yond the usual warning signs, additional warning signs should inform drivers about the cause of the problem.

6. Traffic control plans should be prepared by knowledgeable personnel. If geometric design standards or high volumes appear to indicate problems, past records of accident history should be studied for possible clues. The sites studied in this research did not have very bad accident problems, but all sites were on the rural Interstate system, which usually has low accident rates. On lower standard roads or in urban areas, the problems were more serious.

These suggestions are based on our interpretation of a relatively small set of accident data and on comments received from construction engineers during the data collection phase. More research is clearly needed.

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Ride Quality Criteria of Multifactor Environments

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A comprehensive ride quality model that accounts for the effects of multifrequency and multiaxis vibration inputs as well as the interactive effects of noise and vibration upon passenger discomfort is under development. The model development has been based upon extensive experimental studies utilizing a realistic multi-degree-of-freedom laboratory simulator located at the Langley Research Center, Hampton, Virginia. This paper briefly describes the basic elements of the ride quality model; presents summary data relating to human discomfort response to vertical, lateral, and roll vibrations; and concludes with a description of the results of an initial study of human response to combined noise and vibration stimuli. Results of studies involving vibration stimuli alone are presented in terms of sets of equal discomfort curves for each of the previously mentioned axes of vibration. In addition, a set of noise-vibration criteria curves are included based upon the concept of additivity of the discomfort components due to noise and to vibration. This assumption is shown to be valid for the restricted range of stimuli used in this study.

Passenger comfort within existing as well as future transportation systems is strongly dependent on environmental factors such as vibration, noise, temperature, and pressure. These factors can act to degrade vehicle ride quality and hence passenger acceptance of a particular transportation system. For example, the introduction of advanced transportation systems such as short-takeoff and landing (STOL) aircraft, civil helicopters, and high-speed surface vehicles is expected to be accompanied by more severe levels of vibration and noise than most currently operating systems. The increased levels of these environmental factors are likely to produce additional decrements in ride quality and, therefore, further reductions in passenger acceptance. Consequently, a definite need exists for a valid ride quality model that can be readily used by system designers to estimate the trade-offs between passenger acceptance (or comfort) and the degree of complexity of vehicle ride control systems required to achieve a specified level of comfort. To be useful, such a model should be applicable to any transportation system and should account for both multiaxis and multifrequency vibratory inputs as well as the interactive effects, if any, of noise and vibration. The uses of such a model would be to predict passenger acceptance of any noise-vibration environment, to determine the specific components of the noise or vibration environment or both that most affect passenger discomfort, and to serve as a diagnostic tool in providing a "fix" to a ride quality problem by knowing how much reduction in noise or vibration characteristics or both is required to achieve acceptability.

Numerous studies in the area of human subjective response to whole-body vertical vibration have been conducted (see 1-5). An excellent summary and critical review of a larger number of studies conducted prior to 1970 is given in Hanes (6). Unfortunately, as Hanes points out, the results of these studies show very little agreement with one another. Many of the criteria curves or recommended levels of vibration proposed by various investigators differ by as much as one or two orders of magnitude. Many reasons have been offered for such disparities in results, and these reasons are summarized as being attributable to poor experimental design, the multifactor nature of actual ride environments, use of inadequate adjective rating scales, subject populations used, and a lack of a fundamental understanding of the empirical laws governing human discomfort response to vibration (7).

A systematic and comprehensive effort to develop a general predictive model of passenger discomfort to combined noise and vibration that is free of the above limitations has been under way at the Langley Research Center, Hampton, Virginia. The National Aeronautics and Space Administration (NASA) model (7) recognizes that passenger discomfort results from the interrelationships of many factors of which vibration and noise are the most important. The NASA model approach involves the experimental determination of the basic psychophysical relationships governing human discomfort response to complex vibration and noise stimuli acting singly or in combination. The NASA concept involves the derivation of a scale of vibration discomfort and then inclusion of the effects of noise and other variables in the form of scale correction factors. Various investigations resulting from NASA efforts have been reported (see 8-17). The NASA model studies, however, have not yet accounted for the effect of noise and its possible interactions with the various parameters of vibration such as vibration frequency and amplitude. This paper summarizes the results of experimental studies of passenger response to single axis vertical, lateral, and roll vibrations and presents the results of a study directed toward the extension of the NASA ride quality model to the more general case of predicting passenger discomfort response to a combined noise and vibration environment.

APPARATUS

The experimental apparatus used in these studies is a unique laboratory simulator called the Passenger Ride Quality Apparatus (PRQA). The PRQA is located at the Langley Research Center and has been described in detail in Clevenson and Leatherwood (18). It is a three-degree-of-freedom simulator configured to resemble the interior of a modern jet transport (see Figure 1). Up to six subjects can be simultaneously exposed to field- or laboratory-generated noise and vibration inputs covering the range of frequencies and amplitudes known to affect passenger comfort. Approximately 2000 subjects have been used on the PRQA as part of the NASA Ride Quality Model Development Program.

METHOD

The results presented in this paper are derived from several different studies conducted at the Langley Research Center as part of a program for the development of a ride quality model. The methodology used in the various studies will be discussed in general terms in this paper, and the reader can refer to the designated references in order to obtain details of a particular experimental method.

Figure 1. A view of Passenger Ride Quality Apparatus (PRQA), a laboratory simulator located at Langley Research Center, Hampton, Virginia.

The methodological approach used to develop a family of equal discomfort contours for vertical vibration consisted of a sequence of three studies, employing a method of constant stimuli in the first study and magnitude estimation procedures in the remaining two studies (13). The sequence of studies for this axis used a total of 186 paid subjects (118 female, 68 male). The first study, using the method of constant stimuli, determined the acceleration level at each frequency (1 to 30 Hz) that produced identical values of discomfort. The second study, using magnitude estimates of discomfort as a function of acceleration level at each frequency, generated a family of equal discomfort curves. The third study, also employing magnitude estimation procedures, determined how different frequency components within the vertical axis summate or mask or both to produce the total discomfort response to a ride. The end result of these three studies was the development of a complete model for vertical vibration that accounts for within-axis frequency masking.

Equal discomfort curves and frequency masking for lateral and roll vibration were obtained in a manner similar to that described here for vertical vibration. The only difference in methodology involved the use of the method of constant stimuli to determine equality between vertical axis vibrations and both lateral and roll vibrations. This was necessary in order that the magnitude estimates of discomfort measured within any of the three axes would have similar meaning relative to the total discomfort scale. In other words, identical discomfort ratings of vibrations in each of the three axes would correspond to identical values of subjective discomfort. A total of 84 subjects were used in the lateral vibration study, and 96 subjects were used in the roll vibration study.

The study of the effects of combined noise and vibration used a total of 48 subjects and a magnitude estimation procedure to obtain subjective evaluations of discomfort. Each subject (six subjects concurrently) was required to provide magnitude estimations of successive comparison-ride segments relative to a standard-ride segment assigned the numerical value of 100. The comparison-ride segments consisted of vertical vibrations, either sinusoidal or random, and octave band random noise. The sinusoidal vibrations were at a frequency of 5 Hz, and the random vibrations had a center frequency of 5 Hz and a 5-Hz bandwidth. Root mean square (rms) acceleration levels varied from 0.02 to 0.130 g for both types of vibration stimuli. The octave band random noise was centered at 500 or 2000 Hz and was presented at ambient [$\approx$65 dB(A)], 75, 85, and 95 dB(A). The standard-ride segment was always sinusoidal in nature and was applied at a level of 0.074 g_{rms} in the ambient noise condition. The comparison-ride segments were factorial combinations of the noise and vibrations described above. The experimental design is shown in Figure 2.

RESULTS AND DISCUSSION

Vertical Axis Constant Discomfort Curves

A family of constant discomfort curves for vertical sinusoidal vibration was developed using the methods described in the preceding section. These curves are presented in Figure 3, which shows the peak and root mean square acceleration levels at each frequency (from 1 to 30 Hz) required to produce constant specified levels of discomfort. For example, the curve labeled "1" is defined as the DISC = 1 curve and corresponds to the threshold of discomfort. The acceleration levels of the curve can be considered as defining the boundary at which the subjective evaluations of ride quality change from one of comfort to one of discomfort. The curves noted by the numbers 2, 3, 4, and so on (DISC = 2, 3, 4, and so on) bear a direct ratio relationship to the threshold curve. That is, the DISC = 2 curve provides twice the discomfort of the DISC = 1 curve; the DISC = 4 curve corresponds to twice the discomfort of the DISC = 2 curve and four times that of the DISC = 1 curve. Thus, discomfort is a continuous function of vibration acceleration level at each frequency with the result that various levels of discomfort within each frequency can be readily discriminated. This is particularly pertinent to the development of ride quality criteria, since quantification of the basic psychophysical relationship between perceived discomfort and level of vibration stimuli will allow system designers to reliably estimate the trade-offs between passenger comfort and the ride environment for transportation vehicles. The equal discomfort curves in Figure 3 also display sharp dips at a frequency of 5 to 6 Hz, indicating these frequencies to be the most crucial with respect to ride quality judgments. This is to be expected, since these frequencies correspond to the major whole-body resonances of the human body. Absent from these curves are dips in the 10- to 15-Hz frequency region where local head and neck resonances of the human body have been shown to occur. The reason such effects were not observed in this series of studies was the use of actual cushioned aircraft seats (tourist class) that effectively reduced transmission of floor vibrations to the seated subjects at frequencies greater than 9 Hz. A point of interest regarding Figure 3 is the fact that the increment in floor acceleration required to produce a specified increase in discomfort (doubling of discomfort, for example) at low frequencies is much less than that required at the higher frequencies. This is attributable in part to reduced human sensitivity to vibration at the higher frequencies and also to the effect of the seat transfer function. In any event, vibration frequencies greater than approximately 15 Hz are felt to be relatively unimportant to ride quality for transportation vehicles operating within realistic ride environments. The downturn of the DISC = 1 and DISC = 2 curves of Figure 3 at frequencies greater than 25 Hz is probably due to increased cabin noise levels resulting from harmonic exitation of the cabin structure at the higher frequencies. Such continuous excitation would not normally be present in an actual operational transportation system.

Vertical Frequency Masking and Summation

The equal discomfort curves discussed above were developed by exposing passenger subjects to a series of

Figure 2. Experimental design of combined noise-vibration study.

VIBRATION		NOISE LEVEL, dB(A)						
		AMBIENT	75		85		95	
		Octave Center Frequency						
Type	g_{rms}	--- --	500	2K	500	2K	500	2K
	0.020							
	0.042							
SINUSOIDAL	0.064							
5 Hz	0.085							
	0.106							
	0.130							
	0.020							
	0.042							
RANDOM	0.064							
5 Hz	0.085							
	0.106							
	0.130							

Figure 3. Vertical equal discomfort curves.

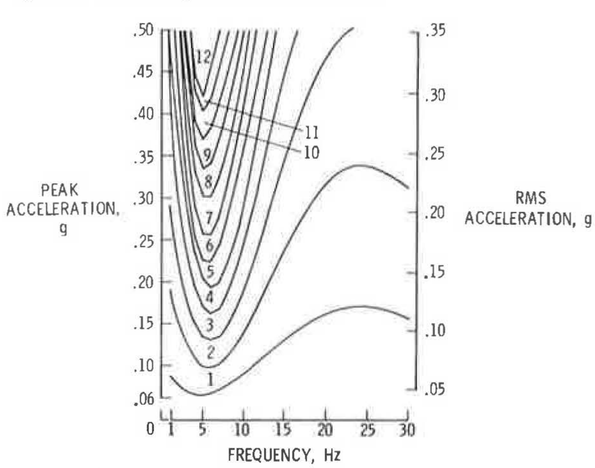


Figure 4. Vertical masking factor as a function of rms vertical acceleration and vibration bandwidth.

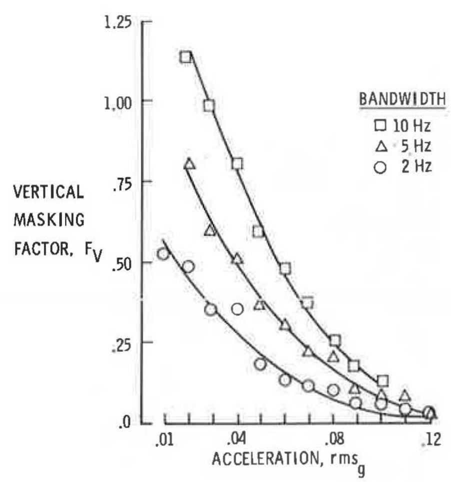
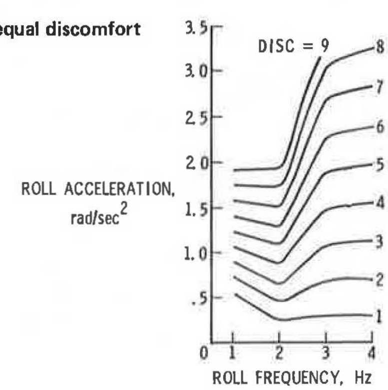


Figure 5. Roll equal discomfort curves.



discrete vibration stimuli, each of which was characterized by specific frequency and acceleration components. Thus, these curves are strictly applicable to the case of single frequency, vertical sinusoidal excitation. The logical question arises as to how to determine passenger subjective discomfort response in the presence of multiple frequency excitation. The results of an experimental study directed toward answering this question for vertical vibrations are presented in Figure 4. This figure presents the vertical frequency masking factor, F_v, as a function of root mean square floor acceleration and the bandwidth of the input vibration. To gain a proper understanding and interpretation of the data presented in Figure 4, it is useful to first consider the basis upon which the masking factor was determined. It was assumed that the masking phenomenon for vertical vibration can be represented by the following equation:

$$DISC_{TOTAL} = DISC_{MAX} + F_V (\Sigma DISC - DISC_{MAX}) \quad (1)$$

Equation 1 is similar to that developed by Stevens (19) to calculate the loudness of complex noise. The terms in Equation 1 are defined as follows:

$DISC_{TOTAL}$ = Total perceived discomfort of an arbitrary ride spectrum composed of 30 contiguous frequency bands, each of which has a 1 Hz bandwidth.

$DISC_{MAX}$ = Contribution to the total perceived discomfort by the 1 Hz frequency band that produces maximum discomfort—determined by use of equal discomfort data.

$\Sigma DISC$ = Summation of the contributions to the total perceived discomfort produced by each of the 1 Hz frequency bands.

F_v = Vertical frequency masking factor.

The masking factor, F_v, is a measure of the degree of additivity (or nonadditivity) of individual frequency components in relation to the total discomfort response. If F_v approaches unity, then the total discomfort response ($DISC_{TOTAL}$) is given by the summation of the individual discomfort components within each frequency band (additive effect); consequently, no frequency masking is present. If F_v approaches 0, then the total discomfort response is given by the particular 1-Hz bandwidth of vibration that produces the maximum discomfort component. In this case, the contributions to discomfort made by the remaining frequency bands is completely masked. If F_v becomes negative, then an antagonistic interaction between the separate frequency components is present with the result that the rated discomfort will be less than the discomfort produced by the dominant frequency component. In Figure 4, the masking factor is seen to be a function of the bandwidth of vibration and the rms acceleration level within a bandwidth. No systematic trends for F occurred as a function of center frequency for any of the bandwidths. Recalling that small values of vertical masking factor, F_v, correspond to a high degree of masking whereas large values of F_v indicate little masking, it is seen that within a particular bandwidth the amount of frequency masking increases substantially with increasing levels of rms acceleration. This means that the separate frequency components of discomfort become less additive as rms acceleration increases. Furthermore, the amount of frequency masking decreases for increasing bandwidth of vibration. Both of these results are consistent with what would be expected to occur based upon purely physical considerations. For example, within a specified bandwidth the increase in masking with increasing rms acceleration level results from the fact that the contribution to total discomfort of the dominant frequency component (i.e., $DISC_{MAX}$) becomes disproportionately larger than the contribution to total discomfort of the remaining component frequencies.

The main point to be made with regard to these results is that the presence (or lack) of frequency masking for vertical vibrations is a function of both bandwidth and overall rms amplitude of the vibration. Hence, development of ride quality criteria should account for these factors. In a practical sense, a knowledge of frequency masking would allow for more accurate diagnosis of the source of ride quality problems and would enable system designers to more effectively select and apply ride control techniques for the improvement of vehicle ride quality.

Roll Equal Discomfort Curves

Constant discomfort curves for roll vibration are presented in Figure 5. The range of frequency (1 to 4 Hz) and roll acceleration level (0 to 2.0 rad/s^2) covered in Figure 5 are considered representative of those that may be encountered in realistic transportation systems. Each curve of the figure indicates the level of roll acceleration that is required at each sinusoidal frequency to produce constant levels of discomfort. These curves show that the lower frequencies result in the greatest discomfort. At these lower frequencies (1 to 2 Hz), large relative motions of the body, head, and trunk occur, and this probably accounts for the increased discomfort. It should also be noted that, for the range of roll acceleration studied, the maximum level of discomfort obtained was 9 discomfort units (DISC = 9) as compared to 12 discomfort units (DISC = 12) for vertical vibration.

Roll Frequency Masking and Summation

Masking information for the roll axis of vibration was obtained from tests in which passenger subjects were exposed to random roll vibrations of various amplitudes within a frequency bandwidth ranging from 0.5 to 4.5 Hz. The resulting masking data are presented in Figure 6 in terms of the roll frequency masking factor, F_R, as a function of root mean square roll acceleration level. These data indicate that values of the roll masking factor are much smaller than those obtained for the vertical masking condition and, furthermore, tend to become increasingly negative as roll acceleration level increases. At the smaller values of roll acceleration (less than 0.40 rad/s^2) used, the roll masking factor is approximately zero, indicating that the subjective response is dominated by the particular frequency component that produces the most discomfort. In other words, the separate frequency components of discomfort do not add, and subjective discomfort can be predicted from knowledge of the maximum discomfort component alone. Negative values of the roll masking factor, however, indicate that increases of roll acceleration level (greater than 0.40 rad/s^2) result in an antagonistic-subtractive interaction between the discomfort produced by separate frequency components. In a sense, this result may be thought of as a case of reverse or reciprocal masking between discomfort components. In this case, the subjective discomfort reported by subjects is less than the discomfort that would be produced by the dominant frequency component acting singly. Thus, predictions of subjective response to roll vibrations that do not account for this antagonistic interaction would tend to overestimate subjective discomfort at the higher roll acceleration levels.

Lateral Equal Discomfort Curves

The equal discomfort curves for lateral (side-to-side) vibrations are presented in Figure 7. These curves cover a frequency range of 1 to 10 Hz, because lateral vibration frequencies in excess of 10 Hz occur relatively infrequently in transportation vehicles and, hence, are considered to be of minor importance to ride quality. These curves indicate that human subjects are most sensitive to lateral vibration occurring at a frequency of 2 Hz with the sensitivity decreasing for frequencies both above and below this value. For the larger levels of discomfort, the sharp upward slope of the curves supports the previous assumption that the higher lateral frequencies are relatively unimportant to ride quality.

Lateral Frequency Masking and Summation

The masking data for the lateral axis of vibration are shown in Figure 8 in terms of the lateral frequency masking factor as a function of root mean square lateral acceleration for several bandwidths of lateral vibration. The data presented in Figure 8 were averaged over lateral vibration center frequency since center frequency had only a minimal effect upon discomfort responses. It is readily apparent that the frequency masking phenomena for the lateral axis differ from the vertical masking results in several important respects. One of the most obvious differences is the fact that lateral frequency masking factors are considerably smaller than those obtained for vertical masking and, most importantly, the 2-Hz bandwidth condition gives masking factors that are negative over the entire range of lateral accelerations investigated. This implies that the effects of individual frequency components within a 2-Hz bandwidth are subtractive, i.e., the total discomfort response to the frequencies contained within the band is less than the discomfort produced by the dominant individual frequency component.

The data for the 5- and 10-Hz bandwidth conditions indicate that lateral accelerations greater than approximately 0.085 g_{rms} result in lateral masking factors that approach zero. This implies that frequency components within a random vibration frequency band do not add, and, therefore, the discomfort response is attributable to the dominant frequency component alone. For lateral accelerations of less than 0.085 g_{rms}, the discomfort responses due to individual frequency components become slightly additive as indicated by the small positive values for the lateral masking factor (recall that F = 1 corresponds to perfect additivity). The overall conclusion to be made from the results of the lateral frequency masking study is that the contributions to discomfort of the individual frequencies within the larger bandwidths (5 and 10 Hz) are, at most, only slightly additive. This means that for these bandwidths the major contributor to discomfort will be the particular frequency that, if acting alone, would produce the most discomfort. For the 2-Hz bandwidth vibrations, however, the effects of the individual frequency components upon total discomfort response are subtractive. The exact mechanism accounting for this antagonistic-subtractive effect of reverse-reciprocal masking is not clear at the present time.

Combined Noise and Vibration

The results of the study of the effects of combined noise and vibration on passenger discomfort were analyzed by computing an analysis of variance based on the de-

Figure 6. Roll masking factor as a function of rms roll acceleration.

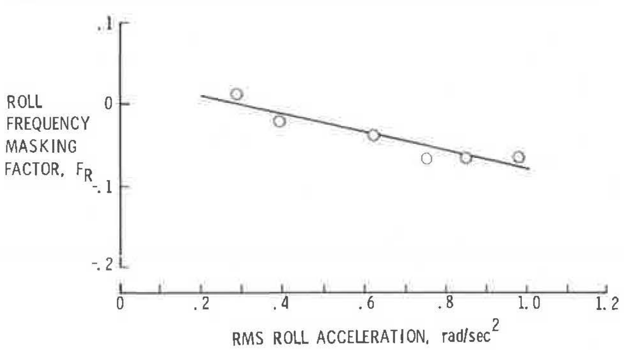


Figure 7. Lateral equal discomfort curves.

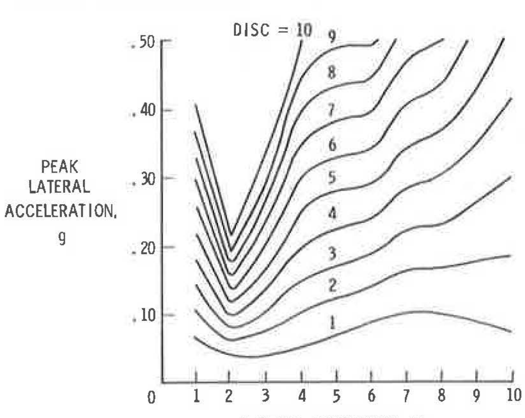


Figure 8. Lateral masking factor as a function of rms lateral acceleration and bandwidth of vibration.

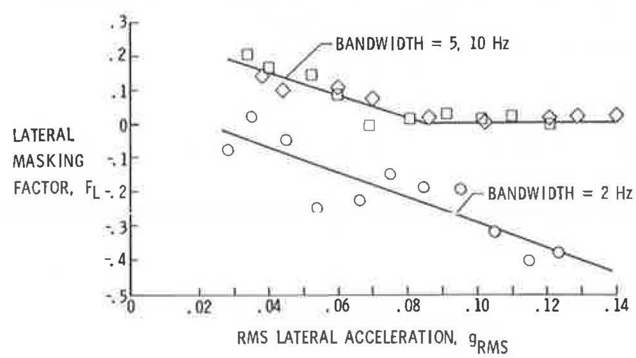
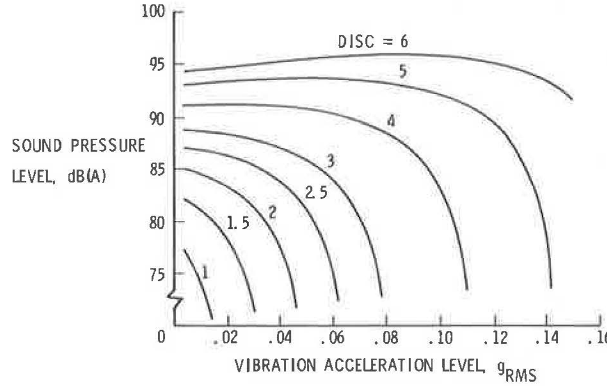


Figure 9. Noise-vibration criteria curves.



sign of Figure 1 (see 20). The analysis of variance (2 × 6 × 3 × 2) consisted of factorial combinations of two types of vibration (random or sinusoidal with each applied at six levels of acceleration: 0.020, 0.042, 0.064, 0.085, 0.106, and 0.130 g_{rms}) and of two different noise octave bands (500 and 2000 Hz center frequencies) presented at one of three noise levels [75, 85, 95 dB(A)]. There were repeated measures on all factors. A summary of the analysis of variance is presented in Table 1. All main effects (except for vibration type) and most double and triple interactions were significant. This implies that knowledge of all four factors is required in order to adequately assess passenger discomfort within the combined environment. Further analysis (Tukey test for additivity) of the interaction, however, indicated that the significant interactions may be an artifact of the analysis of overall reactions due largely to the fact that two different psychophysical laws are embedded in the overall discomfort responses, namely, a linear law for vibration discomfort and a power law for noise discomfort. This concept is discussed in the following section.

NOISE DISCOMFORT MODELING

The experimental design in Figure 1 provided the means for extending the NASA Ride Quality Model to incorporate a discomfort scale correction factor for noise,

Table 1. Summary of analysis of variance for overall discomfort responses.

Source	Sum of Squares	df	Mean Square	F
T Vibration Type	1.10×10^5	1	1.10×10^5	0.68
Error (SXT)				
A Acceleration Level, g_{rms}	609.67	5	121.93	72.62[a]
Error (SXA)				
D Noise Level, dB(A)	837.01	2	418.51	93.89[a]
Error (SXD)				
F Octave Bands	210.94	1	210.94	60.81[a]
Error (SXF)				
S Subjects	1472.19	47	31.32	-
TXA Interaction	9.90	5	1.98	4.15[a]
Error (SXTXA)				
TXD Interaction	2.77	2	1.38	2.58
Error (SXTXD)				
AXD Interaction	35.92	10	3.59	9.18[a]
Error (SXAXD)				
TXF Interaction	1.76	1	1.76	5.54[a]
Error (SXTXF)				
AXF Interaction	9.91	5	1.98	6.27[a]
Error (SXAXF)				
DXF Interaction	179.68	2	89.84	39.96[a]
Error (SXDXF)				
TXAXD Interaction	7.01	10	0.70	1.84[a]
Error (SXTXAXD)				
TXAXF Interaction	0.85	5	0.17	0.39
Error (SXTXAXF)				
TXDXF Interaction	0.68	2	0.34	0.78
Error (SXTXDXF)				
AXDXF Interaction	13.60	10	1.36	3.63[a]
Error (SXAXDXF)				
TXAXDXF Interaction	4.22	10	0.42	1.05
Error (SXTXAXDXF)				

[a] $p < 0.05$.

since the subjects were exposed to the same vibrations both with and without noise. Thus, the discomfort attributable to noise could be represented as the difference in discomfort between a ride with and without noise. The relationship would be given by the following formula:

$$DISC_N = DISC_{N+V} - DISC_V \qquad (2)$$

where

$DISC_N$ = discomfort due to noise only,
$DISC_{N+V}$ = discomfort due to combined noise and vibration, and
$DISC_V$ = discomfort due to vibration only.

Equation 2 is valid only if noise and vibration do not interact, i.e., if their separate effects are additive. As noted earlier in this paper, the analysis of variance of the overall discomfort responses indicated that all interactions were significant. It was also suggested that these interactions may be an artifact of the linear analysis of overall discomfort responses and that two psychophysical relationships may be embedded in the overall discomfort responses. Consequently, two additional analyses of variance were computed based upon (a) noise discomfort responses, obtained from Equation 1, and (b) the logarithm of the noise discomfort responses. Computation of these analyses of variance allowed the determination of the degree of interaction (nonadditivity) of noise and vibration associated with the noise discomfort responses. That is, the assumption of additivity implicit in Equation 2 was tested for a linear and a power law relationship between noise discomfort and noise level. The analyses of variance were for factorial (6 × 3) combinations of vibration (6 levels) and noise (3 levels) with the noise discomfort responses averaged across octave bands. Results are summarized in Table 2. The results in Table 2 indicate a significant interaction of noise and vibration for the linear noise discomfort responses but not for logarithms of these same responses. Thus, the use of logarithmic transformation of noise discomfort responses removed the interaction of noise and vibration. The implication of this fact is that separate but successive noise and vibration criteria may be sufficient for the prediction of ride quality in the combined environment whenever the noise and vibration spectral characteristics are limited, i.e., noise energy, vibration energy, or both are concentrated within single octave bands, discrete frequencies, or both.

A tentative noise-vibration criterion based upon the additivity concept discussed in the preceding section is presented in Figure 9. This figure shows a set of constant overall discomfort curves for the combined noise and vibration environment. These curves were generated from the psychophysical functions relating noise discomfort and vibration discomfort to the physical levels of each stimulus (see 20 for details). The individual curves in Figure 9 indicate the noise level [dB(A)] and vertical vibration acceleration level (g_{rms}) required

Table 2. Summary of analyses of variance for both linear and logarithmic values of noise discomfort responses.

		Linear			Logarithmic		
Source	Degrees of Freedom	Sum of Squares	Mean Square	F	Sum of Squares	Mean Square	F
A Vibration Level	5	2.1602			0.5041		
B Noise Level	2	21.6025			3.9180		
AXB Interaction	10	1.0096			0.1949		
Nonadditivity	1	0.8858	0.8858	64.1884[a]	0.0003	0.0003	0.0139
Balance	9	0.1238	0.0138		0.1946	0.0216	

[a] $p < 0.05$.

to produce constant amounts of overall discomfort for the noise and vibrations of the present study. Discomfort levels range from a value of 1 (DISC = 1), corresponding to discomfort threshold, to a value of 7 (DISC = 7), corresponding to a high level of subjective discomfort. Several important facts and implications of criteria curves such as those in Figure 9 should be noted. These include: (a) the curves supply a single source of information for determining the overall discomfort due to combined noise and vibration, (b) trade-offs between noise and vibration in terms of subjective discomfort can be made, and (c) for noise levels greater than 95 dB(A), the discomfort is relatively unaffected by vibration acceleration level. The reader should note, however, that these statements apply strictly to the factors and factor levels used in this study.

CONCLUSIONS

This paper has presented a summary of the results of experimental studies of passenger discomfort response to single-axis vertical, lateral, and roll vibrations as well as passenger discomfort response to combined noise and vertical vibration. The important facts and implications of this research are summarized below:

1. A family of constant discomfort curves with a direct ratio relationship to one another was generated for vertical sinusoidal vibration. These curves were anchored at the discomfort threshold for vertical vibration.
2. Discomfort is a continuous and readily discriminable function of vibration acceleration level within each frequency of vibration. Quantification of the basic psychophysical relationship between perceived discomfort and vibration stimulus levels provides a very useful tool for determining trade-offs between passenger comfort and ride environment.
3. Vertical vibration frequencies greater than approximately 15 Hz are considered relatively unimportant to ride quality for transportation vehicles operating within realistic ride environments. This is due in large part to the attenuation of vertical vibration by the cushioned seats.
4. The presence (or lack) of frequency masking for vertical vibrations is a function of both bandwidth and overall rms amplitude of the vibration. Thus, the effect of frequency masking should be accounted for in the development of ride quality criteria, since a knowledge of this effect would allow for more accurate prediction, diagnosis, or both of ride quality problems.
5. A family of constant discomfort curves for roll vibration was developed. These curves were anchored at the discomfort threshold for roll vibration.
6. The roll masking data indicated that estimates of human discomfort response to random roll vibrations applied at levels less than or equal to approximately 0.40 rad/s^2 (rms) can be made from knowledge of the single roll frequency component that contributes the largest amount of discomfort. For roll acceleration levels greater than 0.40 rad/s^2 (rms), the separate frequency components of discomfort interact in an antagonistic sense. As a result, estimates of discomfort response that do not account for this antagonistic interaction would tend to overestimate subjective discomfort to the higher levels of roll vibration.
7. A family of constant discomfort curves (anchored at threshold of discomfort) was produced for lateral vibrations.
8. The frequency masking factor for the lateral axis was also found to be a function of both the bandwidth of vibration and the overall rms acceleration level. Furthermore, the relative values of the lateral masking factor were generally much smaller than the values of the vertical masking factor for corresponding bandwidths and acceleration levels. This meant that the discomfort values produced by individual frequency components contained within a lateral vibration spectrum were less additive than the discomfort components contained within a similar vertical vibration spectrum; i.e., for random lateral vibrations a single frequency component of the vibration spectrum is the dominant determiner of the subjective response to that spectrum. Consequently, it is important to account for these effects in the assessment of ride quality within transportation systems having substantial lateral ride motions.
9. Passenger discomfort responses to the combined noise and vibration stimuli used in this study were shown to be additive if a logarithmic transformation of noise discomfort responses was performed. This implies that separate but successive noise and vibration criteria may be sufficient for the prediction of ride quality in the combined environment when the spectrum characteristics of the noise and vibration are relatively uncomplicated, i.e., concentrated within single octave bands or discrete frequencies.
10. A tentative set of noise-vibration criteria curves (constant comfort) based upon the stimulus parameters of this study were generated. These criteria curves provide a single source of information for determining the overall discomfort due to combined noise and vibration as well as the trade-offs between the two stimulus factors in terms of passenger discomfort.

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Resource Impacts of Alternative Automobile Design Technologies

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Automobile production and operation consume energy, materials, capital, and labor resources. Alternative automobile design concepts are examined in terms of their aggregate resource impacts. A computer-based model was developed for generating the resource requirements of alternative automobile technologies. The model goes beyond previous tools in its scope, level of impact disaggregation, and flexibility. It projects the annual energy, materials, capital, and labor requirements of the passenger automobile fleet through the year 2000. The methodology integrates a family-tree technique with an input-output approach that generates the capital and labor information. It tracks 24 major materials, with supply disaggregated among primary and recycled materials, imports, and domestic sources. Net energy consumption is derived, along with capital and labor impacts disaggregated by 90 industries. The model was used to examine a broad range of scenarios, encompassing various automobile design technologies and constraints imposed by safety and emissions regulations. All the scenarios show fleet fuel consumption declining through 1985, as the gains in fleet fuel efficiency outweigh the growth in distances traveled. With a few exceptions, the weight-conscious designs and innovative structures result in a significant reduction in consumption of the major materials used in automobile production. Finally, increased capital expenditures in the automobile industry are offset by capital savings in other sectors of the economy.

As a major consumer of petroleum, the automobile has been the subject of much recent attention. Various techniques have been proposed for improving automobile fuel economy, ranging from simple retrofit devices to advanced engines and innovative structures. Unfortunately, the focus of this attention has been exclusively on petroleum consumption and has tended to ignore the other vital resources consumed by the automobile fleet. Automobile production and operation require energy, materials, capital, and labor resources in delivering a level of service that is usually measured in terms of vehicle distances traveled, or vehicle miles traveled (VMT). Aggregate demand for any of these four resources can be reduced through the substitution of the others. Thus, the selection of fuel-efficient automobile designs should be viewed and evaluated in terms of the trade-offs in aggregate resource requirements that they represent. The increased use of aluminum in automobiles, which would displace materials such as cast iron and sheet steel, is an example of these concepts.

Due to its light weight, aluminum substitution would lower the overall weight of the vehicle and improve fuel economy. However, aluminum production is very energy intensive. Whether or not there is a net energy savings would depend on whether the reduction in propulsion fuel consumption exceeds the changes in automobile fabrication and materials processing energy. Going further, it can be shown that similar trade-offs exist among the other resource categories; additional capital requirements are needed for motor vehicle and aluminum production facilities, but these are offset by investment savings in such areas as refineries, petroleum distribution, and steel manufacturing.

The aluminum example suggests the broad range of options available in the selection of future automobile design concepts and the large number of consequences. There are substitution possibilities within resource categories (e.g., between materials or between energy forms) and trade-offs between resource sectors (e.g., capital displacing energy). These trade-offs raise several critical issues:

1. In the process of lowering petroleum imports, are we creating a vulnerability in another area to a potential cartel?