AN ANALYTICAL METHOD FOR DESIGNING LOW NOISE

HELICOPTER TRANSMISSIONS*

Robert B. Bossler, Jr., and Michael A. Bowes
Kaman Aerospace Corporation

Allen C. Royal
Applied Technology Laboratory
U.S. Army Research and Technology Laboratories (AVRADCOM)

SUMMARY

The internal noise levels in most civil and military helicopters are excessive. In general, the dominant source of noise is the geared power transmission system which operates at a high power level and is in proximity to the cabin. Within this system, vibratory excitations are produced as a by-product of the gear meshing process. These excitations result in vibration of the transmission housing and airframe which ultimately radiate noise into the cabin. This paper discusses the development and experimental validation of a method for analytically modeling this noise mechanism. This method can be used within the design process to predict interior noise levels and to investigate the noise reducing potential of alternative transmission design details. Examples are discussed.

INTRODUCTION

The nature and extent of the internal noise problem in Army helicopters will be illustrated first. In present day civil and military helicopters, the transmission is mounted in a position close to the cockpit. As the single most dominant source of internal noise, it is imperative that the noise producing characteristics of this component be understood. A hypothesis that was investigated and verified under Army-sponsored efforts is that noise is generated by the transmission case as a result of nonuniform transfer of torque from pinion to gear due to tooth profile errors or to the elastic deformation of gear teeth under load. This nonuniform transfer of torque produces a dynamic force at the gear mesh frequency and its multiples, resulting in a coupled torsional/lateral vibration response of the gear shaft. The lateral vibration (bending) produces displacements at the bearings which in turn cause the case to vibrate, thus producing noise. Figure 1 presents allowable and predicted noise levels for the Army Heavy-Lift Helicopter. It is clear that the noise levels will be unacceptable if a means of controlling the acoustic energy at its source is not found.

In order to meet MIL-A-8806, helicopter manufacturers have been using increasing amounts of noise attenuation blanketing with little success and with the penalty of extra weight offsetting the technology advances in light weight gearboxes. See figure 2.

This paper presents the results of a program (reference 1) that is part of 16 years of Army-sponsored programs aimed at understanding and controlling helicopter internal noise and vibration and their effects on personnel and system components.

MODELING METHOD

Problem

A helicopter transmission such as the SH-2D transmission of figure 3 is a complex dynamic system comprised of many interconnected mechanical elements. This system responds, under the influence of periodic forces produced at the various gear meshes, causing vibration of all of the individual elements and noise radiation from the transmission housing. The mechanical elements involved in this response include all of the gears/gearshafts, their support bearings, and the transmission housing. Analytical prediction of the transmission response, and consequently its vibration and noise generation characteristics, requires knowledge of the dynamics of each mechanical element, the nature and extent of coupling among these elements, and the characteristics of the gear mesh induced forcing functions.

The problem of transmission response prediction is compounded by the fact that the gear meshing forces involved occur at high frequency, typically in the hundreds and thousands of Hertz. Since the frequency for which accurate response prediction can be made is a direct function of the degree of detail in which the individual mechanical elements are modeled, very detailed elemental models are required. If the direct approach is taken and the total system is modeled as a whole, such as with finite element modeling methods, the requirement to model in great detail quickly results in an excessively large, complex model which is not amenable to ease of manipulation and use.

Approach

With the method described herein, the transmission is modeled as a fully coupled dynamic system consisting of all rotating gearshafts, the shaft support bearings, and the housing (fig. 4). Individual mechanical elements are modeled separately, in detail, then combined to produce the complete system model. The concept of component synthesis is used to simplify the dynamic modeling task and to reduce the size and complexity of the ultimate system model. Gear mesh vibratory excitations, in terms of relative deflections between mating gear teeth, are calculated independently through consideration of appropriate time-dependent tooth compliances and gear errors. These excitations are introduced in the system model at the proper gear mesh coordinates, and responses are calculated in terms of shaft and housing displacements and radiated sound power level.

Component Synthesis

The technique of component synthesis may be used to calculate the dynamic response of a linear complex structure and modification of it at a relatively small number of discrete frequencies (ref. 2). The component synthesis technique we use has two important features. The first of these relates to the reduction in degrees of freedom. The analysis of each basic component is carried out with as many degrees of freedom as is necessary for a valid analysis. When the resulting analytical model is used, however, the number of degrees of freedom may be drastically reduced and must include only:

(1) Those which interface other components
(2) Those which are to be affected by changes
(3) Those at which a force is applied or dynamic response is specifically desired

This reduction in the number of coordinates is performed only once at each frequency of interest with no loss in the validity of the analytical model, regardless of the extent of this reduction.

The other important feature relates to the ease with which changes may be studied. Structural modifications such as local mass or stiffness changes, the addition of springs or dampers between components, addition of vibration absorbers, and changes in boundary conditions may be exactly modeled at virtually no computer cost and without performing a new modal analysis for each change.

A linear structure is often represented in the frequency domain as an impedance matrix. The starting point for the analysis requires a valid impedance matrix for each substructure for each frequency of interest. The criterion for a valid impedance matrix is that it correctly predict the motion at the coordinates of interest at each frequency of interest. It can be shown that the criterion for a valid impedance matrix is that the elements of its inverse correctly represent the true response characteristics of the structure. This argument leads to a direct method of obtaining a valid reduced impedance matrix as follows:

(1) Perform a structural analysis using conventional methods to obtain a valid, full size, impedance matrix at each frequency of interest, $Z(\omega)$.

(2) Invert Z at each ω to obtain valid, full size mobility matrices $Y(\omega)$. Alternately, a modal approach or a direct integration technique may be used to obtain $Y(\omega)$.

(3) Select elements from Y at each ω corresponding to the coordinates to be retained. These elements are then formed into a new reduced mobility matrix, $Y_R(\omega)$.

(4) The reduced impedance matrix is then formed by inversion of Y_R:

$$Z_R(\omega) = Y_R^{-1}(\omega)$$

It is to be noted that Z_R is valid only at the frequency at which it has been computed. However, Z_R represents a physical system which, at the frequency ω behaves precisely as the system under study.

The reason for specifically obtaining the impedance matrix of the reduced system is that the impedance of a complex structure is obtained by simply adding the impedance matrices of the separate components at coordinates where the deflections are common. The substructures must be modeled as if they were unrestrained at the interface coordinates.

There are several considerations involved in applying this technique to practical analyses:

(1) It is not important how the reduced mobilities are computed as long as they are valid.

(2) For the method to be practical, the number of reduced coordinates must not be so large that matrix inversions become prohibitive.

(3) Local impedance changes due to addition of spring-mass systems or boundary condition changes are simply added to the reduced component impedances.

(4) When adding impedance matrices, the corresponding elements must represent deflections in the same direction.

(5) The impedance elements add when their deflections are equal; thus, when components are separated by spring-damper devices, the impedance of this device must be added to one of the substructures prior to synthesis.

The ability to accommodate structural modifications including local mass or stiffness changes, the addition of springs or dampers between components, addition of vibration absorbers or changes in boundary conditions is a valuable feature of a structural dynamics program. With the impedances of each element at frequencies of interest stored in a data bank, variation in structural parameters of the individual components can be readily made. The technique is described in reference 2.

Application of component synthesis methods resulted in substantial reduction in the size of the impedance matrix ultimately used to represent the transmission system. The SH-2 transmission system consists of four gearshafts and a planetary system. The total number of degrees of freedom used to model these shafts was 445. The case was modeled with 44 degrees of freedom, for a total of 489 individual element degrees of freedom. The synthesized

system impedance model for these combined elements would have been a 473 x 473 term matrix, considering elimination of duplicated interconnection points for the gear meshes and bearings. Reduction of many extraneous degrees of freedom from this impedance model reduced it to a 48 x 48 matrix. Although further reduction could have been achieved, this matrix size was compatible with the available computer capability, and no additional reduction was performed. It is interesting to note, however, that the most commonly performed response calculation, prediction of case surface response to a single gear clash force, could have been accomplished with a 14 x 14 impedance matrix, which represents a 34/1 reduction in size from the unreduced system impedance matrix.

Transmission Case Modeling

Modeling of the transmission housing is accomplished in four steps. First, an approximate, physically valid mass representation of the housing is intuitively derived. Next, the actual housing is shake tested with modal responses measured at all physical coordinates considered in the intuitive mass model. The measured modal data are then used to adjust the approximate mass representation, consistent with the restraints imposed by the orthogonality relationships. Finally, the adjusted or "identified" mass representation and modal data are used to derive the housing impedance, stiffness, and damping matrices. The method is described in reference 3.

In the present program, modal responses of the SH-2D main transmission housing were measured at 44 locations. The transmission case shake test set up is shown in figure 5. Prior to testing, an approximate mass matrix was established, consisting of 44 diagonal and 206 off-diagonal terms. Both real and imaginary case mobilities were measured, with the imaginary mobilities used as an approximation of the normal mode responses. Use of these data with the orthogonality relationships produced only small changes in the approximate mass matrix, typically less than five percent, indicating a high degree of physical as well as mathematical model validity.

Comparisons of measured and calculated real case mobility were made in order to verify the accuracy of the resulting case dynamic model. This represents an independent check of model validity since real data were not used in the derivation. These comparisons showed good correlation, as indicated in figure 6.

Shaft and Bearing Modeling

As for the case modeling, the objective of gearshaft modeling was to obtain valid mechanical impedance representations of each shaft, which could then be joined with the case impedance matrix using component synthesis methods, to form a dynamic response model of the total transmission system. In contrast to the case modeling approach, which is based on physical test data, gearshaft modeling was accomplished by purely analytical methods. An extension of the Holzer-Myklested technique for dynamic modeling of slender shafts was used with local non-slender shaft elements, such as the gears themselves, treated

as lumped masses and inertias (ref. 4). Both shaft flexure (bending) and torsion were considered with a typical shaft consisting of 100 degrees of freedom. All shaft support bearings were modeled as nonlinear orthogonal springs using the method of reference 5.

Excitation Mechanism

The basic mechanism for gear mesh excitation has been incorporated in an analytical calculation technique which permits the determination of local tooth deflections, including fundamental and harmonic components, based on known tooth geometry and loading conditions. This technique was developed through previous Army research efforts and is described in detail in reference 6. Within the present effort, this method has been improved to the extent of incorporating an equivalent spur gear approximation technique for representing helical and spiral bevel gearing. This improvement permits calculation of helical and spiral bevel gear mesh excitations directly from gear data available on gear design drawings. The equivalent spur gear approximation is that described in Appendix III of reference 7.

Acoustic Source Representation

In the present analysis, the transmission case is assumed to consist of a relatively small number of simple, baffled, hemispherical acoustic sources. These sources, which are distributed over the case surface, are assumed to act independently, with the sum of their acoustic outputs equal to the total transmission radiated noise. The output from each source is computed directly in terms of sound power level. Use of this source representation requires only knowledge of case surface motions, amplitude and frequency, and an estimation of the individual source sizes. The total housing radiated sound power is then calculated as the sum of the contribution from each individual source.

TRANSMISSION TESTING

Testing was performed to determine the actual vibration and noise characteristics of an operating helicopter transmission. These data were needed, for comparison with analytically calculated transmission noise and vibration characteristics, to validate the analytical methods used. The test article used in this effort was the SH-2D helicopter main transmission shown in figure 3. Gearbox identities are given in table I. This gearbox is rated at 1695 newton meters (15000 pound inches) of torque (continuous), at an output (main rotor) speed of 287 rpm. Speed reduction through the transmission is 21.3/1. This test article was subjected to simulated operational testing using a regenerative test stand. Measurements were made of all significant dynamic response characteristics including:

      o Shaft bending strain
      o Shaft torsional strain
      o Lateral shaft displacement
      o Housing surface acceleration
      o Radiated sound pressure level

esting consisted of recording data signals corresponding to each of the
dynamic parameters at discrete points over a range of transmission torque
and rpm settings. All test data were recorded on analog tape and reduced
off-line using a real time frequency analyzer.

METHOD CORRELATION

Analytical predictions of the dynamic responses of the SH-2D main transmission
were made for comparison with the measured test data. While predictions
of all relevant transmission responses were made, only case acceleration
and radiated noise proved to be of value in correlating the analytical method.
Since the accuracy of the vibration and noise radiation predictions is very
much dependent on the accuracy of the shaft response predictions, good agree-
ment between measured and predicted acceleration and noise radiation charac-
teristics provides tacit correlation of the shaft response prediction method.

Case Acceleration

Predictions of case surface acceleration at fourteen locations were made and
compared to accelerations measured at these same points. Comparisons were
made at each gear mesh related frequency of interest, including:

    o Planetary system fundamental and second harmonic

    o Spur gear mesh fundamental and second harmonic

    o Spiral bevel gear mesh fundamental

Since two transmission speeds were considered in both the analytical and test
efforts, a total of ten discrete frequency acceleration components were
available for comparison, covering the frequency range of 348 Hz to 3060 Hz.

Examples of comparisons of measured and calculated case accelerations are
shown in figures 7 and 8. The data of figure 7 show the responses to the
planetary system fundamental gear mesh frequency at 80% transmission speed
for 60% torque conditions. Figure 8 illustrates similar data for the
second harmonic of the spur gear mesh frequency at 100% rpm at 80% torque.
The high degree of correlation indicated by these data is similar to that
obtained at the other excitation frequencies considered.

Radiated Noise

Analytical predictions were made for the housing radiated sound power levels
associated with each of the gear mesh excitation frequencies considered for
both conditions of torque. Sound power levels were also calculated from the
measured sound pressure levels. Sound power is not a directly measurable
parameter but must be calculated from sound pressure. Comparisons of measured
and predicted sound power levels are shown in figure 9 at the 80% rpm test
condition. Excellent correlation is shown with the average deviation
between measured and predicted sound power levels less than 2 dB.

METHOD APPLICATION

The transmission dynamic modeling technique developed in the present program permits the rapid and economical evaluation of transmission design changes. Once the individual mechanical element models have been derived, they can be manipulated in various ways without the need for rederivation. This is accomplished through the use of a computer routine, which is an inherent part of the system modeling method and which can be used to perform the following functions:

o Add (or subtract) structural damping to any element or any part of an element

o Add vibration absorbers at any location of an element

o Add (or delete) lumped masses at any location

o Add spring/damper systems between any two elements or from an element to ground

o Change system geometry

The performance of system design studies is further promoted by the fact that changes in individual elements may be made separately. For example, if a change in shaft stiffness or mass distribution is desired, only the shaft model in question need be changed. The remaining shaft and housing models are left alone, and a new system model is synthesized using the new shaft model with these unchanged models.

An applications study was performed using the analytical method. The purpose of this study was to demonstrate the range of transmission design changes which may be investigated with the method. Design changes which were considered in this study effort are given in table II. While a considerable range of design changes was investigated, none of these individual changes were studied in sufficient depth to establish their ultimate practical value or noise reduction potential. The study results do, however, serve as an indication of the relative sensitivity of transmission response to the various design changes which were considered, at least with regard to the particular transmission studied.

Shaft Stiffness Distribution

Stiffness distributions of the input, spur/bevel, and output shafts were analytically simulated by changing the stiffness cross section of the respective shaft models over a limited segment of each shaft. In each case, shaft stiffness cross section was increased by approximately 10% over one-third of the shaft length. Only the central section of the shaft was stiffened, and no mass was added to the shaft.

The effect of increasing input shaft stiffness is shown in figure 10, in

erms of changes in radiated sound power level for each mesh excitation fre-
quency. The changes given are relative to sound power levels calculated for
the baseline transmission. As indicated, increasing the input shaft stiff-
ness caused significant changes in radiated sound power level at several mesh
frequencies and not merely at the spiral bevel gear mesh frequencies of 2448
Hz and 3060 Hz which are most directly associated with the input shaft.
Although the greatest change, an 11 dB reduction, did occur at the 100% rpm
spiral bevel gear mesh frequency of 3060 Hz, a comparable magnitude change
(in this case a 10 dB increase) is shown for the 80% rpm, spur gear mesh
second harmonic frequency, at 2396 Hz. Furthermore, no change in sound power
level was obtained at the 80% rpm spiral bevel gear mesh frequency of 2448 Hz.

The data of figure 10 provide a graphical illustration of the fact that the
analytical model considers the transmission as a coupled dynamic system with
responses determined by all the mechanical elements acting as a unit. This
fact must always be considered in applying this method, particularly when it
is used to evaluate potentially beneficial design changes. Such changes,
although usually predicated on the basis of reducing the response to only
one gear mesh excitation, will normally have an effect on all mesh induced
responses, and furthermore, these effects will be a function of transmission
speed. While a given design change may produce a reduction in response
at the principal mesh frequency of interest, this same change may very well
raise the responses at other mesh frequencies, thus curing one problem and
creating others. In addition, a reduction obtained at one transmission speed
may not prevail at another speed, even if these two speeds are reasonably
close. Because of these considerations, transmission design changes should
always be evaluated with regard to their effect on all gear mesh induced
responses and for pertinent transmission speeds. Although this approach
does require extensive evaluation of each design change, the analytical
method has been set up to perform the required analyses in an economical,
efficient manner requiring a minimum effort on the part of the analyst.

Planetary System Carrier Stiffness

Since, in many cases, helicopter transmissions exhibit their highest responses
due to planetary system excitations, an attempt was made to develop and
evaluate a method for changing these responses. These efforts concentrated
on the effects of planetary system carrier stiffness modification, and an
example of the results of these investigations is shown in figure 11.

The data of figure 11 illustrate the effects of reducing the radial stiffness
of the planet carrier by 50%. This change was considered practical because
only radial stiffness was changed with torsional stiffness held constant.
Since system torque is reacted by the carrier in torsion with little or no
static load reacted in the radial direction, the carrier radial stiffness is
not a primary static design factor and can be changed based on dynamic
requirements. As shown, reducing planet carrier stiffness causes significant
changes in the planetary system responses at 348 Hz, 435 Hz, and 696 Hz.
Further, the effects are isolated to the planetary system excitations with
little or no response change shown for the remaining gear mesh excitations.

While further analytical work is required, it is felt that the beneficial effects of this concept could be readily applied in future helicopter transmission designs.

Transmission Housing Modification

One of the major advantages of the present analytical approach is the ability to model the transmission housing. While a prototype housing is required to develop this model, changes in the housing can be simulated by purely analytical means. In this way, changes in mass and stiffness distributions and housing damping can be considered.

The addition of external damping treatments to transmission housings has often been suggested as a means to reduce housing response and radiated noise. With the present program, this approach has been evaluated analytically by simulating surface damping through increasing the housing structural damping coefficient. Three levels of damping increase were considered, with structural damping coefficient (g) of 0.05, 0.1, and 0.2. The structural damping of the housing itself was determined to be very low, with modal damping coefficients ranging from 0.0015 to 0.03. Increasing damping to the degree considered, then, represents a substantial increase, but one which can readily be obtained with commercial materials.

The effects of increased housing damping are indicated in figure 12. As shown, appreciable sound power level reductions were obtained at several gear mesh excitation frequencies, but the reductions were by no means universal. This is to be expected since the effects of damping are dependent upon the proximity of excitation frequencies and system natural response frequencies. For excitations close to natural frequencies, damping can be effective; while, if excitations are substantially removed from the natural frequencies, damping will have no effect. As shown, damping can also produce an adverse effect since added damping may increase response to excitations which are close to system antiresonant frequencies.

Given the data of figure 12 it is apparent that housing damping is a sensitive parameter which can be adjusted to reduce transmission response. Proper application of this approach, however, requires knowledge of system dynamic response characteristics, most importantly the proximity of gear mesh excitation frequencies and system resonant and antiresonant frequencies.

FUTURE USE

The noise modeling method can be applied in various ways depending upon the development status of the subject transmission. During preliminary design, rough estimates of transmission noise can be made using a simplified noise model which has been derived from the more complex method (fig. 13). The simplified noise prediction method is given in equation form in reference 8. The accuracy of the simplified noise prediction method has been subjected to limited evaluation through comparison of predicted and measured UH-1 internal noise levels, obtained from reference 6. As illustrated in figure 14 the

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calculated UH-1 internal noise spectrum agrees well with the measured data.
The noise prediction method is presented in equation form in the appendix.
When detail design data become available, the complete system modeling method
can be applied using a housing model based on finite element methods. At this
stage, significant efficiencies can be achieved using a substructure approach
with component synthesis. Finally, during hardware development, improved
elemental models may be obtained through the use of mobility test data.

CONCLUSION AND RECOMMENDATIONS

The analytical methods developed in this study represent a significant advance-
ment in the state of the art of helicopter internal noise prediction. These
methods are limited, however, to the prediction of the airborne component
of transmission noise, although the approach used is compatible with the
incorporation of a structure borne noise prediction capability. Extension
of the methodology to include structure borne noise prediction capability is
considered both feasible and appropriate.

APPENDIX

HELICOPTER TRANSMISSION NOISE PREDICTION METHOD

MICHAEL A. BOWES

It is often desirable to have reasonable, though approximate, estimates of transmission noise characteristics. To answer this need, a simplified transmission noise prediction technique has been developed using parametric trending data generated with the SH-2D transmission analytical model. The validity of this simplified method is predicated on the assumption that the SH-2D transmission has dynamic response and noise radiation characteristics which are representative of helicopter transmissions. This assumption is believed to be appropriate, since the SH-2D transmission is similar in design to most existing helicopter transmissions and its operating torque and rpm conditions are near median values for current and planned vehicles.

The simplified transmission noise prediction method is based on a simple parametric relationship between the physical variables of a given gear mesh and the sound power level of the discrete frequency component due to that mesh. The general form of this relationship is:

$$PWL_G = A\, Log_{10}(\tau) + B\, Log_{10}(f) + C + D \qquad (1)$$

where: PWL_G = sound power level - dB re. 10^{-12} watts

A = a constant indicative of the relationship between torque and sound power level

τ = transmitted torque (in-lb)

B = a constant indicative of the relationship between gear clash frequency and sound power level

f = gear clash frequency - Hz

C = a constant indicative of the type of gear mesh

D = a constant indicative of the gear clash harmonic number

A parametric study was performed, using the available SH-2D transmission analytical model, considering three types of gear meshes, all of which were represented in the SH-2D model. Gear mesh types considered were: spur gear, spiral bevel gear and planetary system. Based on this study, three equations of the form of Equation (1) were derived for the three gear mesh types considered. These are:

$$PWL_{SG} = 20\, Log(\tau) + 37.8\, Log(f) - 91 + D_{SG}$$

$$PWL_{SBG} = 20\, Log(\tau) + 37.8\, Log(f) - 100 + D_{SBG}$$

$$PWL_{PS} = 12.8\, Log(\tau) + 37.8\, Log(f) - 59 + D_{PS}$$

$$(2)$$

where: PWL_{SG} = sound power level of spur gear mesh

PWL_{SBG} = sound power level of spiral bevel gear mesh

PWL_{PS} = sound power level of planet system

The constants D_{SG}, D_{SBG}, and D_{PS} in Equation (2) are indicative of the relationship between sound power level and gear mesh harmonic number, which was found to be specific to a particular gear mesh type. Values of these constants were defined to be equal to zero for the gear clash fundamental frequency. Finite values were established for these constants for the second and third harmonics of gear clash frequency, and these are given in Table 1.

Table 1. Values of the Constants D_{SG}, D_{SBG}, and D_{PS} in Equation (2)

Gear Clash Type	Harmonic No.	
	2	3
Spur Gear	-5	-22
Spiral Bevel Gear	+7	+6
Planet System	-10.5	-23

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TABLE I. SH-2 MAIN GEARBOX IDENTITIES

Part	No. of Teeth	Speed - rpm	Excitation Frequency - Hertz
Input Shaft	-	6120	-
Spiral Bevel Pinion	30	6120	3060
Spiral Bevel Gear	47	3906	3060
Spur Gear Pinion	23	3906	1497
Spur Gear	87	1033	1497
Sun Gear	35	1033	-
Planet Gear (6)	28	-	435
Ring Gear	91	-	-
Planet Carrier	-	287	-
Output Shaft	-	287	-

TABLE II. SUMMARY OF DESIGN CHANGES ANALYZED

REDUCED BEARING STIFFNESS
    ALL SHAFTS
        INPUT SHAFT ONLY

INCREASED SHAFT STIFFNESS
    INPUT SHAFT
    OUTPUT SHAFT
    SPUR/BEVEL SHAFT

INCREASED CASE DAMPING

BEARING RELOCATION

SUN GEAR ISOLATION

PLANET CARRIER ISOLATION

INCREASED SHAFT MASS
    INPUT SHAFT
    SPUR/BEVEL SHAFT
    SPUR/SUN SHAFT

INCREASED CASE MASS
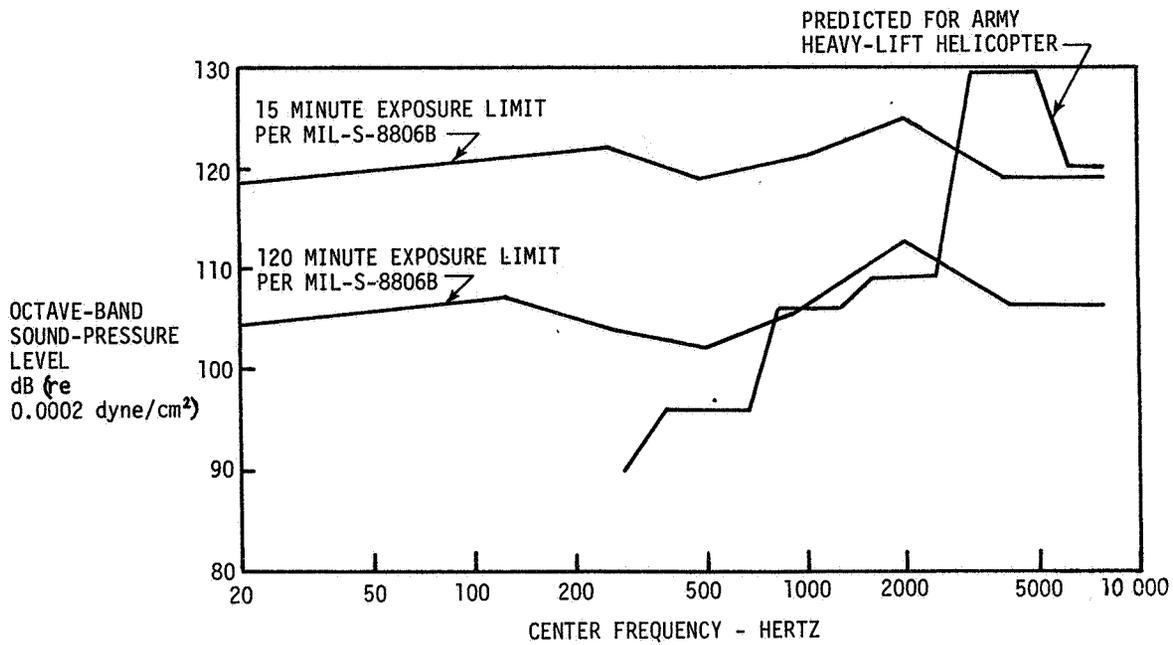
Figure 1.- Allowable and predicted sound pressure levels at personnel locations for crew members wearing SPH-4 headgear.
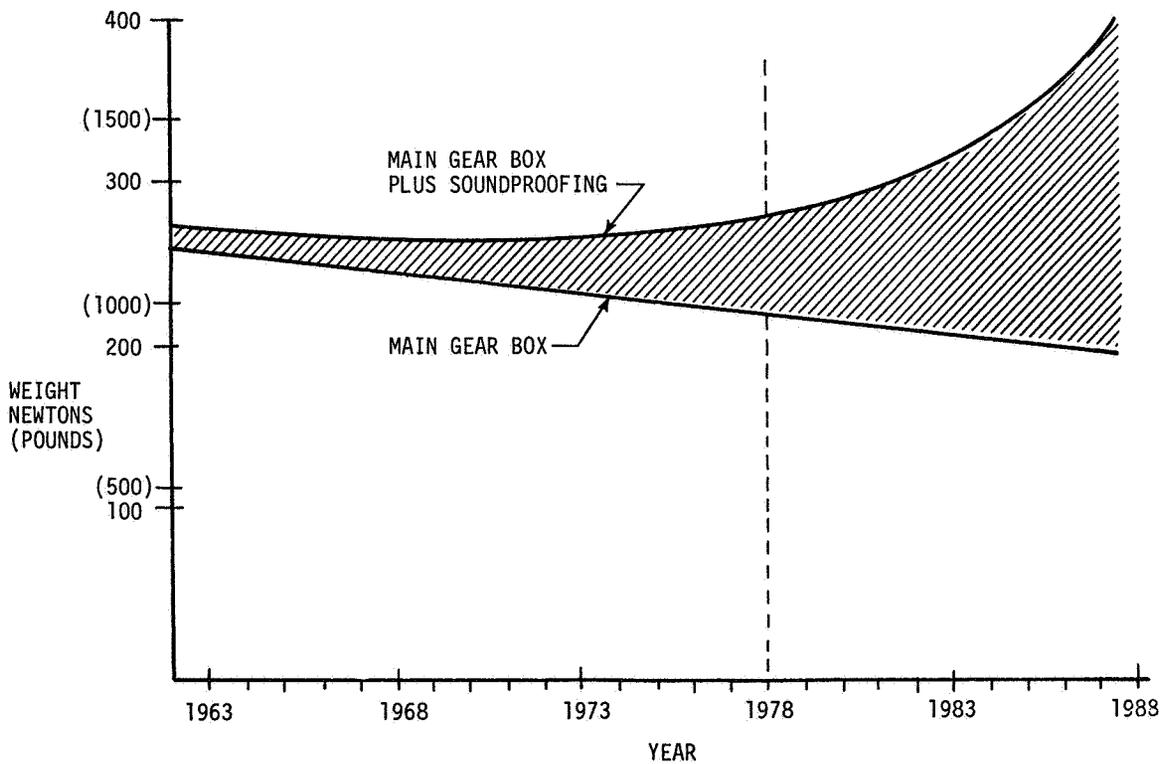
Figure 2.- Weight of gearbox and soundproofing as a function of time.

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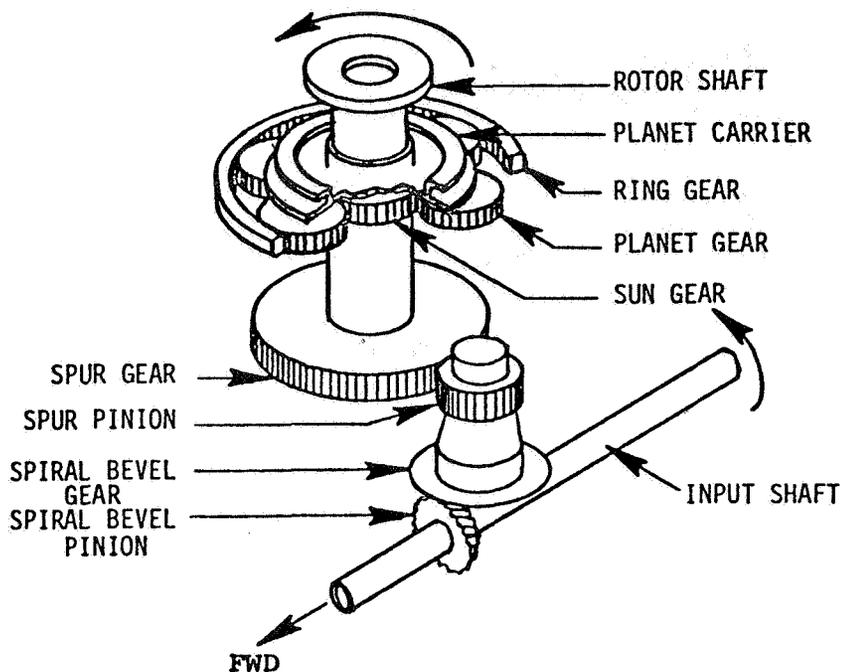
Figure 3.- SH-2 main transmission.

HOUSING MODEL
    TEST DATA OR
    FINITE ELEMENT

SHAFT MODELS
    HOLZER/MYKLESTED
    BENDING/TORSION

MESH EXCITATIONS
    MTI (GEARO)
    SPIRAL BEVEL AND
    HELICAL APPROX.

BEARING MODELS
    B. JONES
    ORTHOGONAL SPRINGS

SYSTEM MODEL
    FULLY COUPLED
    REDUCED SIZE

SHAFT MOTIONS

HOUSING MOTIONS ⟶ SOUND POWER LEVEL

INTERFACE MOTIONS
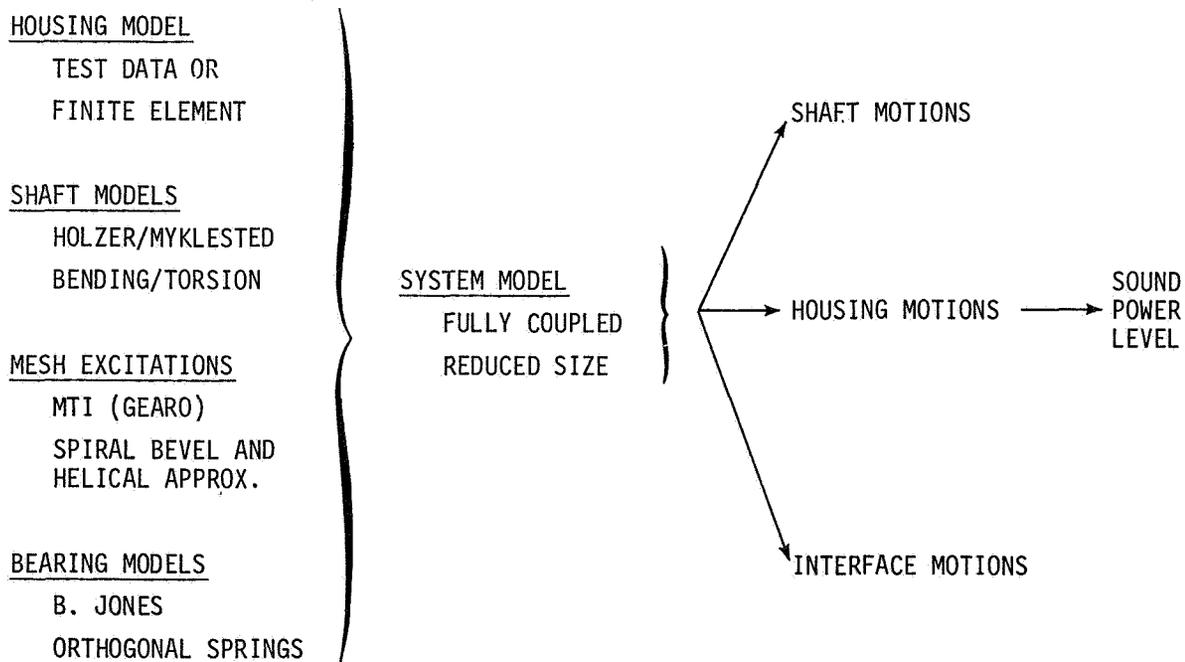
Figure 4.- Transmission noise modeling approach.
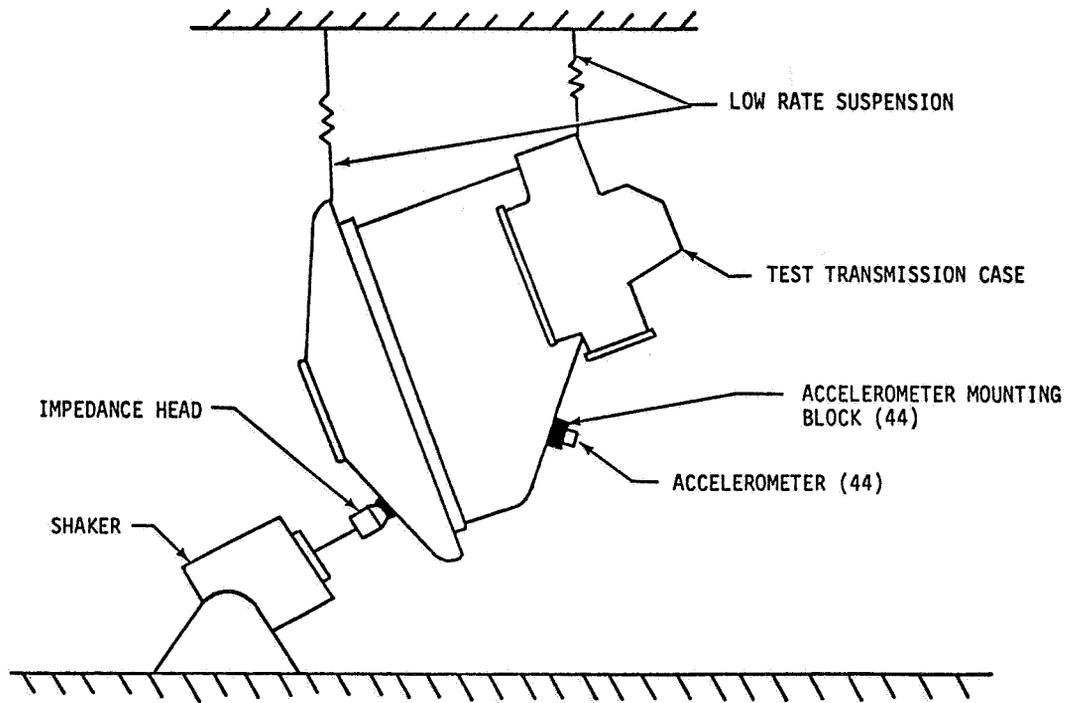
Figure 5.- Transmission case shake-test setup.

FREQUENCY - HERTZ
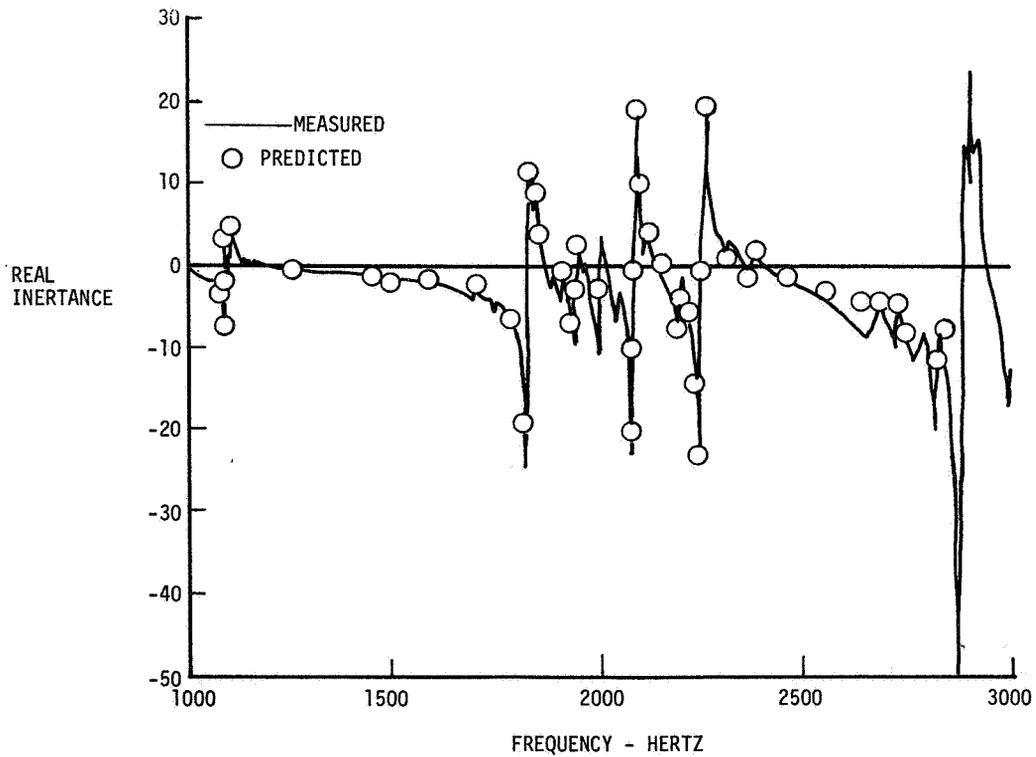
Figure 6.- Measured and predicted transmission case inertance.
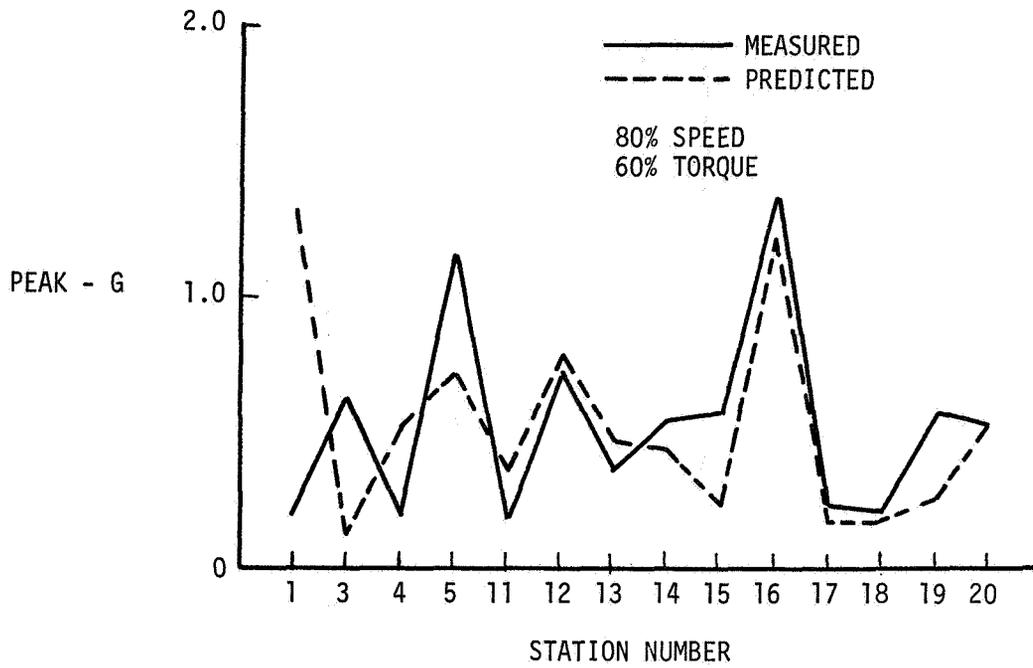
Figure 7.- Measured and predicted case acceleration for planet system excitation (348 Hz).
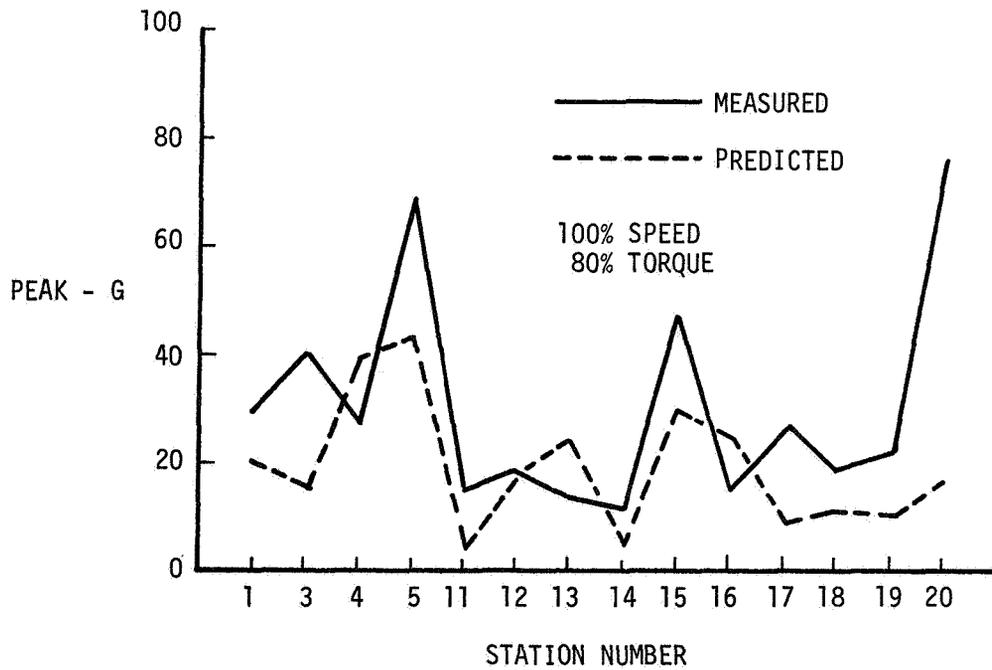
Figure 8.- Measured and predicted case acceleration for spur gear excitation (2994 Hz).
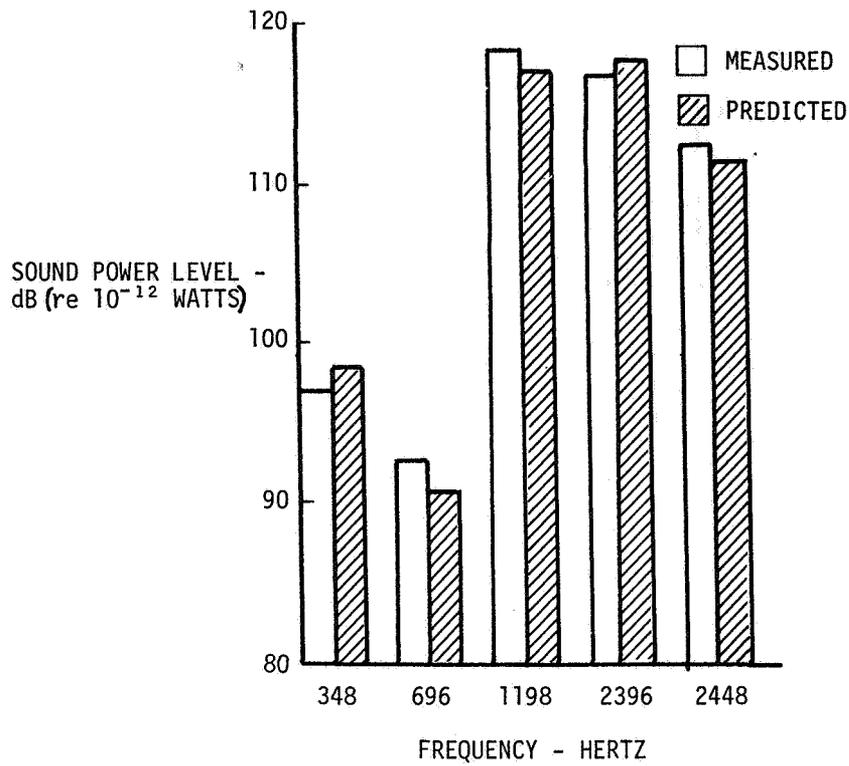
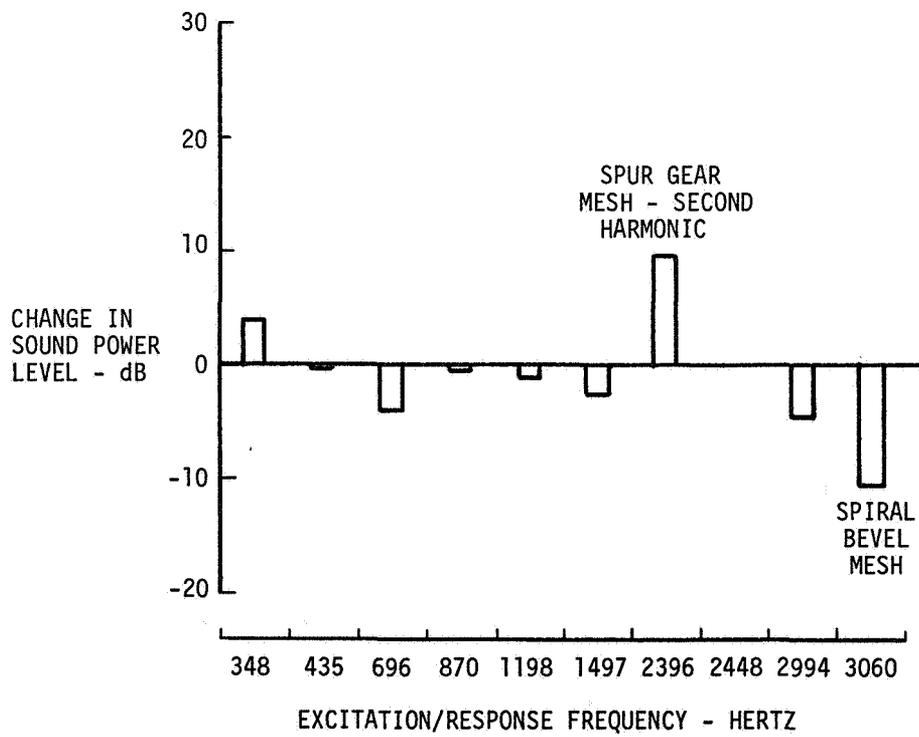Figure 9.- Measured and predicted sound power levels. 80 percent RPM, 80 percent torque.
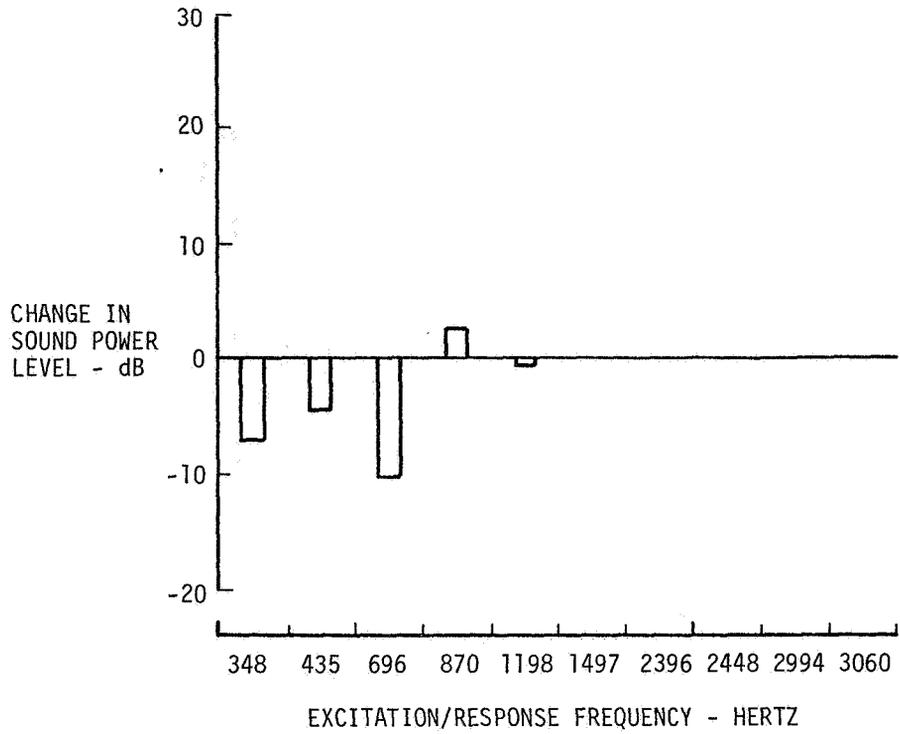

Figure 10.- Effect of increased input shaft stiffness.

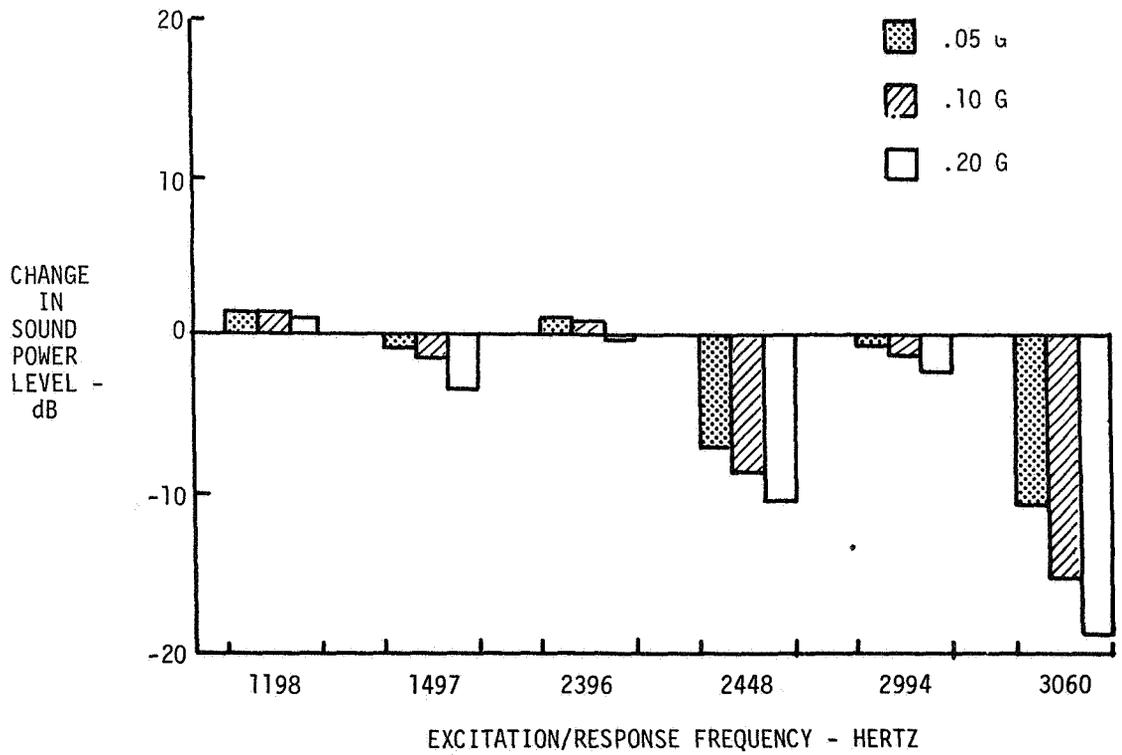Figure 11.- Effect of reduced planet carrier radial stiffness.
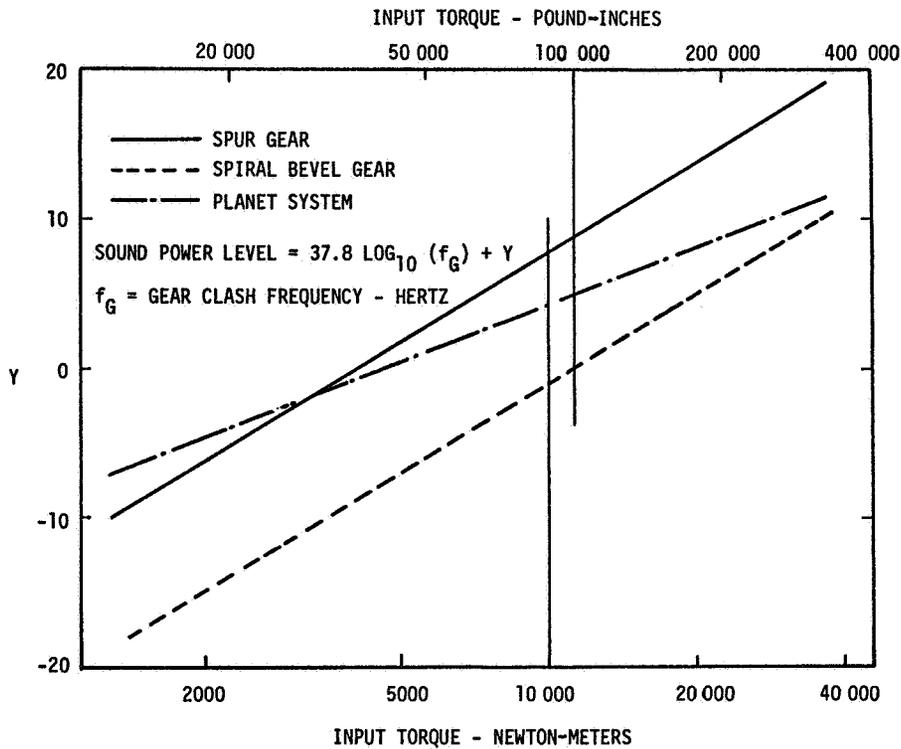

Figure 12.- Effect of case damping.

INPUT TORQUE - POUND-INCHES

Figure 13.- Simplified transmission noise prediction model for
fundamental mesh frequency.

THIRD-OCTAVE
BAND LEVEL IN
dB (re 0.0002
MICROBAR)

- - - - - CALCULATED SPECTRUM

MEASURED DATA, SPREAD OF
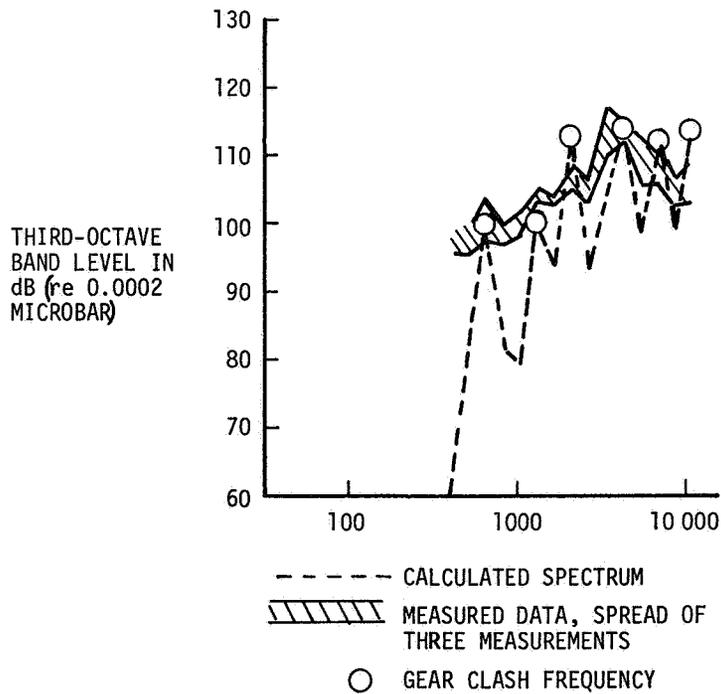THREE MEASUREMENTS

○ GEAR CLASH FREQUENCY

Figure 14.- Measured and calculated UH-1 (B-205) internal noise
using simplified transmission noise prediction model.