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Optimizations of Human Restraint Systems for Short-Period Acceleration¹

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A restraint system's main function is to restrain its occupant when his vehicle is subjected to acceleration. If the restraint system is rigid and well-fitting (to eliminate slack) then it will transmit the vehicle acceleration to its occupant without modifying it in any way. Few present-day restraint systems are stiff enough to give this one-to-one transmission characteristic, and depending upon their dynamic characteristics and the nature of the vehicle's acceleration-time history, they will either magnify or attenuate the acceleration. Obviously an optimum restraint system will give maximum attenuation of an input acceleration. In the general case of an arbitrary acceleration input, a computer must be used to determine the optimum dynamic characteristics for the restraint system. Analytical solutions can be obtained for certain simple cases, however, and these cases are considered in this paper, after the concept of dynamic models of the human body is introduced. The paper concludes with a description of an analog computer specially developed for the Air Force to handle completely general mechanical restraint optimization programs of this type, where the acceleration input may be any arbitrary function of time.

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Optimizations of Human Restraint Systems for Short-Period Acceleration

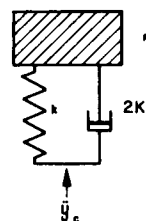
PETER R. PAYNE

The Human Body as a Dynamic System

The central assumption of this paper is that the human body may be represented as a dynamic system. The application of dynamic-model technique to a structure such as a wing is easy to discuss because it has been employed for many years by a great many engineers around the world, and the efficacy of their methods is obvious to everyone. Similarly, we could probably discuss dynamic models for rubber cement, the Queen Mary or extra-galactic nebulae without running into much controversy. Whenever the structure is capable of reproduction, however, we seem to run into communication difficulties, and particularly in the case of the human body it has sometimes proved difficult to gain general acceptance of the fact that the methods of science and engineering still apply. A great deal of controversy has been generated over the past few years since the original (and independent) proposals were made by Latham (1)² in England and Kornhauser (2), (3) in this country. It is probably safe to say that this phase is now largely over, however, particularly because of the excellent results now being obtained with dynamic models of the human body in practical applications. Of particular importance is the fact that dynamic models built up in detail from cadaver test data for individual components of the body [(4) for example] give excellent agreement with the results of tests using live human subjects. Also it is now more generally understood that we are applying the basic approach of the "scientific method" upon which the whole of our civilization is based, rather than making wild and improbable guesses, as was sometimes suggested in the past by workers in the nonmathematical disciplines.

The dynamic models so far developed are limited to the positive spinal (5) and transverse (6) directions, so far as whole-body dynamics are concerned, and to one for the head (7) which is provisionally assumed to apply in any direction; models for other directions are currently being studied. The process of amassing sufficient data to generate these models is much too complicated to permit a simple summary in a brief review of this

nature, except to note that it covers a very broad range involving extensive cadaver data on such elements as individual vertebra (8), AMRL measurements of the steady state and transient impedance of live human subjects (9), numerous measurements of short period acceleration tolerance (10), drop tests (11), and such diverse documents as the records of the American Alpine Club (12). The results of these studies are embodied in Fig.1.



m = mass
 k = spring stiffness
 $2K$ = damper coefficient
 y = input acceleration

Fig.1 Dynamic model of human body

As one would expect, there is an inherent variability in the results of these analyses. The healthy human body is astonishingly standard in some ways - as witness the fact that its temperature is $98.6\text{ F} \pm 0.2\text{ F}$ for millions of specimens - but there is naturally a variation in such areas as muscle tone, bone and artery strength and stiffness, and so forth. By the use of suitable methods of analysis it is found that some of the apparent variability is really a variation with age, and that the residual variation is quite surprisingly small, so far as those factors which influence acceleration tolerance are concerned. Presumably, biologists will identify more specific factors than age in the future, and further reduce the random variation; indeed, many such factors have already been identified, and it is obvious that artery stiffness could be correlated against subject age and Atherogenic Index, for example.

An additional source of variability, when analyzing the results of whole body experiments, is the subject's posture. This is particularly important in the case of spinal acceleration, as pointed out by Latham (1) and Bosee and Payne (13) because canted vertebrae are significantly less able to withstand compressive loading. The dy-

² Numbers in parentheses designate References at the end of the paper.

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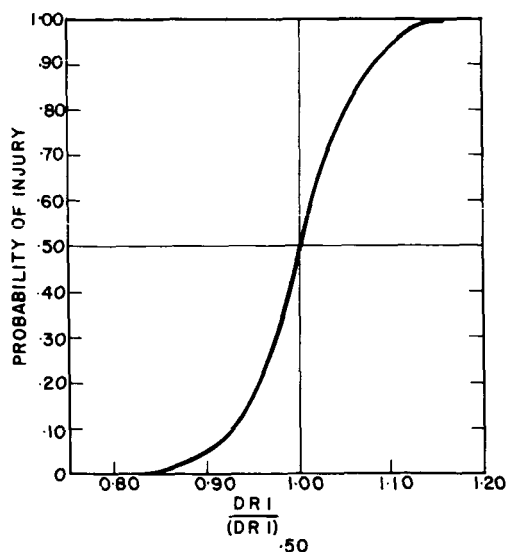


Fig. 2 Probability of minor spinal injury as a function of dynamic response index

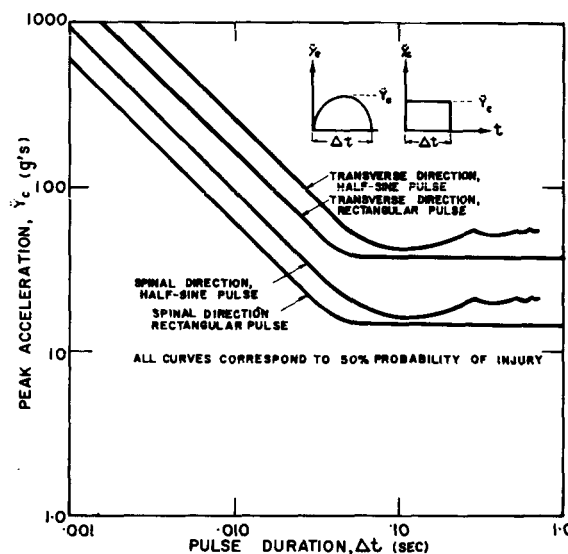


Fig. 3 Fifty percent probability of injury curves for positive spinal and transverse acceleration

dynamic models so far developed are for "well-supported" subjects and no one is better aware than we of the unsatisfactory nature of this vague qualification!

By far the best estimates of variability are for the positive spinal direction and the appropriate curve, taken from Stech's work (14) is given in Fig. 2. Since the critical "Dynamic Response Index" (DRI) given in Fig. 1 is for a 50 percent probability of (minor) injury, Fig. 2 enables us to determine the critical DRI for any other probability of injury (P.I.).

The physical basis for Fig. 2, and a detailed discussion of its limitations, is given by Stech (14).

In concluding this section on dynamic models we will briefly look at some of the results obtained with them, in order to relate the theory to practical cases within general experience. In Fig. 3 the 50 percent injury curves are plotted for rectangular and half-sine acceleration pulses, as a function of pulse duration, for both positive spinal and transverse directions, and ignoring head limits. In other words, the head is assumed to be sufficiently well restrained to raise its tolerance level to that of the whole body. In practice this is achievable in every vector direction except spinal, where a sufficiently short rise time will result in pressure waves being transmitted directly up the spine to the head. (While this is admittedly still an hypothesis, anyone can test it by slipping on ice, making a "one-point" landing on the buttocks and examining the resulting headache!)

Thus in the case of positive spinal, very

short rise times will cause head involvement, and the 50 percent injury curves will be modified as shown in Fig. 4. It should be emphasized that the head involvement in positive spinal acceleration has a considerable amount of experimental verification, as is shown in Stech's analysis of Swearingen's pioneering drop tests (11), involving 162 drops with live human subjects.

It is interesting to compare the injury curves in Fig. 5 with the "tolerance limits" suggested by earlier investigators, and this is done in Figs. 5 and 6 for transverse and positive spinal.

Finally, we may ask whether the models give reasonable agreement with established tolerance limits to steady-state sinusoidal vibration. Part of the answer to this question is illustrated in Fig. 7 where we see that there is rather a broad range of experimental curves, depending upon the details of the test and the actual end-point being investigated. None of these end points corresponds to the vertebral injury for which the dynamic model was developed, so it is somewhat surprising to see that reasonable agreement can be obtained at low frequencies merely by taking a lower (an arbitrary) DRI value. Above a frequency of 8 cps it is evident that another mode of deformation becomes important, however, and for a fuller discussion of this, reference should be made to the work of von Gierke (16).

We have now established, albeit with a number of reservations, dynamic models which describe the human body as accurately as the available experimental data permit. Thus we are now in a position to examine the effect of placing another dynamic system - the restraint system - in series with it.

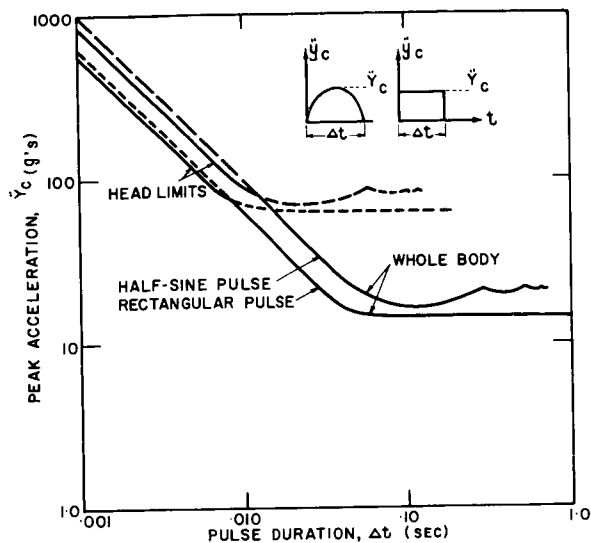


Fig. 4 Fifty percent probability of injury curves for positive spinal showing head involvement

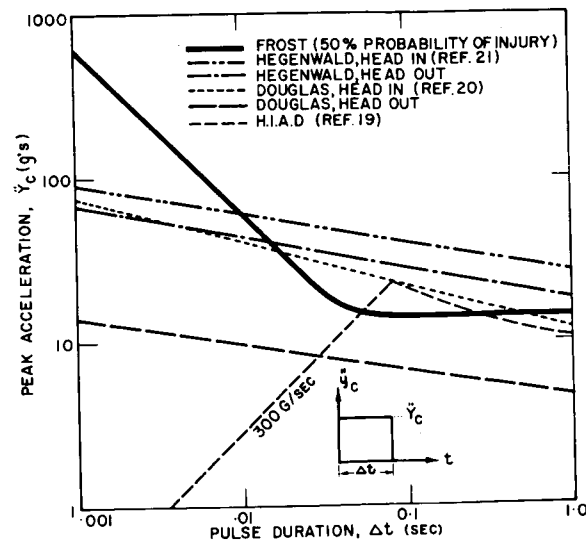


Fig. 6 Comparison of spinal-injury curves with "tolerance limits" suggested by earlier investigation (rectangular pulse)

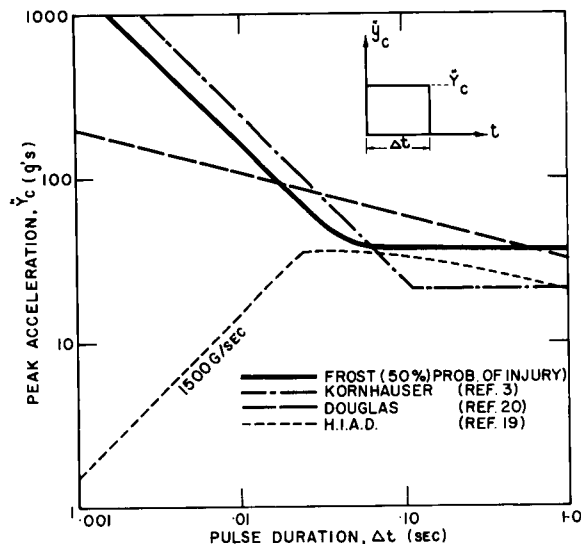


Fig. 5 Comparison of transverse injury curves with "tolerance limits" suggested by earlier investigation (rectangular pulse)

In most cases we shall ignore the influence of damping in the human body, in the interests of mathematical simplicity. Although this obviously influences the results, it does not have a serious effect upon the general trends.

Function of a Restraint System

A restraint system has three distinct functions to perform, although, until recently, only the first two have received any real attention.

Obviously, it must restrain its occupant under conditions of high acceleration and maintain his

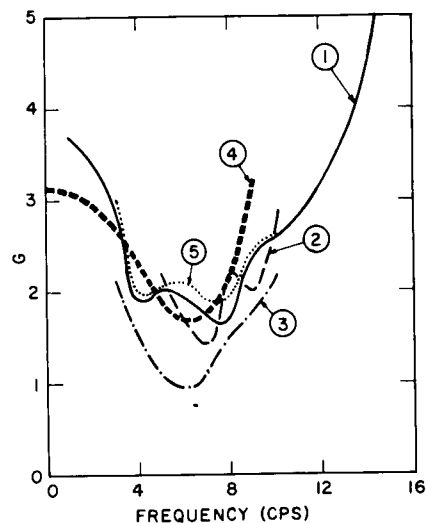


Fig. 7 Human tolerance to sinusoidal vibration in a spinal direction

Legend for Fig. 7

- 1 Zeigenruecker and Magid, WCDC TR 59-18
- 2 Mandel and Lowry, AMRL-TDR-62-121
- 3 Magid, Coermann, and Zeigenruecker, JAM, vol. 31, pp. 915-924, 1960
- 4 Frost Engineering dynamic model of human body, spinal seated
- 5 Same source as 3

NOTES: 1 involves variable tolerance times, from 18-208 sec; 2 50-70 sec tolerance (1 min tolerance); 3 based upon respiratory difficulties; 5 "short time" tolerance

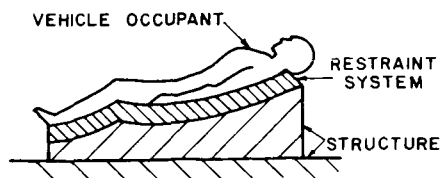


Fig. 8 Definition of elements

posture in his seat or couch in such a way that he is best able to perform his tasks and withstand the loads applied to his body.

Equally obviously, it must be comfortable under normal conditions, when its occupant is not in need of its restraining abilities. In a negative sense, a shoulder harness should not obtrude on the consciousness of its wearer, and should not inhibit any of his body movements associated with the execution of his normal tasks. In a positive sense, restraint elements such as cushions are introduced to increase comfort under normal conditions, and do not necessarily play a significant role in restraining the seat or couch occupant when subjected to acceleration.

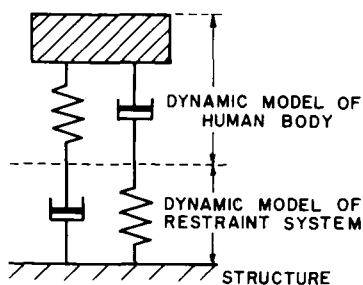


Fig. 9 Dynamic model of Fig. 8

Finally, the fact that a restraint system is resilient means that it will modify the nature of any acceleration applied to it. Obviously, it should attenuate rather than magnify the acceleration, and in the context of this paper a dynamically optimum restraint system will yield the maximum degree of attenuation (or minimum amplification) possible within its space limitations. It is with this aspect of restraint system design that we are concerned in this paper.

MODIFICATION OF INPUT ACCELERATION BY A RESTRAINT SYSTEM

An "input acceleration" is defined as the acceleration of the vehicle's structure. This is a rather loose definition, since the flexibility inherent in any structure will result in somewhat

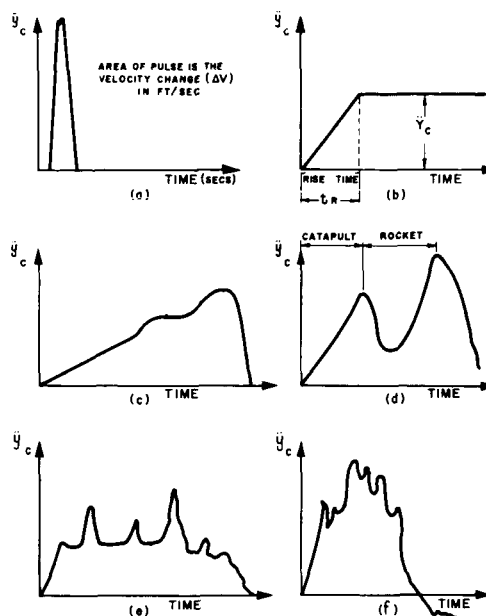


Fig. 10 Some typical acceleration - time histories

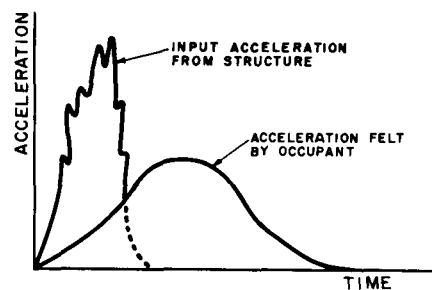


Fig. 11 Modification of input acceleration by a soft-restraint system

different accelerations being measured at different points. Since we are interested in the physiological effect of an acceleration, however, we are obviously talking about the structural elements immediately adjacent to the vehicle's occupant and his restraint system, this being usually his seat or couch. In addition, we can make the initial generalization that frequencies in excess of 100 cps will have no physiological effect, particularly since in practice the amplitude associated with such high frequencies is almost always negligible, so that we can ignore the fine-grain details of the structure's acceleration-time history.

One arrangement of the three separate elements under consideration is shown in Fig. 10 and their dynamic equivalent in Fig. 8. Note that the mass of the restraint system can almost always be ne-

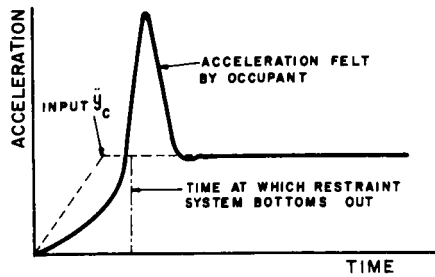


Fig.12 Modification of a short-period acceleration by a restraint system which bottoms out

glected; a generalization which we can formalize by requiring its natural frequency to be above 30 cps when unoccupied.

The input acceleration-time history, measured in the structure, may have almost any form, and some typical ones are sketched in Fig.10. In

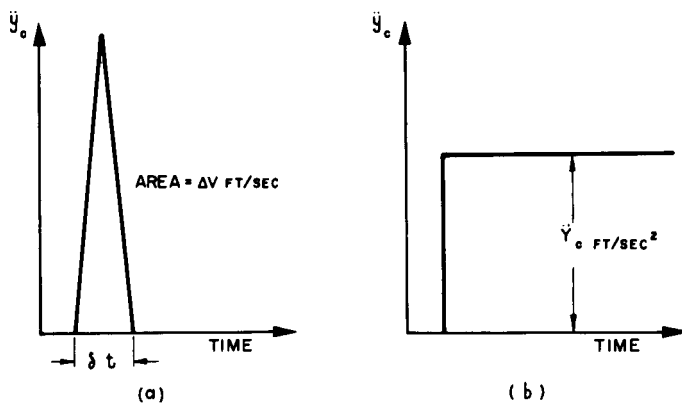


Fig.13 Idealized acceleration inputs

mathematical analysis it is referred to as y_c , the units being ft/sec^2 , while in more general work the notation G is used:

$$G = \frac{y_c}{g} = \frac{y_c}{32.2}$$

Thus G is nondimensional. Note that we often refer to y_c or G as the "input acceleration," or more simply, as the "input."

If we now take the case of Fig.10(f) and assume that this is applied to a very soft restraint system, with a very long travel before it bottoms out, then the acceleration felt by its occupant is as sketched in Fig.11. Obviously, we have modified the input very significantly and to the bene-

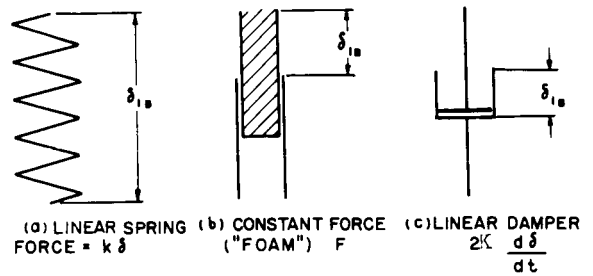


Fig.14 Restraint system idealization

fit of the occupant, in this particular case. We can equally well experience the opposite, however, and Fig.12 illustrates a case in point. Here the restraint system bottoms out before it can carry the full load applied to it, so that its occupant not only has to withstand the full plateau acceleration but also has to absorb an impulsive acceleration at the same time.

When examining these effects mathematically, we obviously cannot begin to encompass the variety of input pulse shapes sketched in Fig.10, but must confine ourselves to very simple idealizations. It is found that the two most useful ones are those depicted in Fig.13, since they encompass the

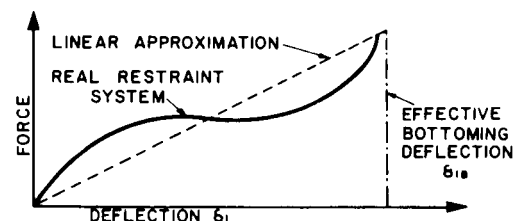


Fig.15 Linear spring approximation to a real system

two extremes of the pulse shapes in which we are interested.

A great deal of theoretical work has been accomplished using these two forcing functions, the first of which will of course be recognized as the Dirac impulse function.

In general, the case of Fig.13(a) will apply to any pulse whose duration is less than $(1/\omega)$ seconds, where ω is the natural frequency of the dynamic system in radians/sec, while Fig.13(b) applies to durations in excess of $(3/\omega)$ seconds. Using these forcing functions we can derive some general guide lines and basic theorems which will help in the selection of optimum restraint systems for the more irregular acceleration-time histories

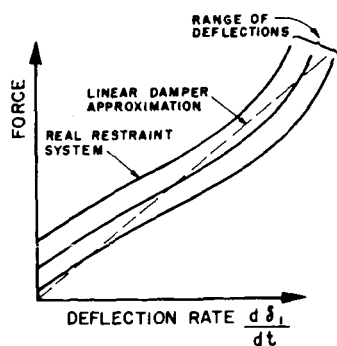


Fig. 16 Linear damper approximation to a heavily damped restraint system

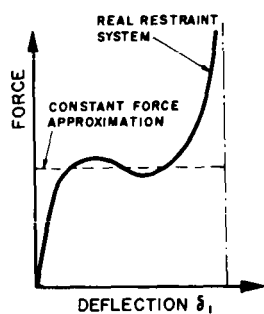


Fig. 17 Constant force approximation to a real system

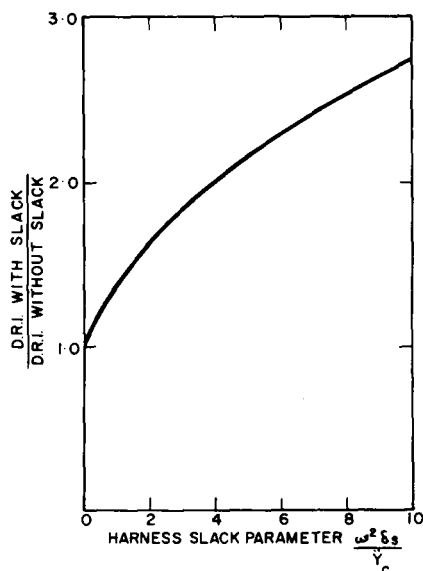


Fig. 18 DRI magnification by slack harness, for a short-period acceleration Y_c (rigid restraint system)

which are experienced in practical applications. It should be realized that the true optimum can only be obtained by solving the equations of motion for the actual input under consideration, however, and that the guide lines deduced from idealized inputs are nothing more than general indications.

Simple Idealizations of a Restraint System

We have seen that mathematical analysis is only possible when a very simple acceleration input is employed. The same is true of the mathematical representation which is used to describe the dynamic characteristics of the restraint system, and the three idealizations used in this report are shown in Fig. 14.

The deflection required to bottom the system

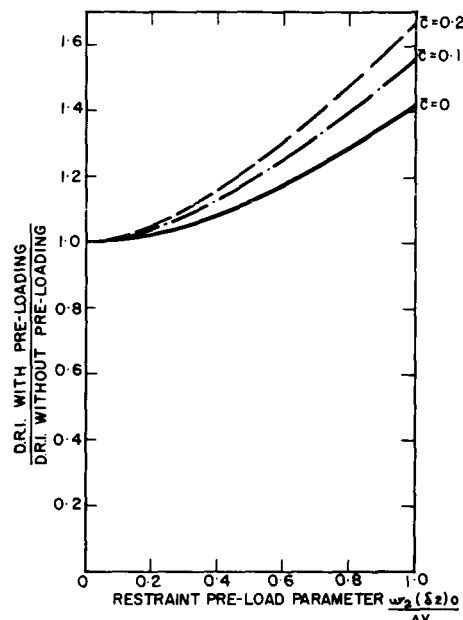


Fig. 19 DRI magnification due to harness pre-tension for impact acceleration

is defined as δ_{1B} , the suffix (1) always denoting that this parameter refers to the restraint system.

The spring illustrated in Fig. 14(a) is representative of materials with negligible damping. Thus it could well apply to any closed cell foam cushion, for example. The fact that such restraint materials are essentially nonlinear so far as their force-deflection curves are concerned, does not necessarily invalidate the use of this approximation, as Fig. 15 shows, so that it can be applied to many practical cases. In the same way, the constant force and linear damper approximations can also be substituted for many real systems, as shown in Figs. 16 and 17. Thus the idealizations of Fig. 16 are very much more useful than might appear at first sight.

It can be shown that the "quality" of a restraint system, irrespective of its physical nature, is defined by a single parameter for each type of acceleration input. For an impulsive velocity change (Δv) this parameter is

$$\frac{\Delta v}{\omega_2 \delta_{1B}} = \frac{\text{impulsive velocity change}}{\text{human body natural frequency} \times \text{bottoming depth}}$$

Obviously, if

Δv is in fps

ω_2 is in radians

δ_{1B} is in ft

The smaller the value of this parameter, the

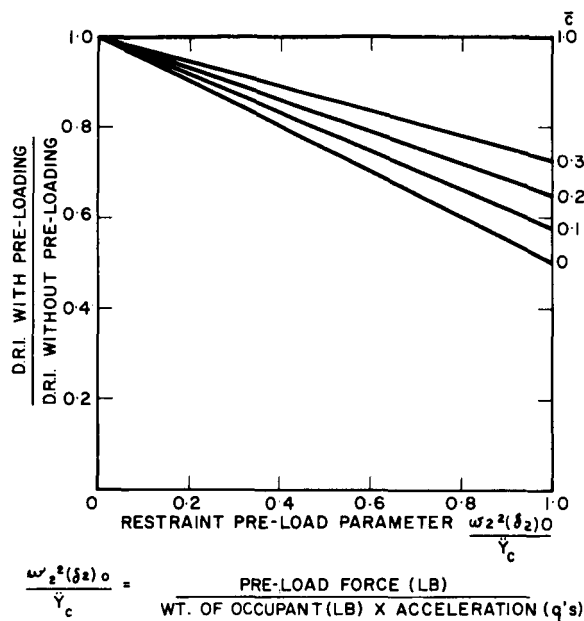


Fig. 20 Attenuation of DRI due to a short-period acceleration by preloading the restraint system

greater the attenuation of a given impact.

For a short period acceleration ($\ddot{Y}_c$) the quality parameter is

$$\frac{\ddot{Y}_c}{\omega_2 \delta_{1B}}$$

$$= \frac{\text{acceleration magnitude}}{\text{human body natural frequency} \times \text{bottoming depth}}$$

the units being as before, and $\ddot{Y}_c$ being in ft/sec². The variation of attenuation with this parameter is more complex, and will be discussed in later sections.

INFLUENCE OF SLACK OR PRELOAD IN A RESTRAINT SYSTEM

When a harness or restraint system is slack, so that the occupant has to travel a distance δ_s before contacting it, the dynamic response to a short-period acceleration will be magnified. The magnification factor is given in Fig. 18, which is based upon the theory of reference (17) for a very stiff (in relation to the human body) restraint system.

When the restraint system has a lower stiffness, the more complicated analysis in (17) shows that the DRI magnification will usually be less than the value given in Fig. 18.

The DRI due to an impulsive acceleration is unaffected by harness slack. Pretension, on the other hand, is harmful in this case, and causes the magnification shown in Fig. 19. This harmful

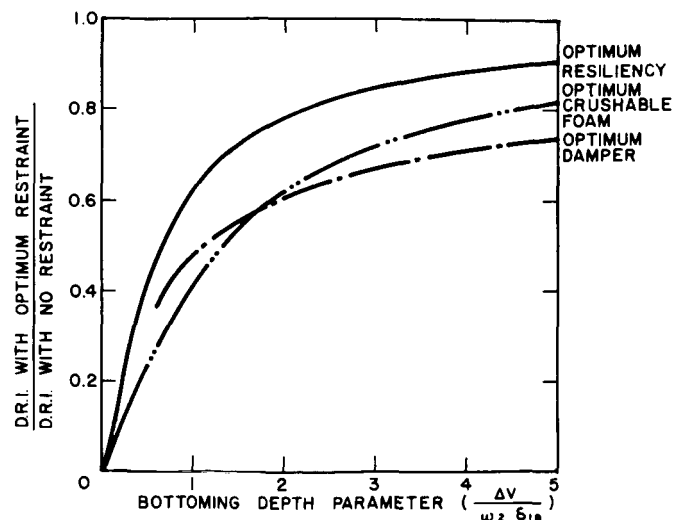


Fig. 21 Attenuation of an impulsive acceleration by idealized restraint systems

effect is quite small, however, for practical pre-load values.

For short-period accelerations, preloading reduces the dynamic response, but once again the effect is only a few percent in magnitude for practical values, as shown in Fig. 20.

In summary, then, we can draw the following conclusions, so far as the DRI is concerned:

	<u>Slack harness</u>	<u>Preloaded harness</u>
Impulsive acceleration..	no effect	slightly harmful
Short-period acceleration..	very harmful	slightly beneficial

Since a slack harness can lead to very large magnification in the short-period case, it should always be avoided. The dynamic effects of preloading, both favorable and unfavorable, are so small that we can probably neglect them in relation to two much more cogent reasons for pretensioning: viz,

- It guarantees that no slack exists,
- It enforces good body posture in a properly design restraint system.

OPTIMUM RESTRAINT SYSTEMS FOR IMPACT ACCELERATION

The optimum restraint system for an impulsive velocity change (Δv) is one which just bottoms-out as the maximum load is reached. In this way, we obtain the benefit of the maximum deceleration distance available.

The performance of the three idealized restraint systems is plotted nondimensionally in Fig. 21, where it is seen that for

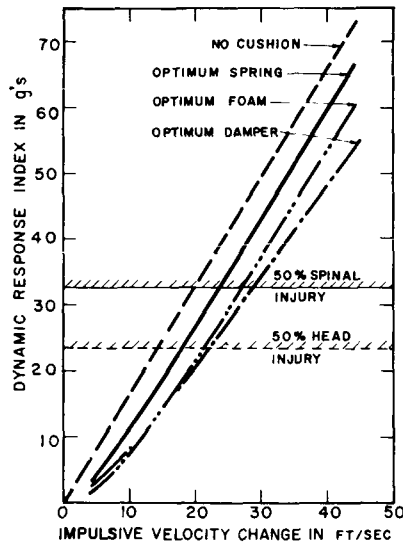


Fig. 22 Variation of DRI with impulsive acceleration for various optimized cushions

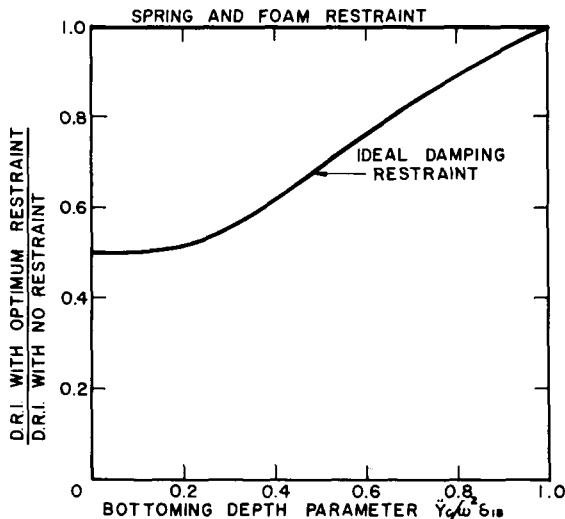


Fig. 23 Attenuation of a zero-rise time acceleration for various optimized cushions

$\frac{\Delta v}{\omega_2 \delta_{1B}} < 1.75$ constant force deceleration is optimum

$\frac{\Delta v}{\omega_2 \delta_{1B}} > 1.75$ viscous damping is optimum.

A typical calculation for a practical case is given in Fig. 22, where DRI is plotted against impulsive velocity change for a bottoming depth of two inches. The critical velocity change for 50 percent probability of injury is seen to be

- 23.7 fps for spring-restraint system
- 27.0 fps for foam-restraint system
- 28.7 fps for viscous-damping case

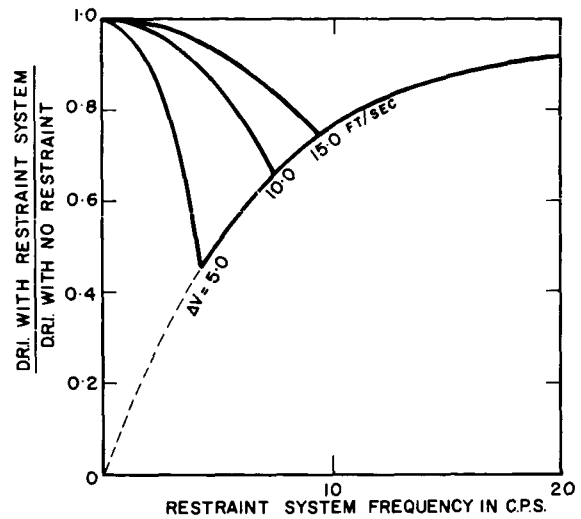


Fig. 24 Effect of restraint system stiffness upon DRI due to an impulsive velocity change

These figures compare with 19.8 fps for a 50 percent probability of spinal injury without any restraint system at all, but the improvement is better than would be indicated by this comparison. As shown in Fig. 22, the limiting velocity change for head injury is much lower than for spinal injury, and this would be the limiting factor if no restraint system were employed. Assuming linear deceleration from 30 fps, over a distance of 2 in., however, we see that the restraint system will take about 11 millisecc to absorb the shock. From Fig. 1, we see that this is in the "short-duration" regime for head injury, so that the restraint system has effectively eliminated this particular injury mode.

Thus the use of an optimum damper restraint system, with a bottoming depth of 2 in., effectively doubles the tolerable impulse for the human body in the positive spinal direction.

OPTIMUM RESTRAINT SYSTEMS FOR SHORT-PERIOD ACCELERATION

Unless they are so rigid that they cannot deflect at all, both foam and spring restraint systems will magnify a short-period acceleration. Thus the optimum foam system is defined as one which requires a crushing force greater than

$$2 \ddot{Y}_c m$$

i.e., 2 * acceleration input * mass of occupant or 2G(mg)

Similarly, the optimum spring restraint system will require a force larger than this value to bottom it. This aspect is discussed in more detail in the next section, and the minimum re-

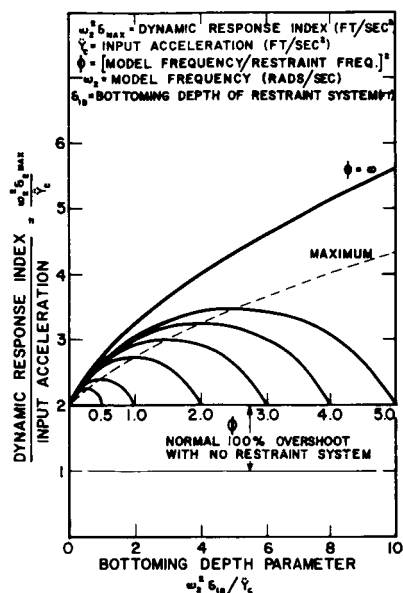


Fig. 25 Influence of a linear spring restraint system on the DRI of an undamped dynamic model

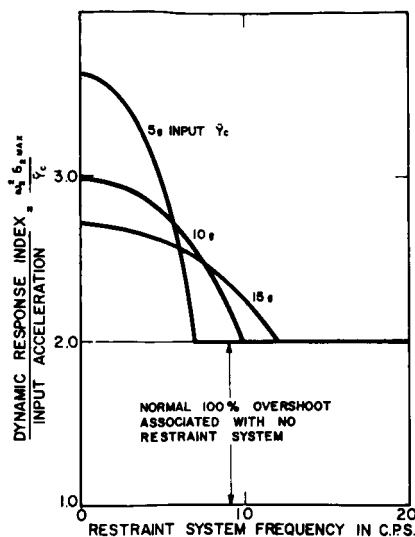


Fig. 26 Variation of DRI with restraint for 2-in. bottoming depth - spinal acceleration

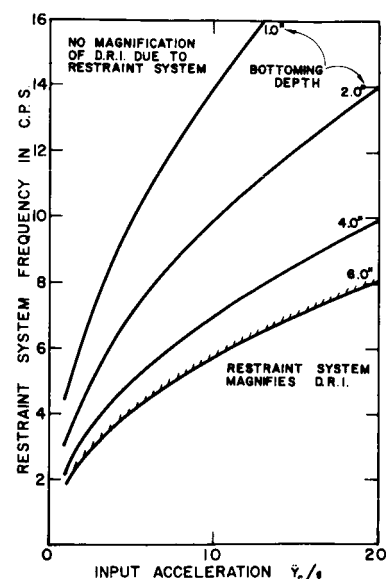


Fig. 27 Critical restraint system frequency for zero magnification of zero rise time acceleration

straint system frequency for optimum performance is plotted in Fig. 27.

Only a damped restraint system can attenuate a zero rise time, short-period acceleration, the attenuation function being that plotted in Fig. 23. Since a damper is also optimum, or near optimum, for impact accelerations, it seems likely that a damping system will be optimum overall.

NONOPTIMUM RESTRAINT SYSTEMS

Bottoming Spring Restraint

The theory of a spring restraint system is given in reference (17).

As an illustration of the use of these equations, the effect of restraint system frequency on spinal DRI is examined in Fig. 24, using a bottoming depth of 2 in. It is obvious that the minimum DRI occurs when the restraint-system stiffness is such that it just bottoms out as the occupant reaches his maximum deflection. It should be noted that any restraint resiliency is beneficial in the impact case, however, and that the measure of its effectiveness is the work required to bottom it.

This picture is reversed for a short-period acceleration, in that a restraint resiliency with zero damping can never attenuate the effects of the input acceleration. Indeed, if the restraint stiffness is less than a certain critical value, it will magnify the DRI, relative to that obtained with a rigid support.

This is illustrated in Fig. 25. Using this chart, the effect of any bottoming resiliency on the DRI can be rapidly obtained for a constant acceleration with zero rise time.

As an illustration of the use of Fig. 25, Fig. 26 presents the results of calculations for similar conditions to those used in Fig. 24; with a 10-g input acceleration, for instance, the restraint frequency must be greater than 10 cps if magnification of the DRI is to be avoided. If the restraint frequency were only 4 cps, the DRI would be nearly 50 percent greater than the value obtainable with a "rigid" system.

The critical frequency which must be exceeded, if this magnification is to be avoided, is plotted in Fig. 27. This result is independent of the frequency of the dynamic model of the human body, being a function only of the bottoming depth δ_{1B} and the input acceleration $\ddot{y}_c$. In the general case it is obviously best to forego the impact attenuation of a soft restraint system and design for a frequency in excess of the minimum value specified in Fig. 30, so that at least we avoid magnification of the short-period acceleration.

Bottoming Foam Restraint

As shown in Fig. 28, the DRI due to impact varies in rather a complex manner when a constant force restraint system is employed. The equations are so simple, however, that the use of generalized solution charts is hardly worth while.

For short-period accelerations with zero rise

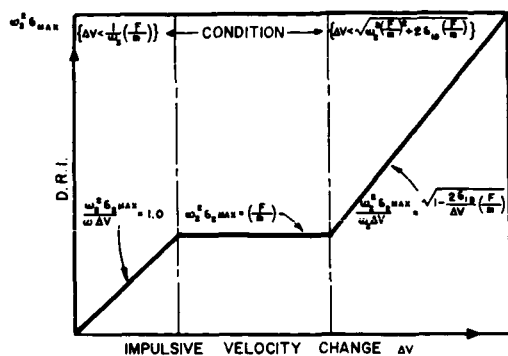


Fig. 28 Variation of dynamic response index with impulsive acceleration when a constant force ("foam") restraint system is used

time, the dynamic magnification of a foam system is plotted in Fig. 29, based on the theory given in (17).

Viscous Damping Restraint

There is no general closed-form solution to the problem of a bottoming, viscous restraint system. Optimum solutions are possible, however, since we know that bottoming must not occur for this condition.

The appropriate optimum solutions, based on the theory of (17), are plotted for impact and short-period accelerations in Figs. 30 and 31. The required damping coefficients for optimum restraint can be obtained from these curves.

RESTRAINT SYSTEM OPTIMIZATION FOR PRACTICAL ENGINEERING PROBLEMS

In this paper we have discussed restraint optimization under the handicap of two serious limitations:

- We have generally ignored the damping in the human body, for analytical simplicity;
- We have considered only two very simple acceleration inputs.

Neither of these simplifications can be justified for general engineering use, where the true optimum is required, at least to within the accuracy of the problem and our knowledge of the dynamics of the human body. Thus, most practical problems have to be solved on a computer, either analog or digital.

A suitable digital program is given in (17), but the nature of the problem is such that it is better suited to the application of analog computer techniques.

In 1961 we looked at this problem with a view to providing a simple yet accurate means of enabling the design engineer to estimate the sever-

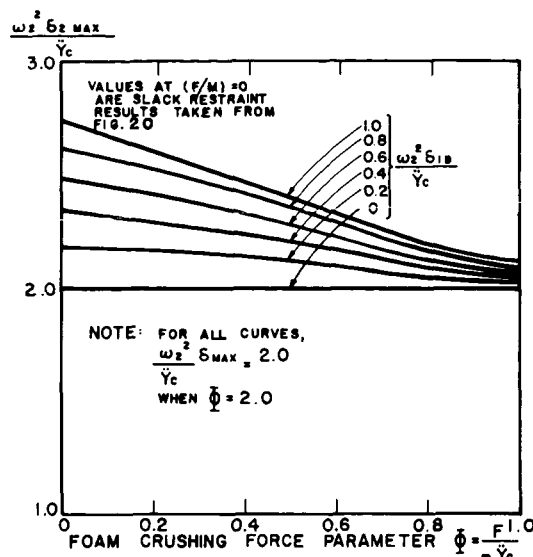


Fig. 29 Magnification of DRI by a constant force restraint system, for a short-period acceleration input

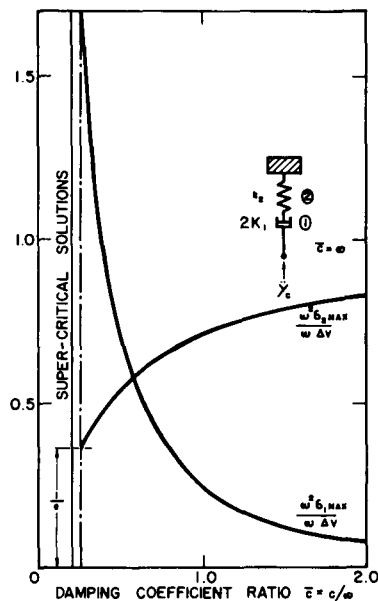


Fig. 30 Effect of restraint system damping upon DRI due to an impulsive velocity change Δv

ity of a given acceleration-time history, from a physiological point of view. It was obvious that we could not define simple rules (such as a critical "rate of onset," and so on) without recourse to a mathematical description of the system, even if the restraint system were ignored. On the other hand, engineering answers to this sort of problem are usually required with a minimum of cost and delay (which ruled out general purpose computers) and by engineers who are generally un-

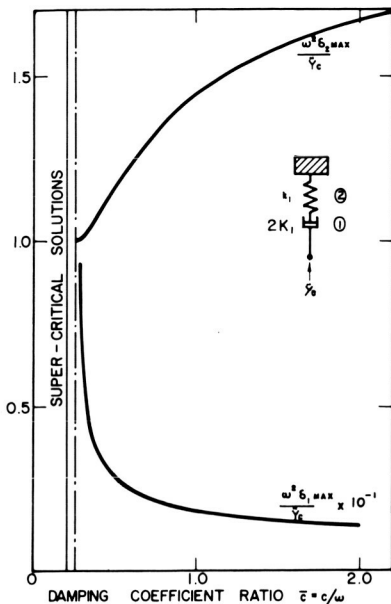


Fig. 31 Influence of a damper restraint upon DRI due to a zero rise time acceleration Y_c

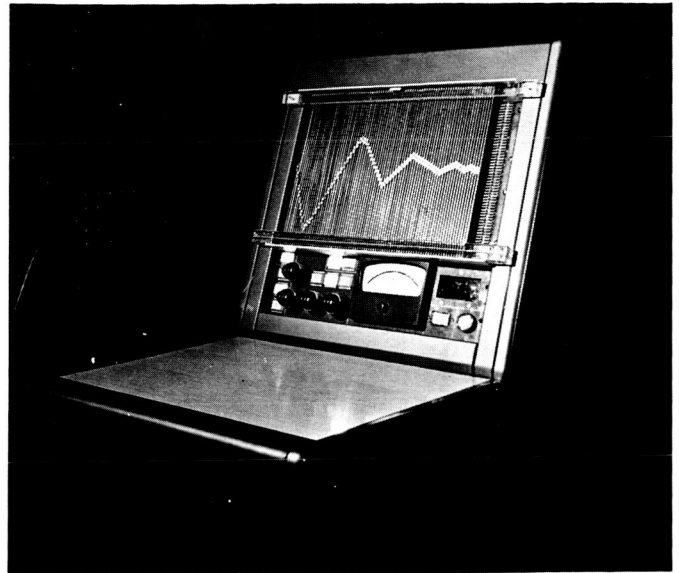


Fig. 33 Restraint analyzer built for the Aerospace Medical Research Laboratories, USAF

dynamics, while the outside would be as simple to use as a tube tester, requiring no computer or electronics experience on the part of the user!

We demonstrated a small prototype of such a computer (equipped with vacuum-tube amplifiers) at the 1961 National Academy of Sciences Symposium on "Acceleration Impact Stress" (18). Illustrated in Fig. 32, this machine was funded by Frost Engineering and gave excellent service before it was retired early this year.

In 1962 the Air Force ordered a "restraint analyzer" to the same general specifications as the prototype, but with transistorized amplifiers, and with a large number of detailed improvements. This unit is illustrated in Fig. 33, and is a much more sophisticated device, capable of giving very precise results. It is designed to be used by itself, or connected up to a large general purpose computer as part of a larger investigation when, for example, the dynamics of the human occupant influence the stability of a vehicle.

An adequate description of the restraint analyzer would obviously require a paper by itself; particularly since it has peripheral capabilities, such as checking the accuracy of accelerometer traces by integrating the input with respect to time. We merely observe here that it is capable of giving the DRI for any input, and that the dynamic characteristics of the restraint systems can be rapidly varied to find the optimum, within engineering limitations, for a given input.

Fig. 34 shows the details of the packaging in the rear of the analyzer. All logic elements are mounted on plug-in cards, so that they can be

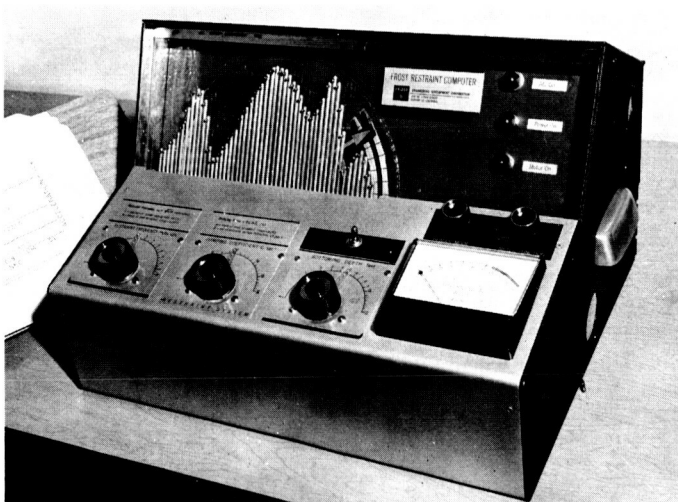


Fig. 32 The Frost restraint analyzer prototype

familiar with the language of dynamics.

We concluded that the best answer to the problem was a "black box" which effectively packaged all our mathematics, and which had buttons marked "go" and "stop" on the front! We further decided that the input acceleration-time history should be set up by plotting it on a mechanical graph, because of the fact that it could have literally any shape, and that the "answer" - the dynamic response index - should be read out on a meter. Thus our preprogrammed computer was intended to present two facets to the customer; inside it would be an electronic synthesis of all our research in body

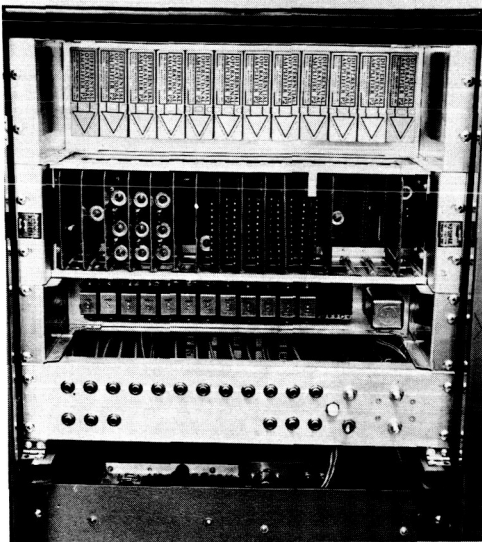


Fig. 34 Rear view of analyzer

readily removed. The dynamic models of the human body are each on separate cards so that when continuing research results in modifications, they can be replaced easily by up-dated circuits. A typical logic card is illustrated in Fig. 35, and it is intriguing to reflect that this very simple assembly of resistors and capacitors summarizes the results of several decades and many millions of dollars worth of research in body dynamics.

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- 6 "A Dynamic Model of the Human Body Subjected to Transverse Acceleration," by P.R. Payne

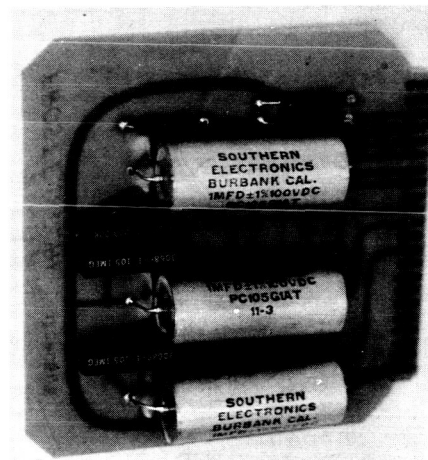


Fig. 35 Typical logic card

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APPENDIX A

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A.2 "An Analog Computer Which Determines Human Tolerance to Acceleration," P. R. Payne, Frost Report No. 101-1, Paper presented at the National Academy of Sciences Symposium on "Impact Acceleration Stress," Nov. 27-29, 1961, San Antonio, Texas.

U. S. NAVY CONTRACT NO. 167019747K

A.3 "Preliminary Study of Acceleration Protection for Crew Members of a Ship Subjected to an Underwater Explosion," P. R. Payne, Frost Report No. 115-1, March 1962.

A.4 "Human Body Dynamics under Short-term Acceleration," P. R. Payne, E. L. Stech, Frost Engineering Report No. 115-2, June 1962.

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- A.9 "Static and Dynamic Properties of Some Restraint System Webbing,"
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When Sitting Erect," P. R. Payne, Frost Report 122-103, January 1963.
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- A.19 "The Design of an Optimized Restraint And Support System for a Semi-
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April 1963.
- A.20 "An Analysis of Free Fall Accident Data Involving the Human Body in the
Transverse Direction," E. L. Stech, Frost Report 122-108, April 1963.