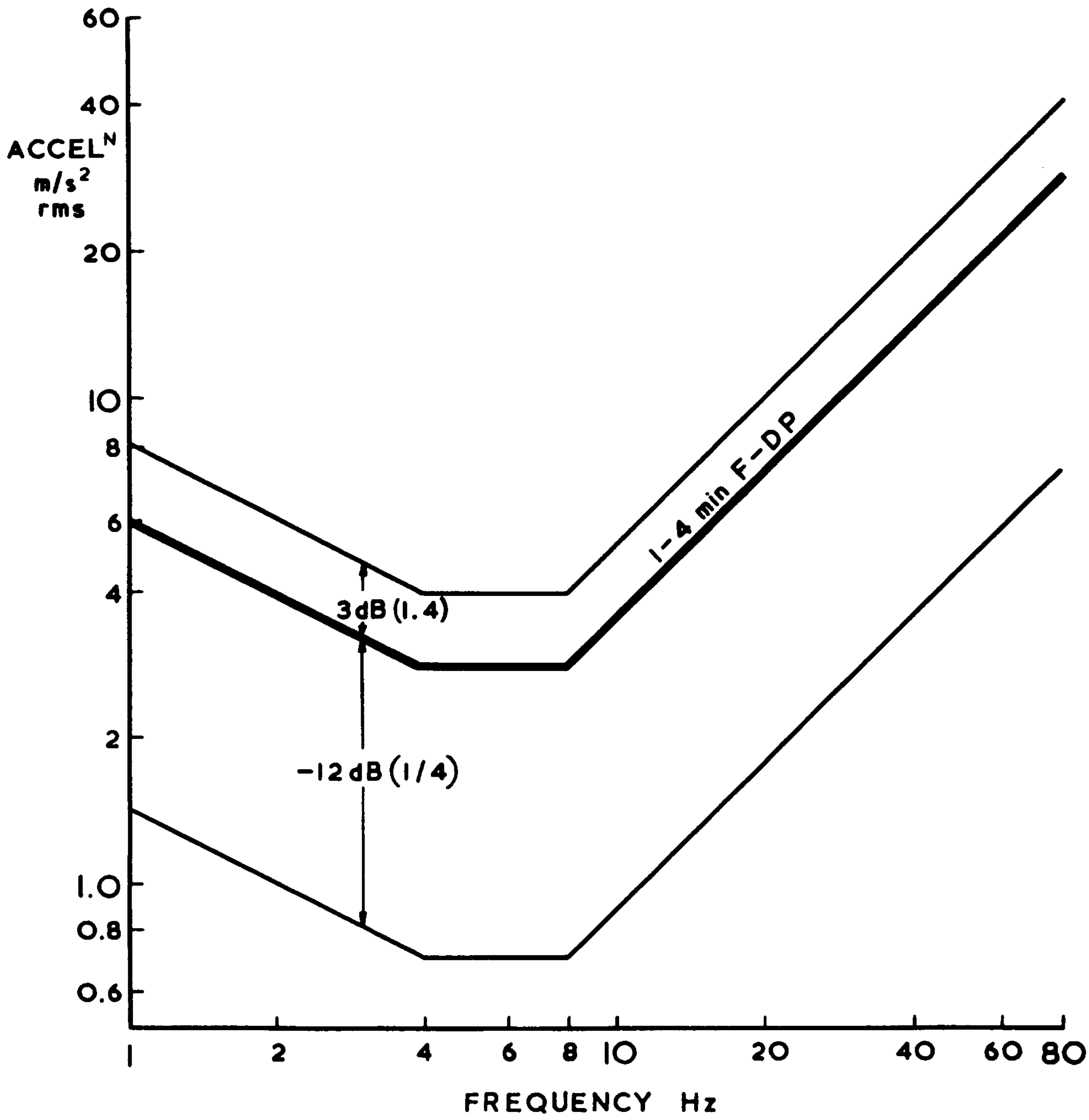

Figure 6.- ISO 2631: Tentative correction factors (a_z, F-DP).

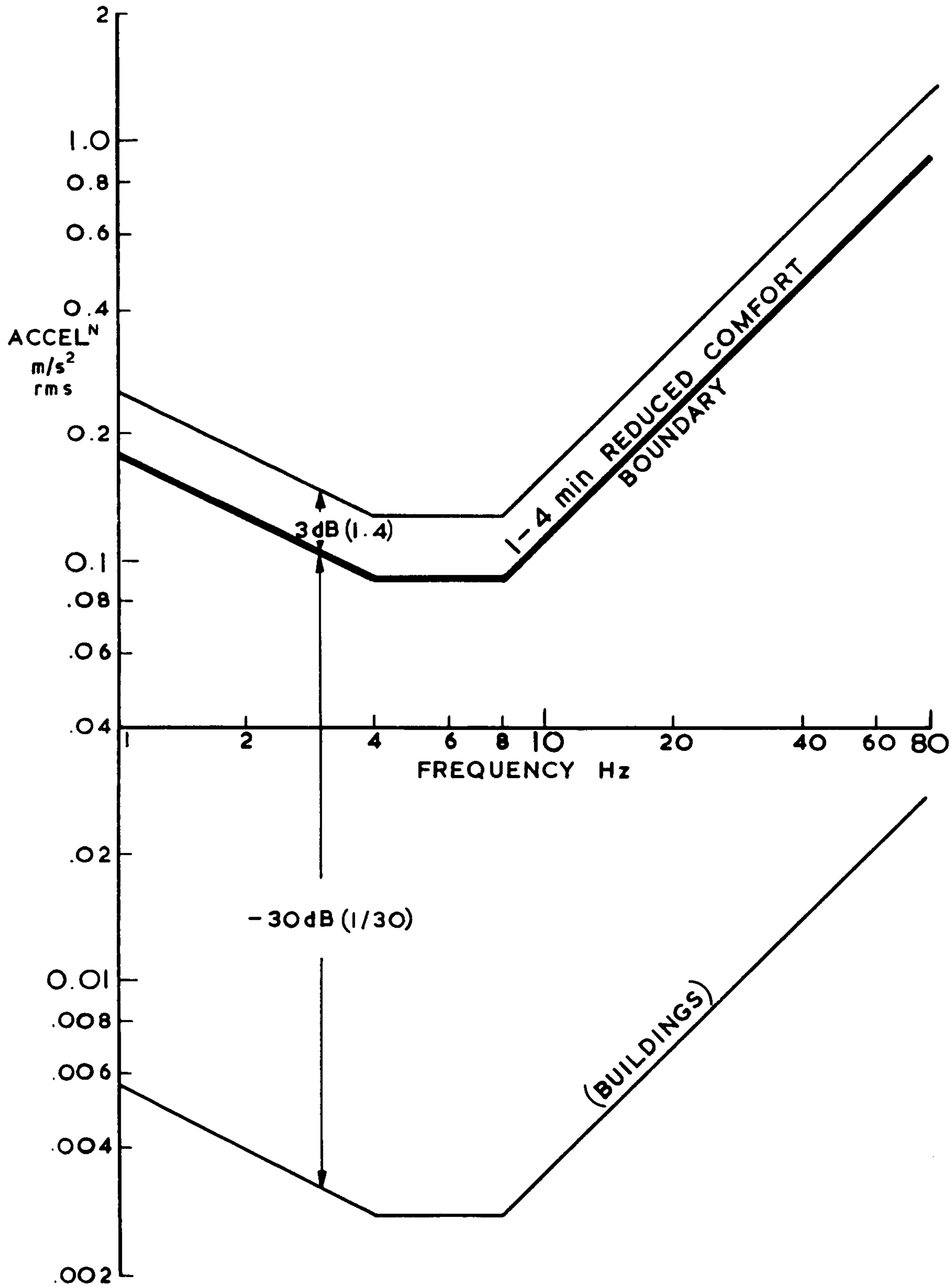

Figure 7.- ISO 2631: Tentative correction factors (a_z, reduced comfort).

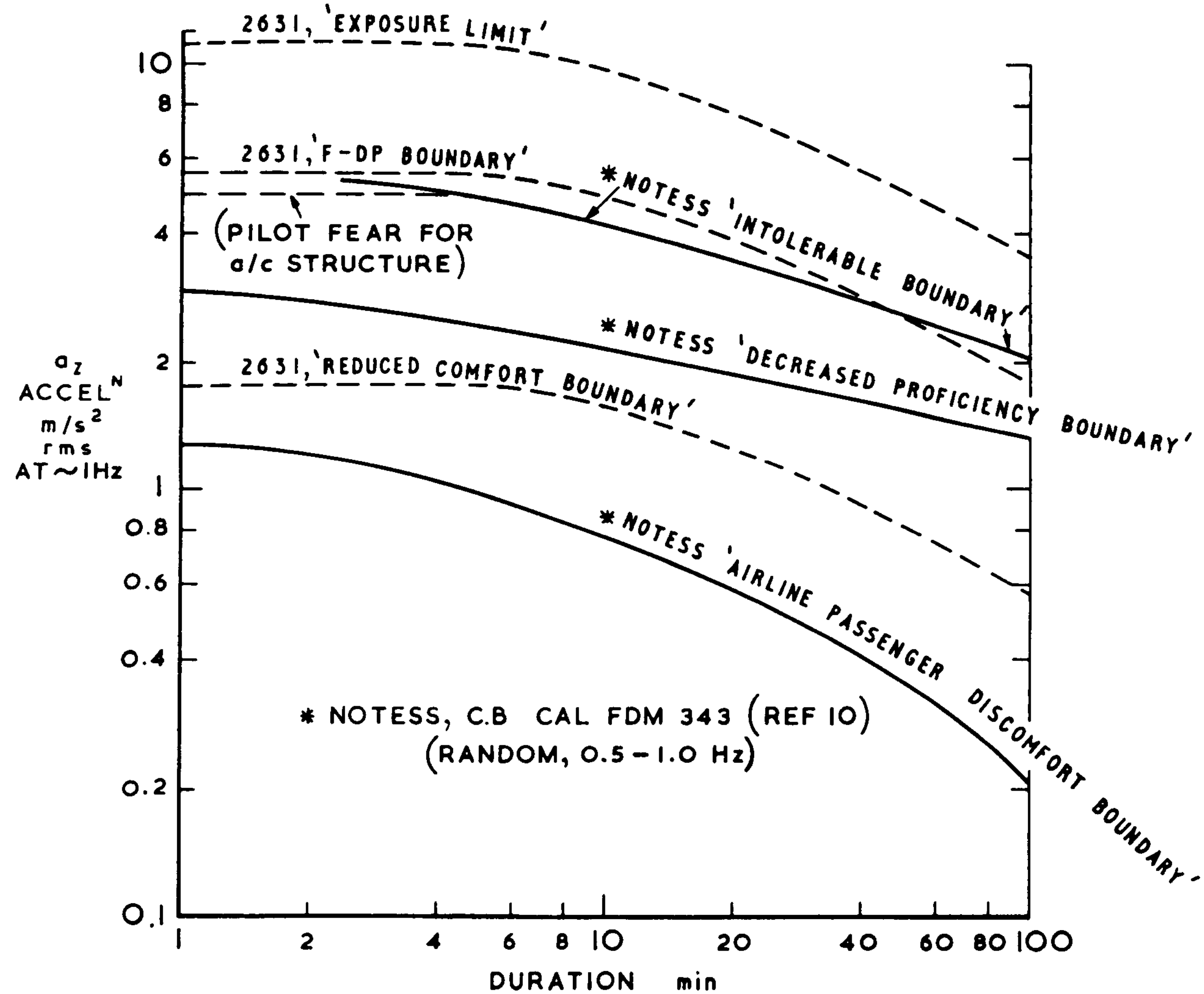

Figure 8.- Acceleration versus time from Notess and ISO 2631.

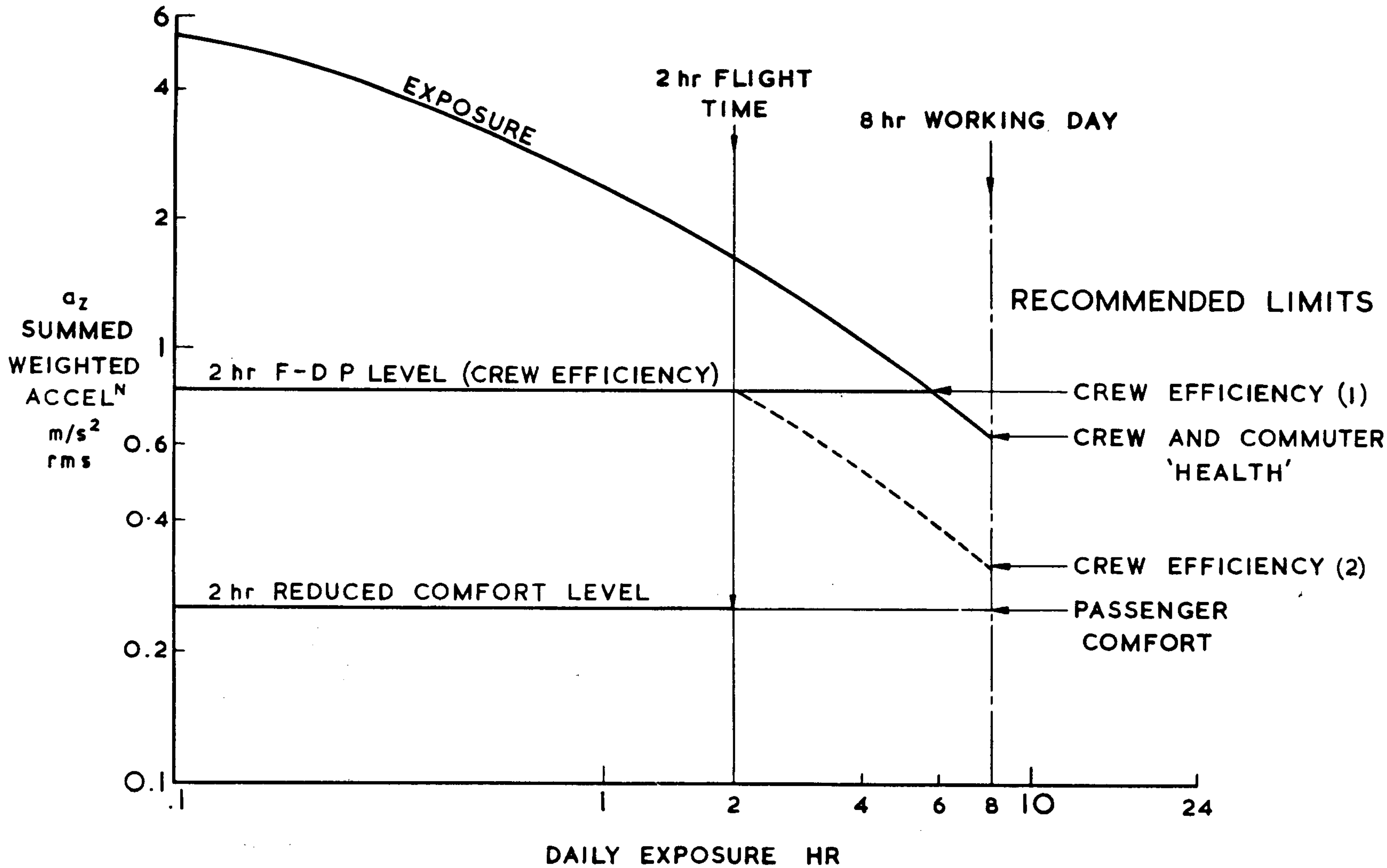

Figure 9.- ISO 2631: Application to aircraft ride quality requirements.

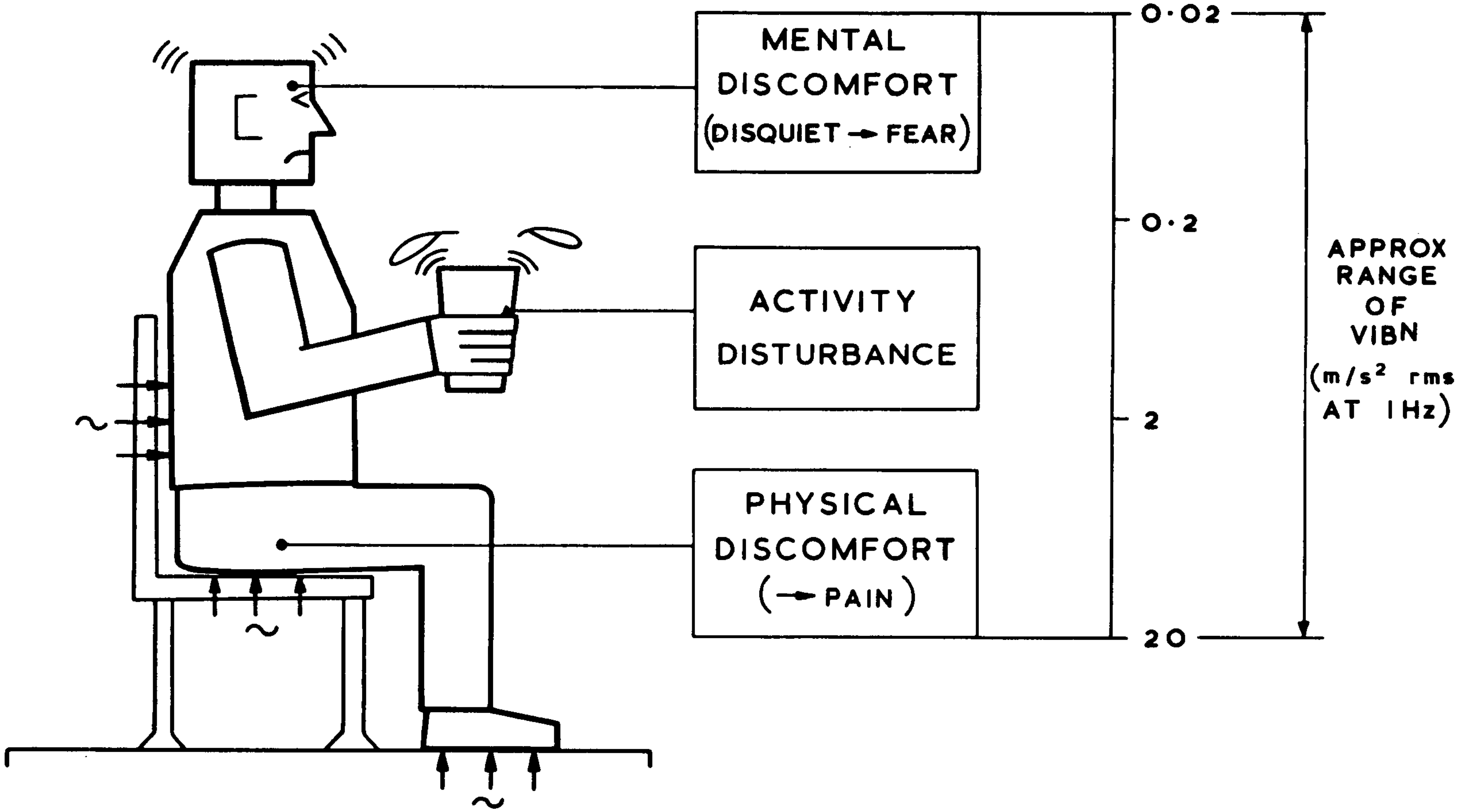

Figure 10.- Vibration: Discomfort.

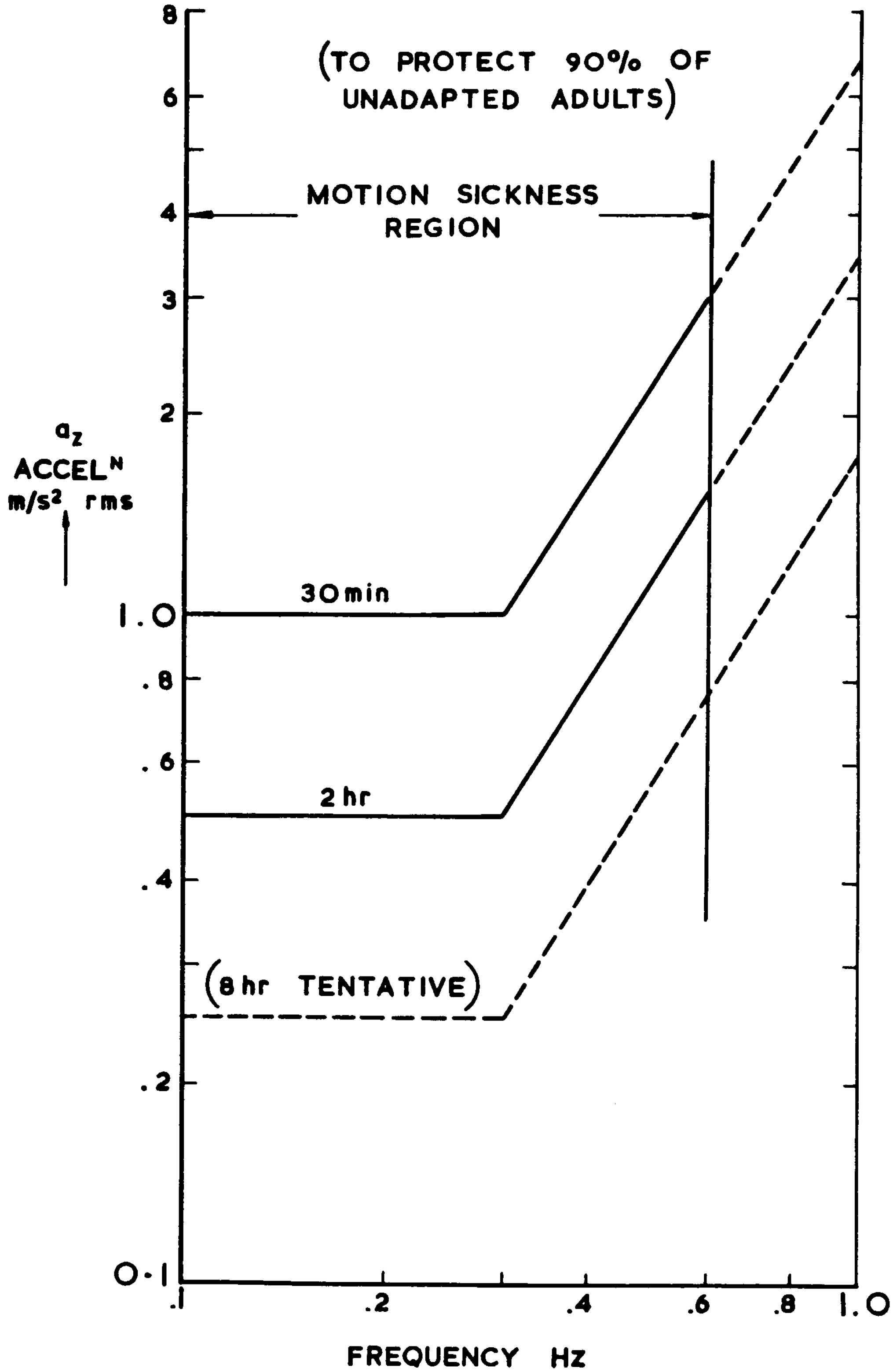

Figure 11.- Proposed 'severe discomfort boundaries', 0.1-1.0Hz.

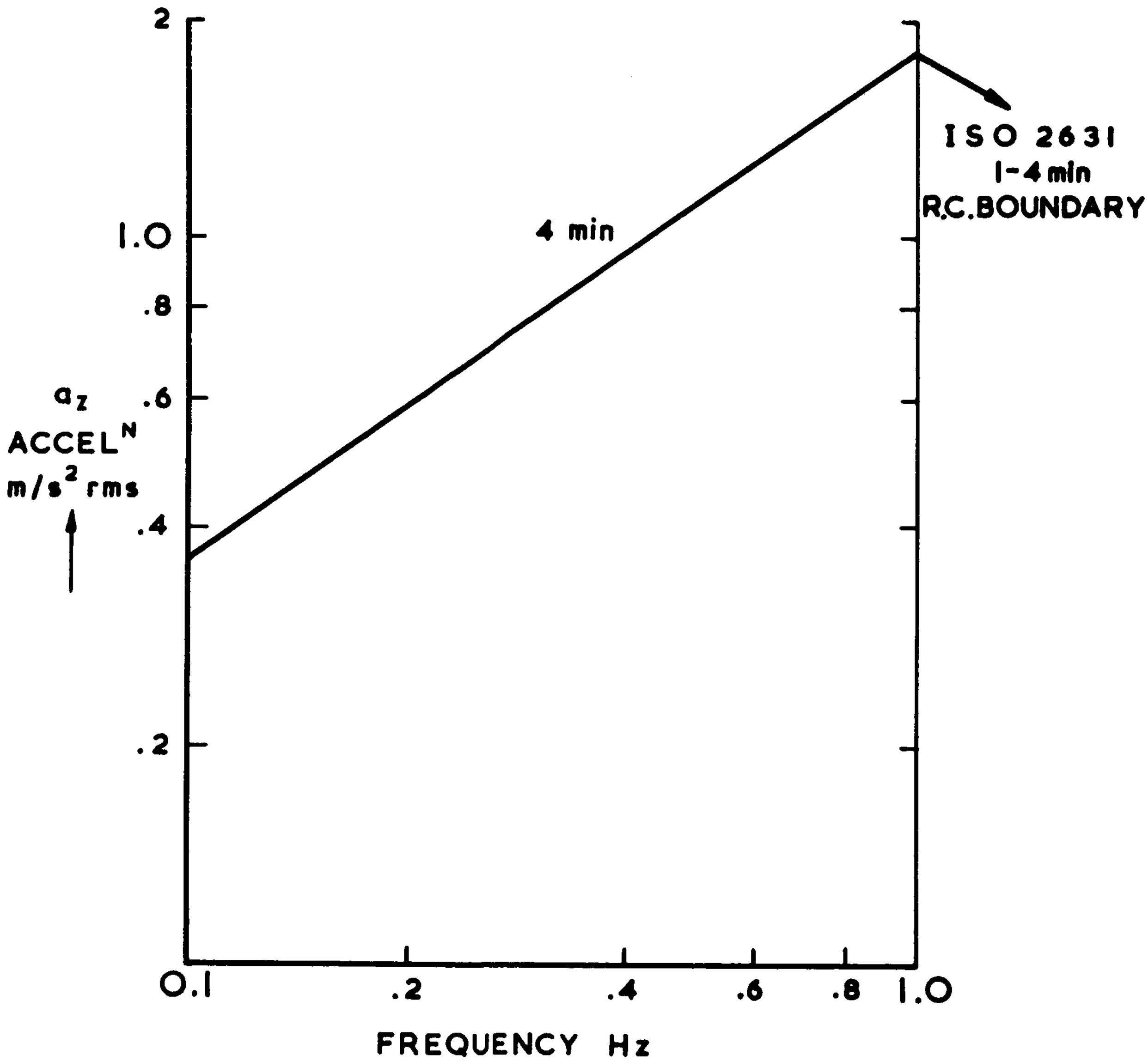

Figure 12.- Possible 'reduced comfort' boundary, 0.1-1.0Hz.

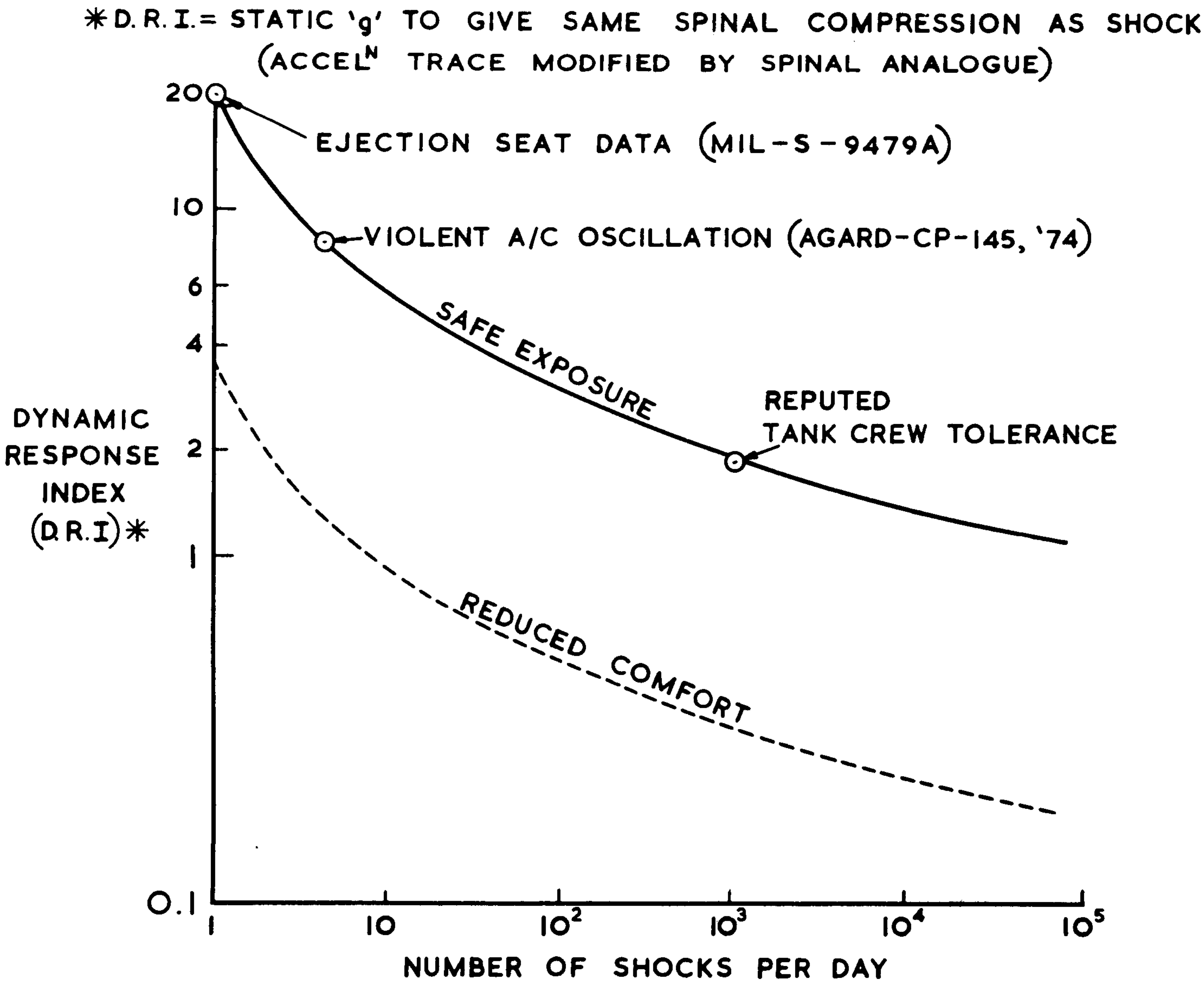

Figure 13.- Possible approach for acceptable levels of repeated shocks.

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ANALYSIS OF PROPOSED CRITERIA FOR HUMAN RESPONSE TO VIBRATION

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SUMMARY

The development of criteria for human vibration response is reviewed, including the evolution of the ISO Standard 2631. The latter document is analyzed to show why its application to vehicle ride evaluation is strongly opposed.

The criticisms are directed not only at the specific limits given for comfort; they extend also to the setting of arbitrary limits for "fatigue-decreased proficiency," together with their decrements as a function of exposure time.

Alternative vertical and horizontal limits for comfort are recommended in the ground vehicle ride frequency range above 1 Hz. These values are derived by correlating the 'absorbed power' findings of Pradko and Lee with other established criteria. Special emphasis is placed on working limits in the frequency range of 1 to 10 Hz since this is the most significant area in ground vehicle ride evaluation, as well as that of greatest diversity from the ISO Standard.

INTRODUCTION

It is most opportune that this Symposium should be called at this particular time. On the one hand, the demand has crystallized for a reliable guide to vehicle ride quality; on the other, there is a pronounced risk that improper criteria, once adopted, may become firmly rooted in practice.

It is this writer's belief that such a risk is presented by the now adopted ISO Document ISO/DIS #2631 (ref. 1) as an International Standard and by the efforts under way to have it adopted as a U.S. Standard. Already, it is evident that the ISO criteria are being applied in a number of important current ride development projects.

One of the objectives of this paper is to discourage the use of the ISO Standard in its present form; the reasons for this opposition are fully documented. The primary objective, however, is to present an alternative set of criteria for application specifically to ground vehicle ride evaluation.

The writer has been actively engaged in the analysis of data on human response to vibration for many years. For the last fourteen years, as SAE representative on the ANSI S3 Committee and a member of its Working Group on Vibration Levels, he has participated in the consideration of all the material submitted by the ISO and others, and has contributed new data to the Group.

Most recently, in February of this year, the writer presented to the SAE a paper on "Human Vibration Tolerance Criteria and Applications to Ride Evaluation" (ref. 2). Some of the latter material is included in this paper, but other data are presented here for the first time. In all cases, attention will be concentrated on the frequency range of 1 to 20 Hz, since this is the principal area of interest in ground vehicles, as well as the area of greatest diversity among the various criteria proposals.

SYMBOLS

a vibration displacement amplitude, m

A vibration acceleration, g

f frequency, Hz

g acceleration of gravity

j jerk (peak rate of change of acceleration), m/sec^3

K absorbed power constant, $watts/m^2/sec^4$

n exponent, $y = x^n$

t time, min

v vibration velocity

W watts

P absorbed power, watts

BACKGROUND OF CRITERIA DEVELOPMENT

Vertical Vibration

Janeway Analysis, 1948.- In 1948, this writer presented a paper to the SAE on "Vibration Limits to Fit the Passenger" (ref. 3; see also ref. 12, pp. 25-26). This was based on an analysis of all the experimental data on human subjective response to vertical vibration that had been published up to that time.

Superficially, these data appeared to be largely contradictory, with no consistent trends. Analysis based on correlation according to the various derivatives of the imposed sinusoidal vibration revealed definite characteristics depending on the frequency range. The discrepancies among the results of the different investigators were evidently due to wide variations in the precision of their experimental equipment and, especially, in the intensity levels imposed on their subjects.

Figures 1 and 2, reproduced from the 1948 paper, show individual plots against frequency of the peak jerk and acceleration, respectively, obtained for constant response.

In figure 1 it will be seen that, in only one instance, did the jerk increase at all in the range of 1 to 6 Hz, namely, in the Meister extreme discomfort curve. In all other cases, the jerk either remained constant or decreased in this frequency range. The Meister data were judged to be the most reliable (a) because all observations were given, and (b) because he was the only investigator who actually ran his tests down to 1 Hz. The three lines labeled "Meister" were given as the boundaries determined for the most sensitive subjects, rather than for the average responses.

In arriving at a recommended 'jerk' limit, the established practice of the Otis Elevator Company in controlling elevator acceleration and deceleration proved most useful. Figure 3 shows the characteristic patterns developed by Otis to ensure the comfort of elevator passengers, separately for each derivative of the motion. Note that the maximum jerk is limited to 10.9 m/sec^3 over 1/4 sec. (This compares with Meister 'uncomfortable' at 18.3-21.3 m/sec^3.) The corresponding maximum acceleration limits were 1.8-2.7 m/sec^2.

Reference to figure 4 for simple harmonic (sinusoidal) motion shows that at 1 Hz a 5.1 cm amplitude produces "jerk" peak of 12.5 m/sec^3, but a mean jerk of 10.9 m/sec^3 over 1/4 sec, and a peak acceleration of 2.0 m/sec^2, both values being close to the Otis criteria. This jerk value was also well below the Meister "uncomfortable" level. Therefore, this condition (5.1 cm amplitude at 1 Hz) was adopted as a safe limit to recommend for vehicle passenger comfort in the 1 to 6 Hz frequency bracket, with allowable amplitude diminishing as $a = 2/f^3$.

Reference to figure 2, showing the plot of the peak acceleration data, indicates a general trend to a constant acceleration in the 6 to 20 Hz range, for a constant response at the "uncomfortable" or higher level. The acceleration value corresponding to the constant 'jerk' peak of 12.5 m/sec^3 at 6 Hz is $\text{acc} = \frac{12.5}{2\pi \cdot 6} = 0.33$ m/sec^2 or $af^2 = \frac{2}{6} = \frac{1}{3}$. This compared with $af^2 = 0.5$ at 6 Hz for Meister uncomfortable response. Thus it appeared to be a safe margin and was adopted as a recommended constant peak acceleration limit from 6 to 20 Hz.

It is of particular interest to note that Meister's "strongly noticeable" line shows a minimum acceleration at 5 Hz and a sharply increasing value as the frequency increases. It will be seen that this characteristic has been demonstrated by later data to be valid at the higher response intensities.

A possible explanation for the observed leveling off of allowable acceleration at the higher intensities is the additional input due to noise generated by mechanically driven vibrating platforms as the amplitude increased at the higher frequencies. The acoustic input to the subject has been eliminated in the later experiments with sophisticated electromagnetic or hydraulic-driven vibrating platforms.

The effect of the constant acceleration value selected in the original recommendation is shown in figure 2 to have increased the margin of safety bel the Meister "uncomfortable" level until at 20 Hz the indicated response was at the "strongly noticeable" level. This was a deliberate judgment factor, in th absence of exposure data, on the basis that vibration at the higher frequencie tends to be longer sustained.

Above 20 Hz all investigators showed a definite tendency for a constant vibration velocity to produce a constant response. This has been borne out by the later work, at least for vertical vibration.

ISO Standard Evolution.- Starting in 1964, the International Standards Association initiated a proposal for a standard on "Thresholds of Mechanical Vibration and Shock Acceptable to Man." Figure 5 shows graphically the several steps in the evolution of the vertical criteria, as now adopted.

The initial criterion was based on Dieckmann and characterized by a constant acceleration line from 1 to 11 Hz at 0.063 g (rms), followed by a constant velocity line for frequencies above 11 Hz. As a result of strenuous objections to this proposal, which was patently erroneous at low ride frequencies, the ISO Group offered a series of compromises, as follows (referring to figure 5):

1. At 1 Hz, the short term comfort level was raised to 0.12 g (rms); from 1 to 2.8 Hz, the tolerance was reduced according to the relation, acc $\propto 1/\sqrt{f}$, giving a value of 0.07 g at 2.8 Hz. This value was then carried as a constant criterion up to 11 Hz; a constant velocity line was then extended from 11 Hz to the higher frequencies (von Gierke, proposal 166).

2. The final step raised the allowable comfort acceleration to 0.14 g at 1 Hz but retained the previous relationship with frequency, which was extended to 4 Hz where it intersected the previous constant acceleration line. The latter was then cut off at 8 Hz, at which point the acceleration again rose with increasing frequency along a constant line (acc $\propto v_c f$).

The latter modification obviously increased substantially the indicated tolerance above 8 Hz. What is more important, however, the ISO vertical criterion, by rejecting the constant jerk response at 1 to 5 Hz, is believed to be much too permissive at precisely the frequencies to which the human body is most sensitive. In this connection, it may be noted that the vertical vibration frequency range of 4 to 6 Hz has been established experimentally as the area of resonance with the human viscera (ref. 4).

For purposes of comparison, the vertical criterion recommended by the writer has been superimposed in figure 5. The so-called "comfort" limit of the

ISO Standard is shown to be at least twice the "recommended" value, in the range of 4 to 6 Hz. This will be discussed more fully in a later section.

Absorbed Power Concept.- In the 1960's, Fred Pradko and Richard Lee, of the U.S. Army Tank-Automotive Command, Mobility Systems Laboratory, carried out a unique series of vibration experiments on human subjects, using the most sophisticated equipment and experimental techniques. It is this writer's belief that this investigation and, especially, the brilliant analysis of their results, constitute the greatest contribution to date to our knowledge of human responses to vibration.

The equipment consisted of a hard, flat chair which could be vibrated in five separate degrees of freedom, namely vertical, horizontal (lateral and longitudinal) and angular pitch and roll, about axes through the seat base. The vibratory motion was imposed by servo-mechanisms controlled by an oscillator which could generate any desired wave form and intensity either periodic or aperiodic. The instrumentation measured not only the subject's acceleration, but also the phase relation between the input and output motions, and the "absorbed power" or rate of energy absorption internally by the subject's body in resisting the induced vibratory motion.

By correlation with the subjective responses, Pradko and Lee arrived at the conclusion that the subjective response is a function of the "absorbed power" and consequently, that a constant absorbed power corresponds to a uniform degree of subjective response. To the extent that absorbed power is a valid indicator of human response, its discovery means that the objective measurement of response now becomes possible for the first time. It follows that the quantitative findings are bound to be much more precise than any subjective determination.

Moreover, since energy and power are scalar quantities, as opposed to vector quantities, they proposed that the resultant response to a complex vibration can be measured by directly summing the power absorbed by each directional component of the imposed vibration. This is another of the great practical advantages of the absorbed power concept.

Their findings were reported in a series of papers presented to the SAE and ASME from 1964 to 1966. In particular, a paper entitled "Analysis of Human Vibration (ref. 5; see also ref. 6) contains the definitive quantitative data for each of the basic degrees of freedom of motion of a seated person, as illustrated in figure 6. In addition, separate data are given for vibration transmitted vertically to the person's feet. These values are given in tables of absorbed power constants, varying with frequency. It should be noted, however, that constants which were extrapolated below 1 Hz should be disregarded. Lee has since disavowed the validity of 'absorbed power' in this very low frequency range because of the dominance of a different human response mechanism, namely, that associated with motion sickness.

The absorbed power in watts for any given acceleration is:

$$P = K\,(A)^2$$

where K = a constant for a given frequency and direction of motion

A = acceleration, m/sec^2 (rms).

Note that, by basing the constants on the rms acceleration value, they are applicable either to sine wave or random vibrations.

Figure 7 shows a comparison of the Pradko-Lee vertical characteristic with the original Janeway comfort criteria on a displacement-amplitude basis over the frequency range of 1 to 50 Hz. It is evident that up to 5 Hz the agreement is remarkably close indicating that constant absorbed power is virtually synonymous with a constant jerk level in the range of vehicle ride frequencies.

In a previous paper (ref. 7), this writer derived a mathematical expression for absorbed power in a simple vibrating system with viscous damping. By assuming a natural frequency of 5 Hz and 50% critical damping, the calculated amplitude vs. frequency for a constant absorbed power was found to agree closely with the Pradko-Lee data for constant response to vertical vibration over the entire frequency range of interest. The calculated points are also plotted in figure 7. Incidentally, other investigators, notably Coermann (ref. 4), have found that human body resonance to vertical vibration occurs in the vicinity of 5 Hz and generally exhibits the characteristics of a damped vibrating system.

It will be noted in figure 7 that a minor resonance is indicated in the Pradko-Lee data at 11 Hz that is not in evidence in the calculated values for the simple systems. Thus, the assumed analog to the human body is an evident oversimplification. The fact remains, however, that the basic mechanism involved in the human response to vertical vibration must be that of forced vibration of elastically supported organs having inherent viscous damping.

It will also be seen in figure 7 that the constant absorbed power criterion above 5 Hz corresponds generally to a constant velocity characteristic (linearly increasing acceleration with frequency).

Other Published Data.- Independent experiments on subjective responses to vibration have been conducted by a number of other investigators. The most noteworthy are Miwa in Japan (ref. 8), Coermann with the ISO Working Group 7 in Prague in 1965, and Volkov in the U.S.S.R. (see item 10 of reference list of ref. 8). Their findings on vertical vibration tolerance are summarized in table 1. (See next page.)

The recommended values are included in table 1 for comparison.

Where the response is designated (allowable to unpleasant), the minimum recommended acceleration at 4 to 6 Hz is seen to be no higher than 0.030 g. In the case of Coermann, the values are taken from the constant response curve nearest to the allowable acceleration at 1 Hz.

Also, it will be noted that the predominant slope in the 1 to 5 Hz range is (-1), corresponding to a constant jerk criterion (acceleration $\propto 1/f$).

Table 1

Source	Response	Freq. at Min. Accel., Hz	Min. Accel. g (rms)	Accel. at 1 Hz, g	Slope n at 1 to 5 Hz
Miwa	Unpleasant Boundary	5	0.029	0.056*	-
Coermann**	Uniform	4-5	0.05	0.15	-1
Volkov	Allowable (Comfort)	6	0.018	0.147	-1
Janeway	Allowable (Comfort)	5	0.030	0.145	-1

NOTES: *Maximum acceleration limit of equipment at 1 Hz (see figure 10)

**Contour rounded at 1 and 5 Hz

No hard evidence has been found to support the ISO assumption that acceleration $\propto 1/\sqrt{f}$ for constant response, or that an acceleration of 0.07 g (rms) is allowable for comfort at 4 to 6 Hz, even for very short time exposures.

The constant jerk criterion of response in the low vertical ride frequency range has also been confirmed by vehicle field tests, as reported by Van Eldik Thieme (ref. 9), M. Haack (ref. 10), and R. Fine (ref. 11).

Horizontal Vibration

'Absorbed Power' Results.- Until Pradko and Lee published their findings in 1966, information on response to vibrations in the horizontal plane were too meager to provide a basis for reasonable judgment. It was generally agreed that human sensitivity to horizontal vibrations is markedly higher than to vertical vibrations at low ride frequencies. However, it was commonly assumed that the horizontal response contour should parallel the vertical but at a lower level and at a constant ratio. For example, the SAE Ride and Vibration Manual (ref. 12) contained a note recommending that the horizontal criteria be taken at a uniform reduction of 30% from the vertical comfort criteria.

The Pradko-Lee results threw entirely new light on the nature of human response to horizontal vibrations (relative to a seated or standing person). They showed that the greatest sensitivity to these vibrations is at the lowest ride frequencies, namely, 1 to 2 Hz. At 1 Hz, the tolerance was found to be only one-seventh of that in the vertical plane, for the same absorbed power. Above 2 Hz, however, the tolerance increases rapidly and continuously. These characteristics are illustrated in figure 8, with relative acceleration as the ordinate scale, and frequency as the abscissa. The lateral and fore-and-aft responses are shown separately and are seen to be closely the same. They become equal to the vertical response at frequencies of 3 and 3.3 Hz, respectively. Above 3.3 Hz, the horizontal and vertical contours diverge rapidly, with sensitivity becoming much greater in the vertical direction as the horizontal tolerance increases according to: accel. $\propto f^{1.5}$.

Miwa Results.- In 1967, T. Miwa in Japan published the first of a series of papers on his subjective experiments, which included response to horizonta: as well as to vertical vibrations (ref. 7).

In a general way, Miwa's results resemble those of Pradko-Lee to the extent that the tolerance is a minimum at the lowest ride frequencies, 1 to 3 Hz, and then increases abruptly as the frequency rises. As shown in figure 9, the minimum value at the 'unpleasant' boundary is 0.048 g (rms), and the accel eration increases linearly with frequency beyond 3 Hz (slope = 1). In figure which includes Miwa's equivalent vertical response contour, it will be noted that the vertical acceleration averages the same as the horizontal over the 1 to 3 Hz range, with the two intersecting at 2 Hz.

The indicated equal response to vertical and horizontal vibrations at 1 to 2 Hz is contrary to common vehicle ride observations.

This discrepancy can be accounted for by Miwa's admission that his observations were quantitatively questionable below 6 Hz because of

(a) The difficulty of subjective judgment in matching response to the standard frequency of 20 Hz (his experimental procedure)

(b) The limitation of his equipment to a maximum vertical amplitude of 2 cm.

It is believed that the latter restriction prevented the acceleration fro: reaching the level for equal response. (The acceleration value of 0.056 g at 1 Hz corresponds to the full 2 cm amplitude).

In a recent exchange of correspondence with Miwa, he advises that recent findings indicate his horizontal limit curve should be extended at the same slope from 3 Hz to 2 Hz and then maintained constant to 1 Hz, as shown by the dotted line in figure 9. This reduces his minimum limit value by one-third, t‹ a point between the ISO and recommended values.

ISO Standard Criterion.- Following publication of the Pradko and Lee findings, the ISO commendably decided to adapt these new data to horizontal vibration criteria, comparable to those they had already developed for vertical vibrations.

What they arrived at was a composite of the Pradko-Lee and Miwa results. From Miwa they adopted his minimum acceleration value and the constant velocity contour at the higher frequencies. From Pradko-Lee, they took the cut-off of minimum tolerance at 2 Hz. Also, in combination with the previously adopted vertical contour (as seen in figure 9), this made it possible for the vertical and horizontal response lines to intersect at 3.2 Hz, as shown by Pradko-Lee.

As pointed out in connection with the Miwa results, the ISO minimum horizontal acceleration value is open to question. This is not a matter of hair splitting; the ISO allowable acceleration is fully 2.4 times the recommended minimum value derived for the same constant absorbed power as that established for the vertical comfort criterion.

The question may be raised whether the same absorbed power value will necessarily ensure comfort in the horizontal as well as in the vertical plane. An affirmative answer is indicated by comparison of Miwa's horizontal 'unpleasant' boundary with the recommended comfort criterion based on the constant absorbed power of 0.2 watts. Figure 9 shows, in the frequency range of 3 to 10 Hz, the two lines average closely the same. This is the range of presumed greater reliability for Miwa's results, and the same area in which the two vertical criteria so closely agree.

Pitch and Roll Vibrations

Pradko and Lee also investigated the effects of pitch and roll vibrations about axes through the seat-subject interface (see figure 6).

The isolated effects of these angular accelerations were found to be negligible. This can be accounted for by the observed ability of the subject, <u>when not confined in the seat</u>, to remain in a substantially vertical position during such vibrations.

It should be noted that the imposition of a seat belt and shoulder harness may seriously change this isolation from angular motions per se. This does not apply (to the same extent) to the horizontal linear accelerations on the seat produced by angular accelerations about remote axes. However, confinement in the seat would impose more direct linear accelerations on the driver's head than would otherwise be the case, and, therefore, make for greater discomfort.

Summary: Criteria Comparison

The foregoing review is believed to amply justify the conclusion that the Pradko-Lee absorbed power results are the most reliable and have the greatest applicability to the evaluation of vehicle ride. The reasons may be summarized as follows:

1. The excellence of the equipment and instrumentation ensured precise control of input vibration wave form and absence of extraneous noise disturbance.

2. The thorough execution of the subjective experiments provided a solid basis for the analytical conclusions.

3. The brilliant mathematical development of the absorbed power concept which, for the first time, provided a meaningful objective measure of human vibration response.

4. The greater quantitative accuracy inherent in objective measurement compared to exclusive reliance on subjective responses.

5. The excellent correlation with previously established criteria and human body characteristics.

6. The ability to combine all components of a complex vibration input into a resultant measure of response.

RECOMMENDED COMFORT CRITERIA

For the reasons given in the previous section, the Pradko-Lee concept of "absorbed power," as an objective measure of subjective response to vibration has been adopted by the writer as the most reliable guide to valid criteria c human tolerance.

Although Pradko and Lee have defined the characteristic variations in acceleration for a constant response (constant absorbed power), they have not attempted to fix limits for comfort or for any other degree of subjective response. Their position is that tolerance is a relative term, and that the tolerable intensity level will change with the environment in which the vibra tion exposure takes place.

Consequently, the principal task in applying the Pradko-Lee findings is to arrive at a viable limit of absorbed power for comfort. Fortunately, we have an accumulated body of observations from experiment and field experience on which to base such a correlation. First of all, the one point of agreemen between the ISO Standard and my previous recommendations is the vertical acceleration limit for comfort at 1 Hz. This limit of 0.2 g (peak), 0.145 g (rms), first proposed in 1948 for short term sine wave vibrations, has with-stood the test of time.

By sheer happenstance, this acceleration is equivalent to an absorbed power of 0.2 watt, as calculated in appendix A from the Pradko-Lee table of vertical vibration constants. This rate of energy dissipation is equivalent 0.02 kg-m/sec (1.8 in-lb/sec).

Vertical Limits

Figure 10 shows a plot of the vertical acceleration values corresponding to a constant absorbed power of 0.2 watt over the frequency range of 1 to 50 H For frequencies over 5 Hz, the acceleration values are also given if the powe limit includes that absorbed by the feet of a seated person exposed to the sam input vibration. As a frame of reference, the boundary of the original Janewa recommended comfort criterion is also shown. It is evident that the constant jerk line approximates the Pradko-Lee values from 1 to 5 Hz. Above 5 Hz, a constant velocity (shown broken) is a perfect fairing of the Pradko-Lee values averaging with and without the power absorbed in the feet. The resultant com-fort boundary is an unsymmetrical 'V', which is very simply described mathe-matically as follows:

$$\text{For } f = 1 \text{ to } 5 \text{ Hz accel.} = \frac{0.145}{f} \qquad \text{g (rms)}$$

$$f = 5 \text{ to } 50 \text{ Hz accel.} = \frac{0.145\ f}{24} = 0.066\ f \qquad \text{g (rms)}$$

It is important to note that a constant absorbed power implies a constant duration of exposure as well as of response. Referring to the original data sources, a constant duration of 16 minutes has been assigned to the boundary values of figure 10. In ground transportation vehicles, this would correspond to traversing a 24.1 km rough stretch of highway at 85.5 km/hr.

It is evident in figure 10 that the original Janeway criteria incorporated an offset at constant acceleration, from 6 to 20 Hz, between the constant jerk and constant velocity phases. This has already been accounted for in the discussion on the development of the earlier criteria.

Horizontal Limits

It is evident in the Pradko-Lee curves of relative horizontal accelerations vs. frequency for constant absorbed power (figure 8) that the lateral and fore-and-aft values are closely the same at most frequencies. Certainly, the differences are not of an order to warrant separate criteria in the two directions.

Figure 11 is a plot on log-log coordinates of the calculated values of the horizontal accelerations in the two directions, for a constant absorbed power of 0.2 watt, over the frequency range of 1 to 20 Hz. The acceleration limits are virtually identical in the critical range of 1 to 2 Hz, and again at 6 Hz and higher frequencies. The divergence in the 2.5 to 5 Hz range, however, is reasonably well reconciled by a straight line connecting the values at 2 Hz and from 6 to 10 Hz. This line has a slope of 1.5, indicating the relationship: $\text{accel.} = 0.02 \left(\frac{f}{2}\right)^{1.5}$ g (rms).

Above 10 Hz, the slope increases to 2, corresponding to a variation of limiting acceleration with the square of frequency. For practical purposes, however, the range of 1 to 10 Hz comprises the important range of ride disturbance.

The direct comparison of the recommended limits with the ISO Standard comfort boundaries is shown in figure 12. The outstanding differences are obviously at the most sensitive frequencies for human vibration response, both vertically and horizontally. In both cases, the maximum ratio of acceleration between ISO and recommended values is 2.4, corresponding to an absorbed power ratio of nearly 6.

Vibration Exposure Time

In the course of the deliberations of our ANSI Vibration Tolerance Working Group, it became clear that criteria of exposure limits, in addition to specifications of frequency and intensity, needed a third dimension, namely, the time duration of exposure.

In the search for experimental data on this aspect of vibration tolerance, this writer found only two fragments in the literature, and these from widely different fields. One was in a report by Notess (ref. 13) to the Cornell Aeronautical Laboratory in 1963 on reaction to disturbances in aircraft, in the frequency range of 0.6 to 1 Hz; the other source was a publication of Mauzin and Sperling (see data in ref. 10) on observations in railway passenger cars, at frequencies of 1 to 2 Hz. Referring to figure 13, these data are plotted (semilog coordinates, with relative acceleration tolerance as ordinate, and exposure time as abscissa. The data were submitted in this form to the ANSI Working Group for their consideration.

It must be considered that these observations were not made under controlled conditions but were incidental to typical operating conditions in each case. Also, the levels of vibration intensity were not likely to have been comparable. Consequently, it was impossible to estimate the degree of validity that could be ascribed to the results.

Nevertheless, the ISO Standard's relative acceleration tolerance values vs. exposure time are largely based on these data and are applied indiscriminately to all frequencies, response levels, and to both vertical and horizonta vibrations.

As far as comfortable levels of vibration are concerned, they should theoretically be independent of time duration. Strictly defined, 'comfort' should mean an absence of disturbance; therefore, there should be no cumulativ effect with time. The comfort state, in some ways, may be likened to the condition of dynamic stress in a material below its endurance limit. In the latter case, no damage will occur even after a theoretically infinite number of repeated stress cycles.

This is not to say that fatigue cannot occur in a vehicle over a prolonge period, for reasons of confinement, noise, poor ventilation, etc. The same fatigue could conceivably result even though no vibration were present.

Consideration of fatigue as a result of sustained exposure to vibration i a legitimate concern in environments having vibration intensities well above the comfort level, as a pertinent example, in highway trucks and truck tractor The drawback here is the difficult problem of establishing valid quantitative relationships between the vibration parameters (direction, intensity, and frequency) and exposure time at onset of fatigue.

So far, no one has succeeded in determining the onset of fatigue in human by any objective measurement, or subjective response, and not for lack of trying on the part of many investigators. Consequently, the promulgation by the ISO Standard of elaborate tables of tolerable exposure time for any given vibration condition must be viewed as completely arbitrary.

It should be noted, also, that the primary ISO Standard data are presente as the boundaries of "fatigue-decreased proficiency." From these values, the corresponding 'comfort' levels must be computed by dividing by a constant (3.1 regardless of frequency or direction of vibration. This constant relationship in itself, is an unsupported assumption. That is, of course, how the ISO values cited in this paper were arrived at.

The coupling of so-called 'fatigue' with 'decreased performance' is still another liberty taken by the ISO Standard except it happens that various aspec of human functional performance are subject to quite precise measurement.

Highly pertinent data on this score are revealed by the results of an extensive series of controlled laboratory tests made by Bostrom Research Laboratories for the U.S. Army (ref. 14). Referring to figures 14 and 15, the following findings effectively challenge the ISO's assumed effects of vertical vibration on performance:

1. Performance can be highly sensitive to frequency, without relation to the passive response to vibration. The following values are taken from figure 14:

Table 2

Type of Performance	Frequency, Hz	Peak Intensity, g	Performance Error, %
Visual Acuity	2.5	0.35	4
	3.5	0.30	23
Compensatory Track.	2.5	0.35	10.2
	3.5	0.30	14.2
Foot Pressure	2.5	0.35	42
	3.5	0.30	80
ISO Boundary	2.5	0.38	
(16 min)	3.5	0.32	

It is evident that the ISO designated "boundary of decreased performance" for an exposure of 16 min would be far higher than an acceptable limit under all conditions at 3.5 Hz and for at least one of the performance types (foot pressure) at 2.5 Hz.

2. The ISO assumption that the decreased proficiency boundary becomes progressively lower with duration of vibration exposure is without foundation. Figure 15 shows that no appreciable change in performance occurred over a period of 90 min exposure, even at the high sustained intensities of the Bostrom tests.

With regard to the relative effect of exposure time on the onset of fatigue, is is of theoretical interest that the 'absorbed power' concept provides a rational basis for deriving such a relationship. If a given level of absorbed power, P_o, produces fatigue in an exposure time, t_o, then it may be reasoned that this is the cumulative effect of the total energy absorbed, or $P_o t_o$, It might be logical then to expect an equal amount of energy absorbed at some other rate (power) to produce the same fatigue effect.

If $P_o t_o = E_o$ is a known reference condition for onset of fatigue:

Then: $E_o = Pt = P_o t_o$

Where P = absorbed power at some other given condition of frequency and direction, at which A.P. constant = K and accel. (rms) = A, t = exposure time at (P)

Then, $P = KA^2$, similarly, $P_o = K_oA_o{}^2$

and, $KA^2t = K_oA_o{}^2t_o$

$$\frac{A^2}{A_o{}^2} = \frac{K_ot_o}{Kt}$$

$$\frac{A}{A_o} = \sqrt{\frac{K_ot_o}{Kt}}$$

If the frequency and direction are the same under both conditions, $K_o = K$ and:

$$\frac{A}{A_o} = \sqrt{\frac{t_o}{t}}$$

Figure 16 is a plot of $\frac{A}{A_o}$ on log-log coordinates, taking $\frac{A}{A_o} = 1$ at $t_o = 16$ min, for a constant value of K. Obviously, this becomes a straight line of slope $= -\frac{1}{2}$. Also plotted are the relative acceleration values vs. exposure time for the fatigue boundaries specified in the ISO Standard, again taking 16 min exposure as the condition for accel. = 1. It is apparent that the ISO relative values agree closely with this theoretical relationship.

All that remains is for the base condition for onset of fatigue to be established.

RIDE EVALUATION

The details of ride evaluation are outside the scope of this paper. Nevei theless, such determination is the ultimate purpose of any system of ride criteria. It follows that the ability of the applied criteria to measure the resultant human response to the complex vibration environment of the real worlc is the acid test.

Application of Absorbed Power

The outstanding contribution of the 'absorbed power' concept is that, being a scalar quantity rather than a vector, any number of simultaneous vibration components can be summed directly to obtain an effective one-number resultant. In contrast, a system like the ISO Standard can attempt to evaluate the resultant of a number of component frequencies in the same direction but with difficulty and with questionable accuracy. (See Notes, pp. 8 and 9 of ISO/DIS 2631.) However, for simultaneous vibrations in different directions, the ISO instructions are explicit; the limits must be applied "separately to each component in the three axes."

This advantage of criteria based on 'absorbed power' is enormously important in such real ride environments as rail cars, trucks, and tractors. Here the horizontal vibrational disturbance can produce equal or greater intensity of operator response as compared with the vertical disturbance. Hence, separate evaluation cannot begin to measure the resultant response.

As a case in point, the results summarized in table 3 were obtained in comparative ride measurements on two current model truck-tractors, under identical operating and road conditions. (The details of this evaluation are given in appendix D of ref. 2.)

Table 3

	Tractor A	Tractor B
A_m^2, $(m/s^2)^2$		
Vertical		
On Cab Floor - Susp. Freq.	0.92	1.24
Tire Freq.	0.45	0.074
On Driver - Susp. Freq.	1.29	2.07
Tire Freq.	0.043	-----
Fore and Aft on Driver	0.51	0.77
Absorbed Power on Driver, W		
Vertical		
Susp. Freq., (K = 0.775)	1.0	1.61
Tire Freq., (K = 1.59)	0.07	-----
Total Vertical	1.07	1.61
Fore and Aft, (K = 1.97)	1.01	1.52
Total Abs. Power, W	2.08	3.13
Ratio to Comfort Limit (=0.2 W)	10.40	15.65
Ratio of Equiv. Mean Accel. to Comfort Limit	3.23	3.96

It will be seen that the separately integrated vertical and horizontal vibrations, in terms of 'absorbed power,' are close to the same intensity, in each of the vehicles. Each resultant, obtained by direct addition, is thus about twice as great as the separate components.

By comparing the overall resultant 'absorbed power' with the comfort criterion of 0.2 watt, the true magnitude of the disturbance is revealed, at 10 to

15 times the recommended level. The conversion to 'absorbed power' for any observed acceleration value and vibration direction is obtained from the relation:

$$P = 0.2\ A^2/A_o^{\ 2}$$

Where A = observed acceleration, g (rms)

A_o = comfort limit acceleration, g (rms), for the same frequency and direction

or $$P = KA^2\ (9.81)^2$$

$$\text{Where } K = \frac{0.2}{(A_o \cdot 9.81)^2}$$

A cautionary note should be added. If any system of criteria of human response is to be meaningfully applied, the instrumentation must measure the true vibration input to the subject; and the locations of the measurements relative to the subject must accord with those used in establishing the criter

The recommended criteria are directed to the vibration input at the interface of the seat and the subject's anatomy. Where a cushioned seat is present the vertical transducer must be interposed between the cushion and the subject As an alternative, the vertical input can be measured on the subject's shoulder and a correction made for the amplification or attenuation of the anatomy according to experimentally established average factors (ref. 2). The horizontal input can be measured on the seat frame at the level of the interface with subject. (This assumes no confinement with respect to a cushion seat back.)

The ISO Standard has very explicit instructions as to the location of vertical measurements. It is vague, however, on the matter of horizontal measurement location.

Application to Power Spectral Density

This is a method of analysis that has been widely used with tape recorded accelerometer data. By playing the tape back through a series of narrow band filters and squaring the output, the mean squared acceleration in each frequency band is charted.

In order to apply absorbed power criteria to ride evaluation from power spectral density data, it is obviously necessary to apply weighting factors, according to the criteria for equal response over the frequency range. Since the ordinates of the chart already represent the mean square of acceleration, it is only necessary to multiply the ordinates by the absorbed power (K) value, corresponding to each frequency, to obtain an absorbed power graph. The area under this graph is the total absorbed power.

This type of evaluation has a serious deficiency, namely, that it gives nc insight into the amplitude distribution. Thus, the same mean acceleration square ordinate can be integrated from many small accelerations or from a few large ones. A ride that comprises many small accelerations might be comfortable, while the few large accelerations might make the ride unacceptable.

CONCLUSIONS

The ISO Standard criteria should be rejected in view of the following deficiencies:

1. The 'comfort' boundary values are excessive at the most critical ride frequencies, for both vertical and horizontal vibrations.

2. The primary 'fatigue-decreased proficiency' boundary values are generally at too high a level for reasonable fatigue limits and have no validity with respect to 'decreased proficiency.'

3. The relative change in 'comfort' boundary values should be much less sensitive to time of exposure than the 'fatigue' values, although the same relative sensitivity is assumed for both.

The recommended comfort limits based on absorbed power, for frequencies of 1 Hz and higher, should be adopted because

1. The vertical characteristic vs. frequency is confirmed by the best available subjective data, as well as by experience, especially in the ride frequency range.

2. The horizontal criteria were the first to offer a definitive guide to valid comfort limits for vibrations in the horizontal plane.

3. Absorbed power is an objective measure of human response. As such, it is bound to be much more precise than any subjective determination.

4. The ability to integrate the absorbed power under complex real environments, consisting of simultaneous vertical and horizontal vibrations, means a great step forward in vehicle ride evaluation.

The available experimental data are entirely inadequate to support any valid standards for the following criteria, which are incorporated in the ISO document:

1. Limits of vibration intensity vs. frequency, representing:

   a. Fatigue boundaries;

   b. Reduced proficiency of performance.

2. Relative tolerance vs. exposure time at specific response levels.

APPENDIX A

CALCULATION OF ABSORBED POWER COMFORT CRITERION

Taking 5.1 cm in amplitude (single) at 1 Hz as an established comfort limit, the corresponding rms acceleration is:

$$A = \frac{(2\ \pi\ f)^2\ a}{\sqrt{2} \cdot 100}$$

$$= \frac{(4\ \pi^2) \cdot 5.1}{\sqrt{2} \cdot 100}$$

$$= 1.42\ m/s^2$$

where: A = accel., m/s^2, rms

f = frequency, Hz

a = amplitude, cm

From Pradko-Lee table of vertical constants, at 1 Hz, $K_v = 0.0985\ W/(m/s^2)$

$$\text{Abs. Power} = K_v A^2$$

$$= 0.0985\ (1.42)^2$$

$$= 0.198\ W$$

check at $f \cdot 4.75$: recommended acceleration comfort limit is 0.0305 g (rms). Absorbed power constant (Pradko-Lee) is maximum at $K_v = 2.189$

$$\text{Abs. Power} = 2.189\ (0.0305 \cdot 9.81)^2$$

$$= 0.196\ W$$

$\therefore$ Constant absorbed power corresponding to recommended vertical criteric $P = 0.2$ W.

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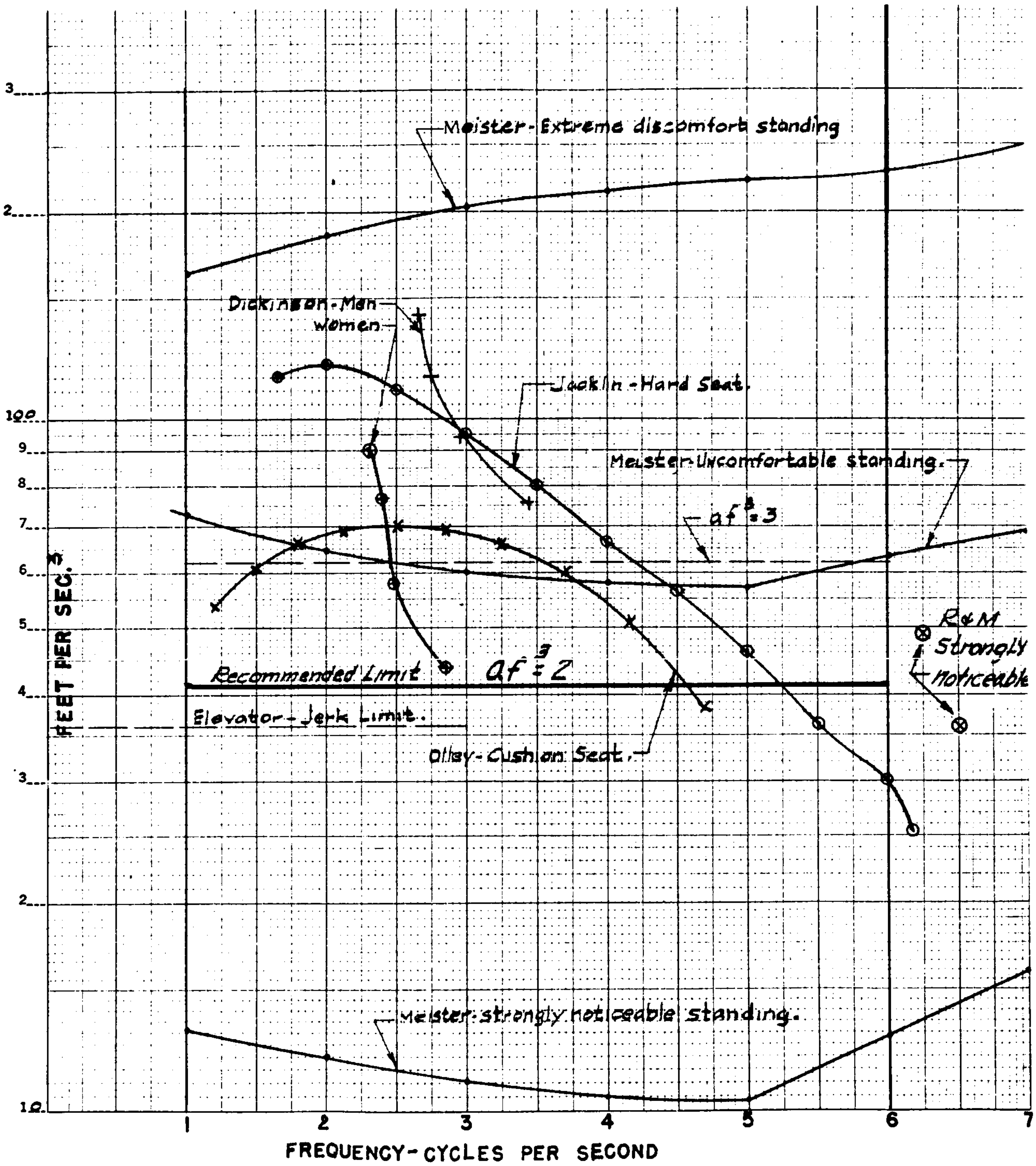

Figure 1.- Correlation of vibration data, jerk as a function of frequency, for constant response. (1 ft = 0.304 m; 1 cps = 1 Hz.)

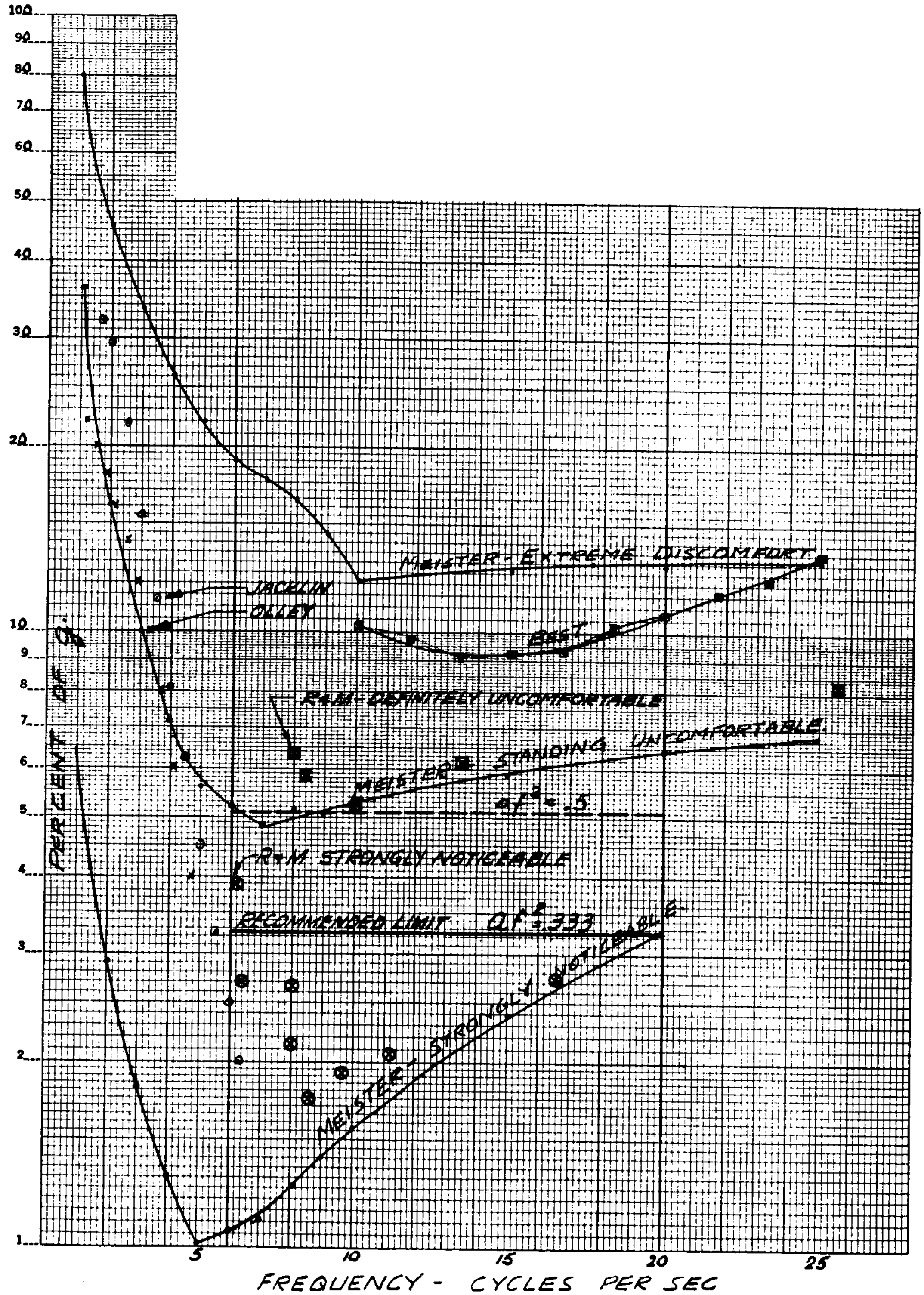

Figure 2.- Correlation of vibration data, acceleration as a function of frequency, for constant response. (1 cps = 1 Hz.)

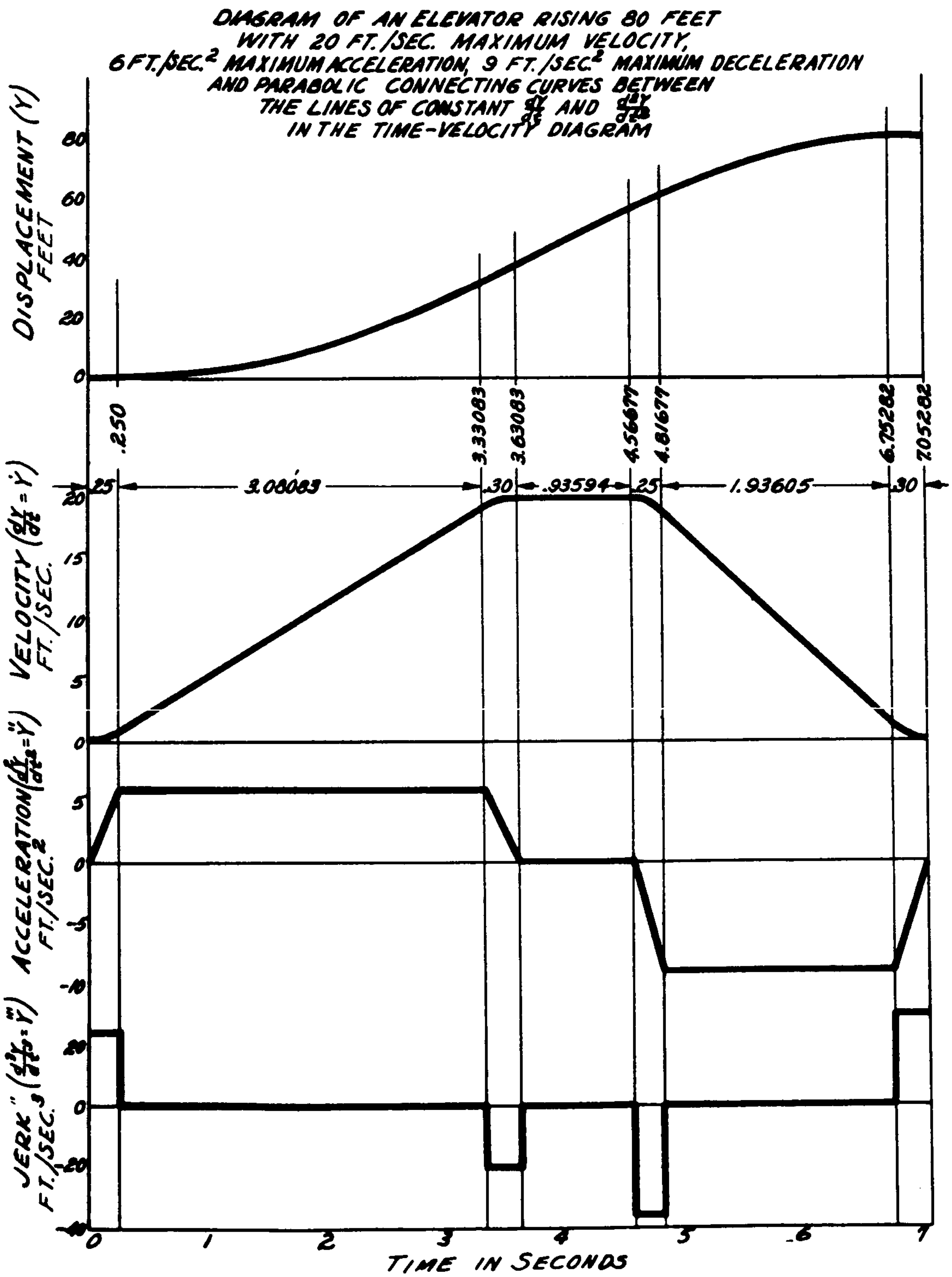

Figure 3.- Desirable limits for comfort in elevator operation.
(1 ft = 0.3048 m.)

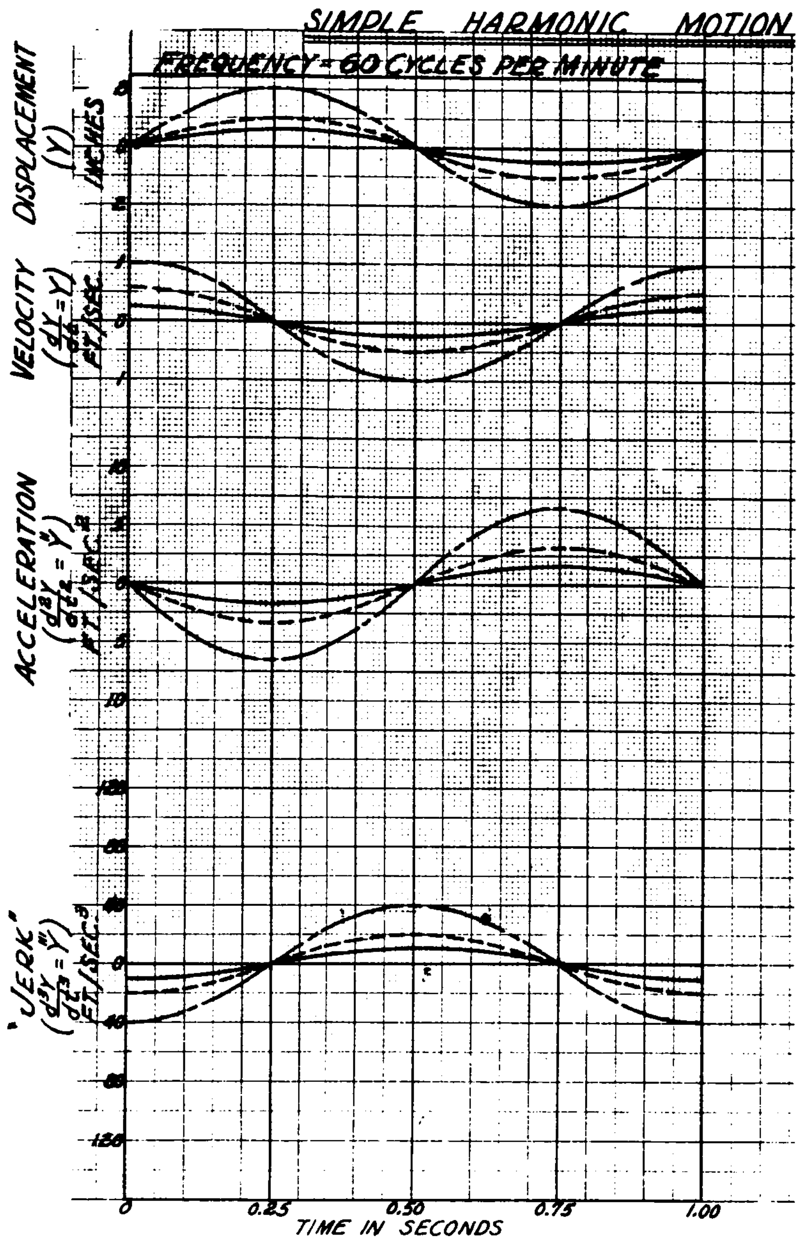

Figure 4.- Simple harmonic motion at 5.1 cm amplitude and 1 Hz. (1 in. = 2.54 cm; 1 ft = 0.3048 m; 60 cpm = 1 Hz.)

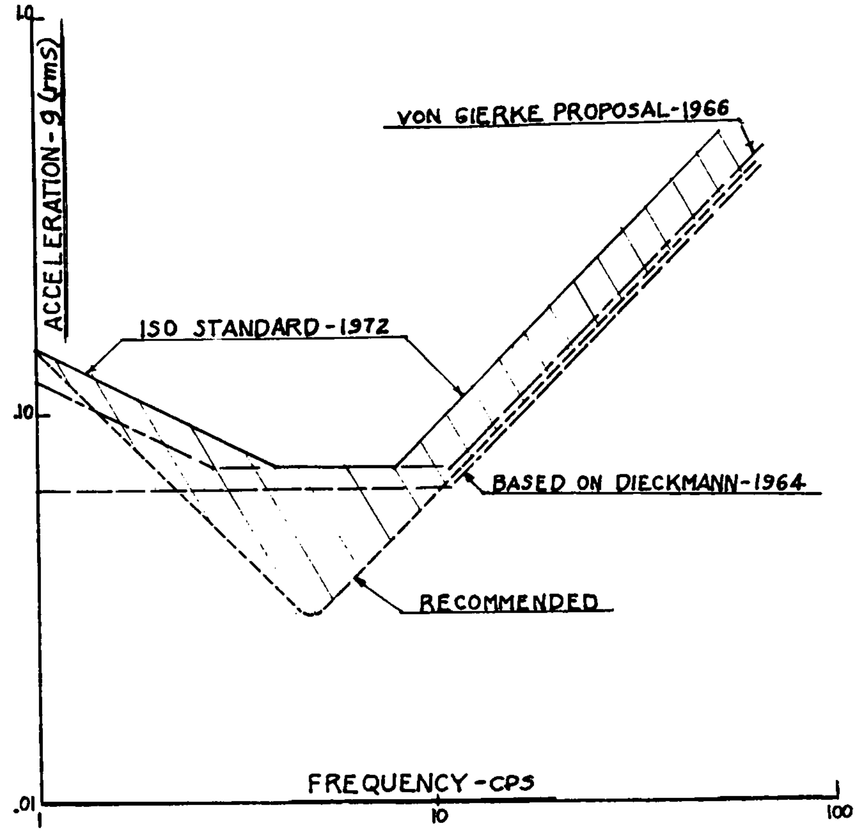

Figure 5.- Evolution of ISO Standard vertical comfort criterion. (1 cps = 1 Hz.)

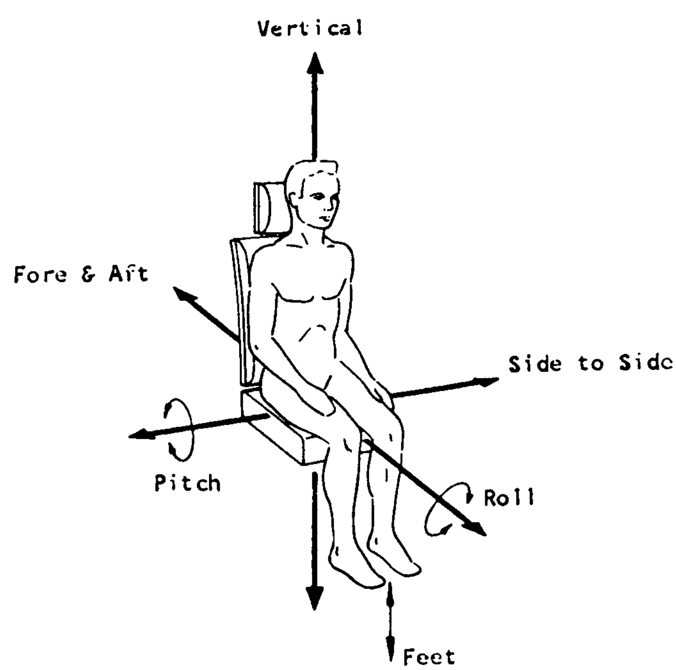

Figure 6.- Vibratory system used in Pradko-Lee tests.

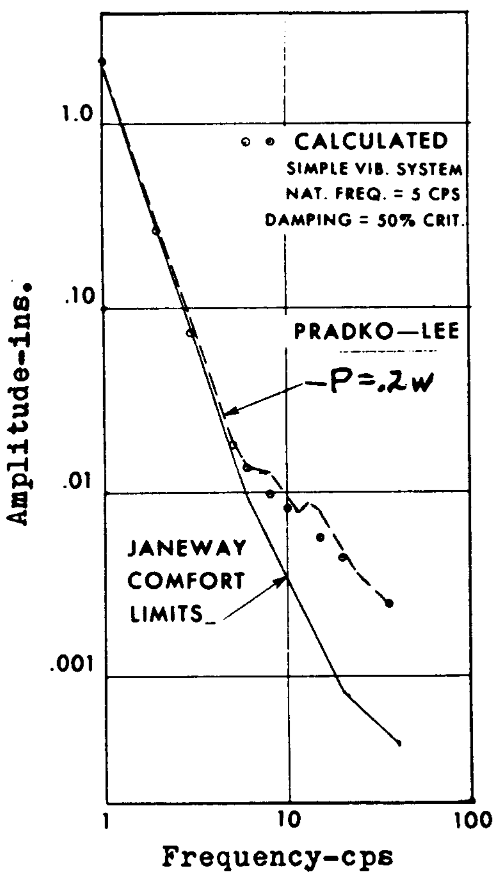

Figure 7.- Comparison of vertical vibration criteria. Calculated points are for constant absorbed power. (1 in. = 2.54 cm; 1 cps = 1 Hz.)

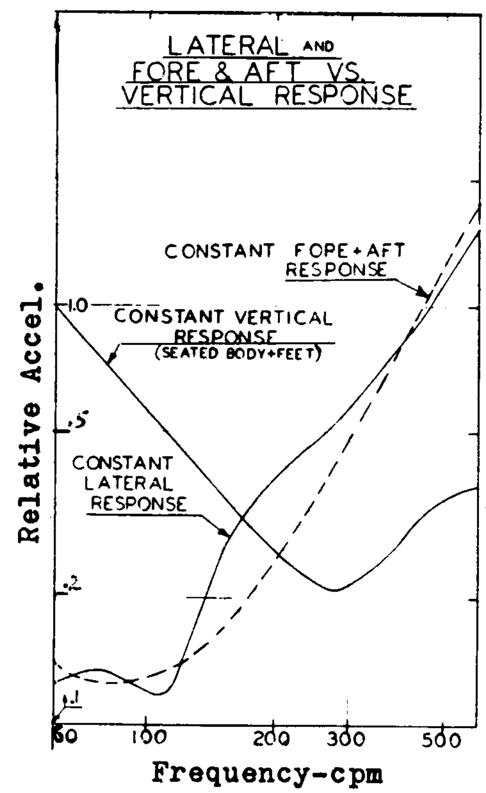

Figure 8.- Relative lateral and fore-and-aft acceleration compared to vertical acceleration for equal response (Pradko-Lee). (60 cpm = 1 Hz.)

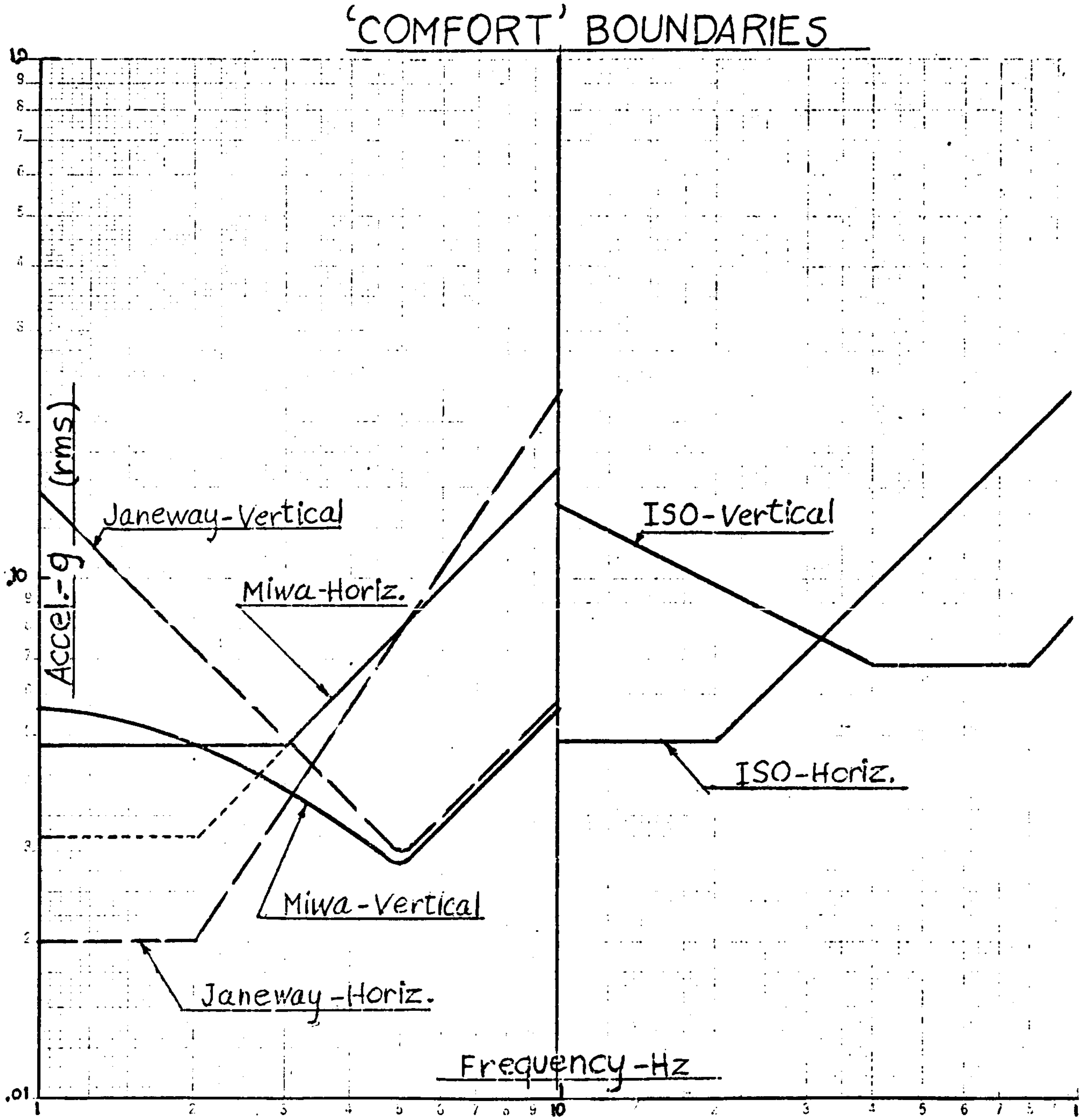

Figure 9.- Miwa criteria compared to Janeway and ISO (16 min exposure).

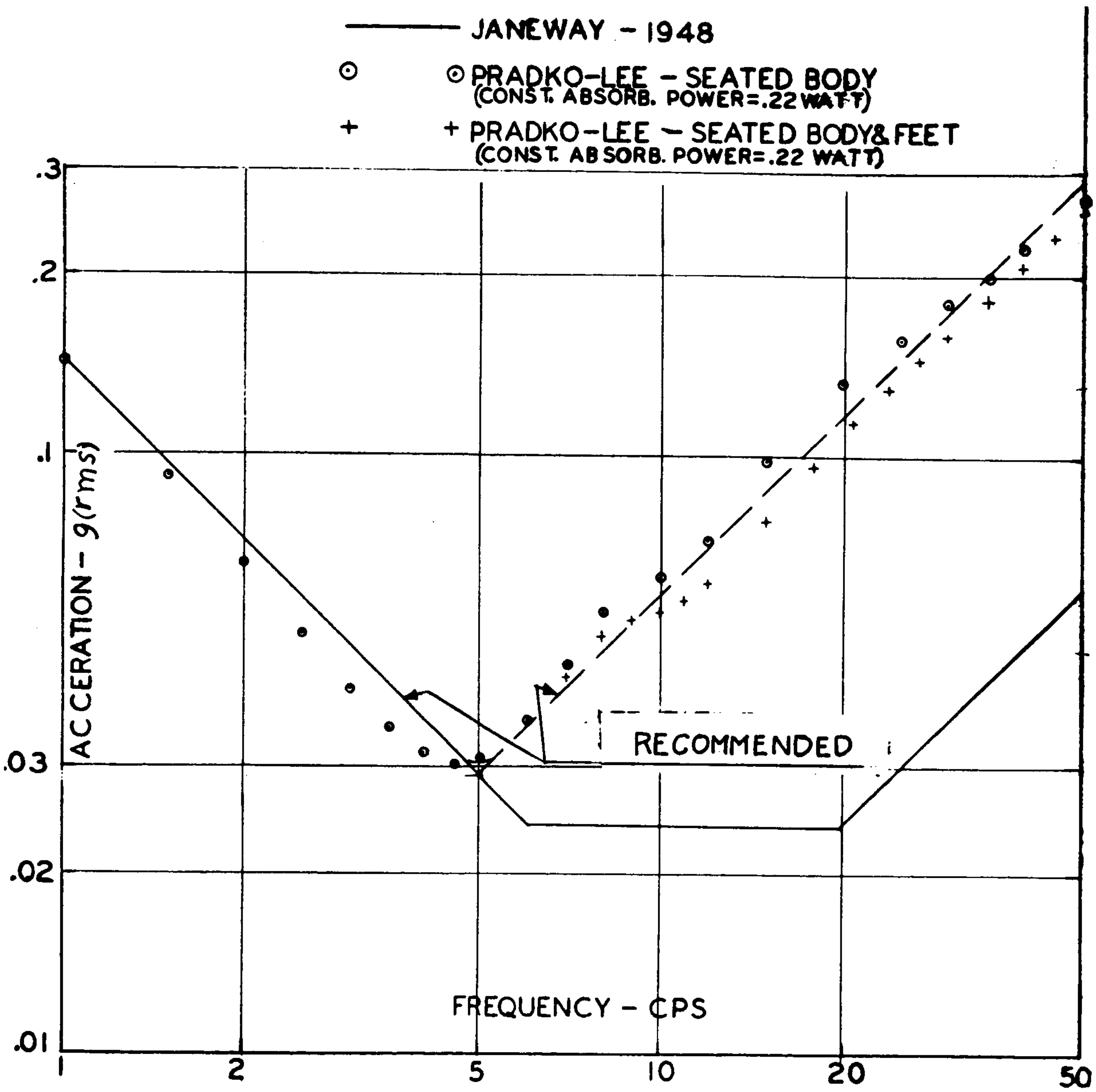

Figure 10.- Correlation of vertical comfort criteria: constant absorbed power and Janeway recommended (1948). (1 cps = 1 Hz.)

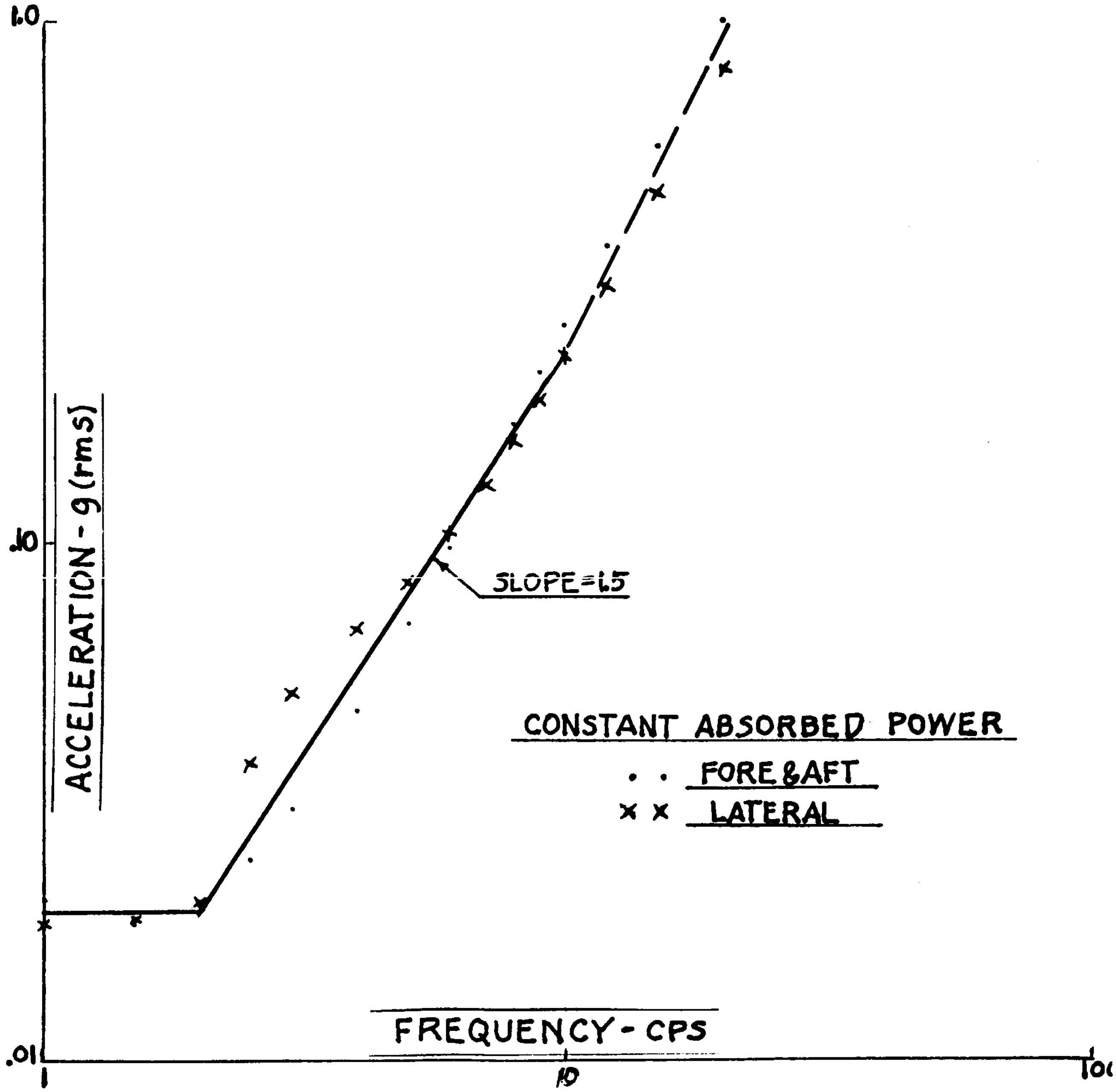

Figure 11.- Recommended horizontal comfort criterion. (1 cps = 1 Hz.)

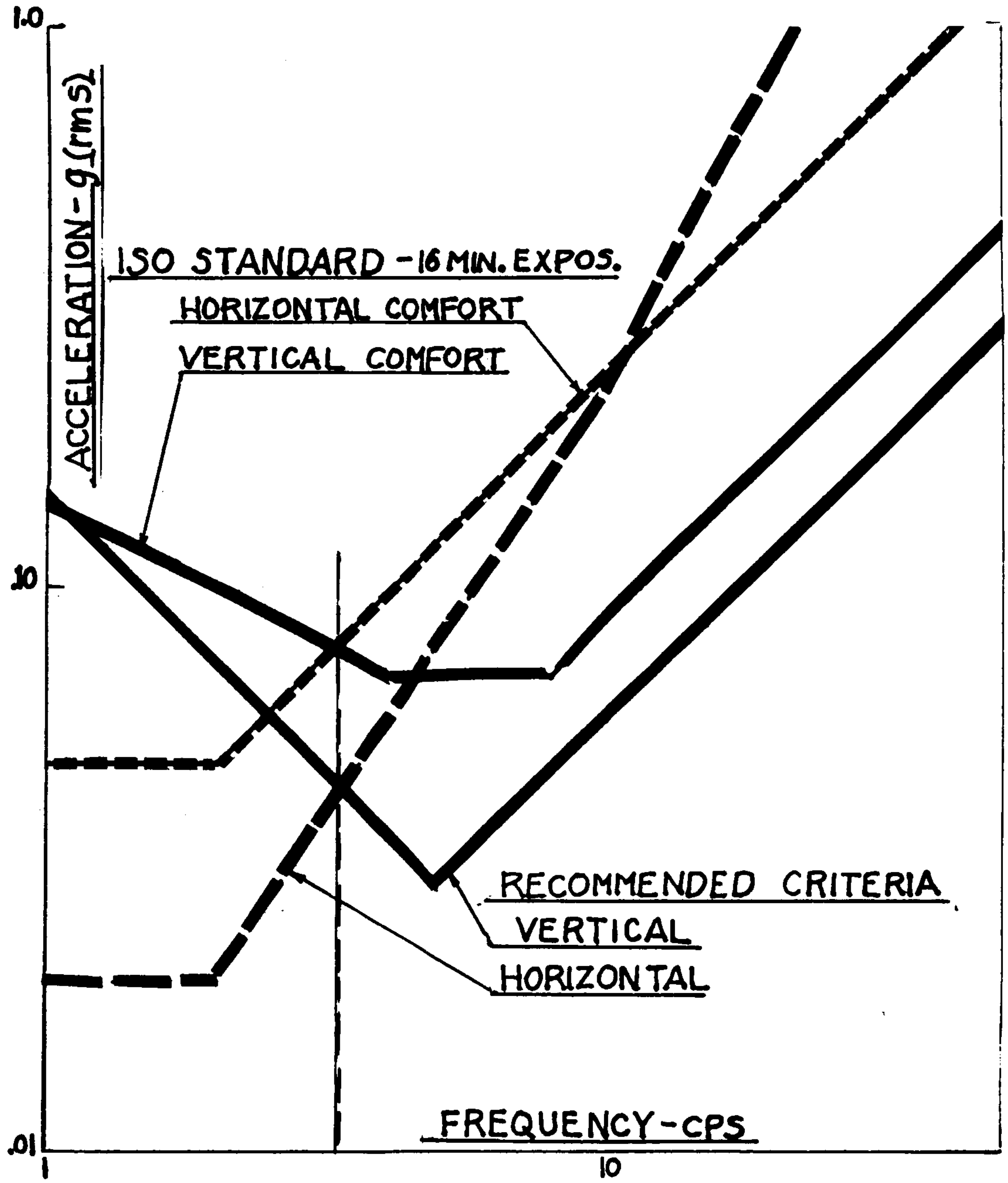

Figure 12.- ISO compared to recommended comfort criteria.
(1 cps = 1 Hz.)

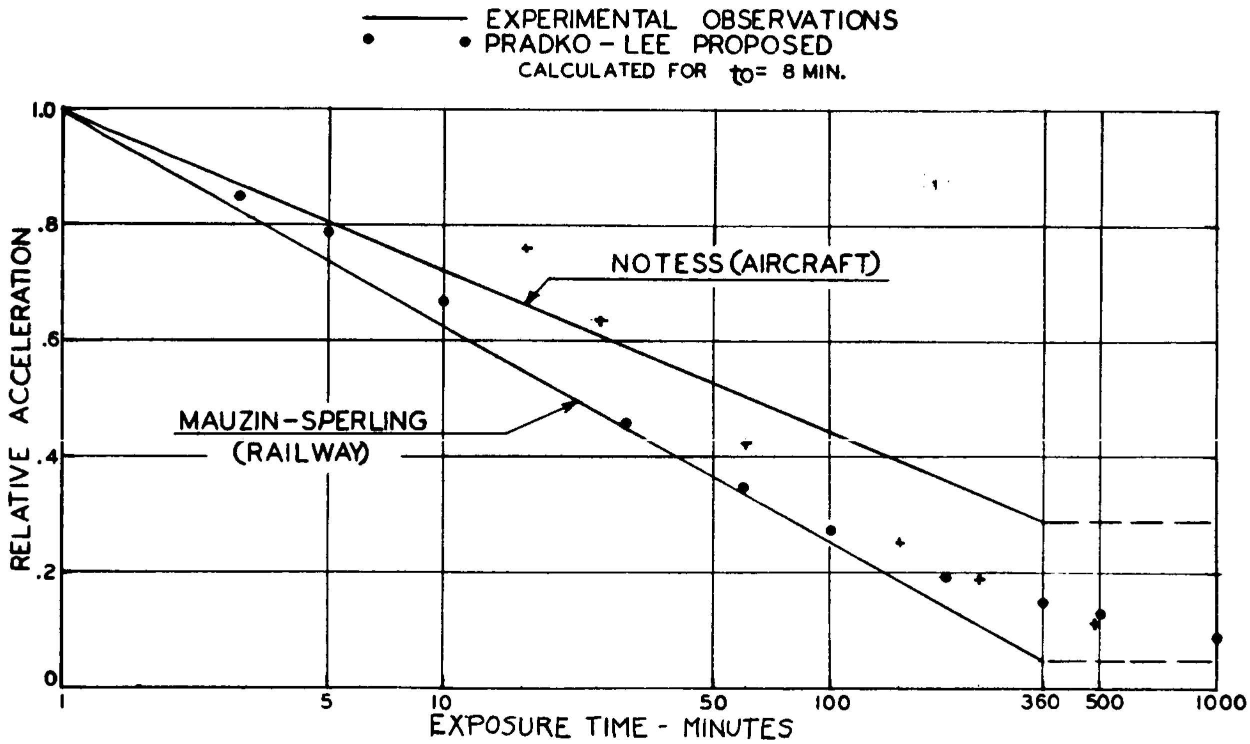

Figure 13.- Vibration exposure time.

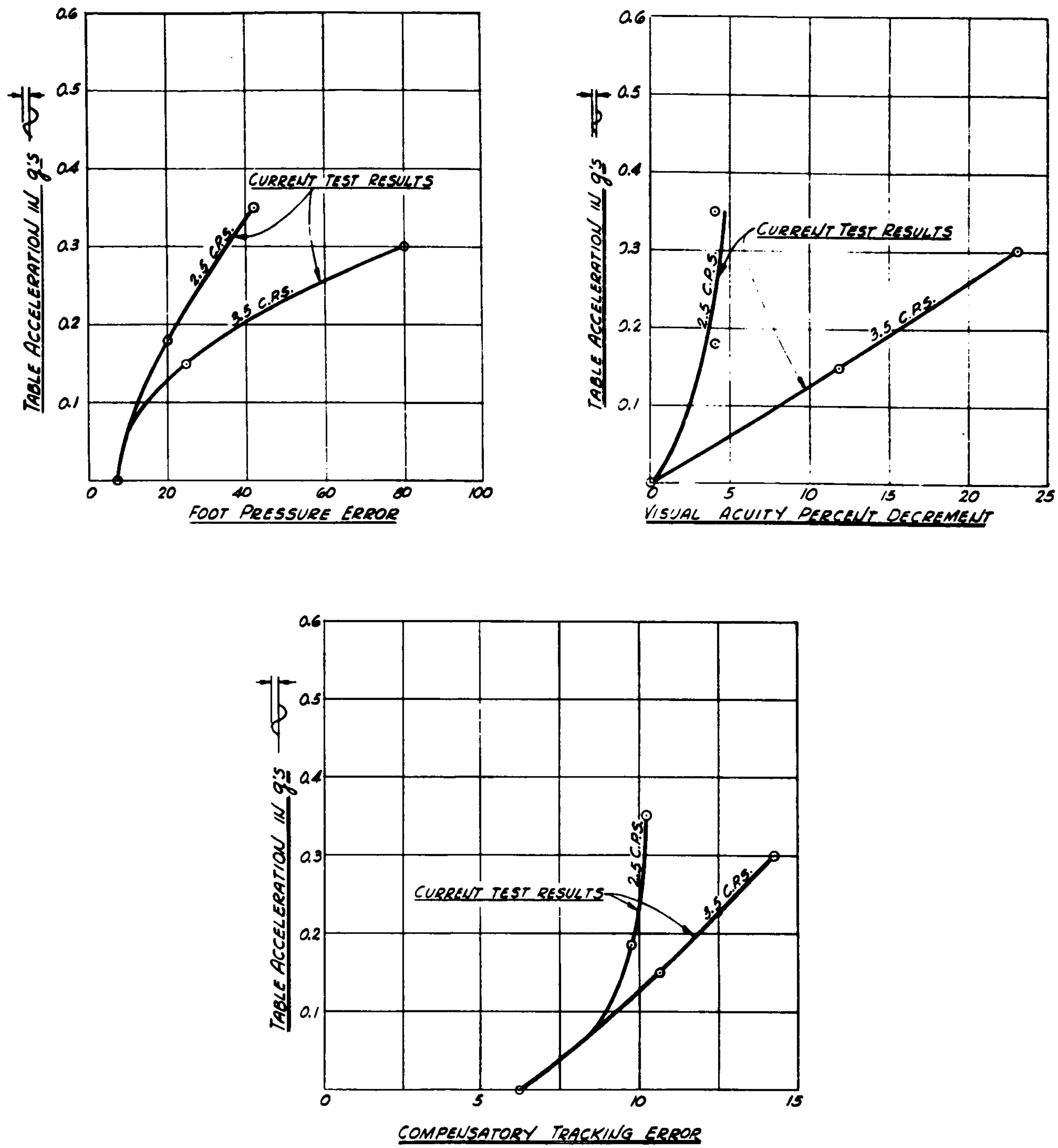

Figure 14.- Performance error as a function of vibration frequency and intensity for foot pressure, visual acuity, and compensatory tracking. (1 cps = 1 Hz.)

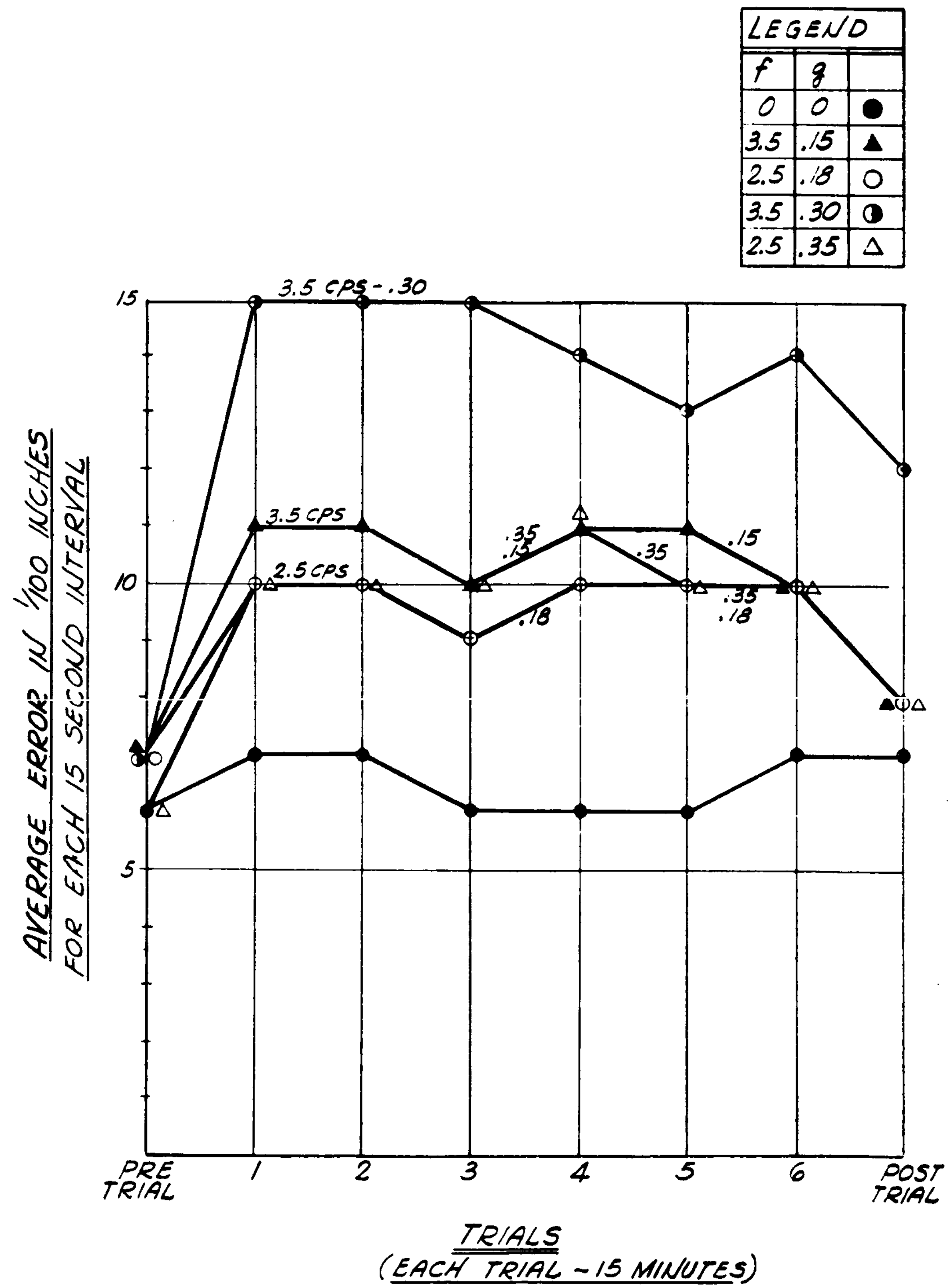

Figure 15.- Performance error as a function of vibration exposure duration. (1 in. = 2.54 cm; 1 cps = 1 Hz.)

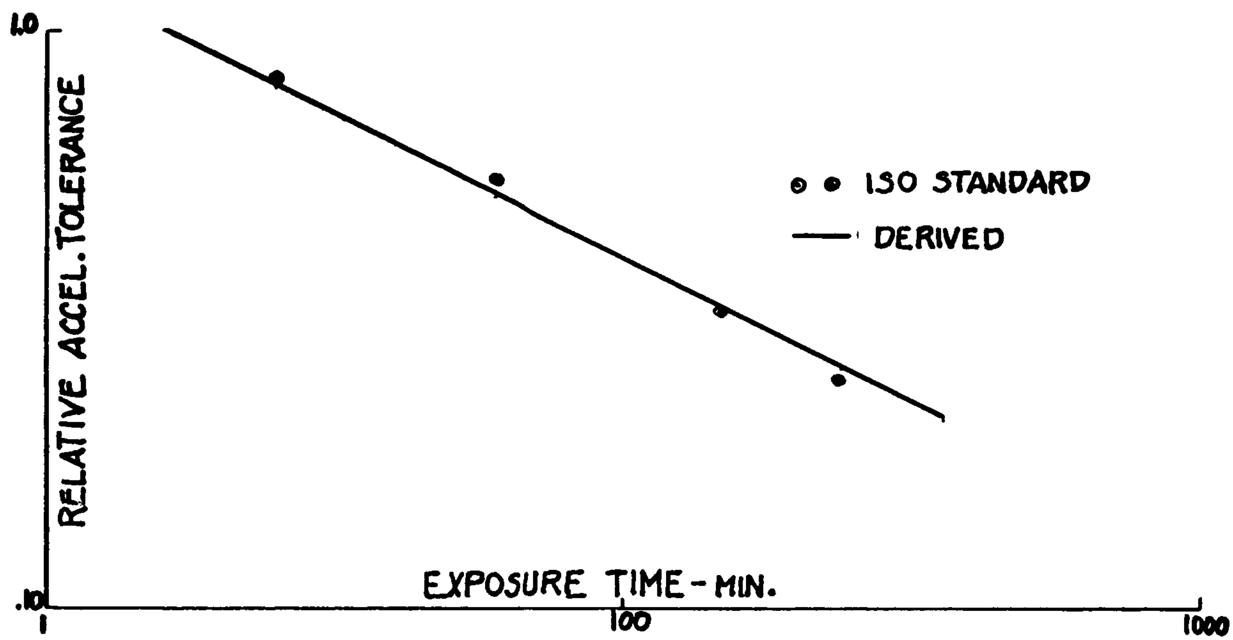

Figure 16.- Relative vibration fatigue tolerance as a function of exposure time, based on constant absorbed energy.

N76-16777

THE ISO STANDARD: "GUIDE FOR THE EVALUATION OF HUMAN EXPOSURE TO WHOLE-BODY VIBRATION"*

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INTRODUCTION

After 10 years of intensive work of Subcommittee 4, "Human Exposure to Mechanical Vibration and Shock," of the ISO Technical Committee 108 "Mechanical Vibration and Shock," the first international standard on human exposure to whole-body vibration has been accepted as an ISO standard (ref. 1). The ISO member bodies of 20 countries interested in this subject, including the United States, voted approval of this standard; two countries (USSR and UK) voted disapproval on technical grounds, although one of them (UK) issued basically the same document as a provisional national standard (BSI DD 32). The United States national vote was strongly in favor of adoption of this document as an ISO standard and as a national ANSI standard (29 in favor, 2 against); however, final submission of the document as an ANSI standard was delayed awaiting the outcome of the international deliberations. All U. S. Government Agencies with an interest in the area of human vibration exposure, including the Department of Transportation, were strongly in favor of the standard. As Chairman of the ISO subcommittee which prepared this document, I am gratified to see such unusually broad support for a document which tries to provide standard guidelines in an area where nothing existed and where data points and opinions were very far apart. The ISO standard stimulated, during its draft stages and during the short time of its existence, a large number of clarifying studies directed to fill in gaps in our knowledge or to support or refute positions adopted in the standard, and fostered international collaboration in this area to an unprecedented degree (ref. 2). This newly accumulated body of information must clearly be taken into account to understand fully the background of the present standard and future standardization plans. I am the first to admit that the present ISO standard is not completely satisfactory in all respects; every standardization, particularly on an international scale, involves technical compromises and compromises between judgements, and some of these will not satisfy everybody. On the other hand, the standard constitutes a tremendous step forward, giving for the first time positive guidance for most vibration exposure conditions incorporating the all-important exposure time as a factor. I can assure you that all decisions and compromises underlying the standard were made after prolonged deliberations taking into consideration the data, or the official comments by the various nations and the expert opinions of the subcommittee members. These

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experts were not only from different geographic locations, but they covered the spectrum of expertise interested in this area from the fields of medicine, physiology, and psychology; the various fields of engineering; the automotive, aircraft, shipbuilding, agricultural, and building industries; and, last but not least, the instrumentation field. Researchers and practitioners from industry as well as from government health departments, were represented. The standard which emerged had to be a compromise, not so much because no adequate data were available but because the large body of data available exhibited a considerable spread (refs. 3, 4, and 5). In most cases the reasons for the differences between the results can be explained by the differences between the experimental conditions and by the differences between the questions asked. Vibrations can act very differently on man; small changes in posture and support can change the effects considerably, and the effects themselves are manyfold and, when it comes to their evaluation, to some extent a matter of judgement. Considering all these variables, agreement of the data from various parts of the world is very good.

The purpose of the ISO effort was to obtain a valid, practical, and safe standard. The first two conditions mean that the standard should cover as many experimental data and practical situations as possible and that simplifications are desirable to facilitate the standard's use. The last condition calling for a safe standard means that if there is any doubt where the exposure limits should be, the more conservative (i.e., protective) interpretation be accepted. The introduction to the standard clearly states these basic philosophies and that it is not to be considered a standard setting firm limits but a general guide for the evaluation of vibration exposure with respect to various human responses. It is obvious that humans judge vibration exposures differently depending on the circumstances and the "benefit" derived from the vibration; they are accepted within certain limits in transportation vehicles because the transportation benefit outweighs the discomfort. The same vibration in a private home is intolerable. The standard tried to average over these differences as much as possible to make it as generally applicable as possible. The "discomfort boundaries" and "fatigue-decreased proficiency boundaries" apply generally to the transportation and industrial environment. For ships - or for the population electing to go on ships - they might be somewhat higher; for residential buildings they must be lower. The ISO working group is presently laboring on amendments to the standard providing much more detailed guidance for specific situations. For example, with respect to desirable vibration limits in various types of buildings such as industrial, residential, and hospitals, we are already close to agreement. But for all these special cases the frequency dependence and the time dependence of the human responses stay the same; the recommended boundaries as a whole are shifted up or down on the intensity scale. The measurement methodology, weighting, and reporting of the data, in other words the overall framework of the standard, remain unchanged. With the overall standard agreed upon as the best guideline available, it is very unproductive not to use it or to exercise any parochialism, be it as an individual, as an industrial branch, or as a country. It is important that the vibration environments of all industries and in all countries can be compared with one and the same measuring stick, even if the absolute boundaries selected as goals or specifications are different. As the standard states, one of its principal aims is to encourage, in comparable and reproducible form, the collection of further and better data. The progress achieved through the development and acceptance of this standard is easily demonstrated by the impressive amount of new information which became available in response to the various unofficial draft stages of the standard and now after its acceptance. This information, gathered in many countries, was essential in changing the drafts, in shaping the final form of the standard, and in getting confidence in its validity. (See refs. 6 to 10 and refs. in ref. 1.) Vibrations on tractors, in tanks, in automobiles, in aircraft, on ships, and in buildings were evaluated by means of the standard (ref. 2); frequently the boundaries recommended in the standard, and in some cases their time dependence, were confirmed by these new data. Any criticism of the standard based on arbitrary restriction to a narrow data base and on personal preference (ref. 11), without considering all evidence available and without being familiar with the recent publications in this field, is unjust and unprofessional.

COMMENTS REGARDING SOME OF THE DECISIONS UNDERLYING THE STANDARD

The basic foundations of the standard are the acceleration limits as a function of frequency and of exposure time. Let me make a few comments regarding each of these.

Curves of equal strength of perception as a function of frequency have been measured by many authors. (See refs. 4, 5, and 7 and refs. in ref. 1.) For longitudinal[1] vibrations they show maximum sensitivity, that is, a minimum acceleration required for constant sensation somewhere in the frequency range between 2 and 8 Hz. The width of this minimum changes with posture of the subject, type of sensation, and exposure time - to name just a few of the variables. Its position appears to change slightly with the same variables, the weaker sensations and the relaxed position having the minimum at somewhat lower frequencies. This range of maximum human sensitivity has been explained generally by the physical resonances of body parts and organs occurring in this frequency range (refs. 12 to 15). Although the range is frequently called the range of "principal body resonance", it has been well shown long ago that several "resonances" are involved. For example, for the erect sitting subject the strain in the lower abdomen peaks around 4.5 Hz; in the upper abdomen, at 5.5 Hz; and in the chest, at 6 Hz (refs. 14 and 15). (For the relaxed subject these curves change again.) A subject being asked to report equal strength of sensation or of discomfort does not report sensations in the abdomen, or chest, or head alone but equates the sensations from all receptors and reports one integrated response. This integrated sensation is at some frequencies predominantly determined by abdominal sensations; at other frequencies, by sensations in the chest; and at still higher frequencies, by sensations in the head (refs. 4 and 12). Therefore, measured curves of equal perception usually are not simple physical resonance curves, but at best - assuming that sensations in the different body areas are proportional to the

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[1] [The term follows terminology used in ISO 2631, where longitudinal is defined as foot (or buttocks)-to-head (vertical).]

physical strain - envelopes to a whole series of resonance curves (ref. 14). Once this fact was established, it made the correlation of subjective tolerance curves with any parameter of a simple oscillator - be it strain or power - unlikely (ref. 13). Therefore, looking at a broad frequency range, curves of equal injury, tolerance curves, or curves of equal sensation are almost always "composite tolerance curves," that is, envelopes of the tolerance curves of several individual subsystems, each by itself having maximum sensitivity in a different frequency range. This knowledge also makes it theoretically very unlikely that curves of equal strength of human perception should be curves of equal mechanical power transmitted to the man and absorbed by him. As soon as more than one reasonating system is responsible for the curve of equal perception, and these systems are in series and not parallel, the absorbed power concept cannot be correct theoretically; for example, a very disturbing head resonance at higher frequencies leading to blurred vision is not appreciably reflected in the mechanical impedance at the seat-buttocks interface, which determines the power absorbed by the subject (ref. 16).

Since sometimes simple concepts work even if they are theoretically not fully correct, the absorbed power concept was tried out very early by Coermann (ref. 13) to explain subjective short-time tolerance curves and was found to be unsatisfactory to explain these curves. When Pradko and Lee (refs. 17 and 18) later revived this idea and made very detailed measurements of the vibratory power absorbed by human subjects, they unfortunately never presented adequate psychophysical or physiological evidence which correlated human response with absorbed power. Without such data the whole approach is a hypothesis not in satisfactory agreement with facts; and all attempts by some individuals to promote this concept have not produced the missing data and agreement and do not change the basic mechanical construction and response of the system man. The limitations and obvious dangers of adopting the concept of constant absorbed power are easily demonstrated by considering the frequency range below 1 Hz: The concept of constant absorbed power predicts increasing human tolerance with decreasing frequency ("constant jerk hypothesis"). Contrary to this prediction, it is well documented that human tolerance has a peak close to 1 Hz and drops off rapidly below 1 Hz (fig. 1) (refs. 19 to 21). Why? Because there is another resonance system in the body which has its maximum response below 1 Hz and results in the complex phenomenon of motion sickness. This system needs much less power to excite it to undesirable responses than the power required to excite the main body system in the frequency range for which the absorbed power concept appeared to be a reasonable approximation. As long as a hypothesis such as the concept of constant absorbed power can lead to such obviously wrong conclusions, it appears unwarranted to make it the basis for a standard.

For all the reasons discussed, the curves of constant perception (or frequency response curves) in the ISO standard are not curves of equal absorbed power or simple resonance curves, but reasonable envelopes to the experimentally observed curves of constant human response. Since these curves change with body posture and support, the final curve was selected as an envelope to most experimental results - arguing that the standard should protect the man against physical harm or undesired psychological responses regardless of whether he sits erect or relaxed on the vibrating seat and free or supported by a backrest.

The final shape of the longitudinal response curve (break points of the curve at 4 and 8 Hz) was then determined by the standardized frequencies for 1/3-octave band and octave band measurements of the vibration spectrum.

The curves of equal vibration perception for transverse vibration were less well defined than the ones for longitudinal vibration at the time the ISO work started. Fortunately, partly in response to the early ISO proposals, excellent and new results by Miwa (ref. 6) became available, supplementing earlier subjective transverse response curves by Dieckmann (ref. 22). Miwa also established the absolute relationship of the longitudinal to the transverse response curve by accurate psychophysical cross-matching of the perceptions in the two directions (ref. 23). (The data by Lee and Pradko (refs. 17 and 18) on transverse curves of constant absorbed power constitute interesting work, but were never considered to contain enough biological evidence upon which to base curves of equal subjective perception.)

The dependence of the acceleration boundaries on the exposure time deserves some comment too. It is true that experimental evidence of the dependence on exposure time of physiological tolerance limits and fatigue and comfort limits for vibration environments was very scarce at the time the standard was first drafted. However, it became obvious that the discrepancies between the absolute levels of recommended exposure criteria in use by different organizations and in different countries had their origin in the fact that exposure time was not taken into consideration (refs. 4 and 24). I compiled available information in 1964 for the ISO group and published it in 1965 (ref. 25): Short-time physiological tolerance decreased with time (observed from 20 sec to 3 min.); subjective judgement of intolerable exposures and working proficiency exhibited a decrease for exposure times from several minutes to 2 hr; subjective "fatigue" of railroad travelers occurred at lower vibration levels with increasing exposure time (reported for 20 min to 8 hr); and similarly, airline passenger comfort required lower levels with increasing exposure time (up to 1 hr). In addition, there was enough evidence that in residential homes the comfort limits are usually exceeded if the vibration levels are above the threshold of perception. This suggested a limit for continuous exposures of 24 hours per day. On the basis of these data, the dependence of the equal perception contours on the exposure time was introduced into the ISO standard. The same time dependence was assumed for physiological limits, fatigue-decreased proficiency, and comfort. The reason for this was that there was not too much latitude for these curves to be drastically different (the comfort boundaries should generally not cross the fatigue-decreased proficiency boundaries, by definition, and similarly the fatigue boundaries are not expected to cross the physiological exposure limits) and the standard had to be not too complex for operational use. One fact which is frequently overlooked has to be kept in mind: The recommended exposure times are for daily routine occupational (habitual) exposures for extended periods, even a lifetime. It is therefore difficult to compare these exposure times recommended for preventive medicine practices or for the prevention of malfunction and accidents due to the vibration environments with experimental laboratory findings. Laboratory tests usually employ a few healthy young subjects not exposed to vibration day in and day out. Therefore in any practical guidance a conservative approach was indicated - boundaries which would not recommend exposures to levels not proven as safe or not presently tolerated in practice. The long-term health effects from chronic exposure to high vibratio levels in some transportation vehicles are still an open question; however, evidence is strong enough to suggest that chronic effects on the musculoskelet system of vibration exposure from presently accepted vehicles cannot be overlooked (refs. 26 and 27). All these arguments spoke in favor of the adopted time dependence of recommended exposure.

Since the publication of the draft standard, several attempts have been made to prove or disprove the time-dependence function (refs. 8, 10, and 28). Data were collected with respect to equal fatigue curves under vibration as a function of exposure time. These data give some additional support to the time dependence selected by the standard (fig. 2). The suggestion to use, instead of the present ISO curve faired through the experimental data, a simil curve represented by a simple analytic expression (refs. 8 and 11), is a good and valid one and might well be adopted in a future revision of the standard without changing the results obtained with it in any appreciable way.

Other experiments concentrated on the one valid question and criticism with respect to the time-dependence concept in which I fully concur and which, by the way, was the main reason for the UK negative vote: It has never been shown by laboratory experiments that task proficiency and performance in general decrease with increasing exposure time. On the contrary, several experiments designed to investigate this question so far obtained a different answer; even 6 hr of vibration exposure did not result in a significant decrease in cognitive or manual performance capability (refs. 5 and 28). In addition, available data clearly show that it is almost impossible to generalize with respect to "performance" under vibration. The nature of the task has been shown to be extremely critical and time dependence must be expected to be related to the nature of the task under consideration. In spite of this it was decided to retain the time dependence not only for the fatigue boundaries, for which it was confirmed, but also for the otherwise identical decreased proficiency boundary. The argument for this decision is based on the experience that laboratory experiments hardly simulate daily, lifelong field exposure with respect to motivation; and if people report increasing fatigue with exposure time, the fatigue can result, at least in some individuals, in decreased motivation and increased error or accident potential. It also appeared unreasonable to recommend boundaries with respect to performance which would be above the fatigue limits and might even cross and be above the exposure limits adopted for health reasons. Such a standard would not fulfill its practical, preventive purpose.

Although stressed in several places in the standard, it might be worth a reminder again here: The disturbance of task performance in the vibration environment depends very much on the task required, and a large body of detaile information exists in the meantime on this subject (refs. 5 and 29). A promising open field in human engineering is the design of controls and display for minimum interference by vibration (ref. 2). Guidelines for this are available. The "decreased proficiency" curves in the standard provide, therefore, some very general guidance only and might have to be moved up or down depending upon the specific task, the man-machine interface, and the reliability required (On the other hand, these curves should not be above the exposure boundaries except for unusual conditions.)

Finally, it is completely misleading to state, as some self-appointed interpreters of the standard do (ref. 11), that the ISO standard presents the "fatigue-decreased proficiency" boundaries as the "primary limits," or that the ISO group first decided on the "reduced comfort" boundaries and then selected the other boundaries "by arbitrarily multiplying these values" with the recommended factors to obtain the other boundaries. Let me assure you, there was nothing "arbitrary" about it. In the evolvement of the overall framework and the recommendation of specific boundaries as a function of frequency and exposure time, equal consideration was given to the data accumulated for each of the three perception criteria selected for characterizing recommended exposure levels: The "exposure limit" for health reasons, the "fatigue-decreased proficiency" boundary, and the "discomfort" boundary. If there was anything arbitrary about this, it was the desire to arrive at a practical, useful, and safe standard on which the majority of the experts could agree.

FUTURE WORK ON VIBRATION EXPOSURE STANDARDS

The ISO standard has found wide application in the shipbuilding, aircraft, automotive, and building industries (ref. 2). The U. S. Department of Defense and many countries made it the basis for their military specifications (ref. 30). The standard is being used as the basis for the international activities on tractor seat testing (proposals from the O. E. C. D. Committee and from ISO/TC23/SC3) and for national recommended practices (refs. 31 and 32). For practical evaluations and comparative tests the single-number characterization (frequency weighting) methodology by way of frequency weighting networks ("ride meter") is generally preferred (ref. 31), which the standard proposes as an approximation. The fear is justified that if this were the sole methodology for assessing vibration exposure (some countries propose already to standardize such a meter), no spectral information on the various occupational environments would become available and further research data on the correlation of human response with the spectrum of the environment would not be forthcoming. For these reasons it does not appear desirable to standardize too early on a general vibration exposure meter (which would probably require narrower tolerances than presently proposed in the standard, an accuracy perhaps not yet justified), although such a meter definitely has its place for the testing in specific industries. When more experience has accumulated, such a vibration exposure meter should be internationlly standardized.

The ISO Subcommittee on Human Exposure to Mechanical Vibration and Shock is presently working on several projects, some of which are closely related to the standard under discussion (ref. 20):

(a) A document on the evaluation of vibration in buildings (including acceptable acceleration ranges for various uses) is nearing completion and will soon be distributed for vote and comment. This document provides for longitudinal and transverse vibration to be evaluated, if desired, by a combined, averaged weighting function. Justification for this was the argument that in buildings the same environment can act on man in all directions (upright and in the horizontal position) and the expressed desire of the industry to have in addition to the existing standard a simple, single-figure evaluation method.

(b) An amendment or appendix to ISO 2631-1974 (E), which we hope is clc to completion, provides some guidance for the frequency range 0.1 to 1 Hz with respect to motion sickness and equal sensation contours (ref. 19). (It will probably be decided as too controversial to continue the reduced comfort or other boundaries from the higher frequency range below 1 Hz with the same designation, since the definition and causes of discomfort due to motion sickness are too different from the phenbmena above 1 Hz.) The guidance might be similar to the information presented in figure 3.

(c) A standard document defining whole-body impedance curves for human subjects will soon be released by the subcommittee for official comments by the ISO countries (ref. 20). The curves are to be used as nominal impedance curves for design and testing and are supplemented by biodynamic models exhibiting the same impedance curves as the human body. Vibration transmission through the body is on the agenda for future working group meetings.

(d) A draft proposal for "Guide for the Evaluation of Human Exposure to Hand-Transmitted Vibration" has seen at least three or four revisions and is being circulated again to the subcommittee for comments and vote (ref. 20). International agreement is difficult to achieve, since several countries alrea have standards or guidance not in agreement with each other or in agreement with present thinking. On the other hand, international guidance and agreement are urgently desired because of the active trade in power tools, such as chain saws, for which the measurement and definition of permissible limits of hand-transmitted vibration, according to an international standard, is highly desirable.

(e) A standard terminology on human shock and vibration exposure is in preparation and is planned for issue as an independent standard and also for incorporation into the general ISO/TC108 "Shock and Vibration Terminology", which is in preparation. This work will be followed by a separate draft document on "Biodynamic Coordinate Systems."

(f) In addition to the efforts listed in the area of human vibration exposure, several documents on human impact testing and evaluation are in preparation.

This list of projects is by no means complete, and I hesitate to predict how soon any of these efforts may result in an approved ISO standard. Comment received through the process of official international circulation can frequently influence or change subcommittee plans and new data might turn up not considered during working group deliberations. All such comments will be carefully evaluated and every effort will be made to obtain as broad support as possible by the international community. This process might not only involve technical changes to the proposed documents but also changes in the overall plans and packaging of the documents. For example, if the planned amendments to ISO 2631, addressing the frequencies below 1 Hz and the vibratio in buildings, should be delayed too long, the subcommittee might consider not issuing such separate documents but incorporating all these amendments into a future revision of the basic guide for the evaluation of human exposure to whole-body vibration. This future revision will also include, in all likelihood, more specific guidance with respect to exposure boundaries for specific situations or industries, such as ships, shops, and tractors. However, more field data on environments and on human responses are desirable to take this step with confidence. In the meantime it is important that these field data be collected and reported uniformly and consistently. The present ISO standard plays an important role in this process. Ongoing investigations will not only result in practical field experience with the standard but will also give new data on problems such as the accuracy of the weighting vs. the rating approach; on single-frequency, multiple-frequency, and random vibrations; on simulataneous multiple-axis vibrations; and on the important problem of impulsive-type vibration, that is, vibrations with a crest factor greater than 3. Data for the establishment of boundaries for rotational vibrations might become consistent enough for inclusion. For all these problems the present standard does not yet give the ultimate answer, but it gives coherent guidance consistent with present knowledge and with sound preventive practices. The ISO subcommittee monitors all these areas and is ready to incorporate any new generally accepted evidence into a future revision of the standard.

Another effort of interest might be mentioned: In its work Subcommittee 4 found it desirable to have standardized environmental inputs available which would be representative of typical field environments imparted by road vehicles, off-the-road vehicles, ships, aircraft, and so forth. At the Subcommittee's request, Technical Committee 108 organized a special working group (ISO/TC108/WG9, "Generalized Road Vibration Inputs to Vehicles"), which has already circulated a draft document for comment proposing standardization of the description and characterization of generalized road, runway, or field inputs into the transportation vehicles. This permits standard description of the quality of these surfaces with respect to vibration generation and will assist in the uniform specification, design, and evaluation, with respect to human response, of such systems and their components.

CONCLUDING REMARKS

In summary, I hope the foregoing remarks provided assurance that

1. The present ISO "Guide for the Evaluation of Human Exposure to Whole-Body vibration" is based on all relevant information presently available and reflects the best judgement of all international experts and all disciplines involved.

2. Advancement in the state-of-the-art is most rapidly achieved by data collection and reporting according to this standard, a proposal which should not stifle parallel research on new approaches and methodologies.

3. Work is continuing to amend and improve this standard as soon as warranted by new data, and everyone is encouraged to submit such data and participate constructively in the standardization process.

4. For the present time the evaluation of vibration environments, occupational as well as recreational, and the testing of equipment and machinery with respect to its effects on man are best accomplished by means of the existing standard to protect man against undesirable effects on his health, safety, performance, and comfort.

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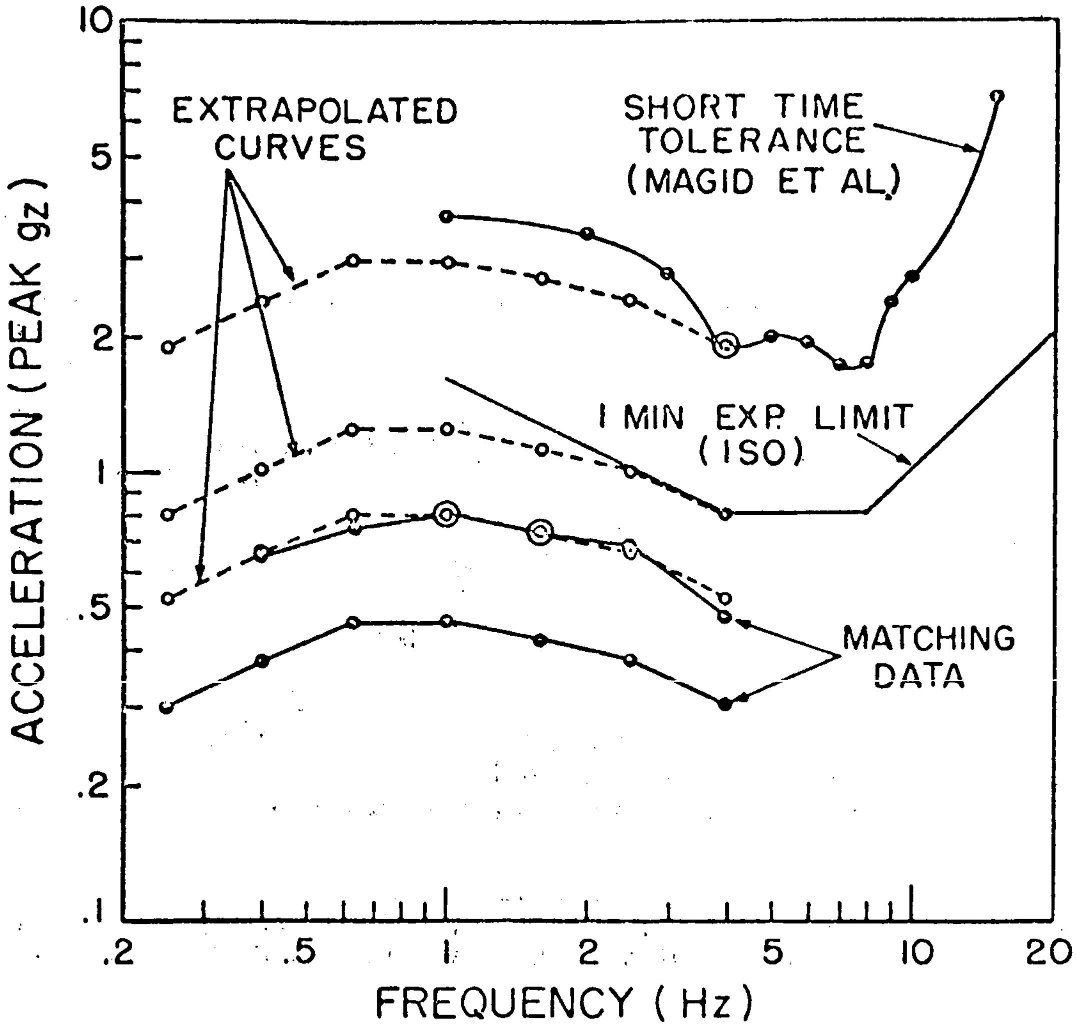

Figure 1.- Curves of equal subjective vibration (g_z) intensity (0.25 to 4.0 Hz). Each match involved a 30-sec exposure at the standard frequency of 1 Hz and at the matching frequency. The extrapolated dashed curves were obtained by magnitude estimation tests at constant frequency. (See ref. 21 for detailed procedures.) The curves are compared with the tolerance data by Magid et al. (ref. 12) and the slope of the ISO Standard curves.

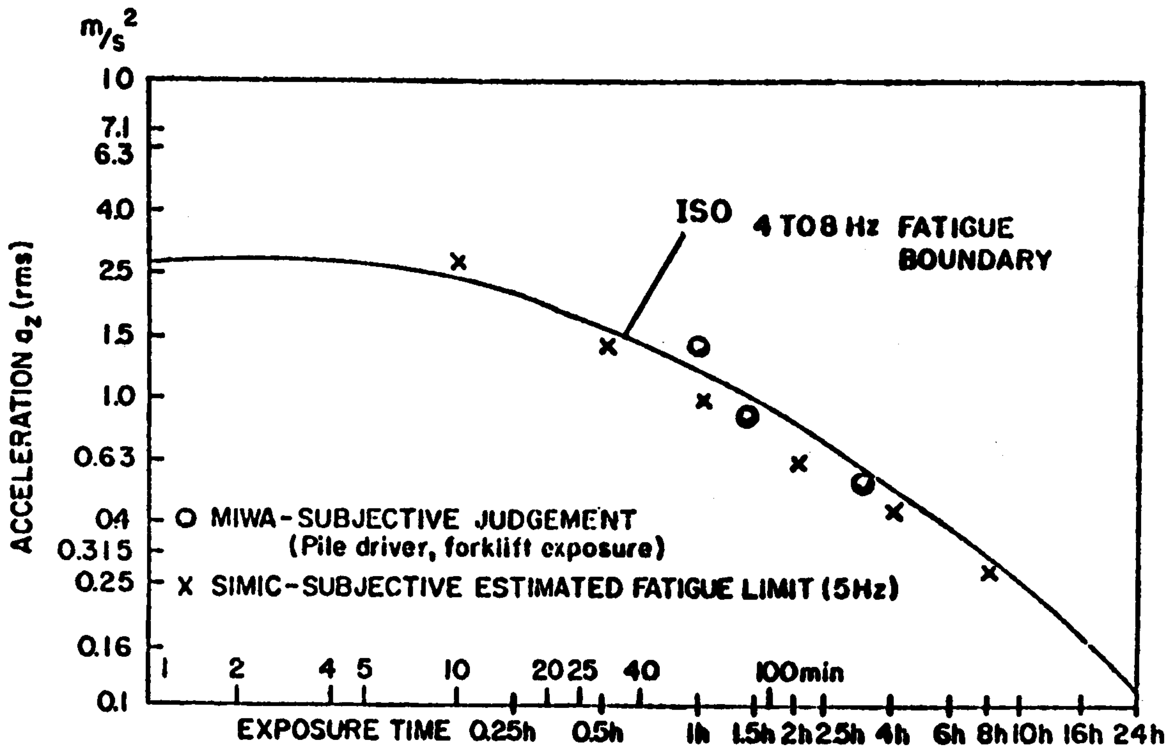

Figure 2.- Subjective judgement of equal fatigue compared with the ISO Standard curves (based on refs. 8 and 10).

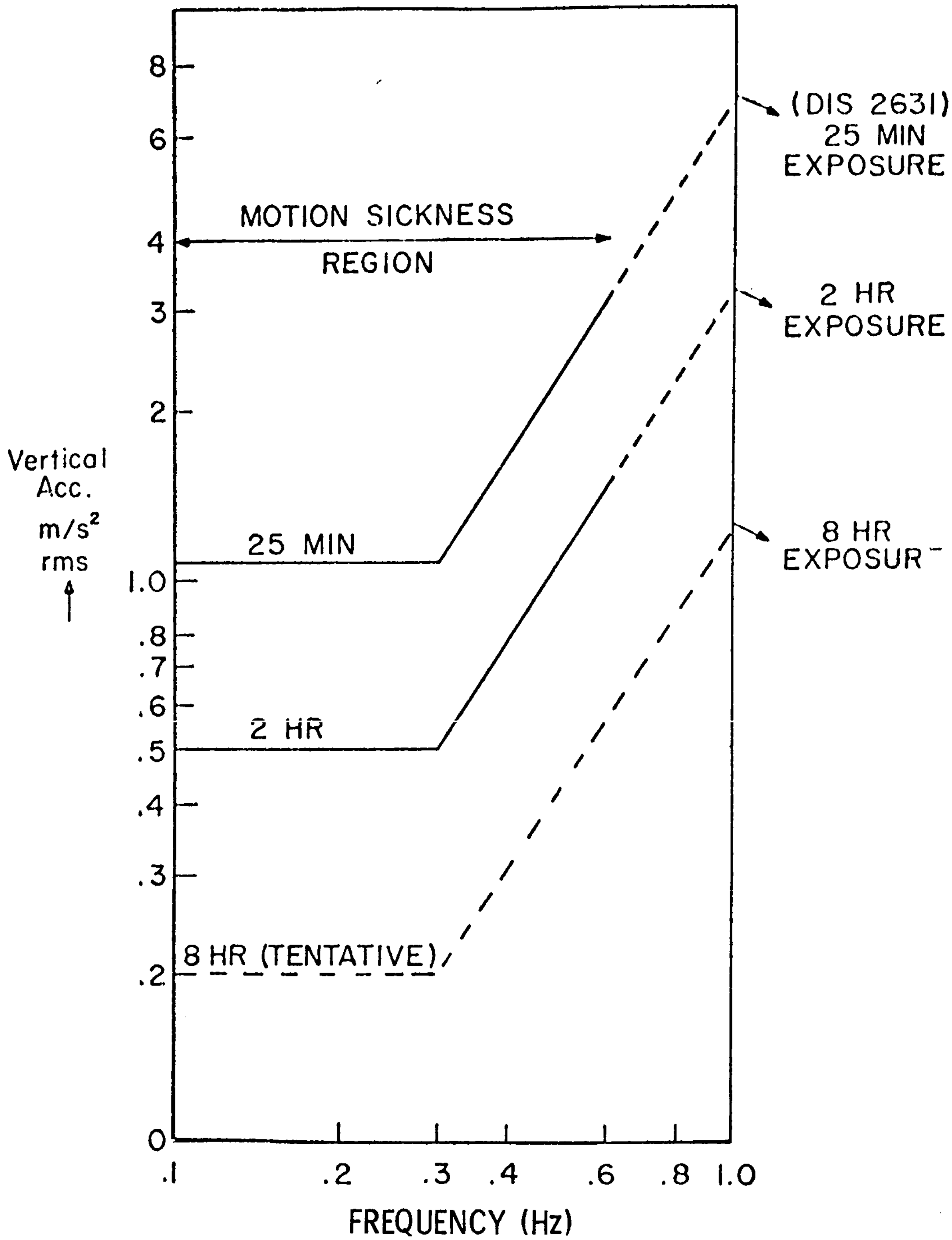

Figure 3.- Proposed "severe discomfort boundaries" for the 0.1 to 1 Hz frequency range (ref. 19). The curves are primarily based upon all available motion-sickness data. (The curves of equal subjective short-time intensity perception (fig. 1) would result in slightly more shallow slopes.)

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AUTOMOBILE RIDE QUALITY EXPERIMENTS CORRELATED TO ISO-WEIGHTED CRITERIA

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SUMMARY

As part of an overall study to evaluate the usefulness of ride quality criteria for the design of improved ground transportation systems an experiment was conducted involving subjective and objective measurement of ride vibrations found in an automobile riding over roadways of various roughness.

Correlation of the results has led to some very significant relationships between passenger rating and ride accelerations. The latter were collapsed using a frequency-weighted root mean square measure of the random vibration. The results suggest the form of a design criterion giving the relationship between ride vibration and acceptable automobile ride quality. Further the ride criterion is expressed in terms that relate to rides with which most people are familiar.

This report deals with the design of the experiment, the ride vibration data acquisition, the concept of frequency weighting and the correlations found between subjective and objective measurements.

INTRODUCTION

Understanding the contributing factors of "good ride quality" and criteria for design of vehicle suspension and isolation systems has been of interest in the transportation community for some time.

Ground transport vehicles ride over road and railways which are, in general, rough. Smooth riding is not always achieved. Vibrational characteristics impressed on passengers are generally random with some dominant harmonics and usually, some form of mechanical isolation system is provided to insure smooth riding while at the same time, secure holding to the road or rail.

Early approaches to classify contributing factors to ride quality involved separate experiments with humans on shake tables. With sinusoidal excitation, subjects were asked to rate various vibration amplitudes at select frequencies. Averages of ratings then yielded curves of constant levels on a plot of amplitude in g's versus frequency as in Fig. 1 from [1]*. The records of measured vehicle accelerations, while random in nature, sometimes contained dominant frequency components. Where obvious components could be identified from measured acceleration records, examination of the record led to average amplitudes for each of these components. Comparison with laboratory shake-table results then yielded an average rating for each component. Components were combined into a composite rating using the tenth root of the sum of rating each raised to the one tenth power [2].

This approach, however, leaves something to be desired in many situations where the riding vibrations are in fact random with broad band characteristics. It is certainly possible to calculate a mean square value for a broad-band random signal but this value cannot be associated with any particular frequency. Alternatively, filtering using a narrow-band filter centered at some particular frequency may be used to compute an equivalent root mean square value at that center frequency, but the question of what band with filter to use is central to the issue and unknown.

Because of the general shape of the lines of constant comfort shown in Fig. 1, it is apparent that humans are more sensitive to frequency components in the 4-15 Hz range than to components in either the lower or higher range. Reflecting this fact for broadband random signals, acceleration power spectra values are weighted in that mid-frequency range by the inverse of a response curve such as is Fig. 1. The root mean square or the frequency-weighted root mean square accelerations are then used as measures of the ride. This approach has been used and is discussed by Van Deusen [3] and Butkunas [4]. In this work, better correlation with passenger rating than previously found is sought.

Discussion of the techniques of acceleration measurement and data reduction follow after a discussion of the general design of the experiment in which r.m.s. and frequency-weighted r.m.s. acceleration measures for a number of rides are compared with passenger subjective responses.

*Numbers in brackets indicate references.

DESIGN OF THE EXPERIMENT

General Considerations

The experiment was based on rides in a late model Buick Century automobile over selected roadways around the Austin, Texas area. A total of thirty-six subject raters and a total of six routes each having three sections of roadway were used in the study. Since each ride involved a driver (same throughout the experiment) and three passenger raters, a total of 216 rides would have to be required if each rater was to rate each section. This would be extremely lengthy to carry out and it was decided on the basis of previous experience dealing with road pavement rating that twelve raters for each ride would be sufficient. In this design three raters were used on each excursion and in all, each rater rated six rides while each ride was rated by twelve raters. The design requires the repeatability of conditions from one day to the next and several test runs were made initially to develop driver technique such that repeatability could be insured.

Background variables such as personality measures, age and type of car normally driven were recorded for each rater so that correlation with ratings could be explored later.

A partial tabulation of the experiment is given in Fig. 2. Six sections were chosen for each excursion. Repetition four times of each excursion in different order with other subjects provided the required number of tests to average within subject variables and ride variables.

Routes

In this experiment the highway sections were divided into six routes. Each route contained three sections of highway and within each route, one of the sections was chosen as relatively rough, one medium and one smooth. These routes were chosen for convenience and were all located near Austin. They were chosen also to have as wide as possible variation of irregularity.

Design

Each subject rode over two routes and made six ratings of ride quality. To avoid the possibility of bias in the ratings of ride quality, sequence of sections within routes was randomized systematically. That is, the three sections in each route were either traversed in a single fixed order (1, 2, 3 or in reverse order (3, 2, 1). In addition, the sequence or relative roughne varied from one route to another such that approximately one-third of the tim the first section rated was smooth, one-third of the time it was medium and o third of the time it was rough. Similarly, the second and third rated sectio were about equally often smooth, medium or rough. In this way if subjective ratings change systematically with time because of such variables as boredom or fatigue, then these sources of bias are distributed approximately equally over the ratings of the various sections within routes.

Subjects

Thirty-six subjects in the experiment were divided in three groups. The 12 subjects of Group 1 rode over two different routes, the 12 subjects of Group 2 rode over two other routes and the 12 subjects of Group 3 rode over still another pair of routes. The 24 subjects of the first two groups were obtained from introductory psychology classes and served in the experiment to fulfill a laboratory requirement. Group three was composed of some non-studer being wives of faculty or female secretaries.

Procedure

The subjects served in the experiment three at a time. In each case the met at the University of Texas and were driven from there to the appropriate site with two in the back seat and one in the passenger's seat. For all tests the car and the driver were the same. Care was taken by the driver to maintai the same standard conditions from one rating session to another. That is, talking between the subjects was not encouraged and the driver remained seriou and as business like as possible to emphasize the serious scientific nature of the project. Upon reaching the section to be rated, the car was driven over it at a standard 80.5 km/hr (50 mph). After completing the ride over the section the subjects rated ride quality using a 1 (rough) to 5 (smooth) rating scale. Ratings were anchored by instructing each subject to use a rating of "1" to indicate "the worst ride I can think of" and to use a rating of "5" to indicate "the best ride I can think of." Fig. 3 shows the rating form used.

Based on the above, each highway section had associated with it a) a number (12) of individual ride ratings and b) a measured acceleration record. During separate tests, vibration records were obtained for each ride. The car was loaded by two passengers and the recording equipment which took the place of the third passenger. The same driver was used. Sufficient number of trials insured the repeatability of data gathered in this manner. At 32.2 km/hr (20 mph) some trouble with repeatability was found but this seemed to disappear at the test speed of 80.5 km/hr (50 mph). Measured accelerations for both vertical and transverse directions were recorded as discussed in [5].

The important variables to be correlated here were the average rating for each ride with r.m.s. and frequency-weighted r.m.s. acceleration records.

ACCELERATION MEASUREMENT AND DATA REDUCTION

For each test section used, a separate recording was made of the acceleration response of the body of the Buick Century. A location below the drivers seat was chosen, close to the pitch and roll axis. With an automobile such as the Buick Century, roll and pitch acceleration components contribute a small amount to the total vertical and lateral motions and variations in rating due to position in the car were not distinguished. The acceleration response was measured using the 3-axis ride accelerometer developed by NASA [6]. This accelerometer system has a bandwidth of about 100 Hz and is accurate to 5% from 0 to 25 Hz.

The measurements of both vertical and transverse acceleration were recorded on a TEAC tape recorder. A test duration of about 15 seconds was used. The analog tape recording was then digitised using the HR2115A processor run by the Texas Highway Dept. with a 100 Hz bandwidth aliasing filter. A sample rate of 434 Hz insured that aliasing errors were minimized in the digitising process [7] and the digitised data were processed on the University of Texas C.D.C. 6600 system.

A data sequence of 4096 points was selected from each record. The mean value was extracted and the power spectral density of each record was computed using a fast fourier transform algorithm.

In the computation of power spectra, the raw power spectral density values obtained on the basis of a finite number of points must be smoothed by averaging adjacent points. In our work, the smoothing was done over three, five and nine times the discretisation frequency (0.106 Hz) for the 0-1 Hz, 1-10 Hz and 10-217 Hz ranges respectively. Thus smoothed power spectral density values were obtained as a sequence

$$\bar{p}_0, \bar{p}_1, \bar{p}_2 \cdot \cdot \bar{p}_k -- \bar{p}_{n-1}$$

Details of the computation procedure are given in [8].

The mean square value x^2 of each record was then obtained by summing individual raw power spectral density values p_k and dividing by the total record time T. Thus,

$$x^2 = \frac{1}{T} \sum_{0}^{n-1} p_k$$

and the root mean square (r.m.s.) value is then $\alpha = (\overline{x^2})^{1/2}$

In dealing with the frequency weighting concept the 8 hour ISO Standard (Fig. 1) was used as a basis. If the inverse curve of Fig. 1 is designated by A(f) then the frequency weighting of each raw power spectral density value p_k is achieved by a weighting constant w_k. Where

$$w_k = \frac{A^2(f')}{\frac{1}{(b-a)} \int_a^b A^2\,(f)\,df}$$

and the argument f' is made to correspond to k times the discretisation frequency of 0.106 Hz. The denominator above was introduced to normalize the weighting constant and corresponds to the mean square value of A(f) in the range of interest. The lowest frequency, a, was taken to be 0.1 Hz and the upper frequency, b, was taken to be 40.0 Hz. The vertical and lateral weighting functions are given in Figs. 4 and 5.

For each ride in the experiment, then, an r.m.s. and a frequency-weighted r.m.s. (w.r.m.s.) value was obtained for both vertical and lateral acceleration.

RESULTS

Initial Data Analysis

Each highway test section was identified by a Texas Highway Dept. number. Rides over each test section gave rise to twelve individual ratings. Table I below gives the mean rating R for each test section as well as the measured r.m.s. and w.r.m.s. values for vertical and lateral acceleration.

Table I.

[All accelerations are in g units]

Highway Section No.	R	r.m.s. Vertical	r.m.s. transverse	w.r.m.s. vertical	w.r.m.s. transverse
1	3.19	0.042	0.022	0.080	0.089
2	2.72	0.063	0.027	0.132	0.136
3	2.94	0.046	0.021	0.094	0.096
5	4.35	0.027	0.014	0.055	0.049
6	2.83	0.060	0.026	0.130	0.104
7	4.31	0.025	0.026	0.051	0.062
8	3.50	0.042	0.016	0.086	0.057
9	4.16	0.027	0.014	0.045	0.065
10	4.25	0.034	0.014	0.049	0.047
12	3.17	0.059	0.023	0.109	0.070
15	2.75	0.068	0.029	0.116	0.097
36	4.25	0.035	0.013	0.045	0.041
37	2.41	0.064	0.024	0.138	0.067
39	2.33	0.054	0.032	0.128	0.164
41	3.00	0.055	0.023	0.124	0.092

Only fifteen section numbers appear above since sections 5 and 37 were used twice in making up routes for two subject groups. They have, as a result, a basis of twenty-four ratings in the computation of the average. Section 35 has been excluded from the analysis since some errors were suspected in the measurement of its transverse acceleration.

The data in Table I has been analysed to yield product moment correlation coefficients defined as

$$r_{xy} = \frac{\frac{1}{N}\sum_{i=1}^{N} x_i y_i - \bar{x}\bar{y}}{\sigma_x \sigma_y}$$

Also, the first order significance and the significance of the quadratic term were computed. The correlation coefficient relates the degree of statistical correlation between two independent sequences and usually indicates significant relationships if values of $r_{xy} > 0.5$ are obtained.

As an added presentation of the data, Figs. 6-9 are provided here in which the r.m.s. vertical and transverse and the w.r.m.s. vertical and transverse accelerations are plotted versus the rating R. Also given is the appropriate correlation coefficient r_{xy}.

Examining Fig. 6 it is apparent that the r.m.s. vertical acceleration is extremely well correlated with the average rating. The correlation coefficient of 0.93 is rarely obtained in statistical experiments. The significance level based on the assumption of normality was better than 10^5.

Comparing Figs. 6 and 7 it is apparent, however, that transverse accelerations generally do not correlate so well with the ratings. The correlation coefficient of 0.779 is still significant but lower than that of the vertical acceleration. The point to the high right hand side which seems to be outside the main concentration of data is from section 7. This is a relatively good quality section of U.S. 71 -- a 4 lane highway. This section has the distinction of giving approximately equal r.m.s. levels of both vertical and lateral accelerations. The roughness characteristics are such that the vertical roughness of each wheel track is low (0.68 cm (0.267 in.) and 0.60 cm (0.237 in.) r.m.s. for right and left respectively) but their differences are relatively important thus inducing larger than expected transverse accelerations. Vertical and transverse r.m.s. accelerations are generally correlated. Here the correlation coefficient was 0.702.

Effect of Using ISO Weighting Function

The correlations with the ISO-weighted r.m.s. vertical and transverse accelerations are shown in Figs. 8 and 9. These show that correlation is definitely improved to 0.95 by use of the weighting function for the vertical acceleration but the data is more spread out for the transverse case (r_{xy} = 0.704). Now, some sections exhibit strong values of weighted r.m.s. acceleration and the same range of values is covered by both components. Apparently, raters respond more to the vertical sensation in the case of the automobile.

Use of Acceleration Vector Measure

There are two ways to combine vertical and lateral accelerations: either as a sum or as a product. Here it seems reasonable to make the vector sum of the two components, combining them into a vector magnitude. The correlation of the r.m.s. magnitude and the w.r.m.s. magnitude is given in Figs. 10 and 11. The correlation coefficients are respectively 0.909 and 0.931.

Again, the use of the ISO weighting functions has improved the correlation coefficients. Also, it is apparent that the magnitude values correlate worse than the vertical w.r.m.s. which in this test was found to be the best correlator with mean ride rating. The latter fact is attributed to the inclusion of transverse accelerations which generally correlate worse than their vertical counterparts.

Significance Levels

Significance levels were computed for each plot shown in Figs. 4-9. The two worst values were found as expected with the transverse accelerations being 0.0004 and 0.0023 for the r.m.s. and w.r.m.s. values respectively. All others had significance levels less than 10^{-4}. This means that the probability, assuming normality, of finding this correlation through a random test is less than 10^{-4}.

Variability of ratings by individual subjects played a relatively small role. That is, the variability of individual ratings accounted for about 5% of the total variability while variability of the means accounted for about 95% of the variability.

Other Analysis

A large amount of information was gathered from each subject. In this section a detailed account of the analyses attempting to find relationships between these variables will be given.

Each subject was given a personality test and a measure of "neuroticism" was obtained. On this basis the subjects were divided into a high-neurotic group--those subjects scoring 10 or above--and into a low-neurotic group--those subjects scoring 5 or lower. After this, ratings of ride quality were again analyzed. However, no difference was found between high and low neurotic groups with means of 3.23 and 3.38 respectively.

Another personality inventory measured degree of extroversion-introversion The hypothesis tested here was that the extrovert being more aware of his surroundings would be more influenced by the quality of the ride with the consequence that a rough ride would be rated rougher and a smooth ride would be rated smoother than would the case for the introvert who would be less influenced by external than internal cues.

Again subjects were divided into two groups with introverts being defined as those who gave scores of 12 or lower on the scale and with extroverts being defined as those who scored 19 or higher on the same scale. As in the previous analysis ratings of ride quality were analyzed and again ratings varied directly as a function of the acceleration. No difference in ratings was found as a function of extroversion-introversion. However, the interaction between extroversion-introversion and rating was in the predicted direction with ride ratings on roads rated as rough being rated as rougher by the extroverts than by the introverts. Unfortunately, this interaction was not reliable, and the hypothesis of random occurrence cannot be rejected. No other sources of variability approached significance.

The Gottschaldt figures test was also administered. As in the previous analyses, subjects were divided into low and high scorers on the test and the two extreme groups were compared on the basis of their ride quality ratings. Again, subjects scoring high or low on the Gottschaldt test tended to give essentially the same ratings.

The intercorrelations between seven variables were determined. The seven variables were total score (the sum of the six ratings made by each subject), the difference between the ratings for the rough rated roads and the ratings for the smooth rated roads, age, size of car (rated on a 1-4 scale) and scores from the Gottschaldt, Neuroticism and Extroversion-Introversion scales. Of the 21 correlations computed only the correlation between age and total rating was significant. In a subsequent analysis all other variables were held constant and again the same correlation was significant. In general, as age increased the overall rating of ride quality increased. This somewhat counter intuitive result suggests that younger adults are more critical of ride quality than are older adults. These data are also somewhat in agreement with the data of Group III--the older non-student group which was not included in subsequent analyses after the initial analysis indicated that this group gave smoother ratings to rough-rated roads than to medium-rated roads. That is, age and rated quality of ride are positively related in Groups I and II while at the same time the older subjects of Group III are unable to differentiate between a rough and a medium surface road. Thus, age appears to be a variable which is related to rating of ride quality and as such needs to be more thoroughly investigated.

Numerous other analyses were made in the course of the analysis of these data. Most of these other analyses were done by holding one or more variables constant to determine what effect this had on some other variable. The relatively few significant differences found in these analyses were roughly equivalent in frequency to the number of significant differences which would be expected on the basis of chance occurence.

Relationships Between Roadway Roughness and Ride Quality

In previous experiments [9] dealing with the rating of roadway roughness (not ride quality) the concept of a roadway servicability index (S.I.) has bee explored. The S.I. value of a section of highway is computed from a formulati of average roughness amplitudes in certain wavelength bands. The roadway roughness in this case has to be measured. Both right and left wheel tracks a included. Each Texas Highway Department test section has thus associated with it an S.I. value. These lie in the general range of 2-3 for secondary roads, 3-4 U.S. highway (good quality) and 4-5 for good quality Interstate highway.

In a separate computation the ratings of the first two groups of raters were compared with test section S.I. values. Extremely good correlation resulted as shown in Fig. 12 with a correlation coefficient of 0.91 and a significance level less than 0.01.

The importance of Fig. 10 is to show generally that ride ratings between 4 and 5 correspond to good quality rides over good quality Interstate highway. Lower values, between 2 and 3, for example, correspond to riding over two lane secondary roads. Thus the ride scale may be translated into meaningful sensations to the majority of our automobile-highway driving population.

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

Several conclusions can be drawn from this work as follows:

1. There is a highly significant correlation between the average rating of a particular automobile ride and the r.m.s. vertical acceleration.
2. Measured lateral accelerations do not correlate as highly as the vertical component with average ride ratings.
3. The use of a frequency weighting function based on the ISO Standard appears to improve the statistical correlation between acceleration level and passenger rating.
4. The use of the acceleration magnitude (i.e. $((\text{vertical})^2 + (\text{lateral})^2)^{1/2}$) appears to improve the correlation slightly.

Recommendation

It is suggested that Fig. 11 be used as a basis for automobile ride quality design. Ride Index values between 4 and 5 correspond generally to the quality of ride found in a Buick Century riding over Interstate quality roadbeds. Ride Index values between 3 and 4 generally correspond to riding over good quality two-lane highways while the 2-3 range corresponds to riding over secondary roadbeds at 80.5 km/hr (50 mph).

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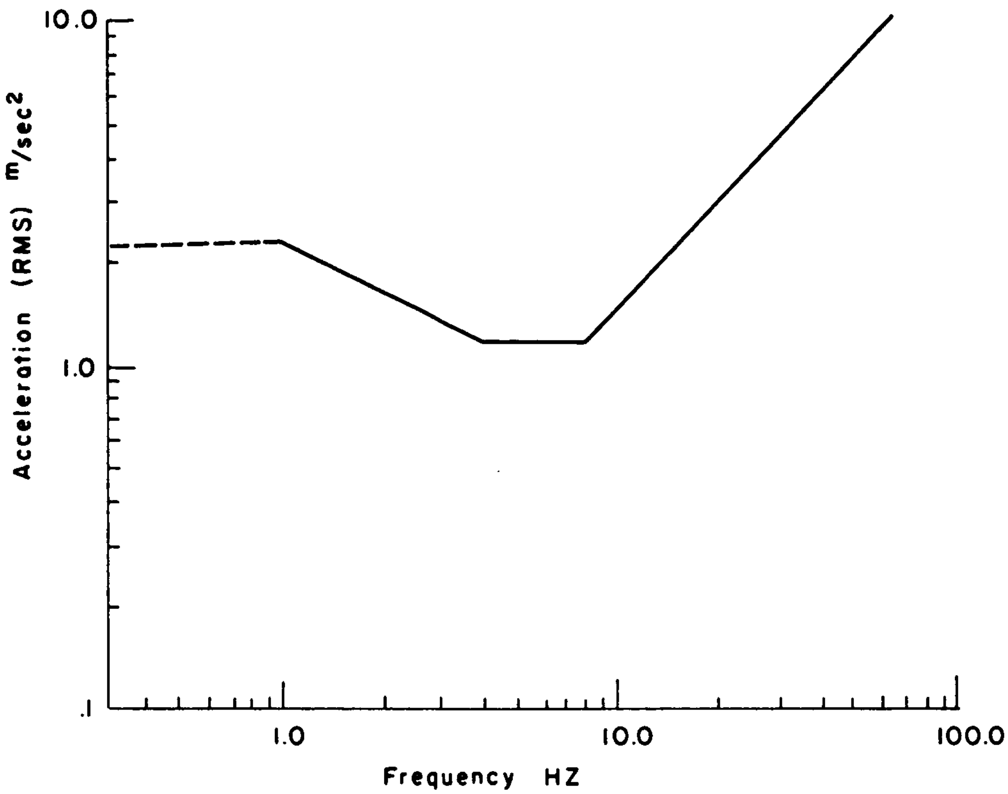

Figure 1.- International Standard 8 hour limit.

		Subject Name	First Route 1st	First Route 2nd	First Route 3rd	Second Route 1st	Second Route 2nd	Second Route 3rd
Route #1 and #2	Repl. 1							
	Repl. 2							
	Repl. 3							
	Repl. 4							

Figure 2.- Partial tabulation of the experiment.

RATING FORM

1. How would you rate the car ride you have just taken?

worse ride I can think of 1 2 3 4 5 best ride I can think of

2. How would you rate your mood right now?

worse mood I can think of 1 2 3 4 5 best mood I can think of

3. How would you rate the weather right now?

worse weather I can think of 1 2 3 4 5 best weather I can think of

your name Thompson, Peggy date 3-6-74 section Number 1-4

Figure 3.- Rating form.

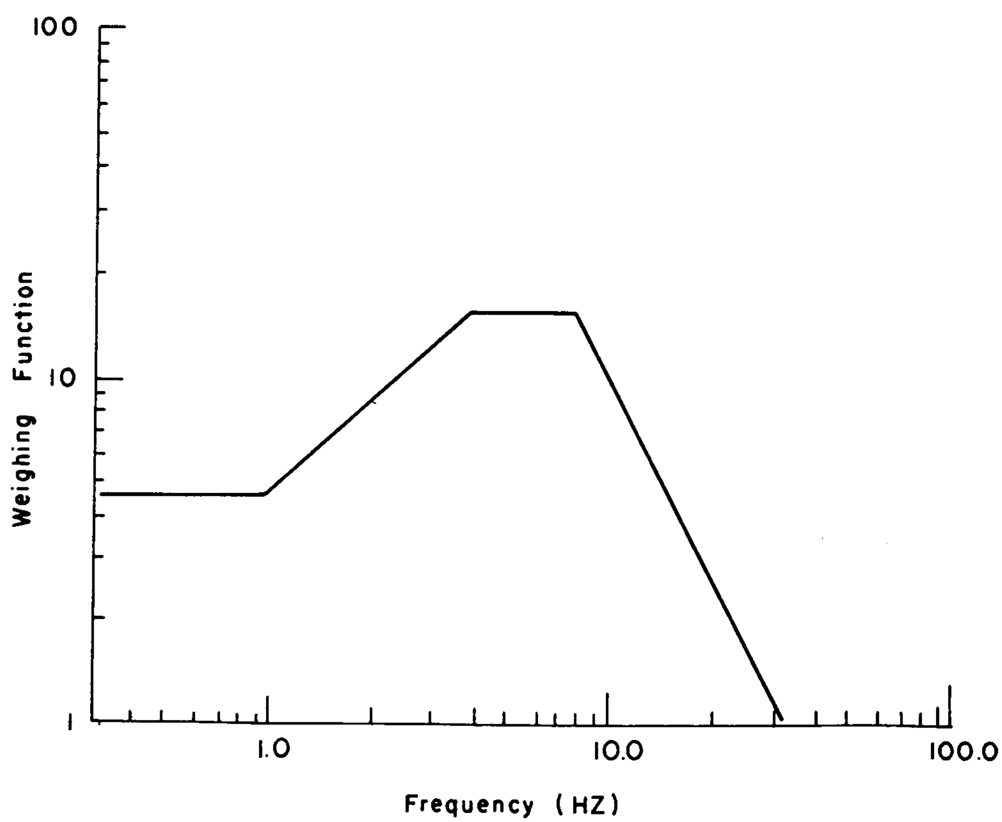

Figure 4.- ISO weighting function - vertical; b = 40 Hz.

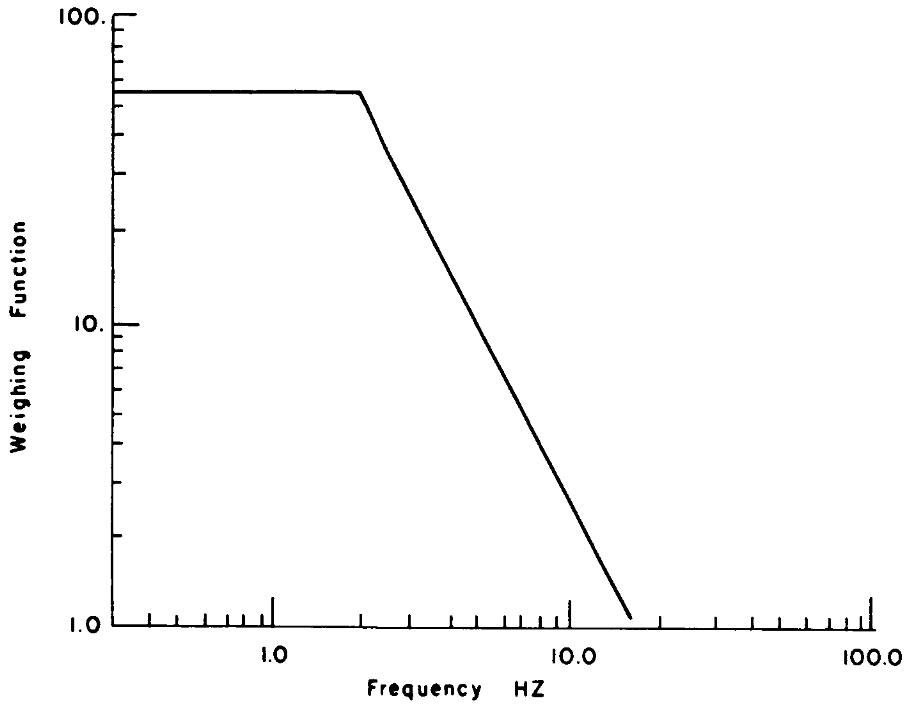

Figure 5.- ISO weighting function - lateral; b = 40 Hz.

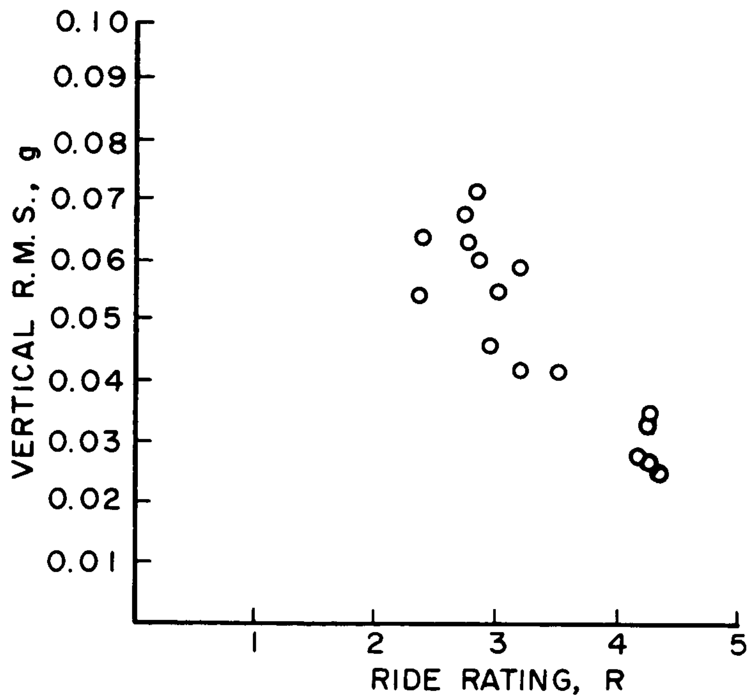

Figure 6.- R.M.S. vertical acceleration versus mean ride rating (correlation coefficient: 0.93).

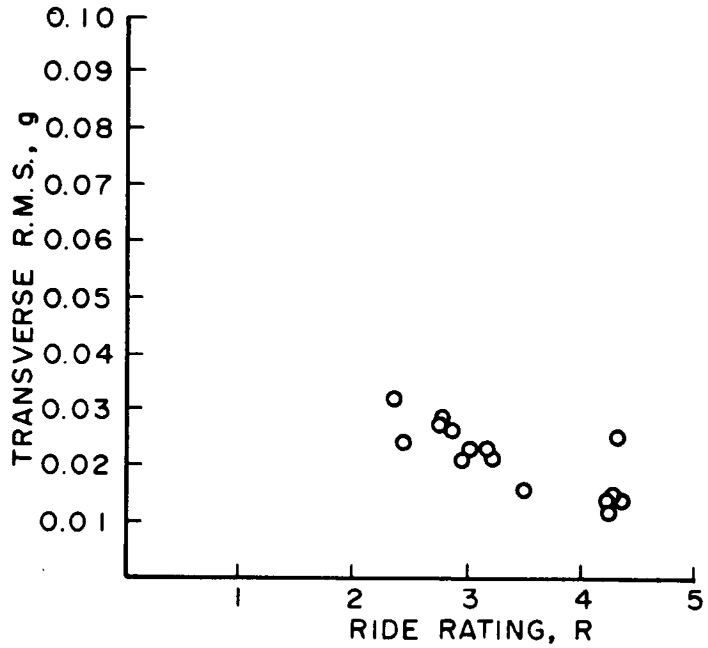

Figure 7.- R.M.S. transverse acceleration versus mean ride rating (correlation coefficient: 0.77).

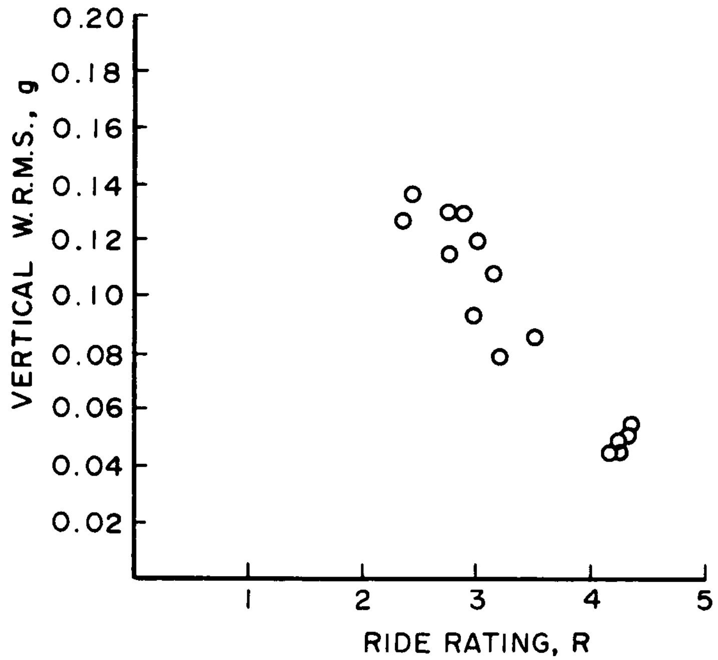

Figure 8.- ISO weighted vertical acceleration versus ride rating (correlation coefficient: 0.95).

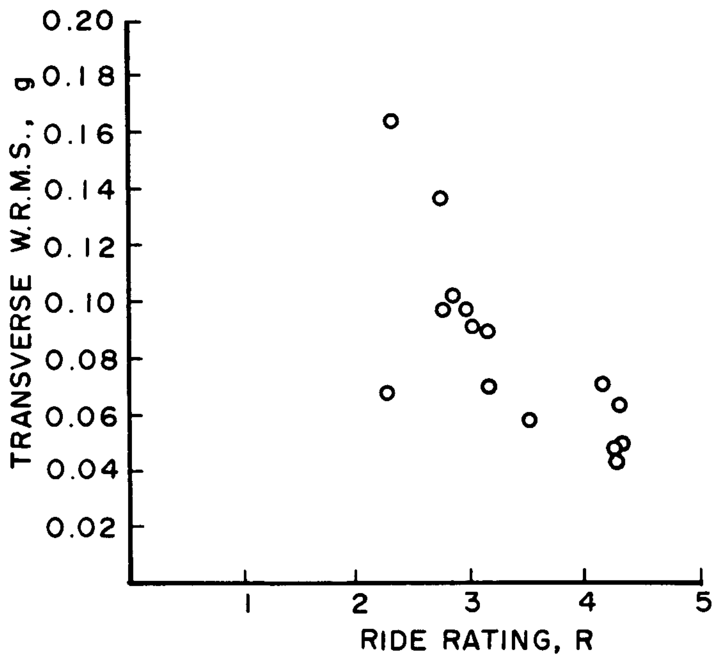

Figure 9.- ISO weighted transverse acceleration versus mean ride rating (correlation coefficient: 0.704).

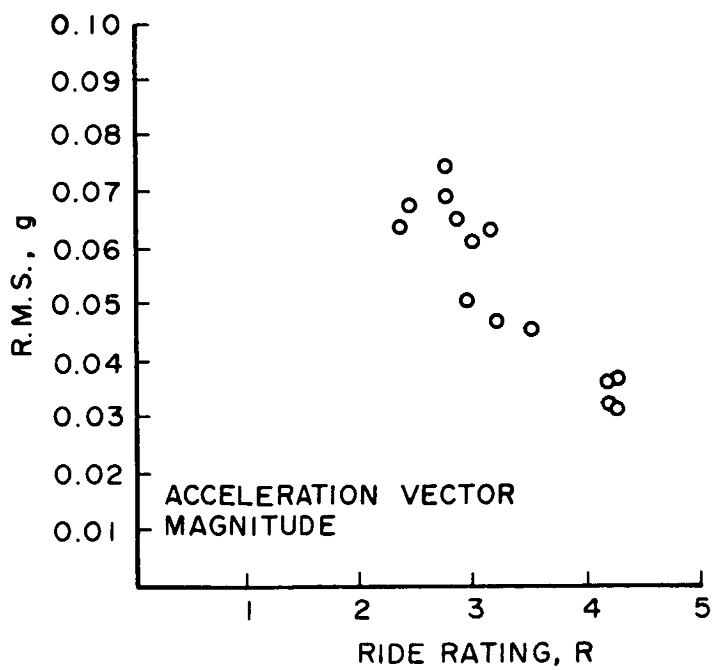

Figure 10.- R.M.S. acceleration vector magnitude (correlation coefficient: 0.909).

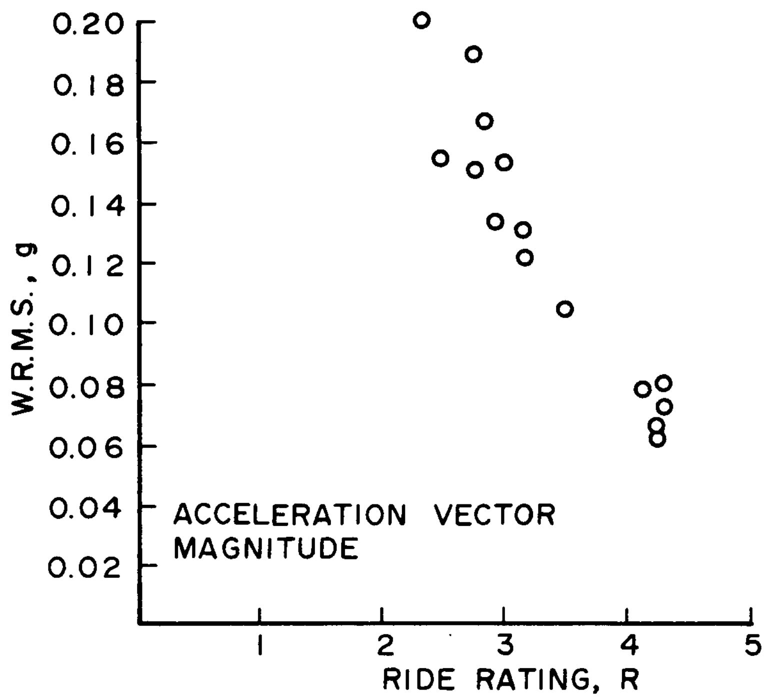

Figure 11.- ISO weighted acceleration vector magnitude versus mean ride rating (correlation coefficient: 0.931).

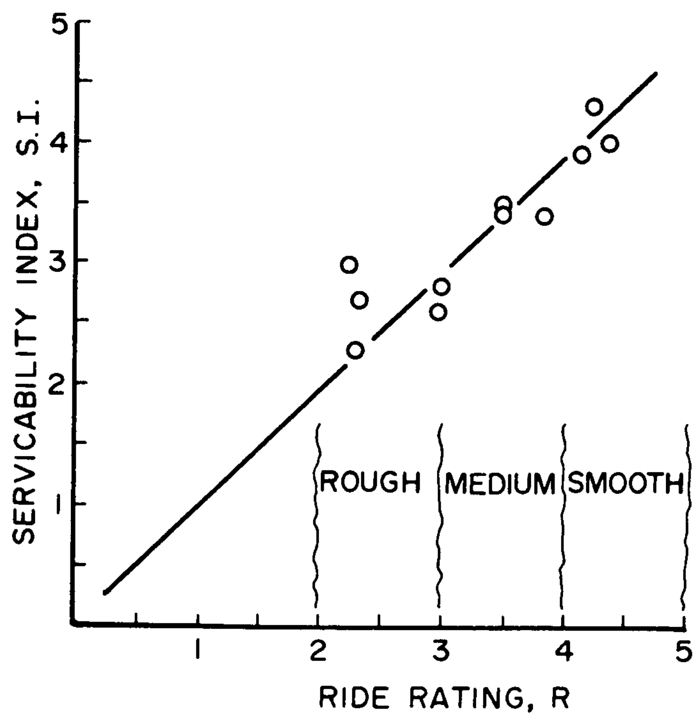

Figure 12.- Roadway servicability index versus mean ride rating.

N76-16779

VIBRATION SIMULATOR STUDIES FOR THE DEVELOPMENT OF PASSENGER RIDE COMFORT CRITERIA

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SUMMARY

A test program to determine the total discomfort associated with vehicle vibration is described. The program utilizes a three-degree-of-freedom vibration simulator to determine the effects of multifrequency and multiaxis vibration inputs. The approach to multifrequency vibration includes a separate consideration of the discomfort associated with each frequency component or band of the total spectrum and a subsequent empirical weighting of the discomfort components of these frequency bands when in various random combinations. Mathematically, this may be represented as

$$DISC_{total} = DISC_{max} + F(\Sigma DISC - DISC_{max})$$

The discomfort (DISC) represents the subjective discomfort associated with the acceleration level of a particular frequency band. The F value or masking factor specifies the fashion in which the discomfort of different frequency bands are added together. Fundamental to this approach is a detailed understanding of human response to discrete frequency inputs. A study has been recently completed that included 186 subjects exposed to frequencies of 1 to 30 Hz and peak acceleration levels from 0.05 to 0.50g. The F value was derived in a second set of tests that systematically explored the passenger discomfort response as a function of various random spectra.

The results are in the form of equal discomfort curves that specify the discomfort associated with discrete frequencies between 1 and 30 Hz and different acceleration levels. These results, in addition to being necessary for the previous equation, provide detailed information of the human discomfort response to increases in acceleration level for each frequency investigated. More importantly, the results provide a method for adding the discomfort associated with separate frequencies to give a total typification of the discomfort of a random spectrum of vibration.

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INTRODUCTION

The development of new transportation systems or the modification of existing systems for improved ride quality requires a comprehensive understanding of human response to whole-body vibration. Specifically, what is needed is a scale of discomfort. The scale would necessitate generating extensive experimental data for the development of constant discomfort curves and the associated empirical laws governing the summation of discomfort responses due to multiple-frequency and/or multiple-axis vibration. A recent review and summary (ref. 1) of the criteria literature points out that many differences and contradictions exist in the various reported investigations. For example, it is not unusual for the vibration levels associated with the various proposed criteria to differ from one another by as much as an order o magnitude. The reasons that have been offered for the diversity of results include such factors as poor experimental design, unrealistic laboratory environments, use of inadequate rating scales or adjectives, small subject samples, and lack of information (e.g., ref. 2) regarding the fundamental psychophysical relationship between human comfort response and vibration. Notable exceptions are the studies reported by Shoenberger and Harris (ref. 3) Jones and Saunders (ref. 4), and Miwa (refs. 5 to 11) which were concerned wi the psychophysics of human sensitivity response to whole-body vibration. However, a recent investigation at Langley Research Center (as yet unpublishe demonstrated that sensitivity (intensity) responses of human subjects were different from discomfort type responses at several different frequencies. Consequently, caution needs to be used in applying results (criteria) from studies of intensity (or sensitivity) to problems related to human discomfort. Therefore, the first objective of this investigation is to develop discomfort criteria (equal discomfort curves) in a systematic fashion that removes the limitations of previous investigations.

A second problem that is encountered in the development of a scale of discomfort with accurate information for vehicle design is the total typification of the discomfort of a random vibration. This problem area necessitates the derivation of the total discomfort of a vibration based upon some combination of the discomfort associated with the frequency components of random ride spectra. Previous approaches to the typification of random vibration for prediction of comfort have concentrated upon measures of (1) power spectral density indices (e.g., ref. 12) based upon either unweighte or frequency-weighted power spectra (e.g., ref. 13), (2) amplitude exceedance counts (e.g., ref. 14), and (3) absorbed power (e.g., refs. 15 to 18).

There are several recent reviews (e.g., refs. 19 to 21) which describe the limited applicability of the use of these measures for the prediction of comfort. A major limitation of these measures is that they are based upon frequency weighting obtained for individually applied sinusoidal vibrations. The measures do not account for the effects of masking between frequencies within an axis but apply frequency weights or coefficients to each individual frequency as if it were acting alone. Thus the second objective of the presen investigation is a determination of the empirical relations governing vibratio masking/summation in order to derive the total discomfort of any random-induced ride spectra. The procedure followed for summation of discomfort components of a ride spectra for the total typification of the discomfort of a random vibration is outlined in reference 2. Experimentally, the procedure involves determining how the subjective assessment of the discomfort of a ride varies when many different frequency components are experienced simultaneously The composite weights for specification of the total discomfort of a ride are thus based upon the discomfort of several frequency components in combination rather than an arbitrary summation (usually algebraic) of the discomfort units associated with these components when individually experienced. A specific result of obtaining equal discomfort contours and empirical information for the summation of discomfort units is a scale of discomfort.

In summary the objectives of the present investigation are

(1) To systematically derive "equal vibration discomfort curves"
(2) To determine the influence of vibration masking in order to account for the total discomfort of any random vibration
(3) To develop a scale of total vibration discomfort

METHOD

The objectives of the investigation were achieved through three separate but interconnected studies hereinafter denoted as studies A, B, and C. Study A was directed at obtaining the acceleration levels of different frequencies that produce identical discomfort responses. Study B was used to obtain the empirical relationship between discomfort responses and acceleration level for each separate frequency. Finally, study C, based on sinusoidal and random vibration tests, was used to obtain a method for adding the discomfort associated with separate frequencies (based on the results of study A and B) for a total typification of the discomfort of a random spectrum of vibration. The following sections provide a review of the Langley passenger ride quality apparatus which was used in each investigation, as well as a short description of the subjects, task, and procedure for each study.

Apparatus

The apparatus used was the Langley passenger ride quality apparatus (PRQA). The PRQA is described briefly in this section, and a detailed description can be obtained from references 22 and 23. The PRQA and associated programing and control instrumentation are shown in the photographs of figure 1. Figure 1(a) shows the waiting room where subjects are instructed as to their participation in the experiment, complete questionnaires, and so forth. Shown in figure 1(b) is a model of the PRQA indicating the supports, actuators, and restraints of the three-axis drive system. A photograph of the exterior of the PRQA is shown in figure 1(c) and it should be noted that the actual mechanisms which drive the simulator are located beneath the pictured floor.

An interior view of PRQA with subjects seated in first-class aircraft seats (tourist-class aircraft seats were used in the present study) is presented in figure 1(d). The control console is shown in figure 1(e) and is located at the same level as the simulator to allow the console control operator to constantly monitor subjects within the simulator. Figure 1(f) is a photograph of tourist-class aircraft seats used in the present study.

Subjects

A total of 186 subjects participated in the three studies. The volunteer subjects were undergraduates from Old Dominion University and were paid for their participation in the studies. The pertinent subject demographics for each study are listed in table 1(a).

Subject Task and Procedure

The subjects involved in study A were required to evaluate successive "comparison ride segments" according to a modified method of limits task. Specifically, a subject's task was to determine if a ride segment provided greater or less discomfort than a ride segment termed the "standard ride." The vibration characteristics of the standard and comparison ride segments are provided in table 1(b). Appropriate counterbalancing of frequencies and acceleration levels was performed for these tests.

The task for the subjects of studies B and C was the evaluation of the discomfort of vibrations through a magnitude estimation procedure. The procedure involves applying a standard ride (vibration that was different than that of study A) to the subjects and assigning the standard ride a numerical value of 100. Comparison ride segments (vibrations that were different from those of study A) were then applied and the subjects were asked to evaluate these vibrations relative to the standard ride segment by assigning it an appropriate numerical value. For example, if the discomfort of a ride was felt to be twice the discomfort of the standard ride, the subjects would give the ride a value of 200. The subjects were instructed not to use zero or negative numbers in making their evaluations.

Although the magnitude estimation procedure was used by the subjects in both studies B and C, the vibration characteristics of the standard and comparison ride segments for the two studies differed. The major difference between the vibrations of the two studies was that sinusoidal vibrations were used in study B, whereas both sinusoidal and random vibrations were investigated in study C. A description of these vibrations is provided in table 1(b). Counterbalancing of appropriate factors was done for testing in both studies.

RESULTS AND DISCUSSION

The results of the three investigations conducted to achieve the objectives listed in the introduction are discussed in this section. The results considered collectively culminate in a scale of discomfort. This scale of discomfort requires an anchor point and a brief discussion of the anchor point selection is presented, followed by a detailed discussion of each study.

Anchor Point: Scale of Discomfort

A previous experimental investigation (ref. 24) concluded that 9 Hz should be selected as the anchor (and standard) frequency for development of the scale of discomfort. The primary reason for selecting 9 Hz as the anchor frequency was that it gave less variability of discomfort responses to vibration stimuli as compared with other sinusoidal vibrations. An additional investigation (ref. 25) provided data from which an acceleration level of 0.08g (g = 0.057) was determined to be the approximate threshold of discomfort at the 9 Hz anchor frequency. Consequently, 9 Hz at 0.08g was selected as the anchor point and was assigned a unit value of discomfort (DISC = 1).

Frequency Equating - Study A

As a first step toward derivation of equal discomfort curves, this study determined the acceleration level at different frequencies that produces identical discomfort. Figure 2 presents typical results of study A for a frequency of 5 Hz. (Similar results were obtained for frequencies from 1 to 30 Hz, excluding the standard frequency of 9 Hz.) Figure 2 shows the z-score (standard normal score) transformations of percentage of responses obtained from comparison rides (5 Hz in this case) that were evaluated as having more discomfort than a standard ride as a function of the acceleration level of the comparison rides. The standard ride for this study was a 9 Hz sinusoidal frequency at an acceleration level of 0.15g. The z-score value of 0.0 corresponds to 50 percent of the 5 Hz comparison ride segments evaluated as having more discomfort than the standard ride. Therefore, the acceleration level at the z = 0.0 point of the 5 Hz ride was taken as equal in discomfort to the standard ride. For the example shown in figure 2, an acceleration level of 0.115g at 5 Hz is equal in discomfort to an acceleration level of 0.15 (precisely 0.1528) at 9Hz. Repeating the procedure described above for all other frequencies gives the curve shown in figure 3. The ordinate of figure 3 is the acceleration level corresponding to z = 0.0 (equal discomfort point) for each frequency along the abscissa. Thus the curve of figure 3 is a constant discomfort curve whose absolute level of discomfort must be determined from study B. The discomfort value for the curve of figure 3 will depend upon the subjective discomfort assigned to a ride at 9 Hz and 0.1528g, given that the value of 1 DISC was assigned to 9 Hz at 0.08g.

Equal Discomfort Curves - Study B

The objective of study B was to derive equal discomfort curves that could be assigned absolute levels of discomfort. The results of this study are in the form of magnitude estimates of successive ride segments for a particular frequency. Figure 4 displays an example of these results and provides a connection of these results with those of study A. Figure 4 shows the magnitude estimations of the discomfort of 9 Hz ride segments as a function of acceleration. Since a discomfort value of 1 DISC was specified for a vibration of 9 Hz at 0.08g, an experimental derived value of 2.47 DISC can be obtained for 9 Hz at 0.1528g. This result is important because it represents the discomfort value (DISC) assigned to each acceleration level and frequency of the curve shown in figure 3. It thus provides an adjustment of ride segments of the various frequencies to the same scale of discomfort.

Figure 5 shows the magnitude estimations of discomfort of 5 Hz ride segments as a function of acceleration level. The results for 5 Hz as well as those for the remaining frequencies investigated (1 to 30 Hz) displayed a strong linear relationship between discomfort and acceleration, as shown in as yet unpublished data obtained at Langley Research Center. As previously mentioned, a discomfort (DISC) value of 2.47 was assigned to a ride segment at 5 Hz and 0.115g and served as a basis for adjusting the magnitude estimations of discomfort for the other ride segments of 5 Hz. Similar adjustments were made to the magnitude estimations of discomfort for the other frequencies investigated (1 to 30 Hz, excluding 9 Hz). Then, using data such as that of figure 5 for each frequency, a set of constant discomfort curves was generated and are presented in figure 6. The individual curves of figure 6 indicate the acceleration level of a sinusoidal vibration required to produce a constant level of discomfort. This figure shows constant discomfort curves ranging from a value of one (DISC = 1), which is approximately the discomfort threshold, to values as high as DISC = 12 corresponding to a very high level of discomfort.

ISO Comparisons

The ISO standards document (ref. 13) contains a tabulation of weighting factors intended to reflect the relative influence of individual sinusoidal vibrations on discomfort for a frequency range of 1 to 80 Hz. The magnitude estimation data generated in this study was also formulated in a frequency weighting factor format and used for comparison with the ISO data as illustrated in figure 7. The ISO weighting curve is represented by the solid line and the NASA weighting curve by the dashed line. The ISO weighting curve is a plot of the tabular data contained in reference 14, whereas the NASA weighting curve was obtained by computing, at each frequency, an average weighting factor based upon a normalization of the magnitude estimates of discomfort corresponding to floor acceleration levels ranging from 0.10g to 0.50g. The normalization factor used was the average magnitude estimate of discomfort where the average was taken over all frequencies in the 4 Hz to 8 Hz (flat, equally weighted part of ISO curve) frequency range.

Inspection of figure 7 shows that the basic trend of the NASA weighting curve is similar to that of the ISO weighting curve. However, there are several important differences which should be noted. First, the ISO data tend to weight the lower frequencies (below 4 Hz) and the higher frequencies (above 7 Hz) considerably more than the present data. For example, at a frequency of 15 Hz the NASA weighting factor is approximately 64 percent of the ISO weighting factor. Another difference between the two weighting factor curves is that the NASA data shows that frequencies of 5 Hz and 6 Hz have the largest weighting, with lesser importance attributed to 4, 7, and 8 Hz. These differences may be important when a researcher or designer decides to select a weighting curve for use in obtaining a weighted measure of a ride spectrum (such as a weighted rms level) or for use as a filter characteristic in a "Ride Quality" meter. The NASA set of weighting factors represent an alternative to the weights of the ISO standards. Future studies will resolve differences in prediction accuracy of the two sets of weights.

Vibration Masking - Study C

Study C addresses the question of how the total discomfort of a ride is affected when different frequency components are combined. Such a knowledge is required for application of these data to operational random ride environments. The total discomfort of a ride as specified in reference 1 is represented in the following formula:

$$DISC_{total} = DISC_{max} + F(\Sigma DISC - DISC_{max})$$

Studies A and B provide the necessary information for computation with the formula, except for F, the masking factor. The derivation of F as a function of bandwidth, center frequency, and acceleration level of vibration is the purpose of study C. At the time this paper was presented for publication, the data analyses for computation of the masking factor(s) were not complete. However, examination of preliminary results for a 10 Hz bandwidth indicated the masking factor to be approximately 0.67. It should be emphasized that this is a rough estimate based upon a single bandwidth and a small portion of the available data. Detailed analyses and results of the masking study will be included in a subsequent publication.

SUMMARY OF RESULTS

The results from this series of interconnected studies can be summarized as follows:

1. Passenger discomfort to whole-body vertical vibration increases linearly with acceleration level for each frequency.
2. A set of constant discomfort curves were generated by accounting for frequency and amplitude effects of vibration upon passenger discomfort.
3. Empirical data from the series of studies provided a mechanism for determining the degree of masking (or summation) of the discomfort of multiple frequency vibration. More importantly, the results, when applied to a mathematical model, provided a method for adding the discomfort associated with separate frequencies to give a total typification of the discomfort of a random spectrum of vibration. Consequently, a scale for the prediction of passenger discomfort was developed.
4. Finally, differences between ISO and NASA derived frequency weighting factors were discussed.

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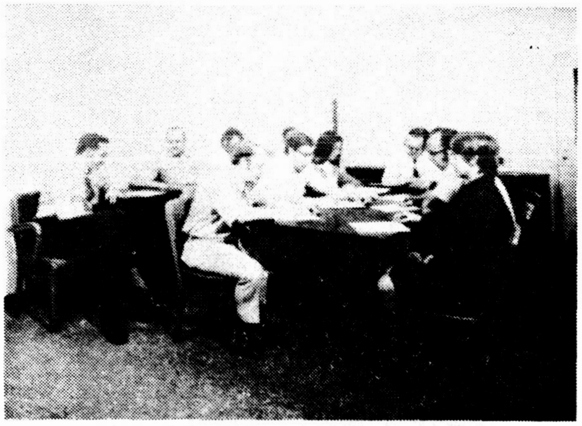
TABLE I.— SUMMARY OF SUBJECT DEMOGRAPHICS AND VIBRATION CHARACTERISTICS OF STANDARD AND COMPARISON RIDE SEGMENTS FOR STUDIES A, B, AND C

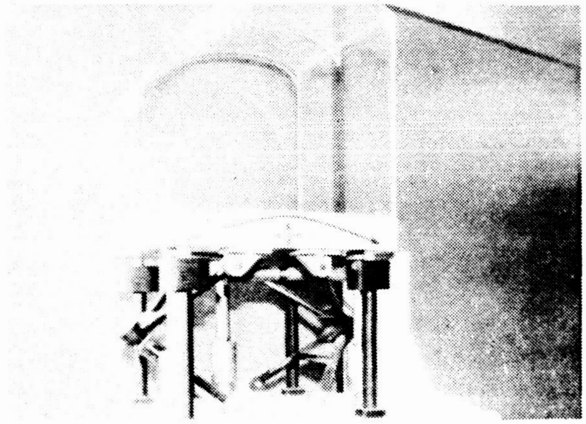
(a) Subject demographics

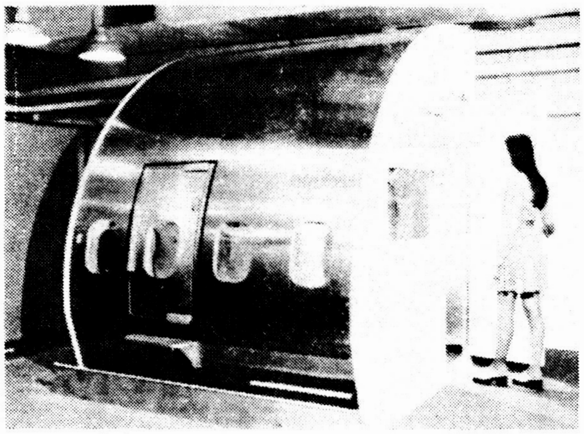
	Number of subjects			Age, yr		Weight, kg (lb)		Subjective task
	Males	Females	Total	Median	Range	Mean	Standard deviation	
Study A	12	42	54	18	18 to 31	60.2 (132.8)	10.3 (22.8)	Method of limits
Study B	41	55	96	21	18 to 55	67.0 (147.8)	14.2 (31.4)	Magnitude estimation
Study C	15	21	36	20	18 to 57	61.2 (135.0)	11.3 (25.0)	Magnitude estimation

(b) Vibration characteristics

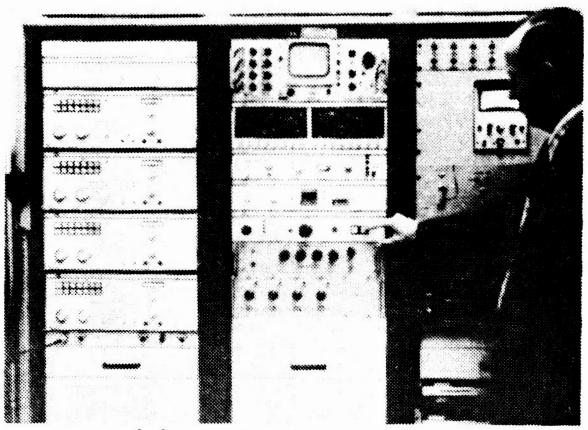
	Axis	Type	Frequency, Hz	Acceleration, g	Onset and offset, sec	Duration, sec	No. of vib.	Time between vibrations, sec
Standard ride								
Study A	Vert.	Sinusoidal	9	0.15	5	10	50	5
Study B	Vert.	Sinusoidal	Variable	Variable	5	10	30	5
Study C	Vert.	Sinusoidal	9	0.10 (rms)	5	20	48	5
Comparison rides								
Study A	Vert.	Sinusoidal	1 to 30	0.05 to 0.475	5	10	100	5
Study B	Vert.	Sinusoidal	1 to 30	0.05 to 0.475	5	10	90	5
Study C	Vert.	Random (2,5, and 10 Hz BW)	Center frequency at 1 to 9 and 13 Hz	0.03 to 0.12 (rms)	5	20	144	5


(a) Waiting room.


(b) Model of PRQA.


(c) Simulator exterior.


(d) Simulator interior.


(e) Control console.

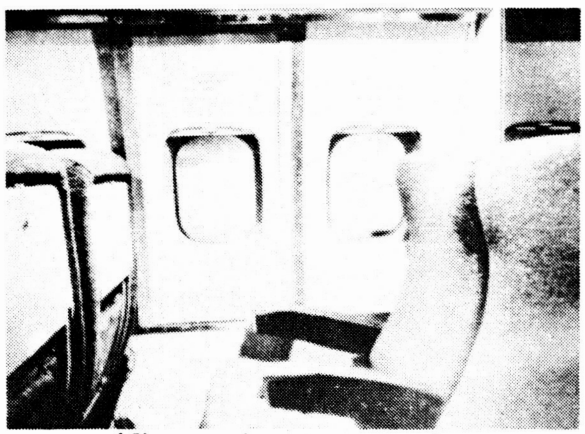

(f) Tourist type seats.

Figure 1.- Langley passenger ride quality apparatus.

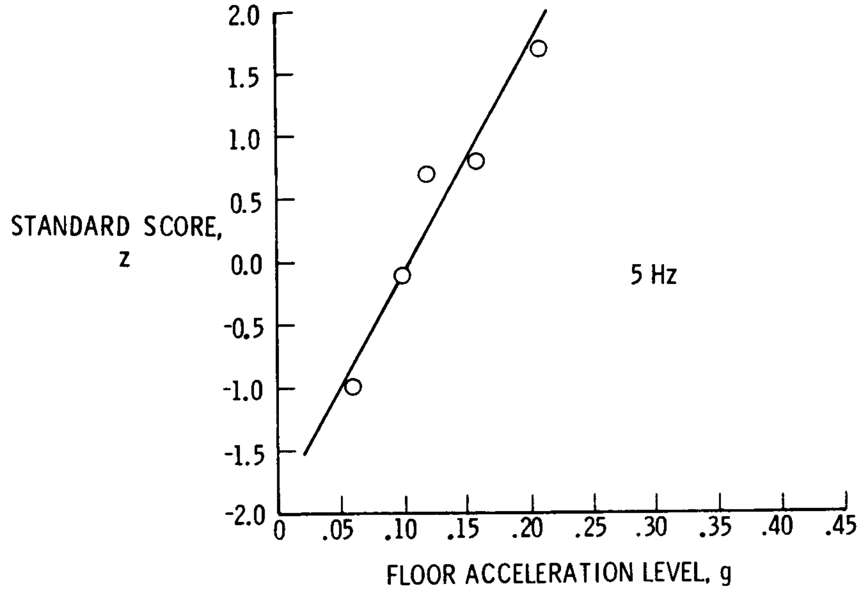


Figure 2.- The z-score transformations of the percentage of comparison rides at 5 Hz evaluated as having greater discomfort than standard ride at 9 Hz and 0.15 g as a function of acceleration level of comparison rides.

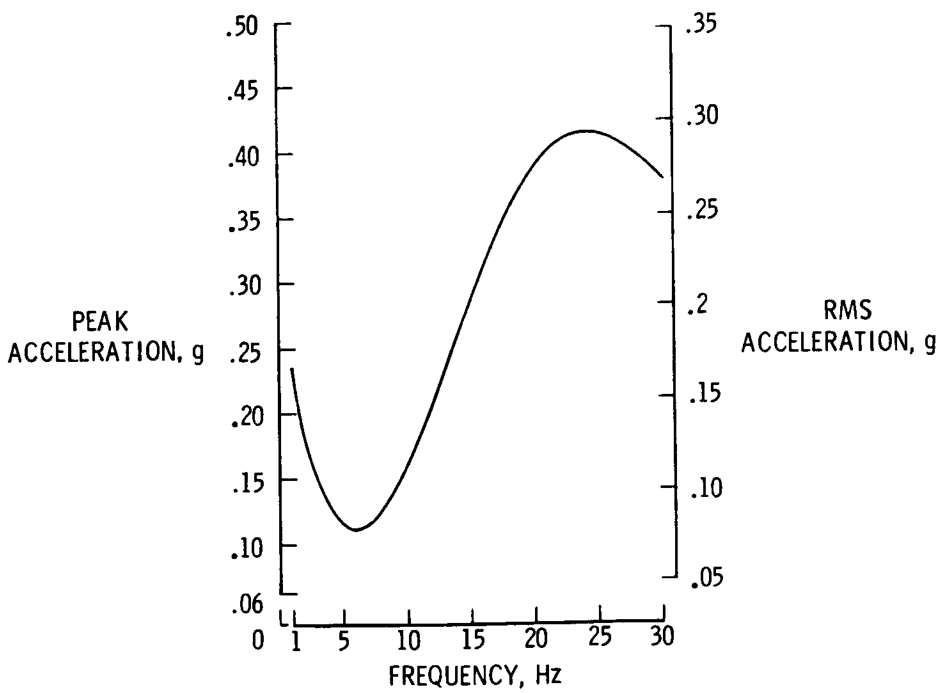


Figure 3.- Peak and rms acceleration levels required to produce equal discomfort as a function of frequency.

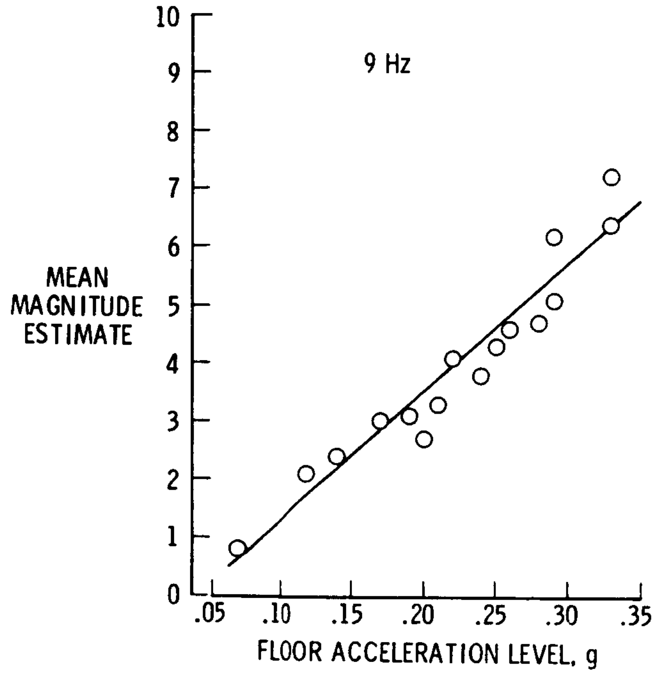


Figure 4.- Mean magnitude estimate of discomfort as a function of floor acceleration level for a 9 Hz sinusoidal vibration.

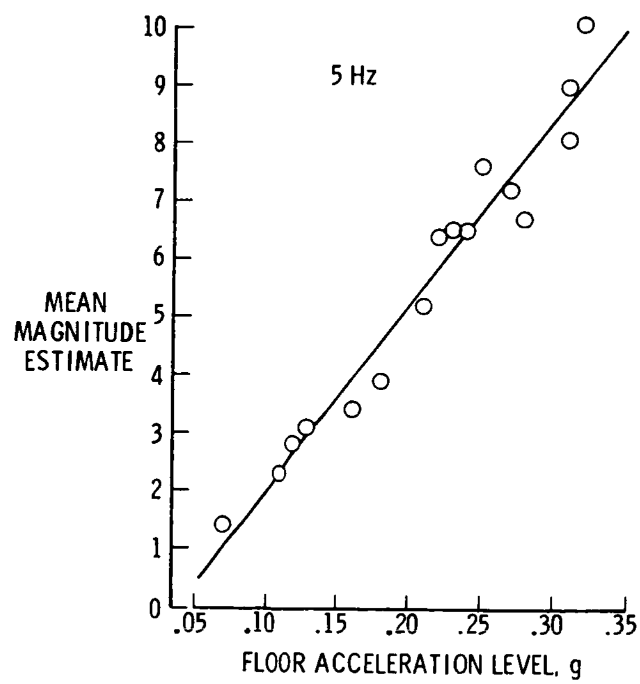


Figure 5.- Mean magnitude estimate of discomfort as a function of floor acceleration level for a 5 Hz sinusoidal vibration.

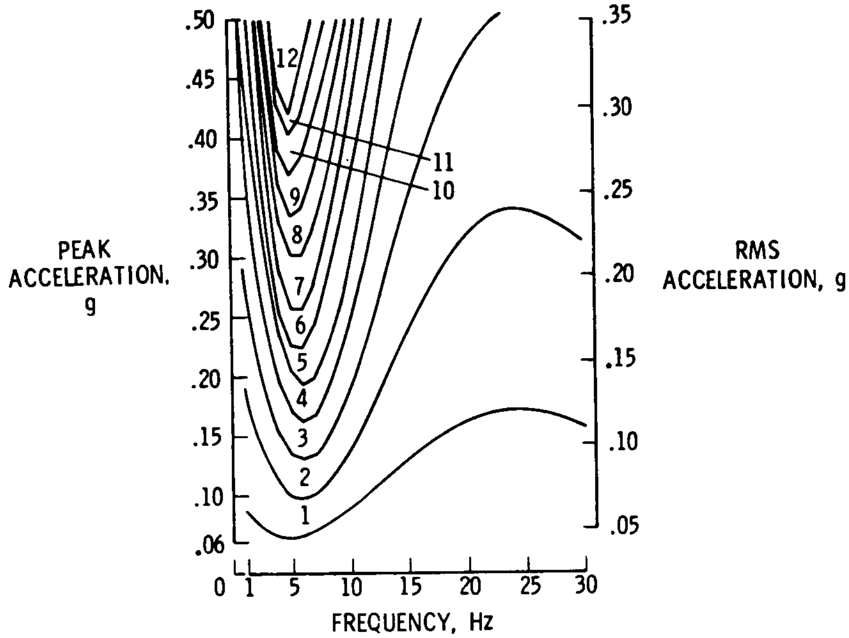


Figure 6.- Peak and rms acceleration levels required to produce successive equal discomfort curves (DISC = 1 to 12) as a function of frequency.

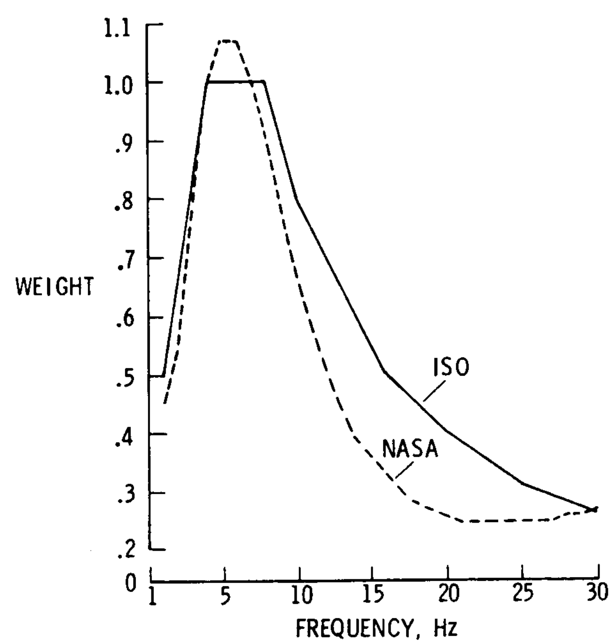


Figure 7.- Comparison of ISO and NASA frequency weighting factors as a function of floor input frequency.

N76-16780

SIMULATOR STUDIES AND

PSYCHOPHYSICAL RIDE COMFORT MODELS

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INTRODUCTION

An elementary psychophysical model to predict ride comfort was developed using flight and simulator data where subjects were exposed to six degrees of freedom. This model is presented in references 1 and 2. The model presumes that the comfort response is proportional to the logarithm of the stimulus above some threshold stimulus. The model further presumes that in a condition of multiple motion stimuli, the ride comfort response is dominantly influenced by the maximum effective stimulus existing and only somewhat modified by the existence of other motion stimuli.

In order to verify this concept of comfort modeling, it was necessary to obtain ride comfort data for single degree of freedom random motions and for combinations of random motions. Accordingly, a simulator program was performed at the NASA Langley Research Center to measure subjective comfort response ratings using one degree of freedom, two degrees of freedom, three degrees of freedom, and six degrees of freedom. Some of the data obtained are presented in references 3, 4, 5, and 6. This paper presents an analysis of the single degree of freedom and two degrees of freedom data. Preliminary models of ride comfort response for single degree of freedom random motions and for certain combinations of two degrees of freedom random motions are developed.

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SYMBOLS

$\bar{a}_i$	Rms linear acceleration, g's
$\bar{a}_x$	Rms longitudinal acceleration, g's
$\bar{a}_y$	Rms transverse acceleration, g's
$\bar{a}_z$	Rms vertical acceleration, g's
$\bar{a}_{i_T}$	Threshold to random linear accelerations, g's
$\bar{a}_{x_T}$	Threshold to random longitudinal accelerations, g's
$\bar{a}_{y_T}$	Threshold to random transverse accelerations, g's

$\bar{a}_{z_T}$	Threshold to random vertical accelerations, g's
$\bar{a}_{max}$	Maximum rms linear acceleration, g's
$\bar{a}_{min}$	Minimum rms linear acceleration, g's
$\bar{a}_{max_T}$	Threshold to maximum linear accelerations, g's
$\bar{a}_{TOT}$	Resultant rms linear acceleration, g's
$\bar{a}_{TOT_T}$	Threshold to resultant rms linear acceleration's, g's
$\bar{\omega}_i$	Rms angular velocity, deg/sec
$\bar{\omega}_p$	Rms rolling velocity, deg/sec
$\bar{\omega}_q$	Rms pitching velocity, deg/sec
$\bar{\omega}_r$	Rms yawing velocity, deg/sec
$\bar{\omega}_{i_T}$	Threshold to random angular velocities, deg/sec
$\bar{\omega}_{p_T}$	Threshold to random rolling velocities, deg/sec
$\bar{\omega}_{q_T}$	Threshold to random pitching velocities, deg/sec
$\bar{\omega}_{r_T}$	Threshold to random yawing velocities, deg/sec
$\bar{\omega}_{max}$	Maximum rms angular velocity, deg/sec
$\bar{\omega}_{min}$	Minimum rms angular velocity, deg/sec
$\bar{\omega}_{max_T}$	Threshold to maximum angular velocities, deg/sec
K_i	Motion sensitivity coefficient
K_x	Longitudinal motion sensitivity coefficient
K_y	Transverse motion sensitivity coefficient
K_z	Vertical motion sensitivity coefficient
K_p	Rolling motion sensitivity coefficient
K_q	Pitching motion sensitivity coefficient
K_r	Yawing motion sensitivity coefficient
K_{max}	Sensitivity to maximum rms linear acceleration, or to maximum rms angular velocity
K_{TOT}	Sensitivity to resultant rms linear acceleration

$\bar{S}_i$	Effective stimulus, $\left(\frac{\bar{a}_i}{\bar{a}_{i_T}}\right)^{K_i}$ or $\left(\frac{\bar{\omega}_i}{\bar{\omega}_{i_T}}\right)^{K_i}$
$\bar{S}_{max}$	Maximum effective stimulus
$\bar{S}_{min}$	Minimum effective stimulus
ϕ	$= \sin^{-1}\left(\frac{\bar{a}_z}{\sqrt{\bar{a}_z^2 + \bar{a}_y^2}}\right)$
θ	$= \sin^{-1}\left(\frac{\bar{a}_z}{\sqrt{\bar{a}_z^2 + \bar{a}_x^2}}\right)$
R_S	Subjective ride comfort response rating
σ_{R_S}	Standard deviation of subjective ride comfort response rating
R_{C_1}	Calculated ride comfort response rating to random motions in one degree of freedom
R_{C_2}	Calculated ride comfort response rating to random motions in two degrees of freedom

TESTS AND TEST CONDITIONS

The program was planned to expose ten subjects to each of several conditions in single degree and multiple degrees of freedom random motions on the Langley Visual-Motion Simulator. The various conditions for any motion component included variations in the magnitude of the rms motion stimulus and variations in the power spectral shape of the motion stimulus. The spectra were varied between 0 and 2 Hz to represent variations of power spectra measured in flight. A discussion of these conditions is made in references 3 to 6. The various segments of "flight" performed on the simulator and presented in this paper were randomly distributed in 10 simulator "flights" each flown five times. Each "flight" was 36 minutes in length and included 24 separate segments having different conditions as noted above. Two subjects rode each "flight." The subjects were supplied generally by Hampton Institute and represented a wide demographic profile (see references 3 to 6).

The subjects responded to each motion segment by rating the ride comfort on a seven-statement scale consisting of the following ratings:

1. Very comfortable;
2. Comfortable;
3. Somewhat comfortable;
4. Acceptable;
5. Somewhat uncomfortable;
6. Uncomfortable;
7. Very uncomfortable.

For correlation with past psychophysical model development and for the analysis of this paper, this seven-statement scale was folded into a five-point scale ranging from 1 for very comfortable to 5 for very uncomfortable.

The actual motions experienced by the subjects were measured by a set of three linear accelerometers and three angular rate gyros installed in the simulator. The subjective ride comfort response ratings are related to these measured motions in this paper.

The Langley Visual-Motion Simulator used in these experiments is shown in figure 1. The simulator is driven by six hydraulic legs which are controlled by a computer. The input signals were on a digital tape and therefore repeatable. Because the simulator is a dynamic system, it is subjected to changes in friction, pressure, etc., and therefore does not precisely duplicate a motion for an identical input signal (see references 3 to 6). For analysis purposes averages of the measured motion components for a given segment were used.

The interior of the Langley Visual-Motion Simulator is shown in figure 2. The subjects rode in the pilot's and co-pilot's seats and the instruments and controls were inoperative.

RESULTS AND DISCUSSION

The subjective ride comfort response ratings presented and analyzed in this paper are the mean values for the ten subjects that experienced each segment. The psychophysical models developed herein were designed to fit the mean subjective ratings and not the total mass of data. The relationships presented are therefore between the models and the mean subjective ratings.

Single Degree of Freedom Responses

The subjective ride comfort response ratings for the single degree of freedom motion tests are plotted as a function of the logarithms of the various stimuli in figures 3 to 8. The standard deviations of the subject ratings are also shown. The vertical, transverse, longitudinal, pitching, rolling and yawing stimuli are shown on figures 3, 4, 5, 6, 7 and 8, respectively.

The mean subjective ride comfort response ratings were fitted with a model of the following format for the linear acceleration degrees of freedom,

$$R_{C_1} = 1 + K_i \log_{10} \left(\frac{\bar{a}_i}{\bar{a}_{i_T}}\right)$$

and for the angular degrees of freedom,

$$R_{C_1} = 1 + K_i \log_{10} \left(\frac{\bar{\omega}_i}{\bar{\omega}_{i_T}}\right) .$$

The threshold stimulus and the constants so established are presented on table I. The thresholds for the linear acceleration stimuli range from 0.00512 to 0.0075 g's. These values are based on the assumption that a ride comfort response rating of very comfortable represents a condition where the stimuli is not sensed essentially or is not considered of any significance. These levels are for rms values of random oscillatory accelerations and are about twice as large as thresholds to constant linear accelerations. The thresholds for rms random angular stimuli range from 0.3 to 0.87 degrees per second. Values for constant angular velocities range from 0.5 to 2.0 degrees per second.

The constants, K_i, represent effectively the ride comfort sensitivity to a given motion stimulus. The subjects were much more sensitive to transverse accelerations than to vertical or longitudinal accelerations. In like manner the subjects were more sensitive to rolling motions than to pitching and yawing motions. These results indicate that from the standpoint of ride comfort, humans are more disturbed by motions whose vectors do not lie in the median plane of the body than by those that do.

On table II are listed the correlation coefficients of the mean subjective ratings and the ratings calculated by the models just discussed. Very good correlation is indicated. The standard deviations of the model ratings for the mean subjective ratings are also shown on table II and are appreciably smaller than the standard deviations of the subjective ratings from their mean values.

Two Degrees of Freedom Responses

It was the intent for the two degree of freedom experiments to combine two of the single degree of freedom tests just discussed. It was not possible, however, to do this precisely because of the nature of the simulator. On tables III through VII are listed the single degree of freedom results intended to be combined and the actual results experienced when the inputs to the simulator were combined. The subjective ride comfort response ratings and their standard deviations are also shown on tables III to VII. The two motions combined on each table are as follows:

Vertical and Transverse	Table III
Vertical and Longitudinal	Table IV
Rolling and Yawing	Table V
Vertical and Pitching	Table VI
Transverse and Rolling	Table VII.

Although not always true, the resultant components of motion in the combined motion experiments were larger than their corresponding individual components in the single degree of freedom tests.

Model Development for Combinations of Like Stimuli

In modeling for combinations of two linear acceleration stimuli, an assumption was made that the response would be to the resultant acceleration and not to its separate components. The most sensitive sensor of the body for sensing linear acceleration is the otolith element of the inner ear which responds basically to the total acceleration vector (reference 7). The otolith organ as a single sensor responds to all components of linear acceleration. Accordingly, the model for combining two linear accelerations has the following format:

$$R_{C_2} = 1 + K_{TOT} \log_{10} \left(\frac{\bar{a}_{TOT}}{\bar{a}_{TOT_T}}\right)$$

where $\bar{a}_{TOT}$ is the vector sum of the two applied components of acceleration and $\bar{a}_{TOT_T}$ is the threshold for accelerations parallel to $\bar{a}_{TOT}$. As the sensitivities and thresholds varied for the separate components of linear accelerations previously discussed, the threshold and sensitivity for combined motions would vary depending on the orientation of the resultant acceleration vector.

For combining vertical and transverse accelerations the following was used:

$$\bar{a}_{TOT_T} = \bar{a}_{y_T} + (0.00059) \sin \phi$$

and

$$K_{TOT} = K_y - (0.775) \sin \phi$$

where

$$\sin \phi = \frac{\bar{a}_z}{\sqrt{\bar{a}_z^{\,2} + \bar{a}_y^{\,2}}}$$

For combining vertical and longitudinal accelerations the following was used:

$$\bar{a}_{TOT_T} = \bar{a}_{x_T} + (0.00238) \sin \theta$$

and

$$K_{TOT} = K_x - (0.147) \sin \theta$$

where

$$\sin \theta = \frac{\bar{a}_z}{\sqrt{\bar{a}_z^{\,2} + \bar{a}_x^{\,2}}}$$

These models are presented as isocontours of ride comfort response rating on a vertical and transverse acceleration grid and on a vertical and longitudinal acceleration grid on figures 9 and 10, respectively. Also shown are the mean subjective response ratings from table III and table IV, respectively. The models show that the sensitivity to the motion varies rapidly as the total acceleration vector rotates from the transverse axis or the longitudinal axis such that larger components of transverse or longitudinal acceleration are more readily tolerated when combined with vertical acceleration. Also shown on figure 9 are isocontours from reference 8 obtained from flight data. The agreement is not startling but it must be remembered that limited data exist and that the phenomenon of ride comfort is one where the standard deviation of the subjective data is of the order of 3/4 of a rating point on a five-point scale.

The rolling and yawing motions are also like stimuli. The most sensitive organs for sensing angular motions are the semi-circular canals. A semi-circular canal measures only that vector component of angular motion perpendicular to the plane of the canal. Each canal has separate sensors and neural pathways and therefore unlike the otolith organ does not measure the resultant vector but its components. It was assumed then that a model based on the resultant angular velocity vector would not be appropriate. A model was therefore developed assuming that the maximum effective stimulus dominated the response rating and that the other component only modified this dominant influence. The model so developed is as follows:

$$R_{C_2} = 1 + \log_{10} (\bar{S}_{max}) - 1.365 \left(\frac{\log_{10} (\bar{S}_{min})}{\log_{10} (\bar{S}_{max})}\right)$$

where

$$\bar{S}_i = \left(\frac{\bar{a}_i}{\bar{a}_{i_T}}\right)^{K_i}$$

Isocontours of response rating on a grid of rolling and yawing angular velocities are shown on figure 11. The data from table V are also shown on figure 11. The negative coefficient in this model represents a synergistic influence of yawing velocity on responses to rolling velocity. Much larger rolling velocities are tolerable when combined with yawing velocity than when not. The data obtained are all in the roll dominant area of figure 11. The model presented may not apply for yaw-dominant conditions.

Model Development for Combinations of Unlike Stimuli

In modeling combinations of unlike stimuli,it is recognized that both the otolith organs and semi-circular canals are involved and are the most sensitive sensors involved. With separate sensors and separate neural pathways it was again assumed that a model responding dominantly to the maximum effective stimulus and being only modified by the second component would be appropriate.

The model so developed for combinations of vertical and pitching motions is as follows:

$$R_{C_2} = 1 + \log_{10} (\bar{S}_{max}) - 0.0112 \left(\frac{\log_{10} (\bar{S}_{min})}{\log_{10} (\bar{S}_{max})}\right)$$

and for combinations of transverse and rolling motions is

$$R_{C_2} = 1 + \log_{10} (\bar{S}_{max}) - 0.1534 \left(\frac{\log_{10} (\bar{S}_{min})}{\log_{10} (\bar{S}_{max})}\right)$$

where
$$\bar{S}_{max} = \left(\frac{\bar{a}_{max}}{\bar{a}_{max_T}}\right)^{K_{max}}$$

or
$$= \left(\frac{\bar{\omega}_{max}}{\bar{\omega}_{max_T}}\right)^{K_{max}}$$

Isocontours of response rating on a grid of vertical and pitching motions are presented on figure 12 and for transverse and rolling motions on figure 13. The data from tables VI and VII are also shown on figures 12 and 13, respectively.

The data on figure 13 are primarily in the pitch-dominant area and the model may not apply in the vertical-dominant area. The model indicates very little influence of vertical motions on the comfort response to pitching motions.

The data on figure 13 are primarily in the transverse-dominant region and the model may not apply in the roll-dominant region. The model shows a slightly synergistic effect of rolling velocity on responses to transverse acceleration.

The relationship between the mean subjective response ratings and the ratings calculated by the various models for combined two degrees of freedom motions are shown in table VIII. The correlation coefficients show relatively good agreement but not nearly as good as those previously discussed for the single degree of freedom models. The standard deviations also are somewhat larger for these combined motions than for the single degree of freedom motions previously discussed. The standard deviations of the subjective response rating from the mean subjective response ratings are, however, somewhat smaller than for the single degree of freedom results. These results imply that additional study of the interactive effects of combined motions will be necessary for improved insight to the problems involved and the characteristics of the models required.

CONCLUDING REMARKS

Subjective ride comfort responses to single degree and two degrees of freedom random motions have been examined. Models with responses proportional to the logarithm of the stimuli are proposed for single degree of freedom motion responses. The data and the models developed for single degree of freedom random motions indicate that the subjects were much more sensitive to random transverse accelerations and rolling velocities than to the other degrees of freedom. For combinations of linear accelerations, a model based on the resultant acceleration is proposed.

For other motion combinations, models based on the concept of a primary response to the dominant stimulus with small modifications from the other stimulus are proposed. Fair correlation exists between the models and the mean subjective ride comfort response ratings. The data and models suggest a synergistic effect of certain motion combinations; for example, the presence of yawing motions for the conditions studied causes greater tolerance to rolling motions.

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Table I.- An Elementary Psychophysical Model for Ride Comfort Responses to Single Degree of Freedom Random Motions

Motion Stimulus, $\bar{a}_i$ or $\bar{\omega}_i$	K_i	Threshold Stimulus, $\bar{a}_{i_T}$ or $\bar{\omega}_{i_T}$
$\bar{a}_z$	2.370	0.00750
$\bar{a}_y$	3.145	0.00691
$\bar{a}_x$	2.517	0.00512
$\bar{\omega}_p$	3.756	0.8740
$\bar{\omega}_q$	2.573	0.3025
$\bar{\omega}_r$	2.679	0.7240

Table II.- The Relation of the Mean Subjective Response Ratings with Calculated Ratings for Single Degree of Freedom Random Motions

Motion Stimulus	Correlation Coefficient	Rms-Standard Deviation	Average Rms-Standard Deviation of Subjective Ratings from Mean Subjective Ratings
$\bar{a}_z$	0.978	0.151	0.747
$\bar{a}_y$	0.977	0.235	0.690
$\bar{a}_x$	0.945	0.286	0.610
$\bar{\omega}_p$	0.948	0.316	0.715
$\bar{\omega}_q$	0.939	0.440	0.708
$\bar{\omega}_r$	0.976	0.216	0.663

Table III.- Ride Comfort Responses to Combined Random Vertical and Transverse Motions

$\bar{a}_y$	$\bar{a}_z$	R_S	σ_{R_S}	$\bar{a}_y$	$\bar{a}_z$	R_S	σ_{R_S}
			Single Degree of Freedom Tests				
0	0.0870	3.500	0.624	0	0.0890	3.950	0.864
0	0.0573	3.000	1.106	0	0.0597	2.900	0.615
0	0.0306	2.200	0.753	0	0.0303	2.500	0.745
0.0608	0	3.500	0.527	0.0612	0	4.000	0.782
			Two Degrees of Freedom Tests				
0.0628	0.0846	4.000	0.577	0.0611	0.0849	3.700	0.258
0.0810	0.0575	3.700	0.746	0.0675	0.0591	4.150	0.784
0.0675	0.0334	3.500	0.333	0.0616	0.0385	3.550	0.725
			Single Degree of Freedom Tests				
0.0857	0	4.450	0.762	0.0890	0	4.450	0.599
0.0608	0	3.500	0.527	0.0612	0	4.000	0.782
0.0330	0	3.250	0.830	0.0341	0	3.100	0.532
0	0.0573	3.000		0	0.0597	2.900	
			Two Degrees of Freedom Tests				
0.0873	0.0575	4.400	0.658	0.0831	0.0607	4.200	0.587
0.0810	0.0575	3.700	0.746	0.0675	0.0591	4.150	0.783
0.0532	0.0634	4.050	0.685	0.0417	0.0622	2.500	0.707
			Single Degree of Freedom Tests				
0.0890	0	4.450	0.599	0.0857	0	4.450	0.762
0.0612	0	4.000	0.782	0.0608	0	3.500	0.527
0.0341	0	3.100	0.532	0.0330	0	3.250	0.830
0	0.0573	3.000		0	0.0597	2.900	
			Two Degrees of Freedom Tests				
0.0845	0.0538	4.350	0.747	0.0920	0.0625	4.100	0.994
0.0649	0.0650	4.100	0.699	0.0663	0.0602	3.100	0.810
0.0396	0.0617	3.150	0.626	0.0385	0.0561	3.600	0.658

Table IV.- Ride Comfort Responses to Combined Random Vertical and Longitudinal Motions

$\bar{a}_x$	$\bar{a}_z$	R_S	σ_{R_S}
	Single Degree of Freedom Tests		
0	0.0870	3.500	0.624
0	0.0573	3.000	1.106
0	0.0306	2.200	0.753
0.0598	0	3.625	0.232
	Two Degrees of Freedom Tests		
0.0636	0.0819	3.750	0.755
0.0670	0.0583	4.250	0.677
0.0548	0.0331	3.250	0.540
	Single Degree of Freedom Tests		
0.0900	0	4.312	0.753
0.0835	0	4.375	0.694
0.0598	0	3.625	0.232
0.0571	0	3.688	0.372
0.0315	0	2.938	0.496
0.0315	0	2.812	0.259
0	0.0573	3.000	1.106
	Two Degrees of Freedom Tests		
0.1008	0.0609	4.650	0.580
0.0840	0.0627	4.250	0.540
0.0670	0.0583	4.250	0.677
0.0655	0.0686	3.500	0.667
0.0354	0.0644	2.600	0.460
0.0327	0.0546	3.000	0.333

Table V.- Ride Comfort Responses to Combined Random Rolling and Yawing Motions

$\bar{\omega}_p$	$\bar{\omega}_r$	R_S	σ_{R_S}
	Single Degree of Freedom Tests		
6.104	0	4.150	0.852
4.048	0	3.375	0.876
2.254	0	2.500	0.667
0	1.328	1.650	0.699
	Two Degrees of Freedom Tests		
7.577	2.496	4.050	0.725
5.564	2.731	3.600	0.775
4.733	3.231	2.800	0.538
	Single Degree of Freedom Tests		
0	4.758	3.100	0.460
0	4.525	2.950	0.896
0	1.328	1.650	0.669
0	1.247	1.600	0.699
0	1.134	1.550	0.599
0	1.070	1.550	0.497
4.048	0	3.375	0.876
	Two Degrees of Freedom Tests		
8.165	4.052	3.950	0.643
7.516	3.301	3.800	0.949
5.564	2.732	3.600	0.744
5.365	2.591	2.850	0.338
4.906	1.797	2.800	0.350
4.089	1.689	3.350	0.338

Table VI.- Ride Comfort Responses to Combined Random Vertical and Pitching Motions

$\bar{a}_z$	$\bar{\omega}_q$	R_S	σ_{R_S}
	Single Degree of Freedom Tests		
0.0870	0	3.500	0.624
0.0573	0	3.000	1.106
0.0306	0	2.200	0.753
0	2.0614	2.938	0.853
	Two Degrees of Freedom Tests		
0.0948	3.0591	3.700	0.632
0.0762	2.8574	3.100	0.460
0.0563	2.4538	3.050	0.497
	Single Degree of Freedom Tests		
0.0890	0	3.950	0.864
0.0597	0	2.900	0.615
0.0303	0	2.500	0.745
0	2.0152	3.375	0.641
	Two Degrees of Freedom Tests		
0.0812	2.8272	3.250	0.540
0.0700	2.6444	4.100	0.843
0.0568	2.4391	3.350	0.784
	Single Degree of Freedom Tests		
0	3.0766	4.250	0.655
0	2.0614	2.938	0.853
0	1.0703	2.750	0.463
0.0573	0	3.000	1.106
	Two Degrees of Freedom Tests		
0.0892	3.6558	4.000	0.745
0.0762	2.8514	3.100	0.460
0.0720	2.4632	3.750	0.791

Table VII.- Ride Comfort Responses to Combined Random Transverse and Rolling Motions

$\bar{a}_y$	$\bar{\omega}_p$	R_S	σ_{R_S}	$\bar{a}_y$	$\bar{\omega}_p$	R_S	σ
			Single Degree of Freedom Tests				
0.0857	0	4.450	0.762	0.0890	0	4.450	0.
0.0608	0	3.500	0.527	0.0612	0	4.000	0.
0.0330	0	3.250	0.830	0.0341	0	3.100	0.
0	4.0481	3.375	0.876	0	3.1771	3.438	1.
			Two Degrees of Freedom Tests				
0.0955	4.2591	4.550	0.599	0.0968	4.3749	4.800	0.4
0.0671	3.1237	3.350	0.416	0.0680	3.9086	4.300	0.6
0.0496	4.4645	3.500	0.408	0.0478	3.9441	3.350	0.4
			Single Degree of Freedom Tests				
0	6.1042	4.150	0.852	0	5.2468	4.200	0.7
0	4.0481	3.375	0.876	0	3.1771	3.438	1.0
0	2.2539	2.500	0.699	0	1.9809	2.750	0.5
0.0608	0	3.500	0.527	0.0612	0	4.000	0.7
			Two Degrees of Freedom Tests				
0.0783	5.9730	4.300	0.483	0.0833	5.9112	4.800	0.4
0.0671	3.1237	3.250	0.416	0.0680	3.9086	4.300	0.6
0.0644	2.8152	3.900	0.658	0.0559	3.0566	3.250	0.4

Table VIII.- The Relation of the Mean Subjective Response Ratings with Calculated Ratings for Two Degrees of Freedom Random Motions

Motion Stimulus	Correlation Coefficient	Rms-Standard Deviation	Average Rms-Standard Deviation of Subjective Ratings from Mean Subjective Ratings
$\bar{a}_z$ and $\bar{a}_y$	0.614	0.325	0.674
$\bar{a}_z$ and $\bar{a}_x$	0.860	0.458	0.569
$\bar{\omega}_p$ and $\bar{\omega}_r$	0.791	0.304	0.582
$\bar{a}_z$ and $\bar{\omega}_q$	0.631	0.390	0.628
$\bar{a}_y$ and $\bar{\omega}_p$	0.716	0.384	0.537

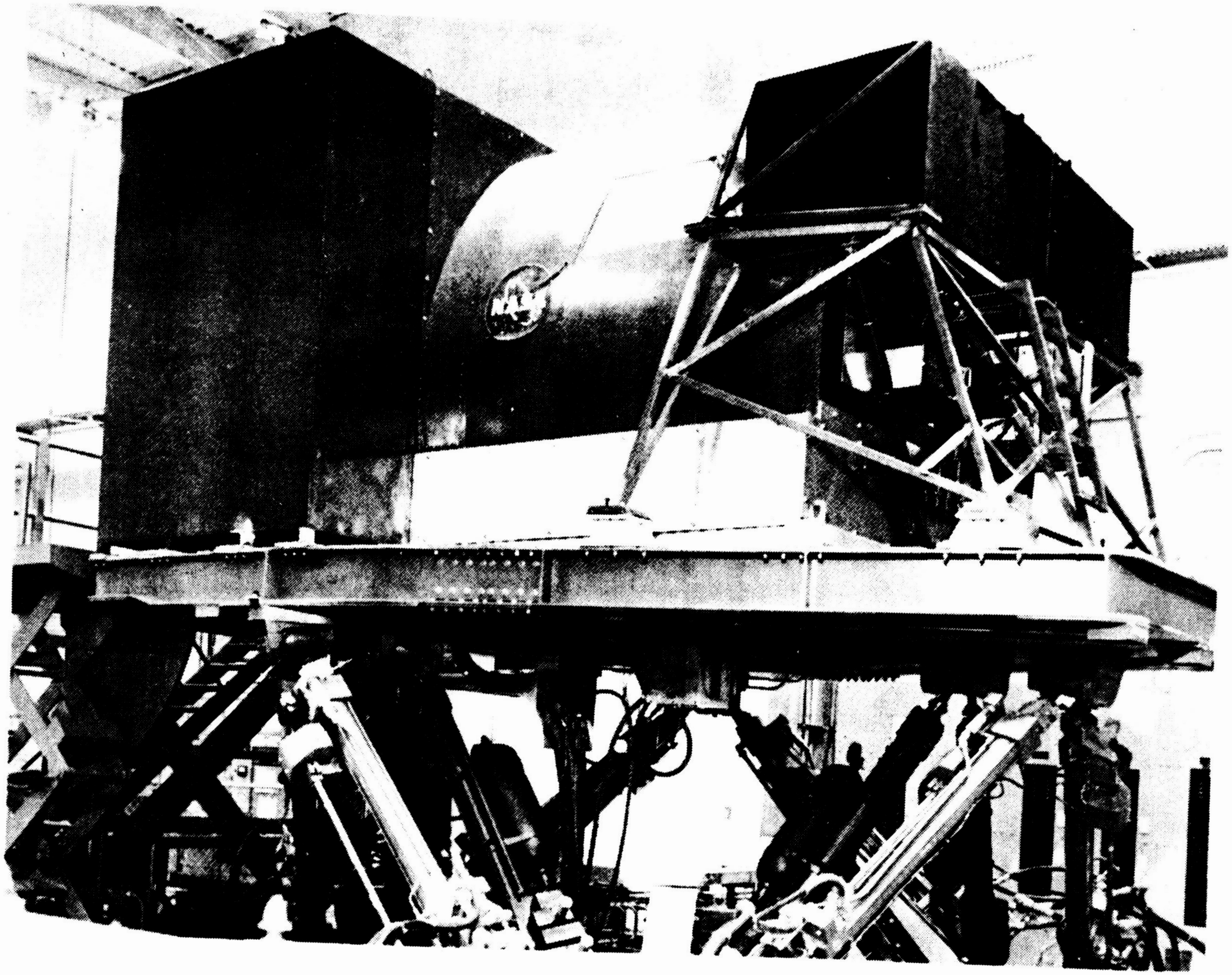

Figure 1.- NASA Langley visual-motion simulator.

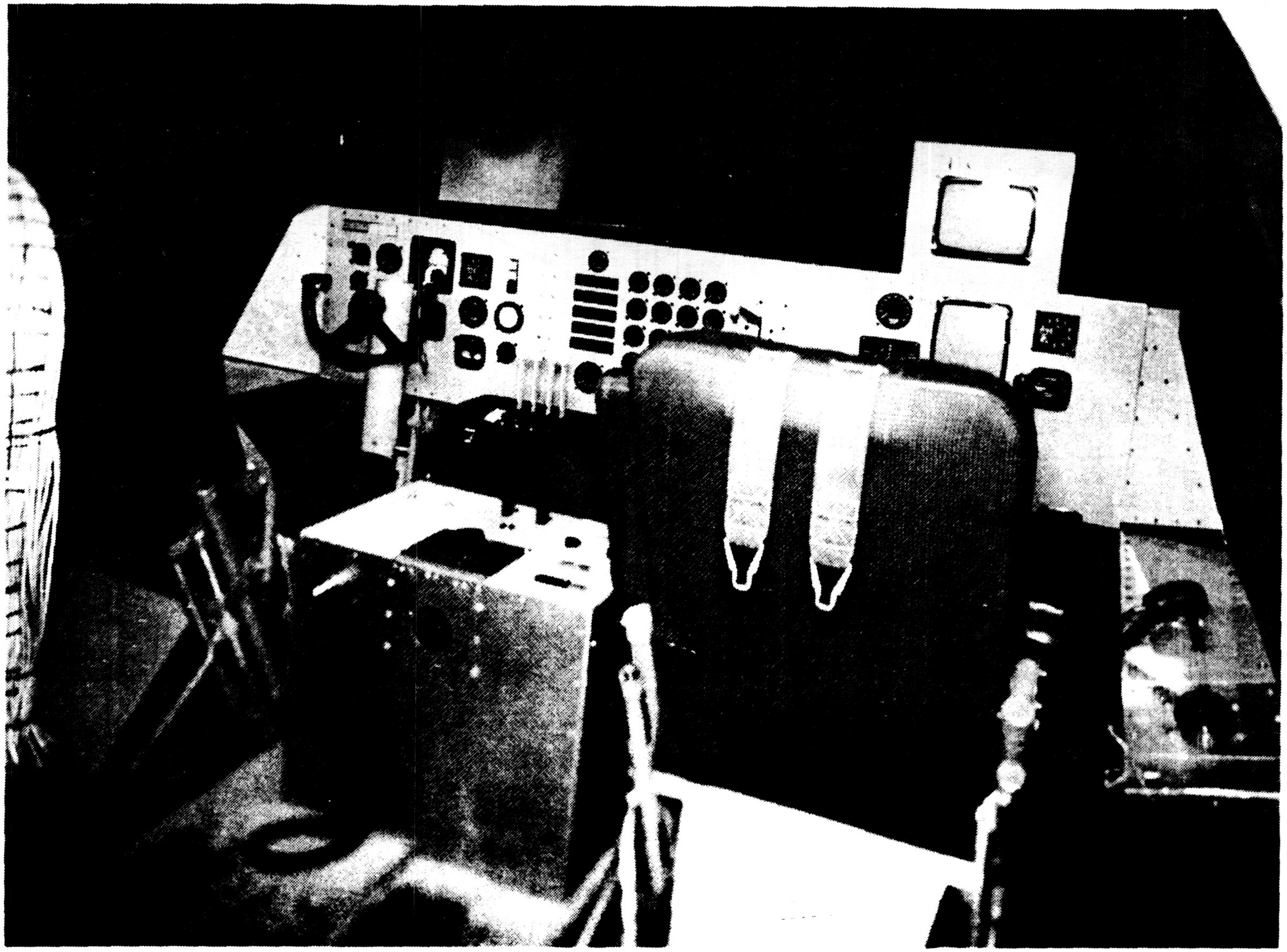

Figure 2.- Interior of the NASA Langley visual-motion simulator.

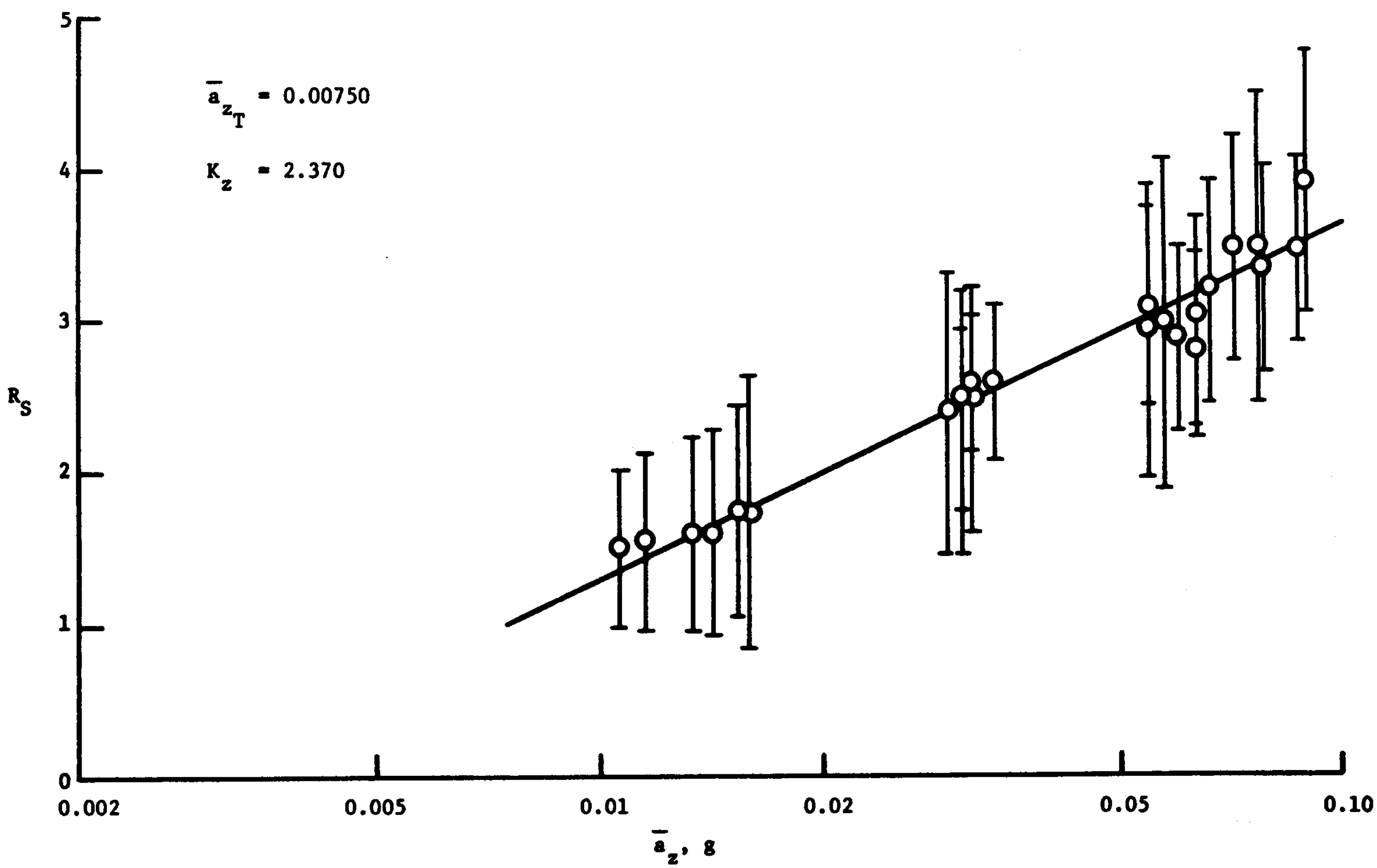

Figure 3.- Variation of ride comfort response rating with rms vertical accelerations.

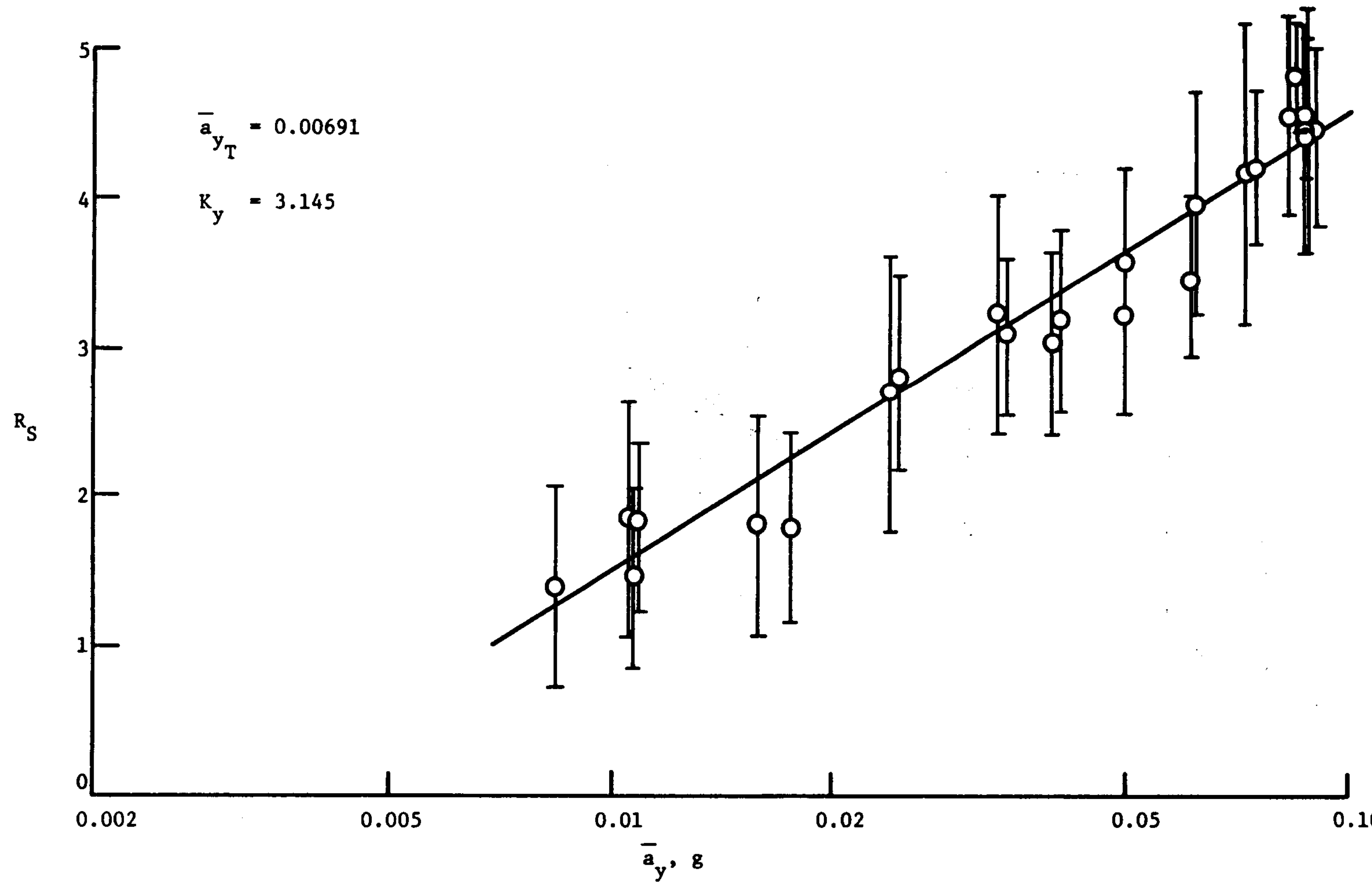

Figure 4.- Variation of ride comfort response rating with rms transverse accelerations.

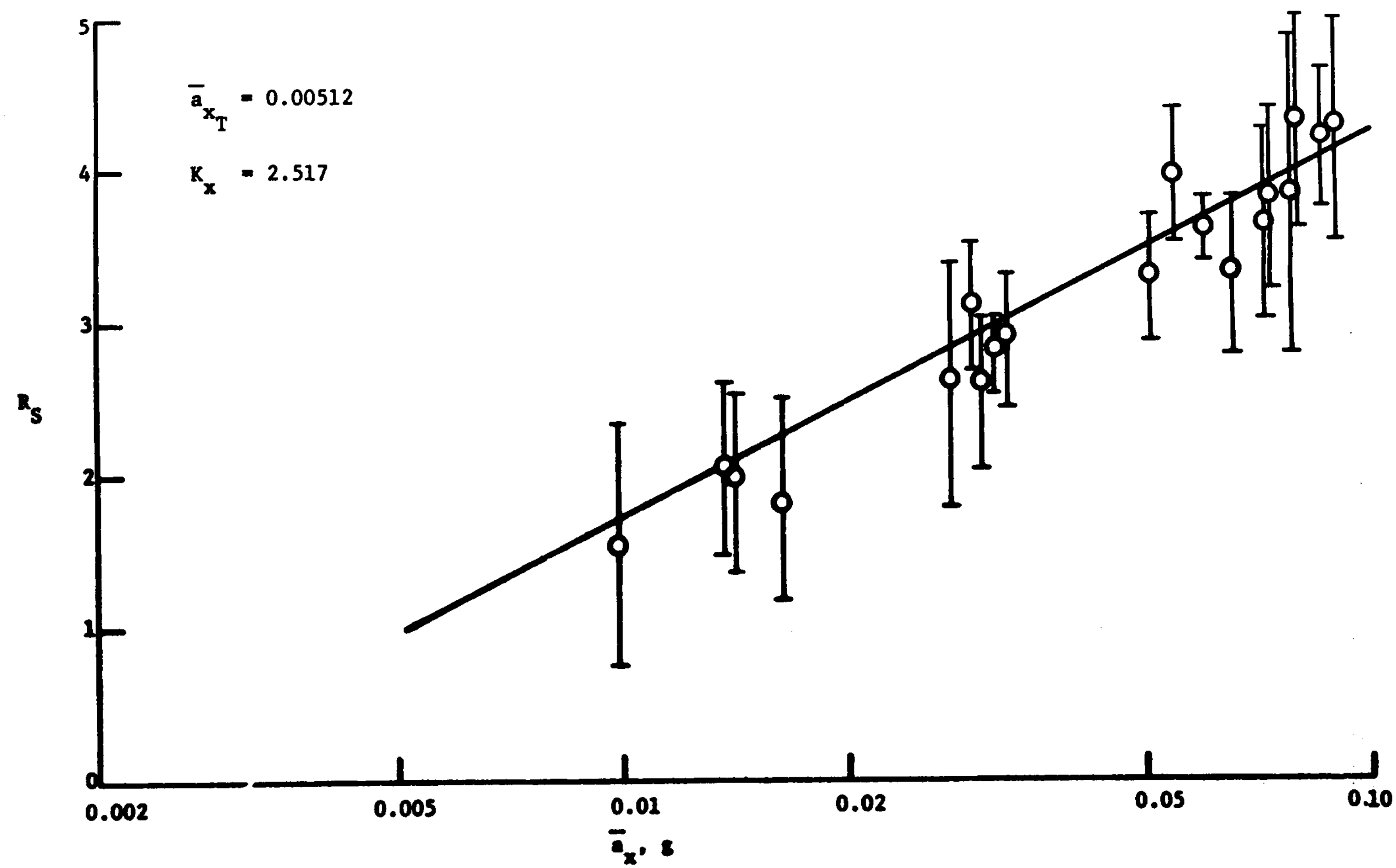

Figure 5.- Variation of ride comfort response rating with rms longitudinal accelerations

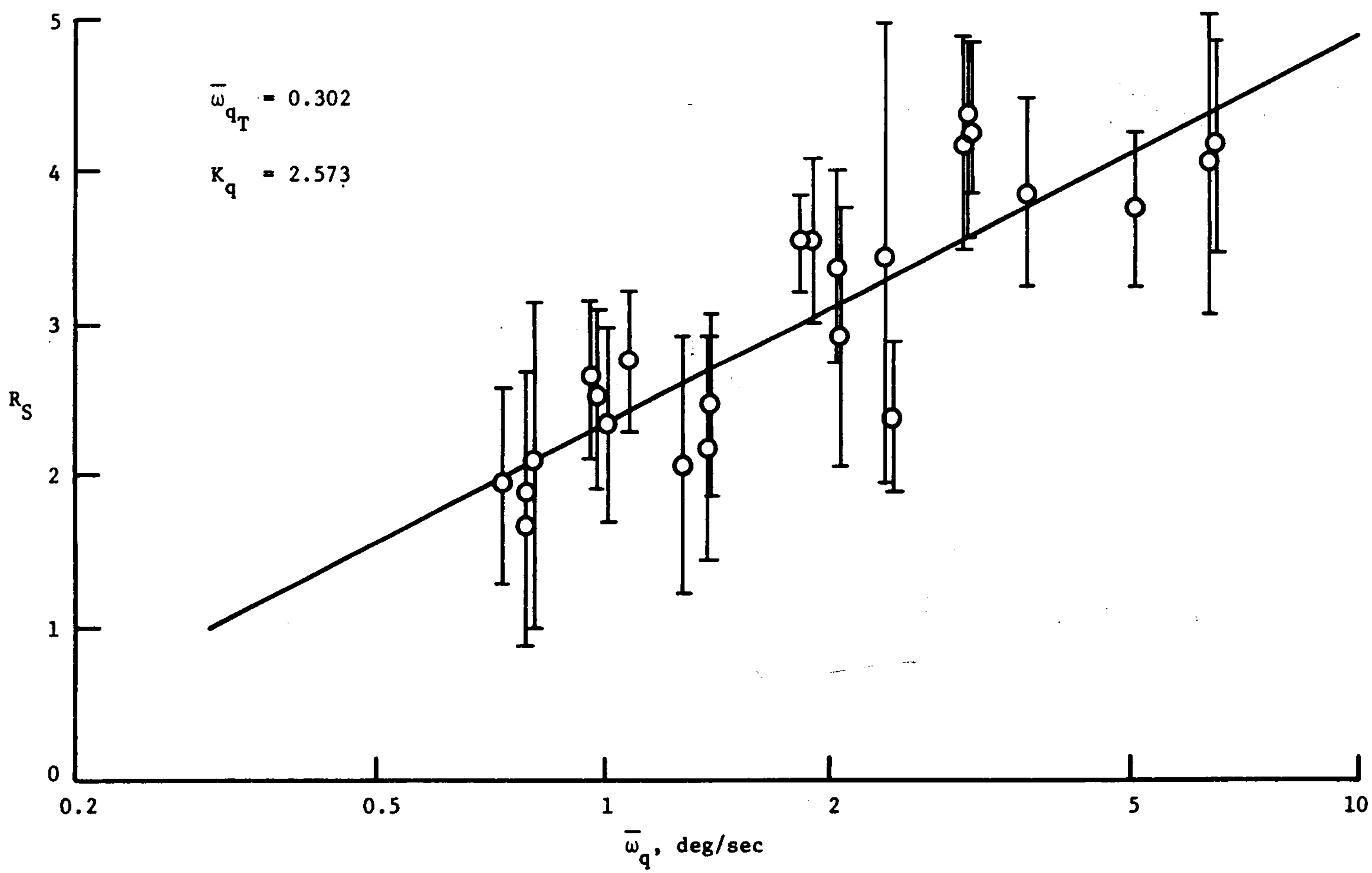

Figure 6.- Variation of ride comfort response rating with rms pitching velocity.

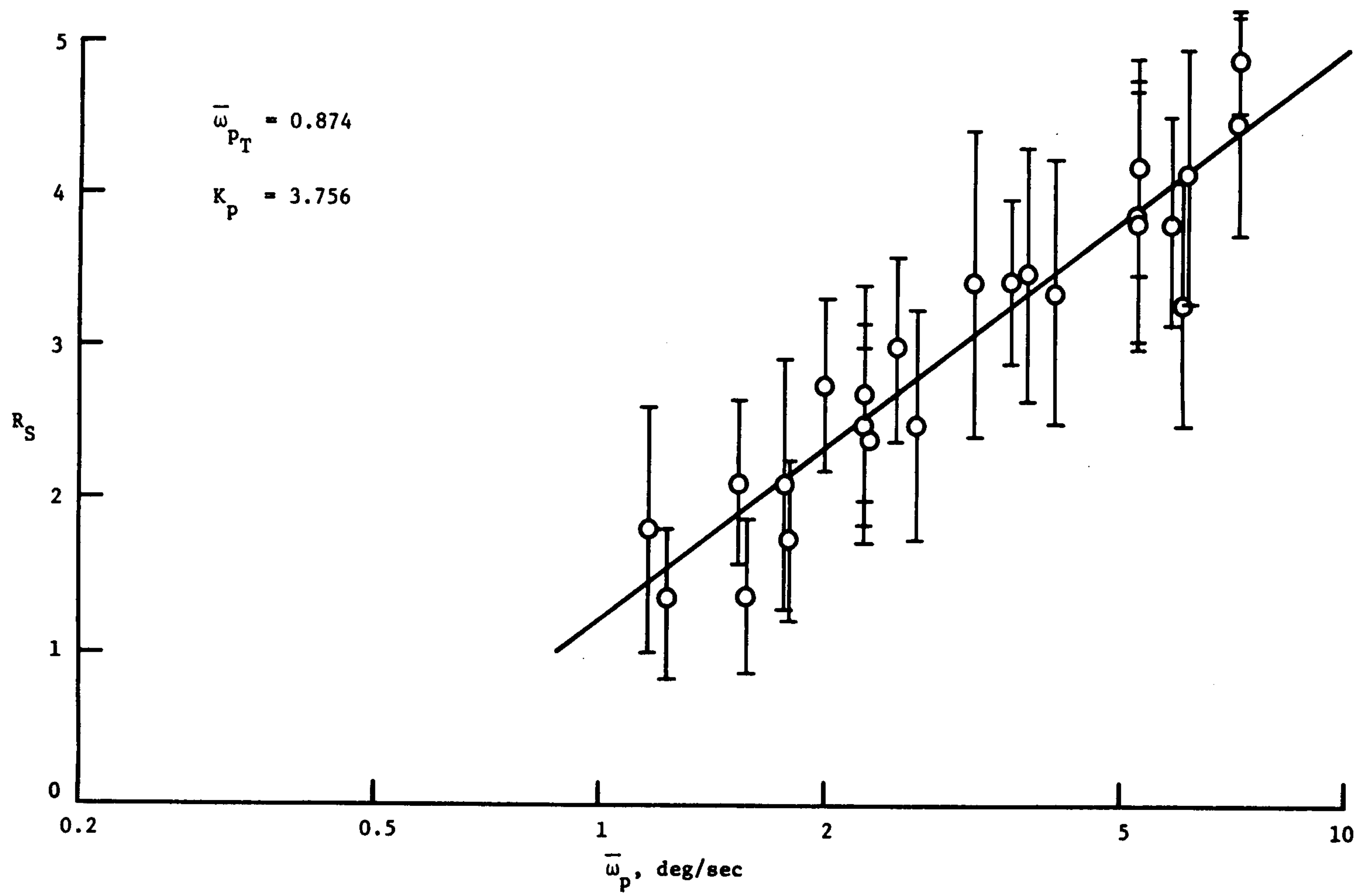

Figure 7.- Variation of ride comfort response rating

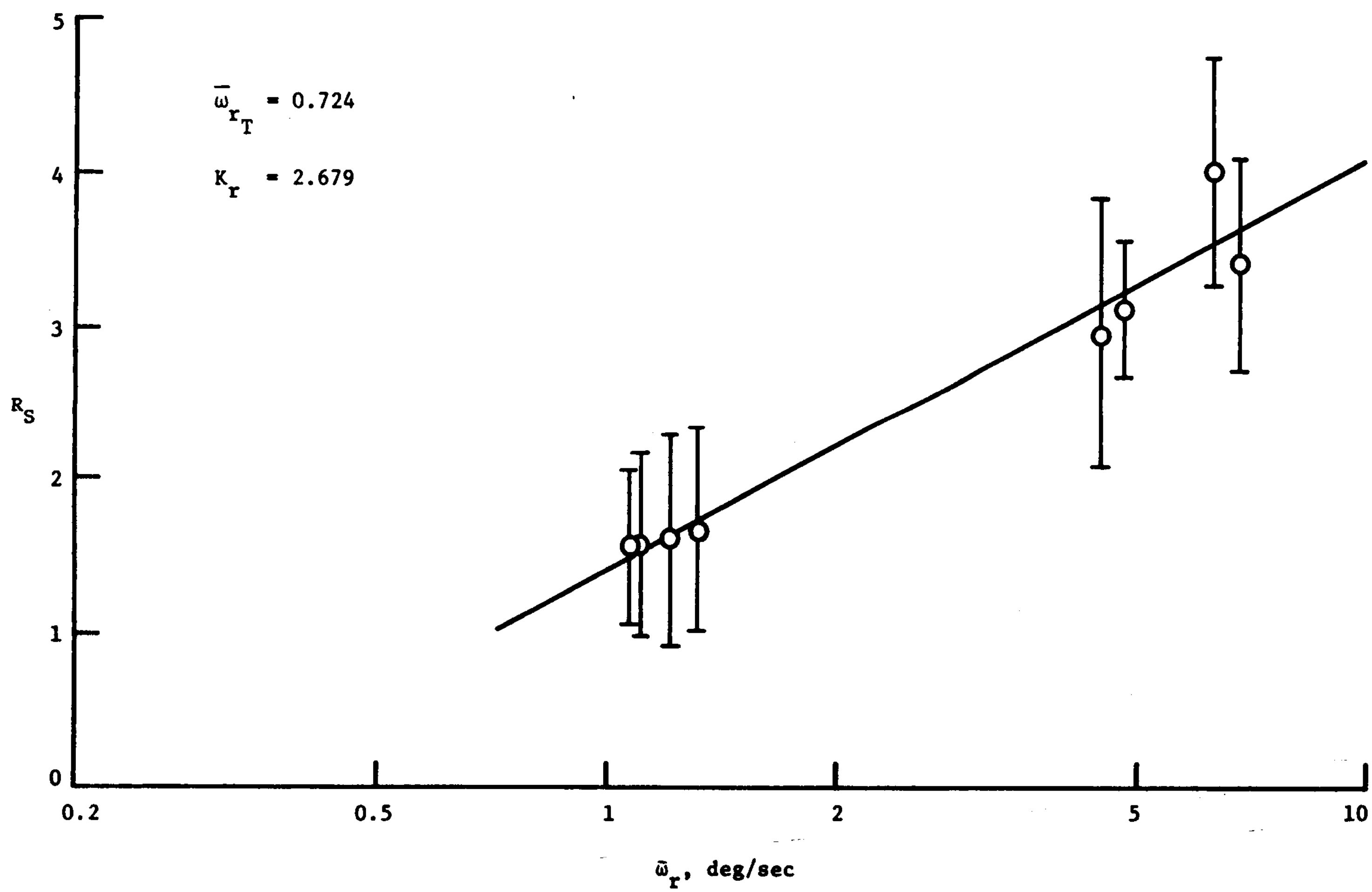

Figure 8.- Variation of ride comfort response rating with rms yawing velocity.

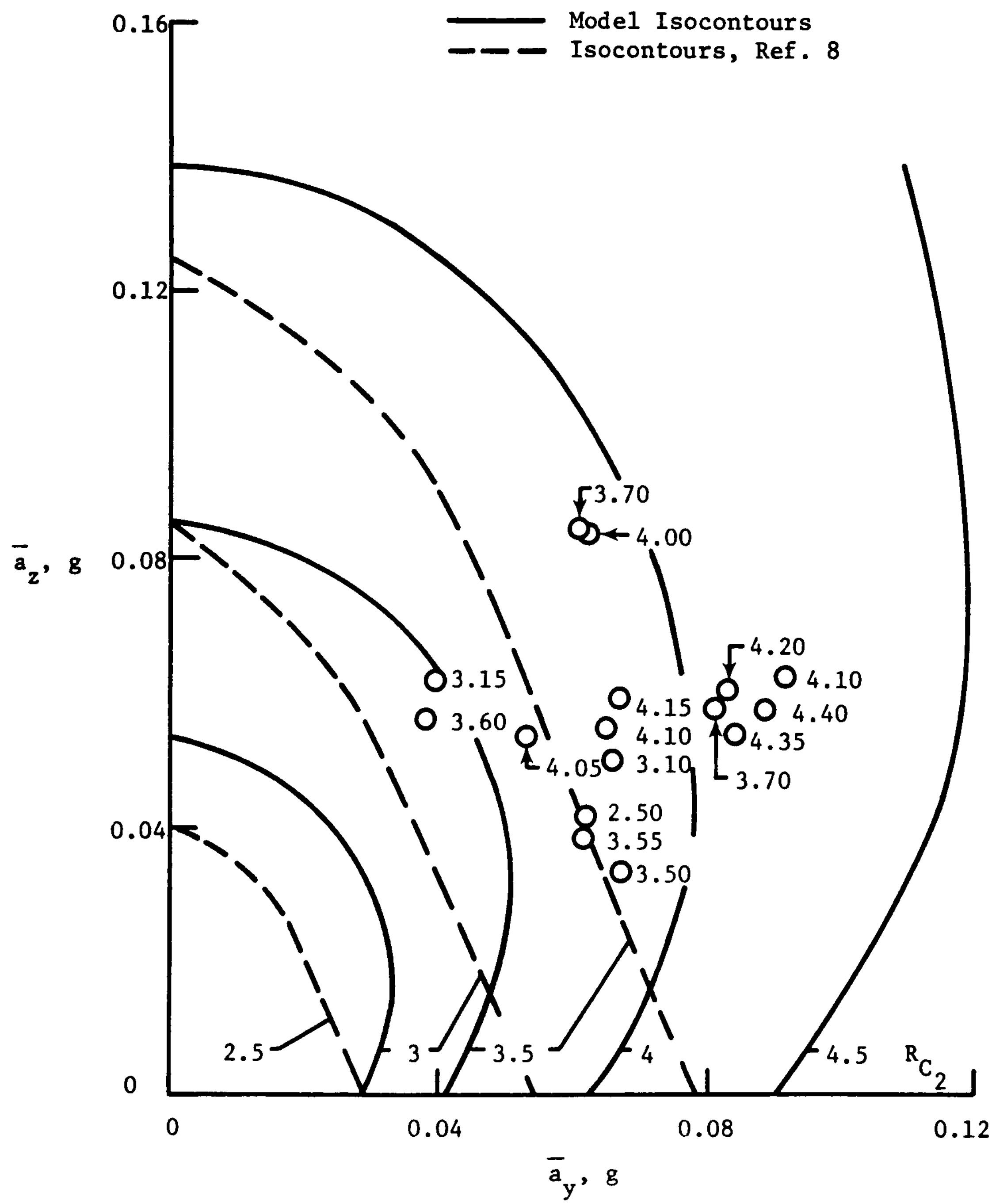

Figure 9.- Ride comfort responses to combined random vertical and transverse motions.

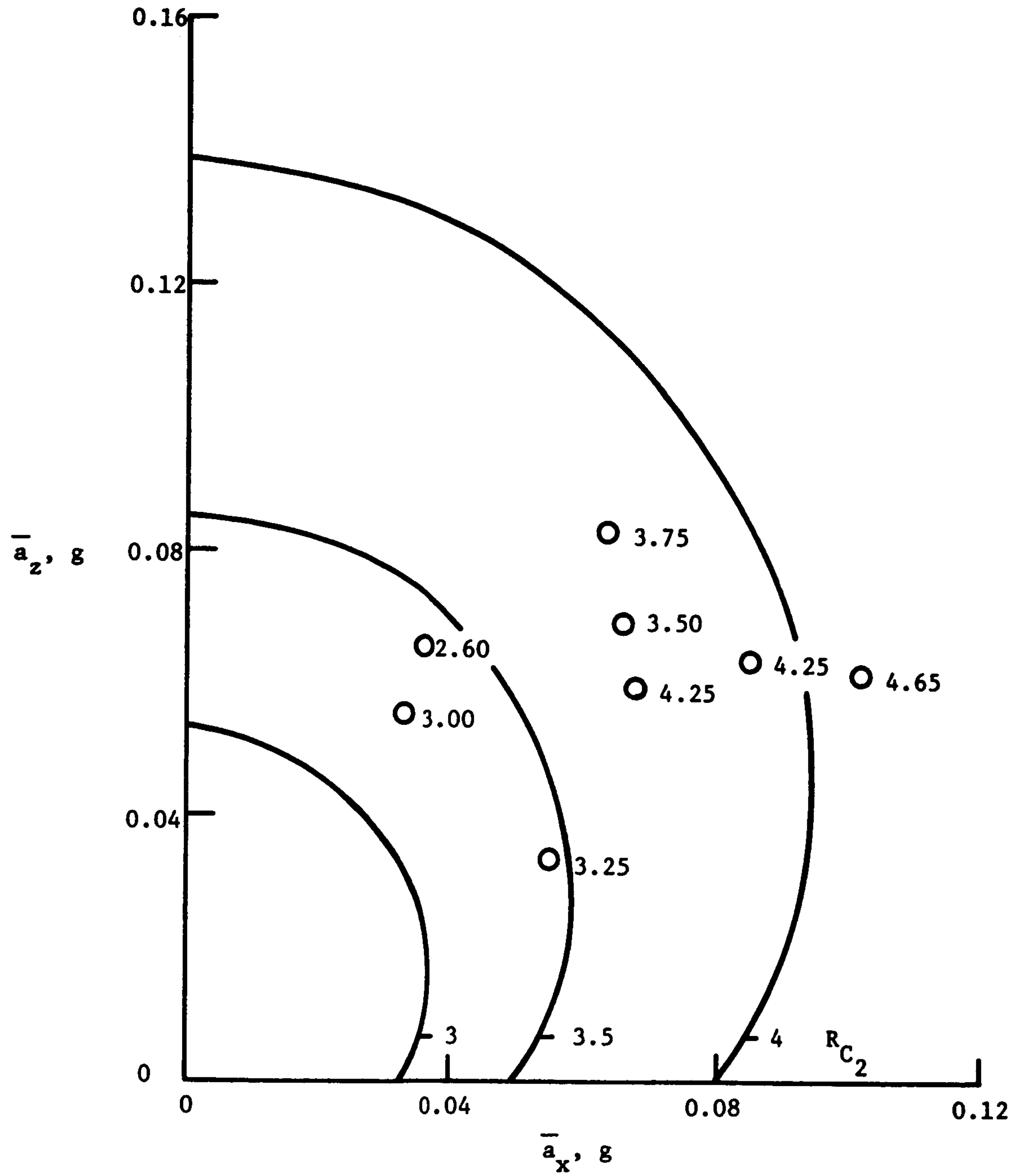

Figure 10.- Ride comfort responses to combined random vertical and longitudinal motions.

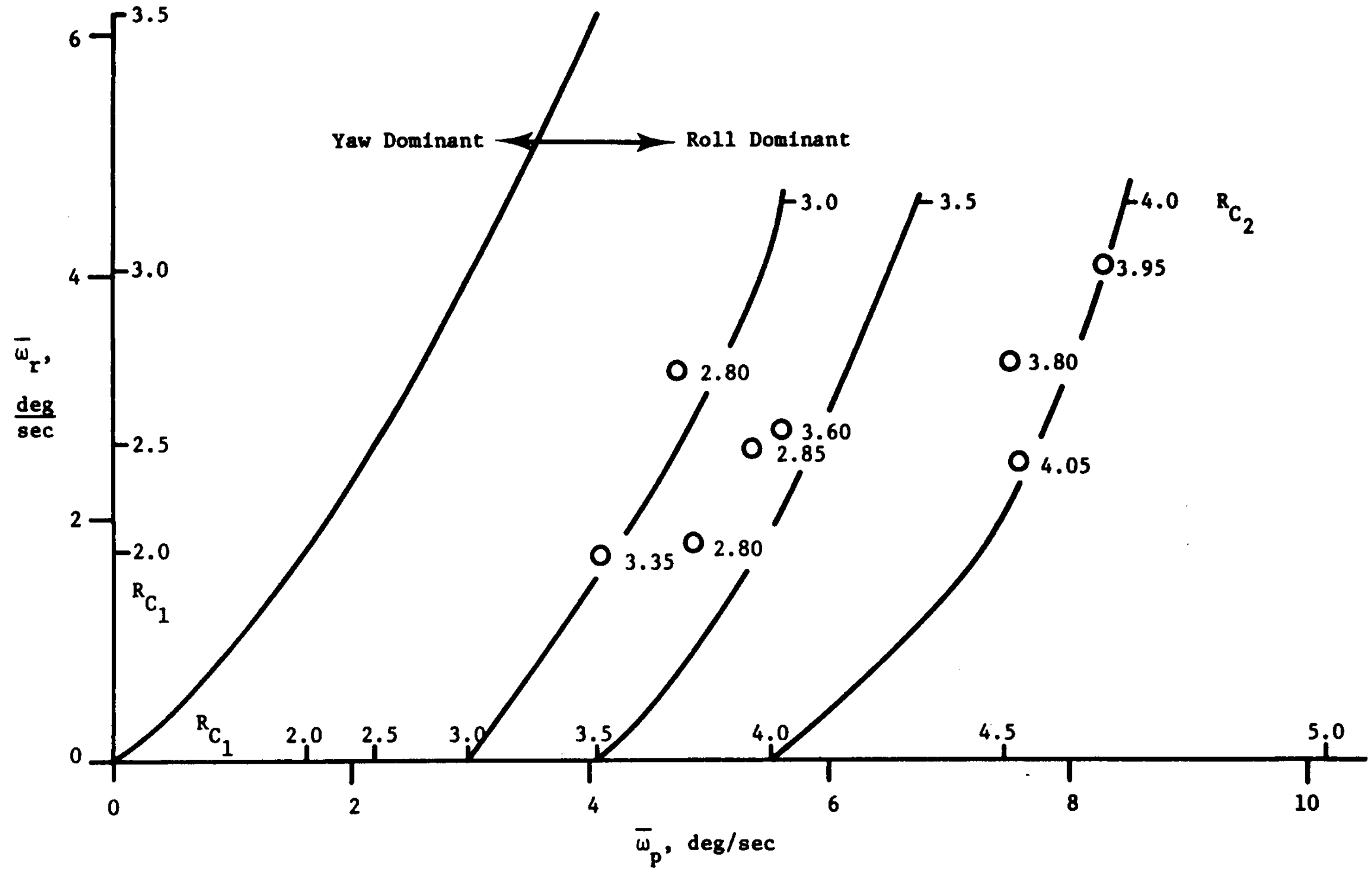

Figure 11.- Ride comfort responses to combined random

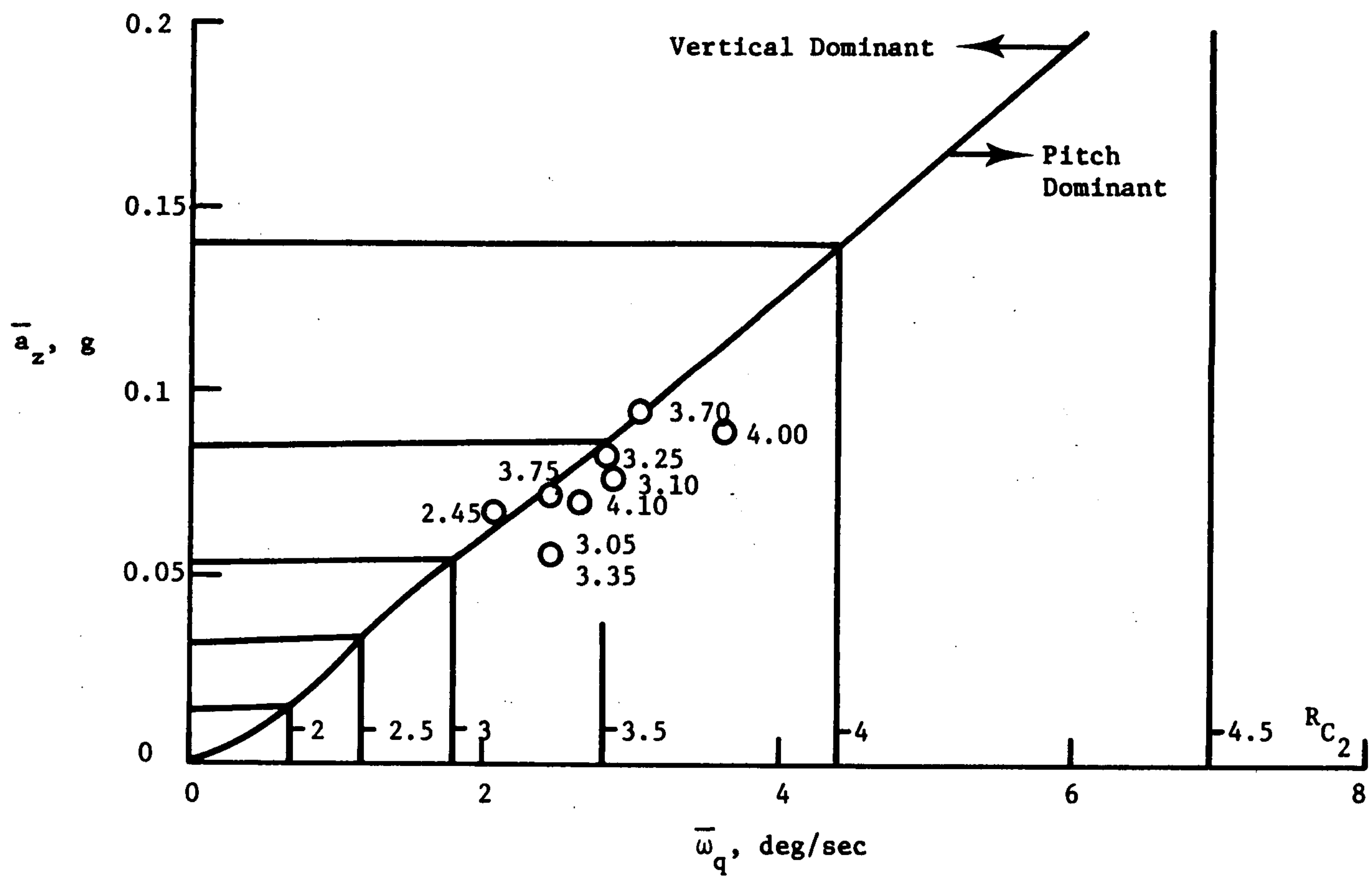

Figure 12.- Ride comfort responses to combined random vertical and pitching motions.

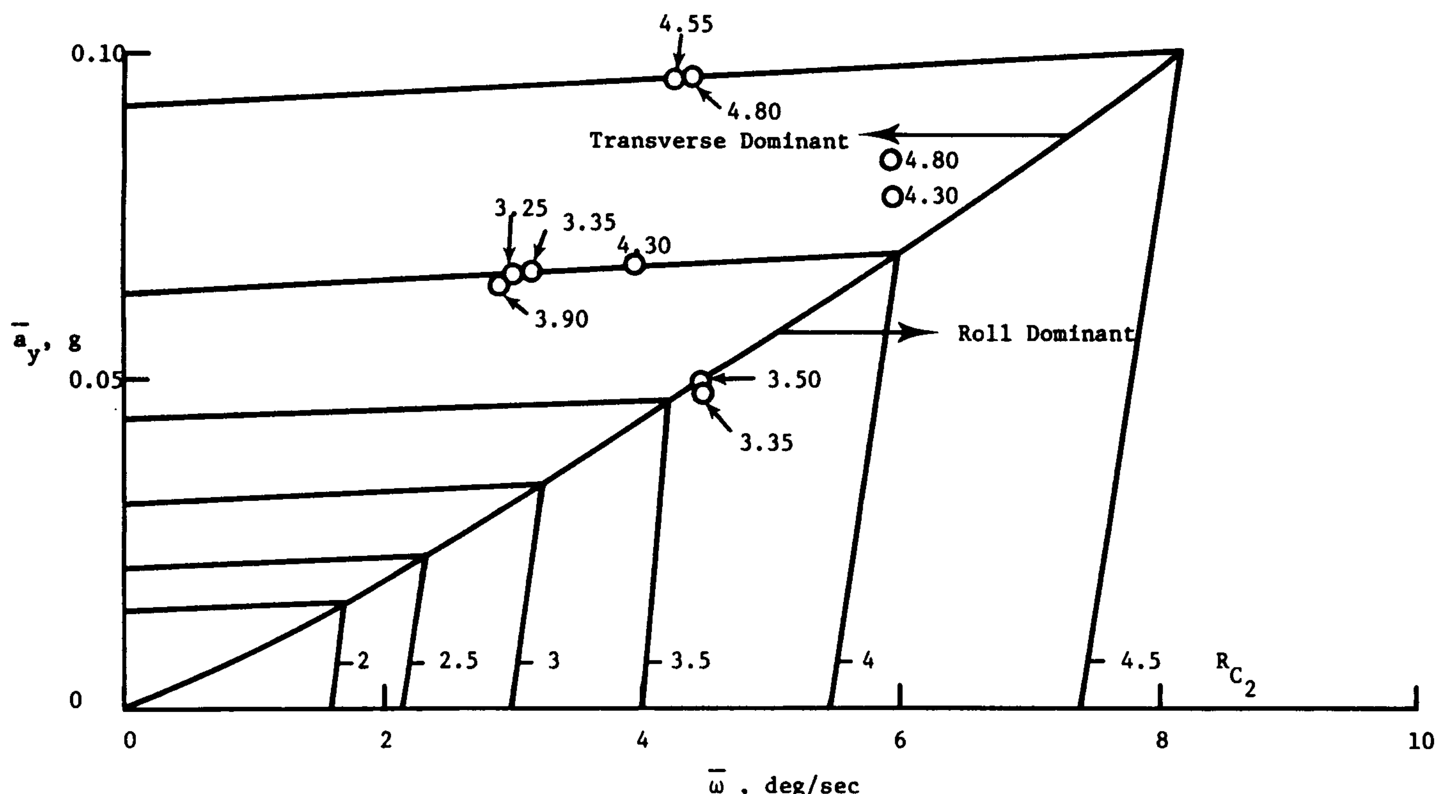

Figure 13.- Ride comfort responses to combined random transverse and rolling motions.

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