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Vibration Testing of Off-Road Vehicle Seats

By John C. Gagliardi and Walter K. Utt

UNITED STATES DEPARTMENT OF THE INTERIOR

BUREAU OF MINES

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UNIT OF MEASURE ABBREVIATIONS USED IN THIS REPORT

cm	centimeter	lbf	pound force
ft/s^2	foot per square second	lbf/lbm	pound force per pound mass
h	hour	lb/ft^3	pound per cubic foot
Hz	hertz	min	minute
Hz/s	hertz per second	mm	millimeter
in	inch	m/s^2	meter per square second
kg	kilogram	N	newton
kg/m^3	kilogram per cubic meter	N/kg	newton per kilogram
kPa	kilopascal	pct	percent
lb	pound	psi	pound per square inch

C

VIBRATION TESTING OF OFF-ROAD VEHICLE SEATS

By John C. Gagliardi[1] and Walter K. Utt[2]

ABSTRACT

The U.S. Bureau of Mines, in cooperation with Carter Mining Co., conducted vibration tests of four off-road vehicle seats. The purpose of the tests was to determine which seat provided the best vibration attenuation under laboratory conditions. Laboratory tests were constructed to simulate the mining vibration environment within the limitations of the test equipment. The acceleration test levels and corresponding response of the seats were low compared to ISO 2631's fatigue-decreased-proficiency time limits. Two of the seats employed mechanical suspensions and two seats employed air suspensions. The seats were tested using a 22,241-N (5,000-lbf) electrodynamic shaker. Mechanical suspension seats were tested with various preload-to-mass ratios (PMR's) and cushion densities. Air suspension seats were tested with various air pressure levels and seat cushion densities. Air suspension seats provided good vibration attenuation if pressurized greater than 552 kPa (80 psi). Mechanical suspension seats' attenuation performance decreased if the PMR exceeded 9.8. Seat cushions of lower density provided less vibration damping.

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INTRODUCTION

A potential health risk to the operators of mobile surface mining equipment is whole-body vibration (WBV) exposure. The final point of transmission of vehicle vibration to the operator is through the seat. Having a seat that provides good vibration isolation and allows good vehicle control is of prime importance to operator safety and health. The U.S. Bureau of Mines performed vibration tests on four off-road vehicle operators' seats in support of the Bureau's goal to enhance the safety of miners. The study determined the optimal adjustment settings (air pressure, preload setting, etc.) for each seat tested. From the best performance from each seat tested, comparisons were obtained among the different seat types. The seats were tested at maximum vibration levels that never exceeded the 4-h, fatigue-decreased-proficiency (FDP) exposure time defined by the International Standards Organization in its guideline ISO 2631. In addition, the measured response of the seat never exceeded a 4-h FDP time. Two seats employed mechanical suspensions and two seats employed air spring suspensions. The seats were tested using a 22,241-N (5,000-lbf) electrodynamic shaker. The seats were subjected to a swept sine acceleration input. The frequency range tested for the seats was from 0.7 to 10 Hz. Mechanical suspension seats were tested with varying seat cushion densities and preload-to-mass ratios (PMR's). Air suspension seats were tested with varying seat cushion densities and air pressure levels. Measured results were transmissibility and the root mean square (RMS) of the peak accelerations of the seat. Results for the mechanical suspension showed that seats with PMR's greater than 9.8 (9.8 is recommended normal ride setting) had increased acceleration levels. Results for the air suspension showed that seats pressurized greater than 552 kPa (80 psi) attenuated vibration over the frequency range tested (0.7 to 10 Hz).

The effect of WBV on operators of off-road vehicles has received considerable attention. Recommendations for field measurement and definitions of acceptable levels of WBV are available (*1-2*).[3]

The International Standards Organization has developed guidelines for the evaluation of human exposure to WBV known as ISO 2631 (*1*). ISO 2631 provides recommendations for performing vibration measurements in the field and analysis in the laboratory. ISO 2631 also provides guidelines or recommendations for exposure limits for WBV in the frequency range of 1 to 80 Hz. The standard is specified in terms of vibration frequency, acceleration magnitude, exposure time, and the direction of vibration relative to the body. Three different human criteria were established: human comfort, FDP, and health risk exposure. The Society of Automotive Engineers (SAE) has defined a SAE Recommended Practice known as SAE J1013 Jan80 for the measurement of WBV of seated operators of off-highway work machines (*2*). SAE J1013 and ISO 2631 compare similarly in terms of measurement procedures and frequency weighting.

A field study was conducted by the Bureau and its contractor that determines the probability of WBV exposure for a large class of off-road machinery operating in surface coal mines (*3*). The study determined that, with the exception of loaders and graders, vertical vibration is responsible for the most severe vibration exposure. It was found that more than 40 pct of off-road equipment operators were exposed to WBV exceeding the FDP level.

The vibration transmission path from source to operator for vehicles is given as follows: (1) tire-road and track-terrain, (2) frame, (3) cab, and (4) operator. Coupling the various transmission paths are suspensions. The purpose of the suspension is to dissipate vibrational energy. The various suspensions are (1) primary suspension, (2) cab mounting, and (3) seat. The primary suspension couples road input to vehicle frame. The primary suspension for vehicles such as loaders, scrapers, and rubber-tired dozers typically consists of tire damping only. Tire damping is provided by the dissipation of energy by the flexing of the tires. Cab mounting is usually an elastomeric material that separates frame from cab. The seat uncouples the operator from cab floor motion. The seat is typically the least expensive element of a vehicle's overall suspension to modify or redesign, and modification of the seat would appear to be the most cost-effective way to achieve better ride quality.

Because of the importance of the seat in protecting the operators of off-road vehicles, guidelines have been established for measuring vibration performance of seats (*4-6*). SAE J1384 and J1385 and ISO 7096 provide recommendations for laboratory measurement and evaluation of seat transmissibility and damping (*4-6*). SAE J1385 and ISO 7096 define classes of earth-moving machines and provide test parameters for each class in the form of power spectral density (PSD). SAE J1384 specifies the measurement of acceleration and maximum transmissibility at the seat-buttock interface.

Laboratory and simulation studies of vibration attenuation characteristics of off-road vehicle seats have been conducted (*7-8*). The vibration transmission performance of five off-road vehicle seats of different makes and manufacturers was investigated for varying amplitudes of excitation and suspension height in the frequency range of 1 to 8 Hz (*8*). The static and dynamic characteristics of the seats' cushion and suspension were also measured (*8*).

[3]Italic numbers in parentheses refer to items in the list of references at the end of this report.

Using the measured parameters, Rakheja developed a general analytical model of a mechanical seat (7).

In the present study, four off-road vehicle seats were tested using an acceleration test spectrum that was swept over the frequency band of 0.7 to 10 Hz. Two seats employed mechanical suspensions and two seats employed air suspensions. The mechanical seats were tested to determine the effect of PMR. The preload is an adjustment in the seat that loads the suspension spring in tension to allow constant static deflection of the seat for various operators' weight. The mass is simply the weight on the seat. PMR is the amount of preload force put into the mechanical suspension divided by the mass of the operator. If the operator adjusts the preload to less than his or her weight, when the operator sits on the seat, the seat suspension will compress beyond the mid-ride static deflection. In contrast, if the preload is set greater than the operator's weight, when the operator sits on the seat, the compressive static deflection of the seat will be less than the mid-ride position. In either case, the operator will increase vibration levels transmitted to him or her by hitting the compression or extension limit stops during high-level shocks because of improper static deflection position of the seat. Mechanical seats were also tested with foam seat cushions of different densities and materials. The air suspension seats were tested at various air pressures. Air suspension seats were also tested with foam seat cushions of different densities. Comparisons among the seats are given in terms of peak transmissibilities and their corresponding frequencies, overall frequency-weighted RMS acceleration, and attenuation frequency. The attenuation frequency is defined as the largest frequency at which the transmissibility decreases to a value of 1.

OFF-ROAD VEHICLE SEATS

Four off-road vehicle seats were tested. All seats were manufactured by the German company Isringhausen and provided through a cooperative agreement between the Bureau and Carter Mining Co. The seat models were two Isringhausen 6000/575, mechanical suspension seats, one 6500/575 air suspension seat, and one 6500/577 air suspension seat. The two model 6000/575's were a rebuilt seat and an unused new seat. The 6500/575 and 6500/577 were both unused new seats. All seat types are commercially available. The major elements of all seats consisted of a suspension, seat cushion, and backrest cushion. Various types of seat suspensions and cushions were tested; they are described below.

MECHANICAL SUSPENSION

The 6000/575 mechanical suspension consists of a scissors suspension linkage. The seat is mechanically sprung by two-coil steel springs mounted horizontally to the top of the scissor suspension linkage. A one-coil spring is an extension spring connected to a preload deflection linkage. The operator's weight adjustment dial is connected to the preload linkage. As the dial is turned for increasing operator weight, the preload linkage extends the coil spring. This extension loads the spring in tension. When the operator sits, the seat lowers, causing the coils to further extend. The sum of the preload tension force and static deflection force of the seat suspension equals the operator's weight. The preload adjustment on the 6000 seats is calibrated so that the seat's static deflection will be the same for all operators.

A shock absorber is mounted asymmetrically from the top front of the scissor suspension to the rear of the seatbase. The shock absorber is oil hydraulic with a dual setting. The 6000/575 has a permitted suspension stroke of 95 mm (3.74 in). The 6000/575 also has load-independent height and inclination adjustment. The original equipment manufacturer's (OEM's) seat cushion is a high-density polyurethane foam.

AIR SPRING SUSPENSION

The 6500 series seat features the same scissor linkage and asymmetrically mounted shock absorber as the 6000/575 seat. The 6500 series replaces the 6000's mechanical spring with an air spring. The 6500 possesses an automatic leveling suspension. The automatic leveling suspension consists of a cam-operated valve that positions the system independently of the driver's weight to the mid-ride suspension stroke location. The 6500 series integrates pneumatic lumbar support into the seat backrest. This feature was not examined as part of this study. The 6500/575 consists of a standard polyurethane foam seat cushion, while the 6500/577 contains air chambers in the seat cushion for individual adjustment. The 6500/577 model also has incorporated a suspension system for longitudinal motion that was not tested. The vertical suspension stroke on the 6500 is 100 mm (4.0 in).

CUSHIONS FOR SEAT AND BACK

In addition to original vendor-supplied seat cushions for the Isringhausen chairs, seat cushions were provided by

Dynamic Systems, Inc. through Carter Mining Co. The cushions provided by Dynamic Systems were developed for high-energy impact absorption and to provide uniform orthopedic support. The two trade names tested were Sun-Mate cushion and Pudgee cushion.

The Sun-Mate cushion's foam is a high-density 100-pct open-cell elastomeric foam with a density of 80.1 kg/m^3 (5 lb/ft^3). It possesses high impact energy absorption. The Sun-Mate cushion is a visco-elastic foam that contours to give uniform pressure distribution and soft springback.

The Pudgee cushion is a visco-elastic gel-foam with a 100-pct open-cell structure. The Pudgee cushion's dough-like consistency contours to a body's outline with low tension-shear force. The Pudgee cushion possesses a high density of 320.4 kg/m^3 (20 lb/ft^3).

VIBRATION TESTING

TESTING HARDWARE AND INSTRUMENTATION

The seat was tested using a MB Dynamics Model C50 electrodynamic shaker[4] over the frequency range of 0.7 to 10.0 Hz. The C50 shaker was limited to a 2.5 cm (1 in) peak-to-peak stroke and a maximum force rating of 22,241 N (5,000 lbf).

The acceleration input to the base of the seat was measured using a Bruel and Kjaer (B&K) 4370 accelerometer. The response of the seat was obtained using a B&K 4322 triaxial seat accelerometer. The B&K 4322 consists of a B&K 4321 triaxial accelerometer mounted inside a semi-rigid rubber pad. The B&K 4322 was attached to the center of the seat cushion using a generous amount of duct tape.

A study was made comparing transmissibility measurements on seat cushions using three different seat interface transducers (*9*). The seat interface transducer measures the vibrations on top of the seat. The seat interface transducer is mounted between the seat surface and human buttocks. The three different seat interfaces were a semi-rigid rubber pad, an aluminum bar, and a rigid nylon disk. It was shown that good agreement for the transmissibilities was obtained for the semi-rigid rubber pad and the rigid nylon disk. In this study, the semi-rigid rubber pad was used for measuring the acceleration on top of the seat cushion.

The acceleration signals from the base and cushion accelerometers were conditioned and amplified using a Timewave Technology 4050 two-channel portable charge-icp (internal circuit preamplifier) amplifier.

Shaker control problems caused by 60- and 180-Hz line noise were eliminated by using notch filters. Each channel employed a Frequency Device 781R1Q2 60- and 180-Hz notch filter.

[4]Reference to specific products does not imply endorsement by the U.S. Bureau of Mines.

COMPUTER CONTROL

The shaker was controlled using a Genrad 2514 vibration control system. The Genrad 2514 is a computer-based, software-controlled closed-loop vibration test and shaker control system. The Genrad 2514 is a Digital Equipment Corp. LSI-11 microcomputer interfaced with analog-to-digital and digital-to-analog subsystems. Software programs for product testing available for the Genrad 2514 are random, swept sine, shock, random-on-random, and sine-on-random.

Initial testing revealed that random vibration testing could not be performed at sufficiently high vibration levels because of the limited stroke of the C50 shaker. Random vibration tests superimpose individual frequencies simultaneously. This superimposition contributes to a large overall vibration test level with low individual frequency levels. A swept sine vibration test was chosen over a random vibration test because of the ability to subject the seat to an individual forcing frequency at any instant of time. By subjecting the seat to a swept sine test, the vibration level for an individual discrete frequency could be set at a significantly greater level than could be accomplished by using a random test.

REFERENCE SPECTRUM

The tests on the seats were performed using an acceleration spectrum. The sine test program can define a test spectrum in units of acceleration only. The spectrum was constructed to produce accelerations at levels as great as possible without having the C50 shaker exceed its stroke limits. For programming purposes, the acceleration reference spectrum was defined using a combination of constant slope segments.

$$A = M(f - f_*) + B, \quad (1)$$

where A = acceleration (m/s^2),

M = slope [(m/s^2)/Hz],

f = frequency (Hz),

B = acceleration intercept (m/s^2),

and f_* = upper frequency limit of the previous band or the lower limit of the current band.

The parameters of the employed acceleration test spectrum of equation 1 are given in table 1 with the resulting spectrum shown in figure 1. This spectrum functions as a reference spectrum for the Genrad 2514 vibration control system. An error spectrum for updating the drive control is obtained from the difference between the reference and measured test spectrums on the shaker table. An actual test spectrum measured at the base of the seat is shown in figure 1.

Table 1.—Slope-intercept values for acceleration test spectrum

f, Hz	M, (m/s^2)/Hz	B, m/s^2
0.7 to <1.0 . . .	0.0	0.1963
1.0 to <1.5 . . .	.5794	.1963
1.5 to <3.0 . . .	.0693	.4860
3.0 to <3.5 . . .	.31	.5900
3.5 to <6.0 . . .	−.86	.745
6.0 to <10.0 . .	−.0455	.53

B Acceleration intercept.
f Frequency.
M Slope.

TEST PROCEDURE

Each seat test condition was performed five times, each time being considered a run. A single linearly increasing frequency sweep was employed in each test run. The swept frequency range was 0.7 to 10.0 Hz. The sweep duration was 10 min per run for a sweep rate of 0.01549 Hz/s. To equalize the system, a startup time of 5 min was employed. The startup was performed at 0.7 Hz with a gradual buildup of acceleration amplitude to full test level. During the swept tests, a total of 450 acceleration values were stored for both the seat-base and cushion.

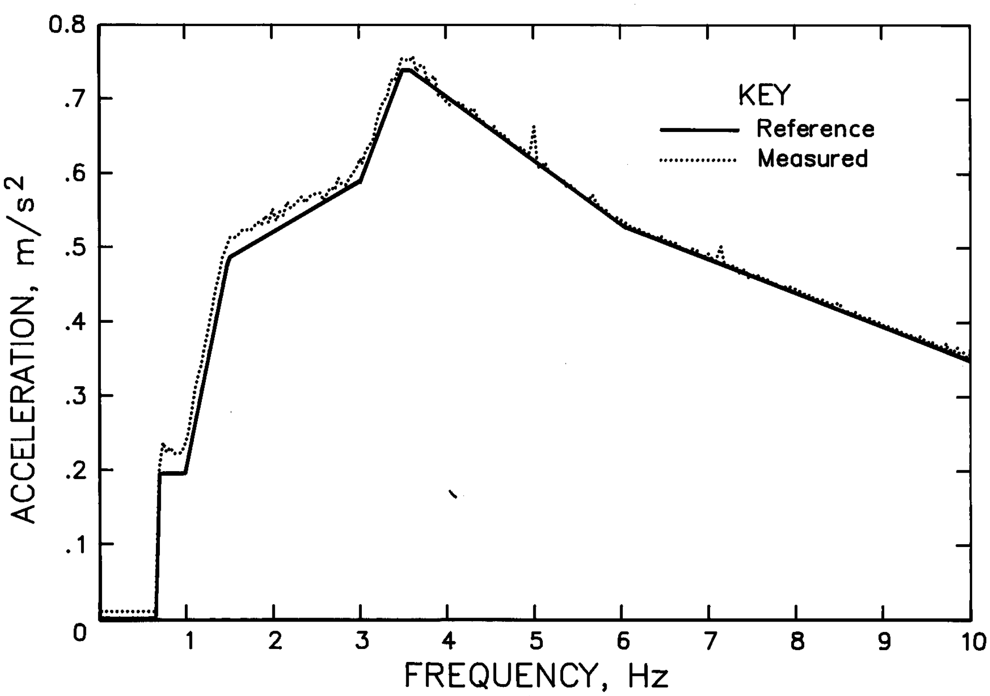

Figure 1.—Acceleration test spectrum.

ANALYSIS

For each run, the stored acceleration values of the seat-base and cushion were transferred to a personal computer for further processing. The computation of the various physical measures was performed on the personal computer using the commercially available interactive signal analysis program DADiSP. The seat-base and cushion acceleration spectra for the five runs per test were imported to DADiSP. From within DADiSP, the five spectra for each common location were averaged together.

Within DADiSP, the reference acceleration spectrum shown in figure 1 and defined in table 1 was generated. In addition, a vertical weighting factor was defined (fig. 2). This weighting factor is a continuous frequency weighting factor that has equal values of ISO 2631 weighting factors at the 1/3 octave center frequencies. The processing procedure computed the following measures:

1. Normalizing of reference acceleration spectra to measured base acceleration spectra:

$$N(f) = A_r(f)/A_b(f). \tag{2}$$

2. Transmissibility of the seat, i.e., ratio of cushion acceleration spectra to the base acceleration spectra:

$$T(f) = A_s(f)/A_b(f). \tag{3}$$

3. Normalized weighted seat acceleration spectra:

$$A_w(f) = A_s(f) \cdot W(f) \cdot N(f). \tag{4}$$

4. Normalized weighted seat mean squared acceleration spectrum ($[m/s^2]^2$):

$$P_w(f) = A^2{}_w(f). \tag{5}$$

5. Overall mean normalized weighted acceleration level:

$$A_a = \frac{1}{N} \sum_{I=1}^{N} A_w(f_i). \tag{6}$$

6. Overall RMS acceleration:

$$A_{rms} = \left[\frac{1}{N} \sum_{i=1}^{N} P_w(f_i) \right]^{1/2}. \tag{7}$$

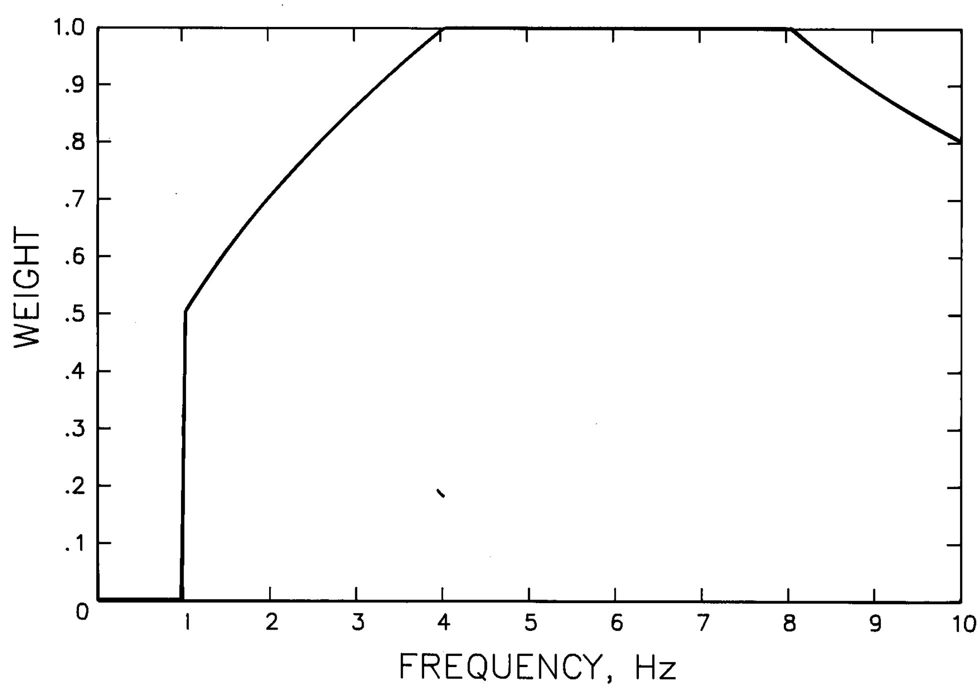

Figure 2.—Vertical vibration weighting.

Where f = frequency (Hz),

$A_r(f)$ = reference acceleration spectra (m/s^2),

$A_s(f)$ = acceleration spectra from seat cushion (m/s^2),

$A_b(f)$ = acceleration spectra of seatbase (m/s^2),

$T(f)$ = transmissibility,

$N(f)$ = normalized factor,

$W(f)$ = vertical weighting factor,

$A_w(f)$ = normalized weighted seat acceleration spectra (m/s^2),

$P_w(f)$ = normalized weighted seat mean square acceleration spectrum ($[m/s^2]^2$),

N = number of acceleration values per spectra,

A_a = overall mean normalized weighted acceleration level (m/s^2),

and A_{rms} = overall RMS acceleration (m/s^2).

RESULTS

6000/575 MECHANICAL SUSPENSION SEATS

Reference Settings

A comparison of transmissibilities, attenuation frequencies, and overall RMS accelerations was made for a new Isringhausen 6000/575 mechanical suspension seat loaded with a human, an anthropomorphic dummy, and inert masses. Table 2 presents the type of load object, its mass, seat preload setting, PMR, and various test measures. The test measures in table 2 are transmissibility peaks and corresponding frequencies, overall acceleration RMS levels, and attenuation frequencies. The peak transmissibilities were selected as being significantly higher than the local measures. The correct setting for a human sitting on the seat is a PMR of 9.8. The value of 9.8 is the acceleration constant of gravity. Following the assumption of other researchers that 2/7 of the seated body mass is supported by the feet, one obtains a PMR of 13.7 (*7*, *10*). Figure 3 presents the transmissibilities of human, dummy, and 51.8- and 68.0-kg (114- and 150-lb) inert masses for PMR's of 13.7 and 9.8, respectively.

Table 2.—Comparison of transmissibility peak values and frequencies, attenuation frequency, and overall acceleration RMS between human, anthropomorphic dummy, and inert mass for new 6000/575

	Mass, kg	Preload setting, N	PMR	Frequency peak, Hz	Transmissibility peak	Attenuation frequency, Hz	A_{rms}, m/s^2
Human	63.6	623.3	9.8	2.1	1.38	4.9	0.52
				3.7	1.51		
Dummy	72.6	712.0	9.8	.9	1.74	>10	.60
				2.7	1.69		
				7.9	1.49		
Inert mass ..	51.8	712.0	13.7	2.8	2.82	9.0	.84
				3.2	2.65		
				6.9	1.80		
	68.0	666.4	9.8	1.0	2.00	7.8	.56
				2.5	1.50		
				5.7	1.30		

A_{rms} Acceleration root mean square.

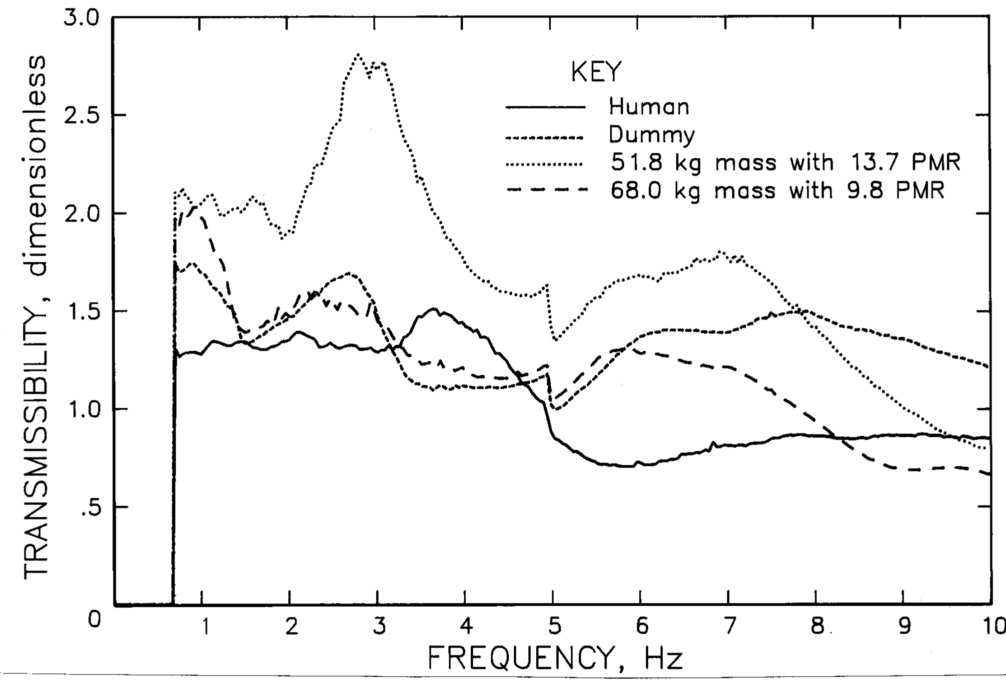

Figure 3.—Transmissibility of seat-base using human, anthropomorphic dummy, and inert mass with new 6000/575 mechanical suspension. Inert mass of 51.8 kg was tested with PMR = 13.7; inert mass of 68.0 kg was tested with PMR = 9.8.

One sees from table 2 that the PMR's of 9.8 produce similar overall RMS acceleration values. The dummy and 68.0-kg (150-lb) inert mass produced high transmissibilities near 1.0 Hz; however, a peak transmissibility at 1 Hz was absent for the human. The PMR of 13.7 produced transmissibilities that exceeded all cases of the 9.8 PMR except at very low frequencies ($f < 1$ Hz) and very high frequencies ($f > 9.5$ Hz). The human attained attenuation at the lowest frequency. The dummy never attained attenuation. There appear to be two major effects of the legs. For low frequency ($f < 3.0$ Hz), long-stroke displacement, the legs resist relative motion between the top of the seat suspension and floor, hence decreasing the transmissibility level. This resistance to relative motion is due to the coupling between seat and floor by the legs. At higher frequencies ($f > 4.0$ Hz), human legs act as a vibration energy absorber, resulting in lower transmissibility levels. The reverse effect is present for the dummy. At higher frequencies, the dummy's legs act as a vibration transmission path to the seat cushion, resulting in higher transmissibility levels.

With the exception of initial comparison of results between human, dummy, and inert mass, for the tests conducted, the seat was loaded with a mass, not a person or dummy. The driver's mass supported by the seat is taken to be (1) equal to and (2) 5/7 of the total human body mass (*7-8*), assuming 2/7 is supported by the legs.

Effect of Preload and Inert Mass

The effect that preload and inert mass have on the vibration attenuation characteristics of the seat for the new Isringhausen 6000/575 mechanical seat suspension was investigated. Changing the PMR is equivalent to changing the static displacement height of the seat suspension. This increases the possibility that the seat will hit the compression limit stop for decreasing preload or hitting the extension limit stop for an increasing preload. If the seat hits a limit stop, the limit stop will contribute to an increase in the stiffness in the seat suspension, hence changing the vibration attenuation characteristics of the seat.

Figures 4 and 5 present transmissibilities for the new Isringhausen 6000/575 with OEM seat cushion, and PMR's of 7.1, 9.9, and 11.3 (fig. 4), and 13.1, 16.4, and 19.6, (fig. 5).

The seat's vibration characteristics appear to be heavily dependent on PMR. Table 3 contains peak transmissibility values and corresponding frequencies, attenuation frequencies, and overall acceleration levels for the various PMR's. Figures 6 through 10 graphically summarize these results.

As seen in figure 6, as the PMR increases, the peak transmissibility increases for all ratios except PMR equal to 19.6. The first transmissibility peak values range from 1.9 to 2.6. Figure 7 shows a second transmissibility peak for all seats in the frequency range of 1.8 to 3.2 Hz. From figure 7, one sees that the second peak transmissibility tends to shift to a higher frequency with increasing PMR and also that the value of the transmissibility increases with the exception of PMR equal to 19.6. Figure 8 shows a third peak transmissibility occurring in the frequency range of 5.7 to 6.9 Hz. As with the second transmissibility peak, the trend is for the third peak transmissibility to shift to a higher frequency and to a greater level with increasing PMR. The attenuation frequencies for the various levels of PMR are given in figure 9. As seen in figure 9, as the PMR increases, the frequency at which the seat transmissibility decreases to below 1 increases. Figure 10 presents the overall RMS acceleration value as a function of PMR. As the PMR increases, the overall acceleration RMS value increases. These results demonstrate that seat dynamics have significant sensitivity to PMR adjustment above and below the operator's weight. If an operator sets the chair preload setting to a value greater than his or her weight, the PMR ratio would be greater than 9.8, resulting in less vibration attenuation of the seat. Also, if the operator sets the chair preload setting to less than his or her weight, the possibility of the operator receiving shocks because of the chair bottoming out would increase.

Table 3.—New 6000/575 peak transmissibilities and frequencies, attenuation frequency, and overall acceleration RMS value for various PMR's

PMR	Frequency peak, Hz	Transmissibility peak	Attenuation frequency, Hz	A_{rms}, m/s^2
7.1	0.9	1.9		
	1.8	1.3	[1]3.4	0.48
	5.8	1.2	7.7	
9.9	1.0	2.0		
	[2]2.5	1.6	7.8	.56
	5.7	1.3		
11.3	.9	2.1		
	2.5	1.6	8.1	.56
	6.0	1.3		
13.1	.8	2.4		
	2.6	2.4	7.9	.81
	5.8	2.1		
16.4	.9	2.6		
	2.8	3.8	8.4	.95
	6.6	2.2		
19.6	1.1	1.9		
	1.7	2.4	>10.0	1.03
	3.2	3.2		
	6.9	2.8		

A_{rms} Acceleration root mean square.

[1]A dip occurred for transmissibilities <1 for frequencies between 3.4 and 4.2 Hz.

[2]The peak transmissibility level was broad in frequency between 2.0 and 2.9 Hz.

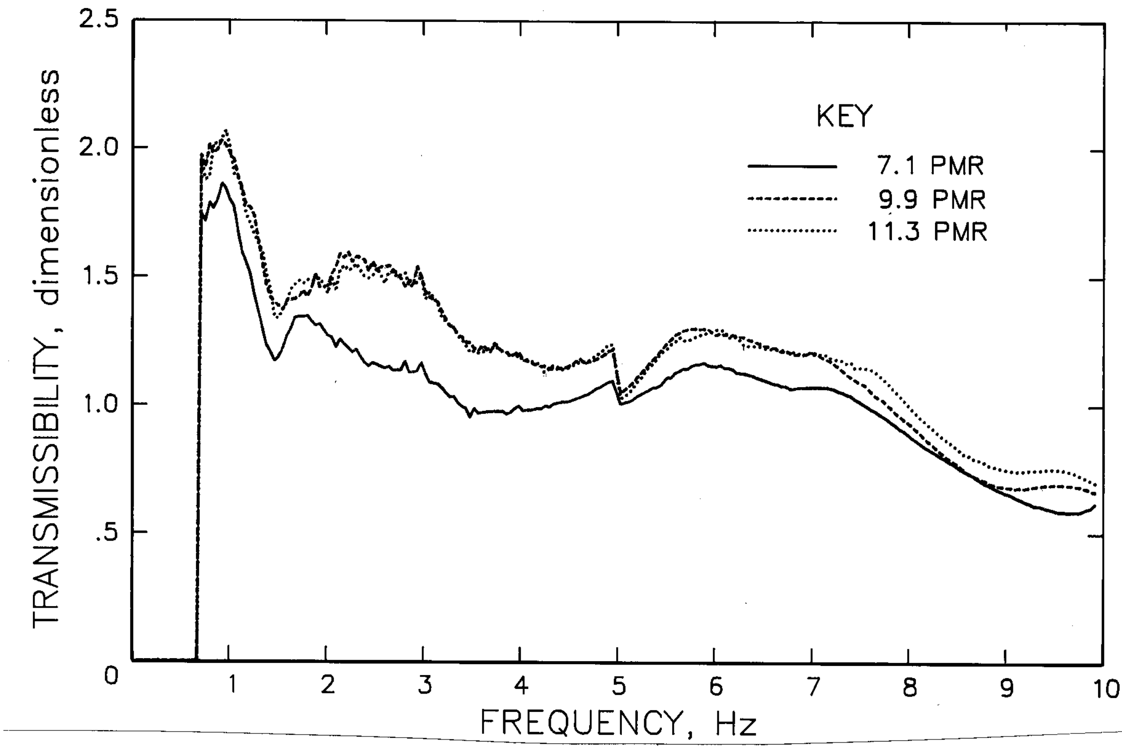

Figure 4.—Transmissibility of seat-base for PMR's of 7.1, 9.9, and 11.3 with new 6000/575 mechanical suspension.

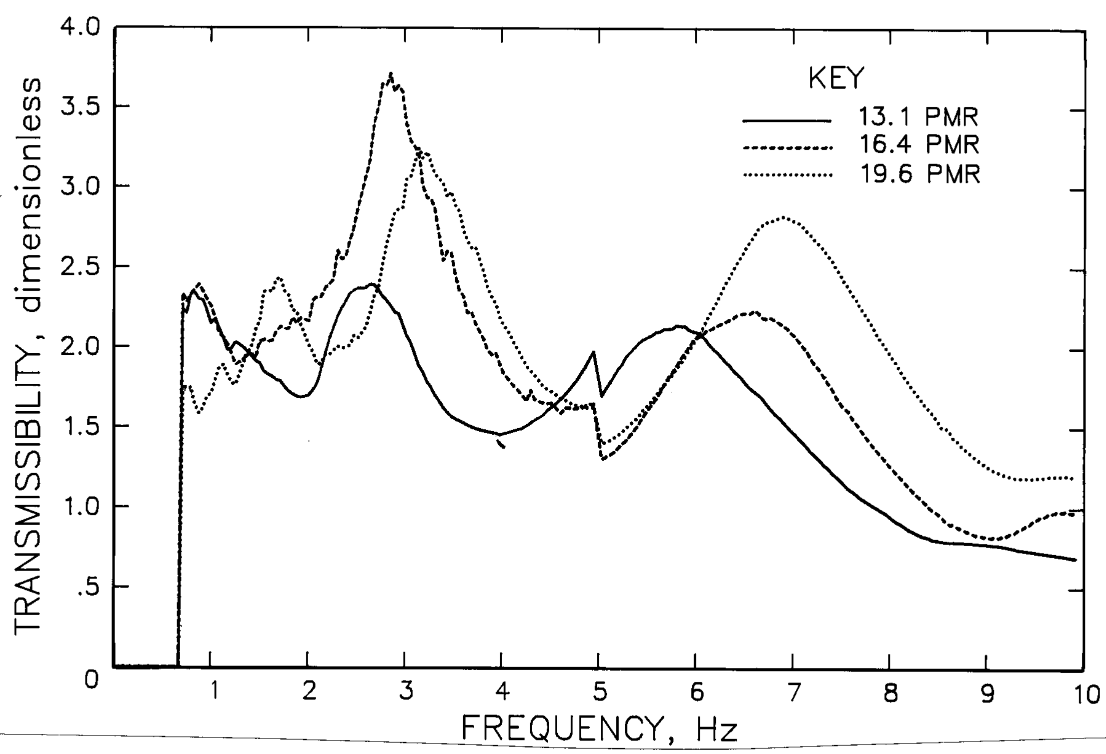

Figure 5.—Transmissibility of seat-base for PMR's of 13.1, 16.4, and 19.6 with new 6000/575 mechanical suspension.

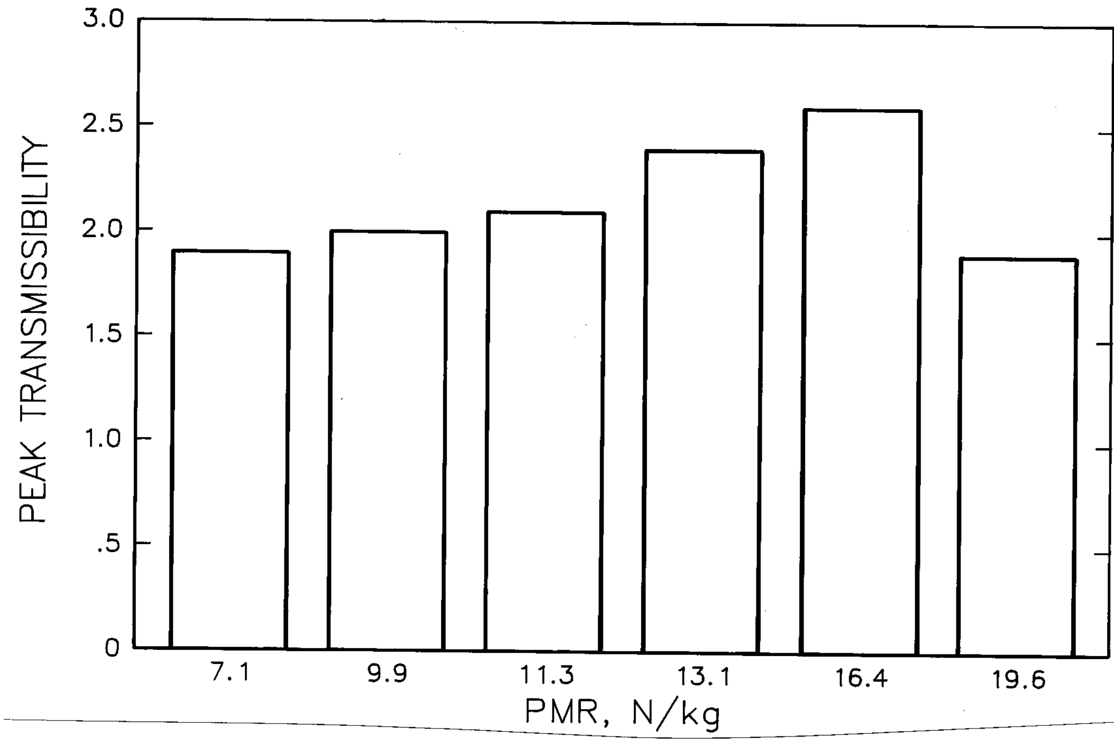

Figure 6.—First transmissibility peak of seat-base for various PMR's with new 6000/575 mechanical suspension.

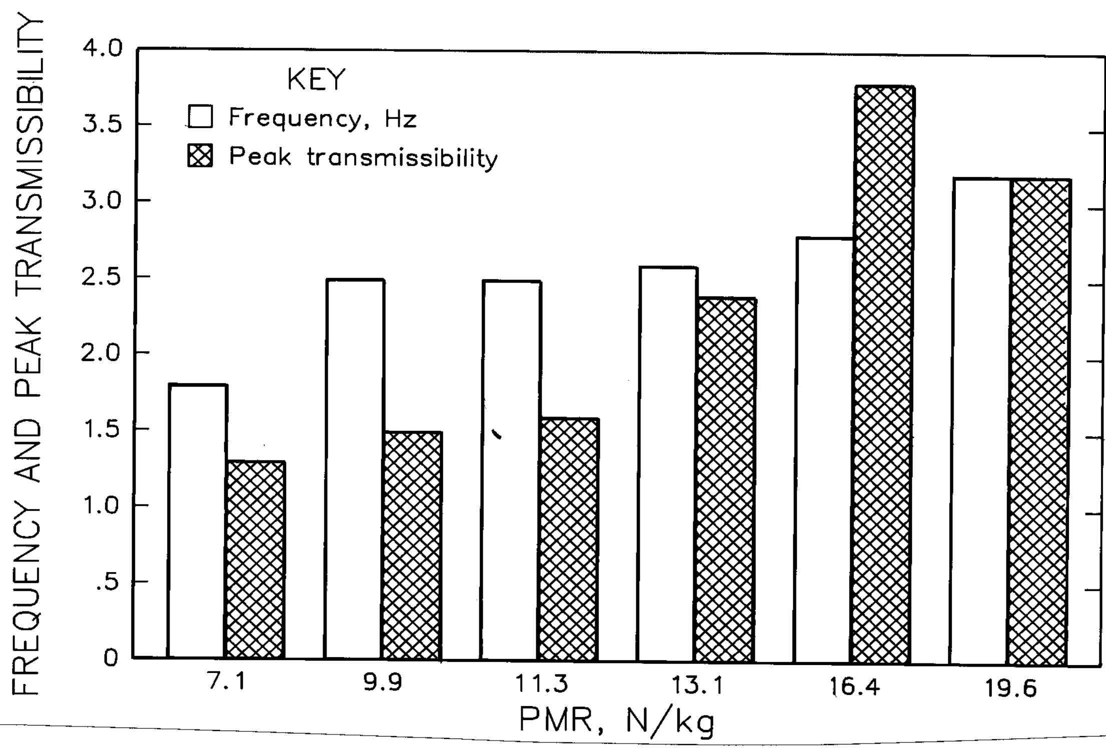

Figure 7.—Second transmissibility peak and corresponding frequencies of seat-base for various PMR's with new 6000/575 mechanical suspension.

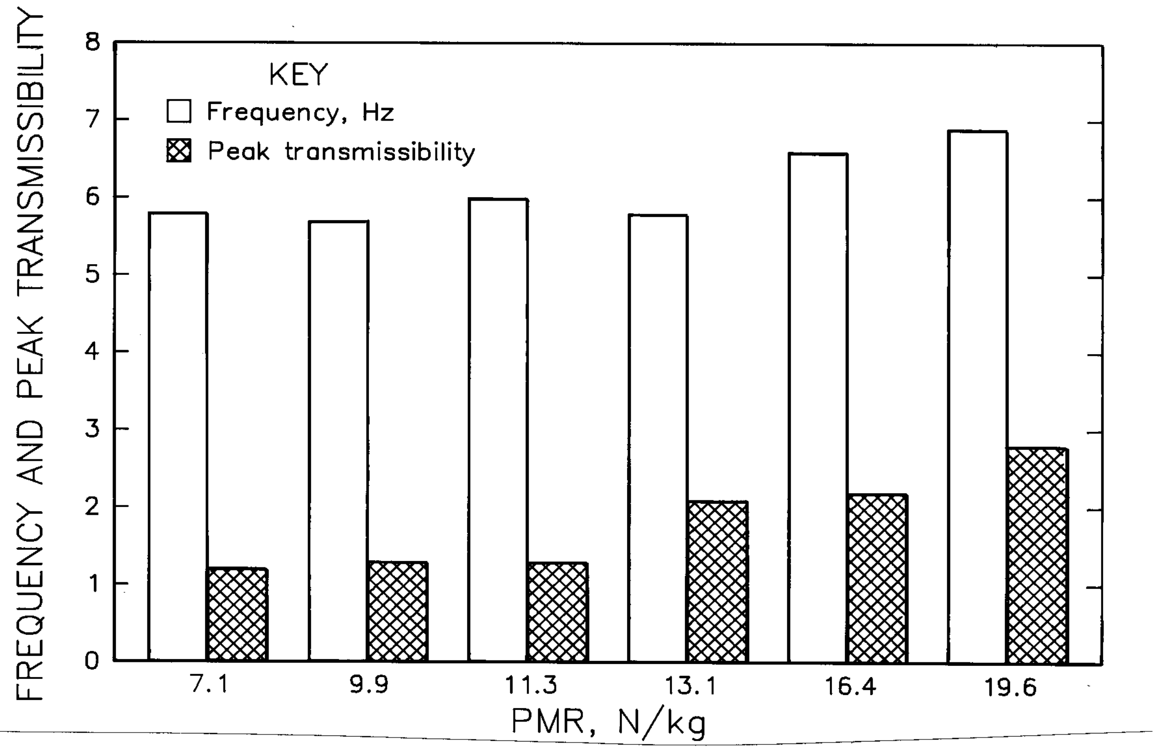

Figure 8.—Third transmissibility peak and corresponding frequencies of seat-base for various PMR's with new 6000/575 mechanical suspension.

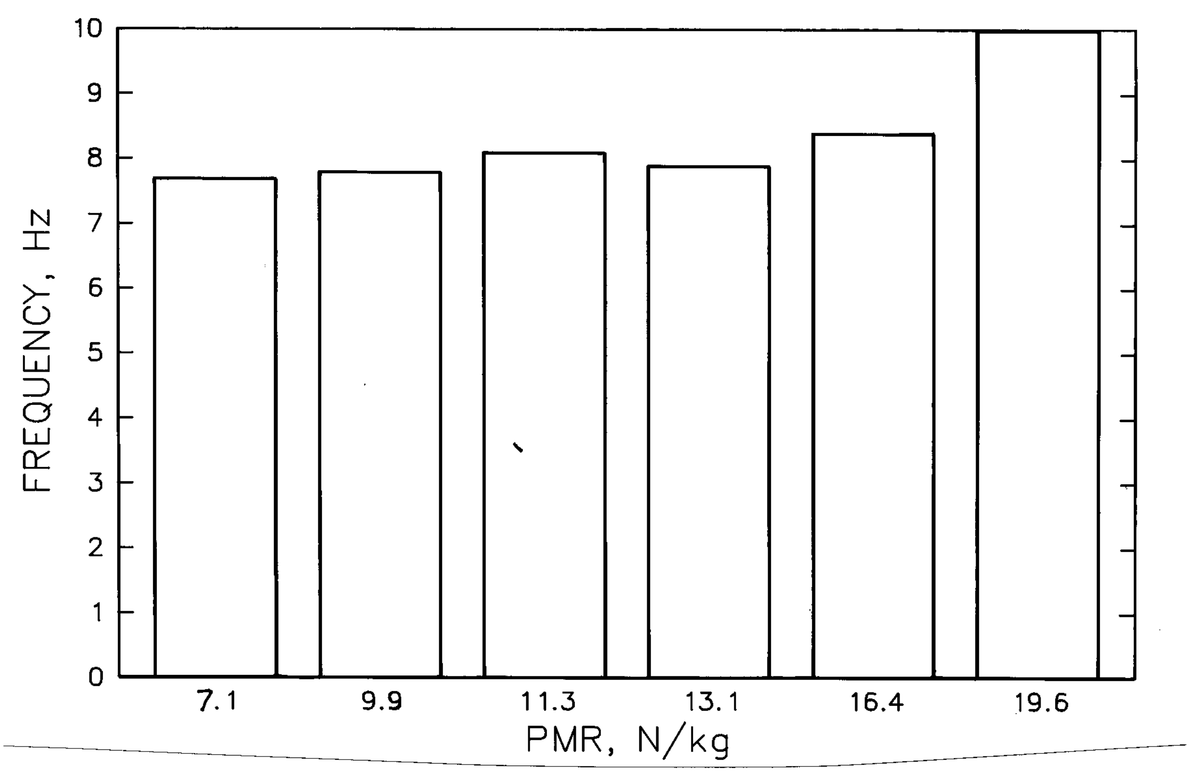

Figure 9.—Attenuation frequency for various PMR's with new 6000/575 mechanical suspension.

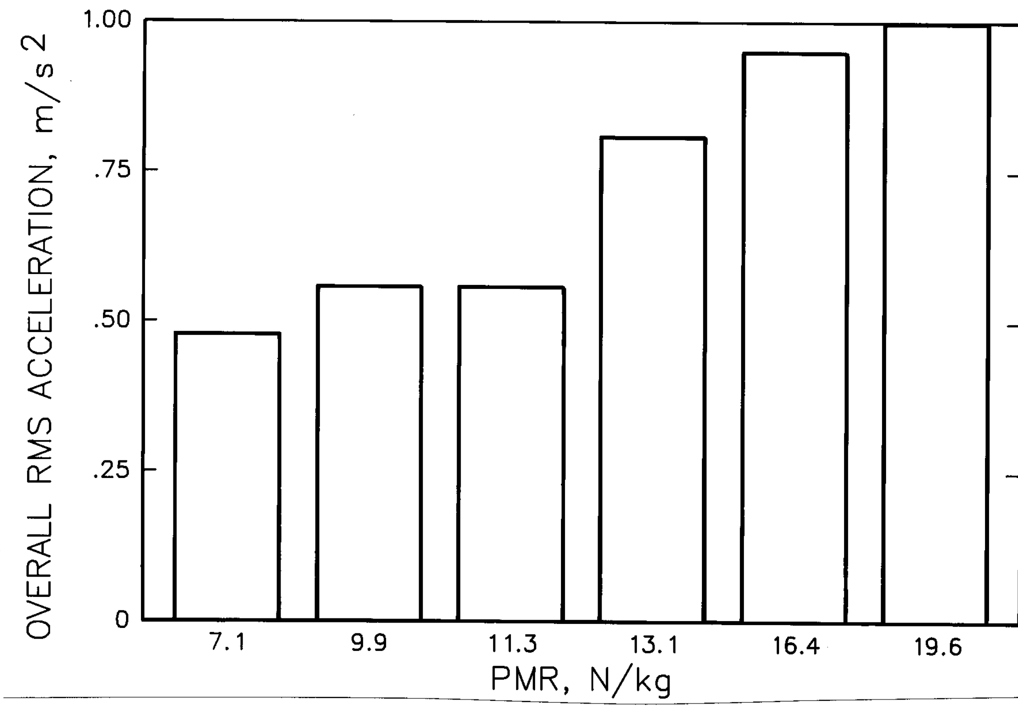

Figure 10.—Overall RMS acceleration for various PMR's with new 6000/575 mechanical suspension.

Effect of Cushions

To determine the effect of the seat cushion on seat dynamics, the rebuilt 6000/575 mechanical seat was tested with a test mass of 68.0 kg (150 lb) and a preload of 889.5 N (200 lbf) (PMR = 13.1). The new 6000/575 mechanical suspension seat was tested with a test mass of 68.0 kg (150 lb) and preloads of 666.4 N (150 lbf) (PMR = 9.8) and 889.5 N (200 lbf) (PMR = 13.1).

Rebuilt 6000/575 and PMR = 13.1

The transmissibility plots for the rebuilt 6000/575 are shown in figure 11. The peak transmissibilities and their corresponding frequencies are given in table 4. Also included in table 4 are the attenuation frequencies and the overall acceleration RMS values. A graphical summary of table 4 is presented in figure 12. As shown by the data in table 4 and figure 12, as the density of the cushion material increases, the transmissibility decreases. Assuming that the stiffness, damping, and density are all directly related, these results correspond similarly to a previous study (*7*). In that work, an analytical model of an Isringhausen mechanical suspension seat was developed to aid in determining optimal suspension parameters. The peak transmissibility value decreases with increasing cushion density (fig. 12). This compares similarly with the study (*7*), which found that stiffer cushions suppress resonant peaks. Table 4 and figure 12 show that transmissibility attenuation is achieved (transmissibility < 1) at lower frequencies for less dense seat cushions. This trend was also reported in the CONCAVE (Concordia Computer Aided Vehicle Engineering) study, which stated that acceleration transmissibility increases at higher frequencies for stiffer cushions (*7*). From figure 12, the overall acceleration RMS value decreases with increasing cushion density.

A sharp decrease in transmissibility occurs at 5 Hz for the 6000/575. This phenomenon occurs to some extent for all seats tested under every test condition. The sudden drop in transmissibility at 5 Hz is due to a fore-aft resonance from the asymmetrical mounting of the shock absorber relative to the scissor suspension (*10*).

Table 4.—Rebuilt 6000/575, PMR = 13.1 peak transmissibility, attenuation frequency, and overall acceleration RMS for various cushions

Cushion	Density, kg/m^3	Frequency peak, Hz	Transmissibility peak	Attenuation frequency, Hz	A_{rms}, m/s^2
OEM	40.0	2.3	2.66	6.9	0.72
Sun-Mate	80.1	2.7	2.20	8.9	.67
Pudgee	320.4	2.5	2.17	9.3	.65

A_{rms} Acceleration root mean square.
OEM Original equipment manufacturer.

New 6000/575 and PMR = 13.1

The new 6000/575 seat was tested with the OEM, Sun-Mate, and Pudgee cushions for fixed preload and mass of 889.5 N (200 lbf) and 68.0 kg (150 lb), respectively (PMR = 13.1). The transmissibility plots for the new 6000/575 are shown in figure 13. Table 5 and figures 14 and 15 give peak transmissibilities and their corresponding frequencies, attenuation frequencies, and overall acceleration RMS's.

Table 5 and figure 14 show that peak transmissibilities in the range of 2 to 3 Hz are similar to those for the rebuilt 6000/575 seat, i.e., seat cushions with increasing foam densities have decreasing peak values of transmissibility. From the transmissibility plots (fig. 13), as the cushion density increases, the peak transmissibility bandwidth increases. For the Pudgee cushion, the peak transmissibility spreads over a frequency range of 2.0 to 3.0 Hz. A second peak transmissibility occurred in the range of 5.8 to 7.9 Hz. The levels of transmissibility for the second peak are in the range of 2.1 to 2.3. The frequency at which the second peak occurs increases with increasing cushion density. Attenuation in the OEM cushion was obtained at 7.9 Hz. However, in the frequency range of analysis (0.7 to 10 Hz), the Sun-Mate and Pudgee cushions never achieved attenuation.

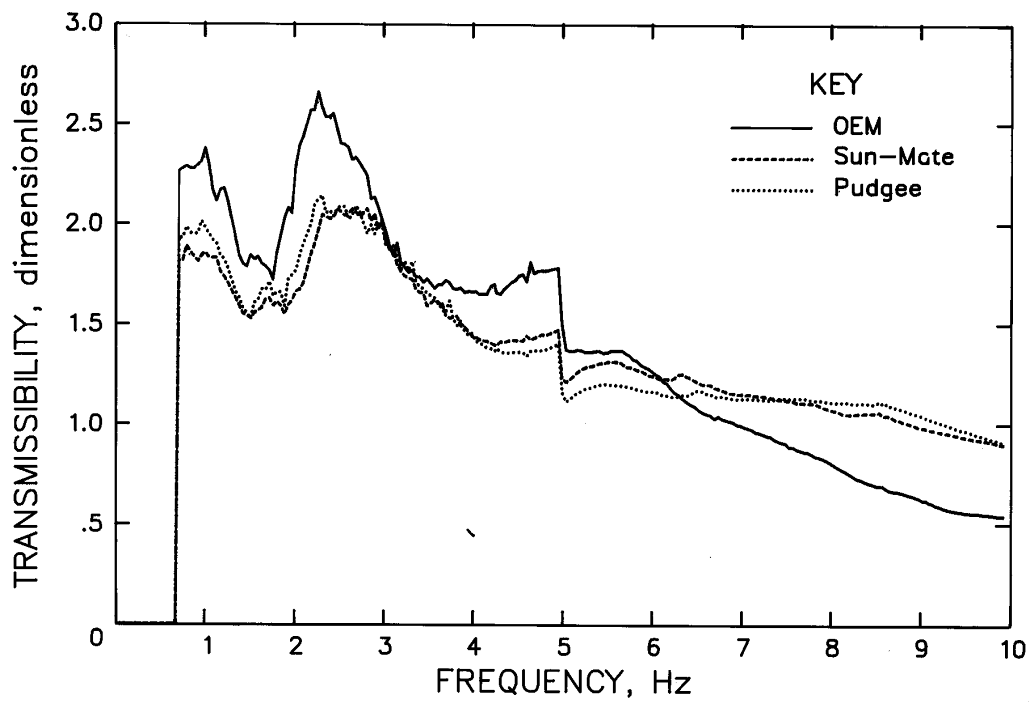

Figure 11.—Transmissibility of seat-base with differing seat cushions.

Table 5.—New 6000/575, PMR = 13.1 peak transmissibility, attenuation frequency, and overall acceleration RMS for various cushions

Cushion	Density, kg/m^3	Frequency peak, Hz	Transmissibility peak	Attenuation frequency, Hz	A_{rms}, m/s^2
OEM	40.0	2.6 5.8	2.4 2.1	7.9	0.78
Sun-Mate	80.1	2.7 7.8	2.2 2.3	>10	.81
Pudgee	320.4	2.0-<3.0 7.9	>1.75 2.10	>10	.75

A_{rms} Acceleration root mean square.
OEM Original equipment manufacturer.

New 6000/575 and PMR = 9.8

The new 6000/575 seat was tested with the OEM, Sun-Mate, and Pudgee cushions for fixed preload and mass of 666.4 N (150 lbf) and 68.0 kg (150 lb), respectively. The transmissibility plots for the new 6000/575 are shown in figure 16. Table 6 and figure 17 give peak transmissibilities and their corresponding frequencies, attenuation frequencies, and overall acceleration RMS's.

From tables 5 and 6 and figures 14, 15, and 17, the frequencies at which peak transmissibilities occur are highly dependent on cushion type but nearly independent of PMR. The general trends in transmissibilities appear to be independent of PMR but highly dependent on cushion type. These most obvious trends are that (1) attenuation is never attained in the seat for the Pudgee and Sun-Mate cushions, (2) the OEM cushion had higher transmissibilities for f < 3.0 Hz but lower values for f > 6.0 Hz, (3) the Sun-Mate cushions generally had higher transmissibilities than the Pudgee cushions, and (4) as the PMR ratio increases, the transmissibility levels and overall RMS acceleration increase.

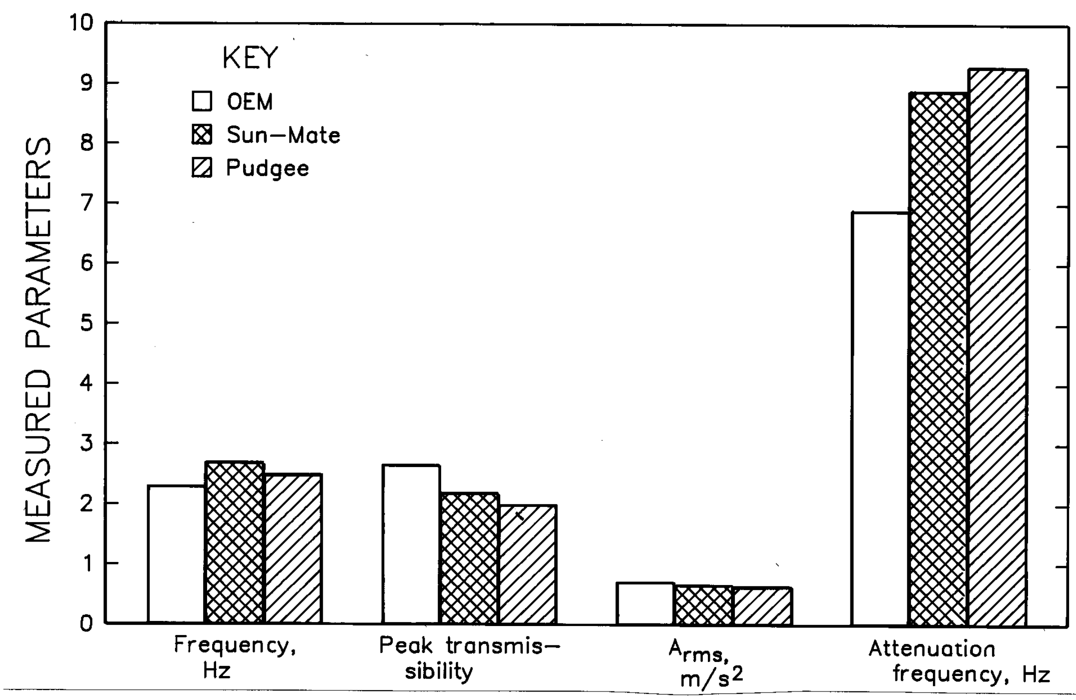

Figure 12.—Transmissibility peak and frequency of seat-base, acceleration RMS (A_{rms}), and attenuation frequency with differing seat cushions for rebuilt 6000/575 mechanical suspension, PMR = 13.1.

Table 6.—New 6000/575, PMR = 9.8 peak transmissibility, attenuation frequency, and overall acceleration RMS for various cushions

Cushion	Density, kg/m³	Frequency peak, Hz	Transmissibility peak	Attenuation frequency, Hz	A_{rms}, m/s²
OEM	40.0	0.95	2.03		
		2.3	1.60	7.8	0.56
		5.8	1.38		
Sun-Mate	80.1	1.1	1.63		
		1.5	1.64	>10	.68
		3.1	1.34		
		7.8	1.80		
Pudgee	320.4	.94	1.69		
		2.9	1.38	>10	.60
		7.9	1.57		

A_{rms} Acceleration root mean square.
OEM Original equipment manufacturer.

6500/575 AND 6500/577 AIR SPRING SUSPENSION SEATS

Reference Settings

A comparison of transmissibilities, attenuation frequencies, and overall RMS accelerations was made for an Isringhausen 6500/577 air suspension seat loaded with a human, an anthropomorphic dummy, and an inert mass. The air pressure for the comparison tests was 503 kPa (73 psi). Figure 18 presents the transmissibilities of a human, dummy, and 68.0-kg (150-lb) inert mass. Table 7 presents the type of load object, its mass, and various test measures. The test measures in table 7 are transmissibility peaks with corresponding frequencies and overall acceleration RMS levels. Figure 19 presents the results of table 7 graphically.

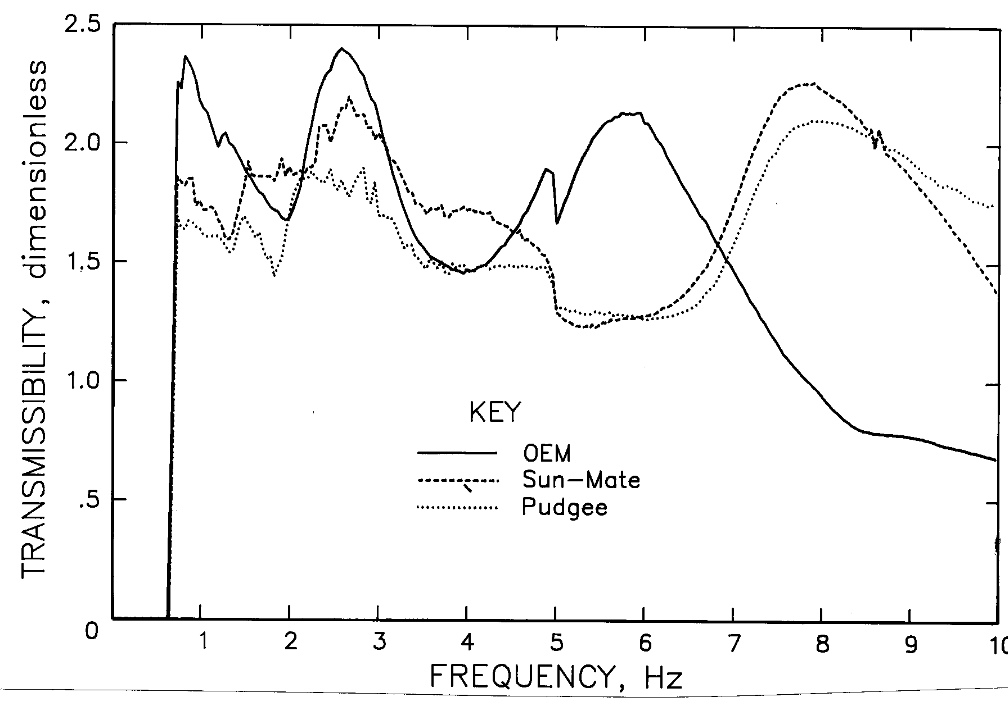

Figure 13.—Transmissibility of seat-base with differing seat cushions for new 6000/575 mechanical suspension, PMR = 13.1.

Table 7.—Comparison of transmissibility peak values and frequencies and overall acceleration RMS between human, anthropomorphic dummy, and inert mass for new 6500/577

Object	Mass, kg	Frequency peak, Hz	Transmissibility peak	A_{rms}, m/s^2
Human	63.6	1.6	0.78	0.19
Dummy	72.6	1.6	.66	.14
Inert mass ..	68.0	1.8	.58	.16

A_{rms} Acceleration root mean square.

From figures 18 and 19 and table 7, the peak transmissibility is greatest for the human. The general shape of the transmissibility curves is similar for the human, dummy, and inert mass. At frequencies greater than 5 Hz, the dummy produces the lowest transmissibility. This is in contrast to the results for the dummy on the mechanical suspension seat (see figure 3). As was the case for the mechanical suspension seat, the air suspension seats were tested with the inert mass of 68.0 kg (150 lb).

Effect of Air Pressure

The Isringhausen 6500/577 air suspension spring was tested for varying air pressures of 393, 448, 503, 552, 621, and 690 kPa (57, 65, 73, 80, 90, and 100 psi, respectively). The resulting transmissibility plots are presented in figures 20 and 21. The summaries of the tests are presented in table 8 and figures 22 and 23.

Table 8.—6500/577 peak transmissibility and overall acceleration RMS for various test pressures

Pressure	Frequency peak, Hz	Transmissibility peak	Attenuation frequency	A_{rms}, m/s^2
393	1.5 3.1	4.12 3.66	4.0	0.61
448	1.2 2.3	1.85 2.38	2.3	.30
503	.9 1.8	.67 .58	(1)	.16
552	1.0 1.7	.65 .58	(1)	.16
621	.9 1.7	.65 .56	(1)	.15
690	.9 1.6	.63 .56	(1)	.15

A_{rms} Acceleration root mean square.
[1]Transmissibility < 1, so there is no attenuation frequency.

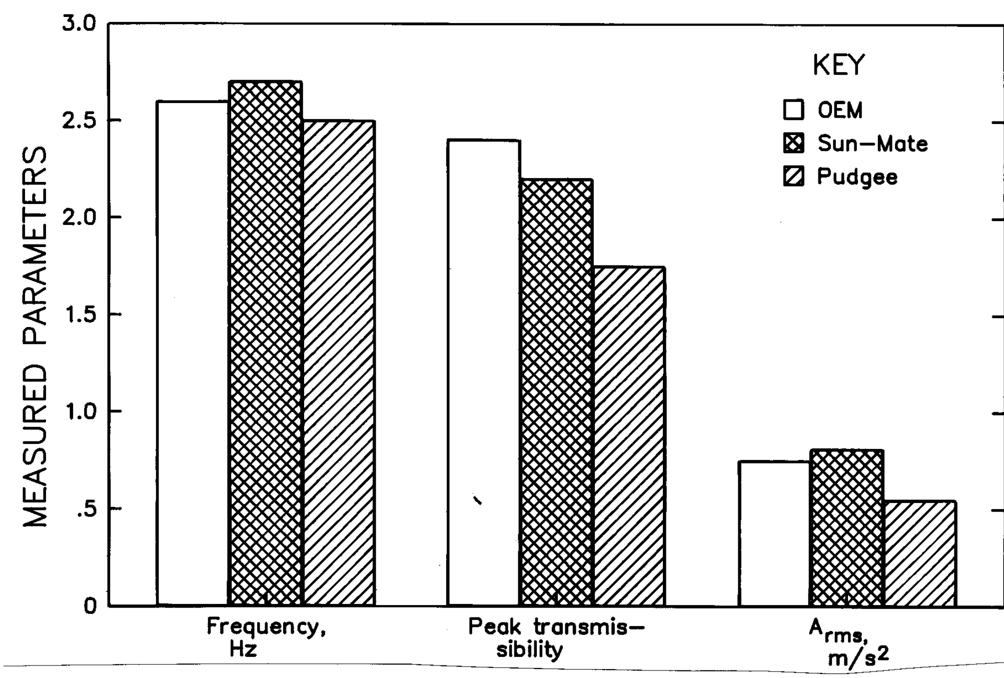

Figure 14.—First transmissibility peak of seat-base and corresponding frequency and acceleration RMS (A_{rms}) with differing seat cushions for new 6000/575 mechanical suspension, PMR = 13.1.

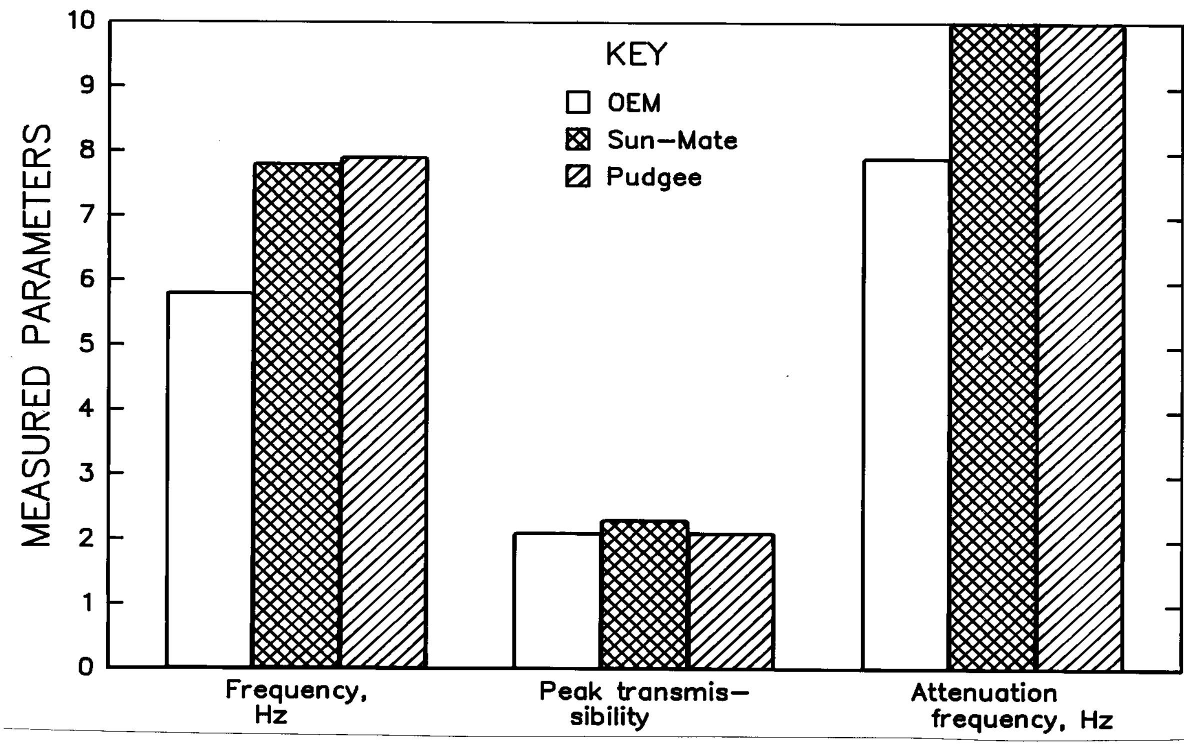

Figure 15.—Second transmissibility peak of seat-base and corresponding frequency and attenuation frequency with differing seat cushions for new 6000/575 mechanical suspension, PMR = 13.1.

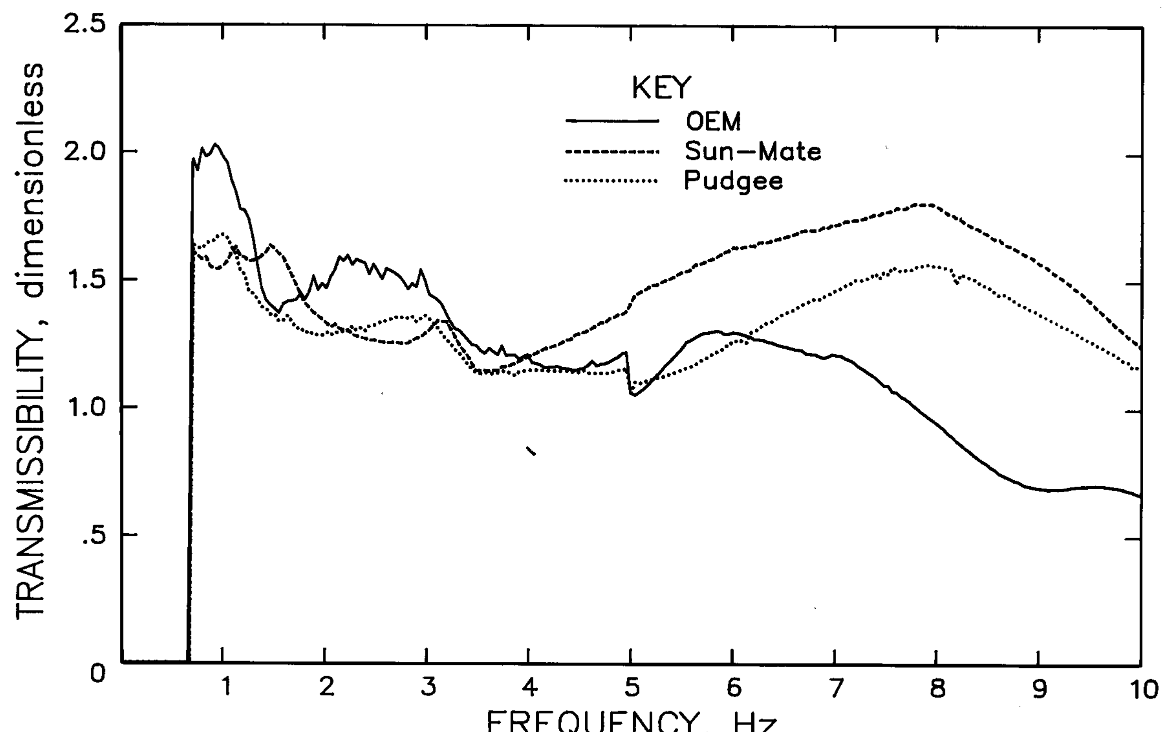

Figure 16.—Transmissibility of seat-base with differing seat cushions for new 6000/575 mechanical suspension, PMR = 9.8.

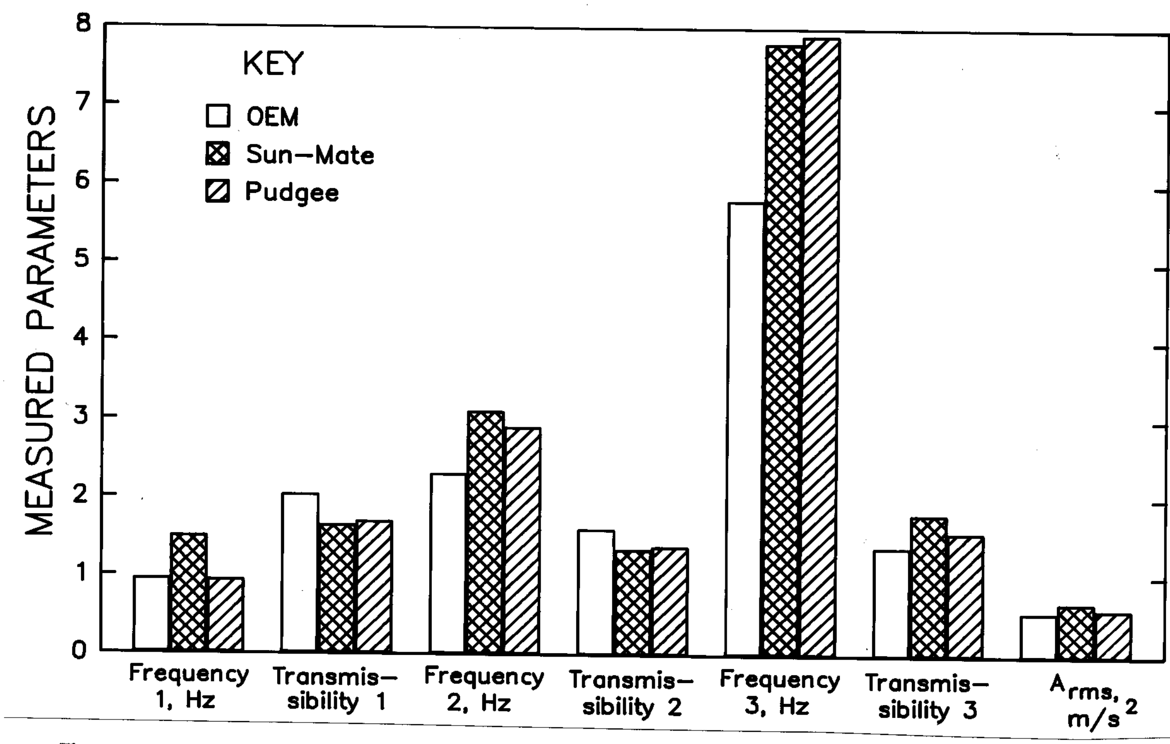

Figure 17.—Transmissibility peaks and frequency of seat-base and overall acceleration RMS (A_{rms}) of seat with differing seat cushions for new 6000/575 mechanical suspension, PMR = 9.8.

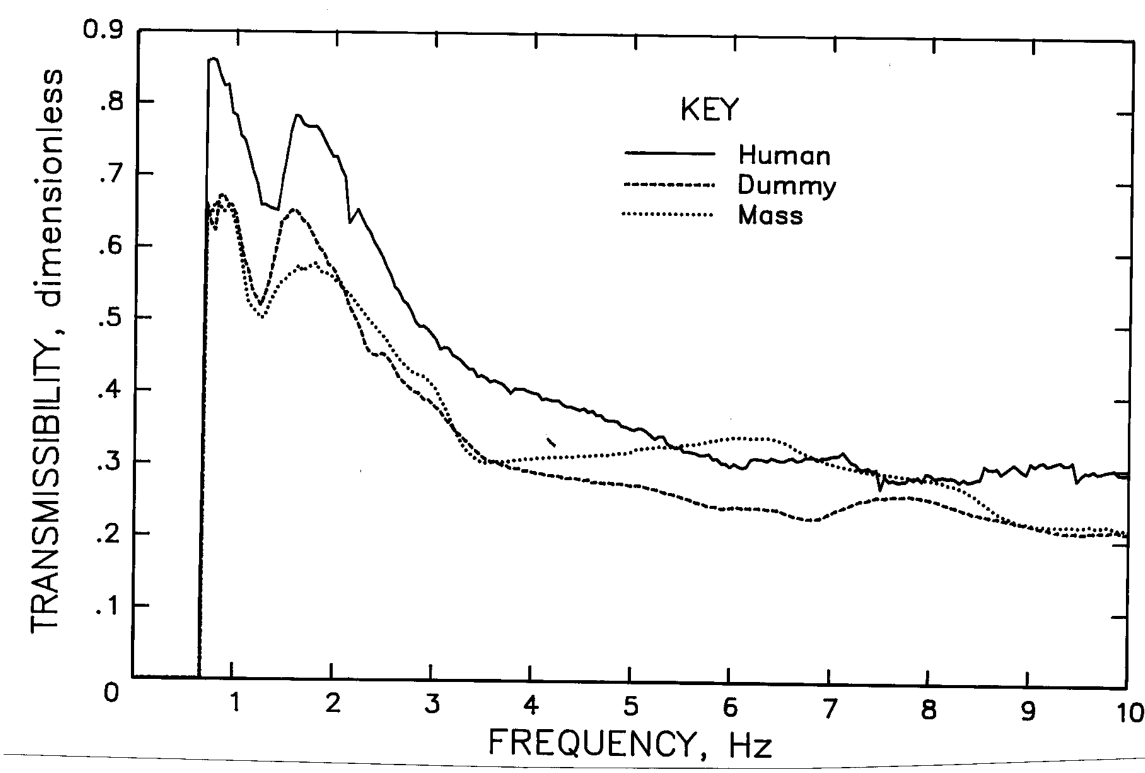

Figure 18.—Transmissibility of seat-base for human, anthropomorphic dummy, and inert mass for new 6500/577 air suspension, 503 kPa.

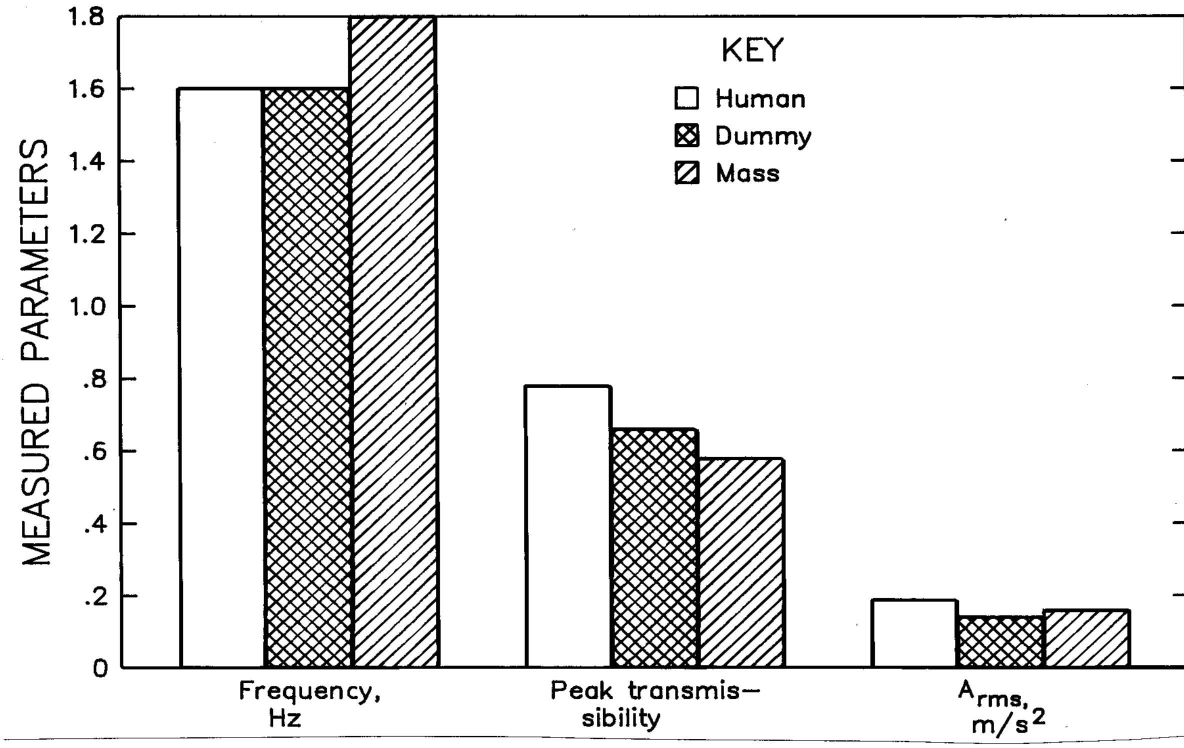

Figure 19.—Transmissibility peaks and frequency of seat-base and overall seat acceleration RMS (A_{rms}) for human, anthropomorphic dummy, and inert mass for new 6500/577 air suspension, 503 kPa.

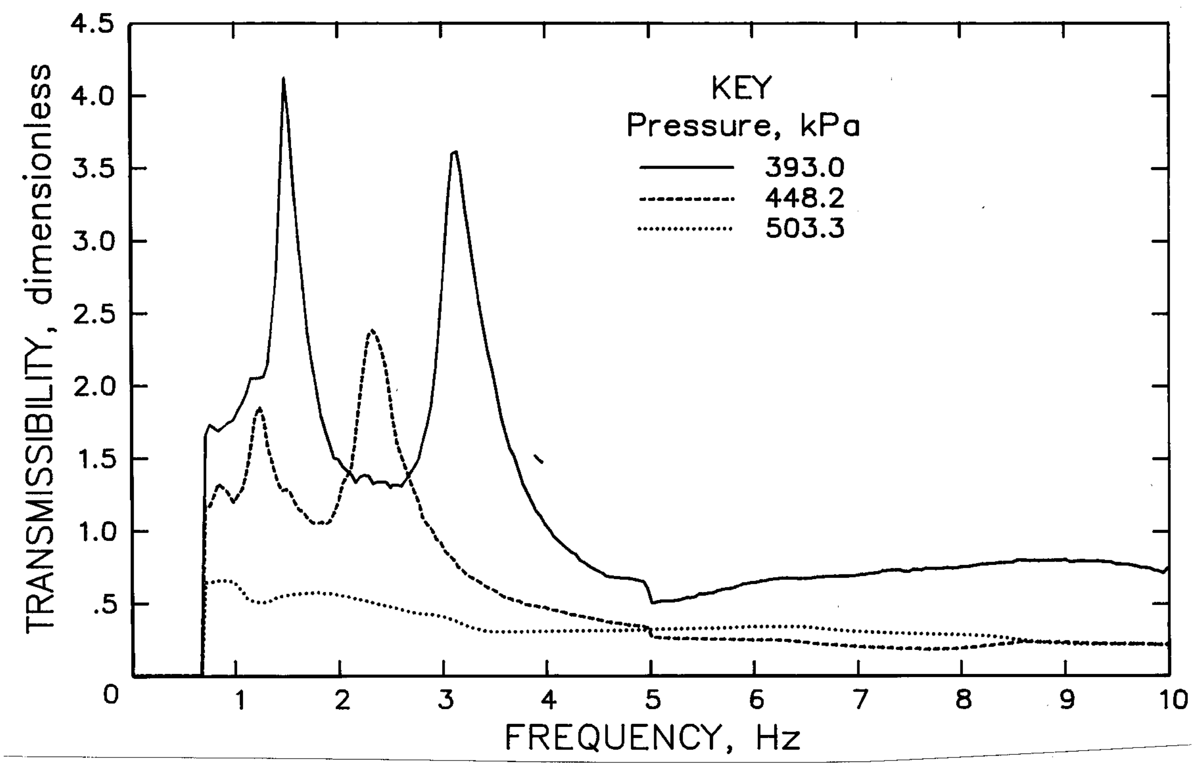

Figure 20.—Transmissibility of seat-base with air pressures of 393.0, 448.2, and 503.3 kPa for new 6500/577 air suspension.

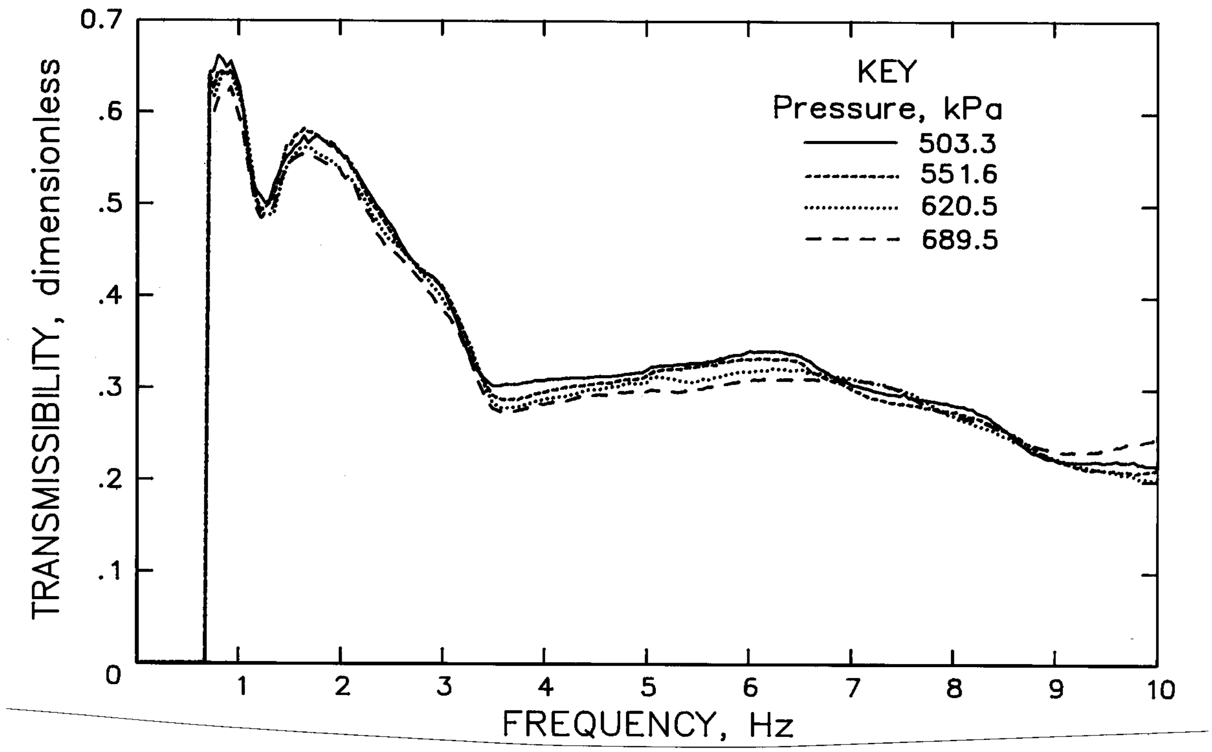

Figure 21.—Transmissibility of seat-base with air pressures of 503.3, 551.6, 620.5, and 689.5 kPa for new 6500/577 air suspension.

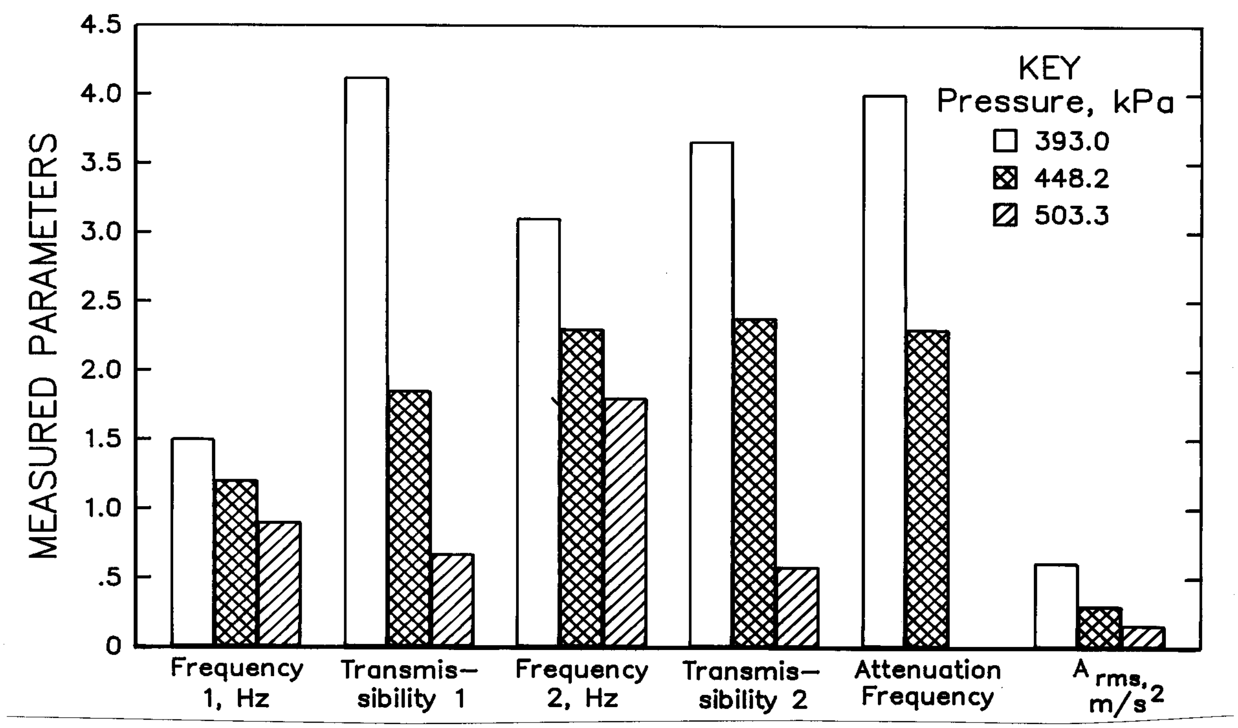

Figure 22.—Transmissibility peaks and frequency of seat-base, attenuation frequency, and seat RMS acceleration with differing air pressures for new 6500/577 air suspension, 68.0-kg mass.

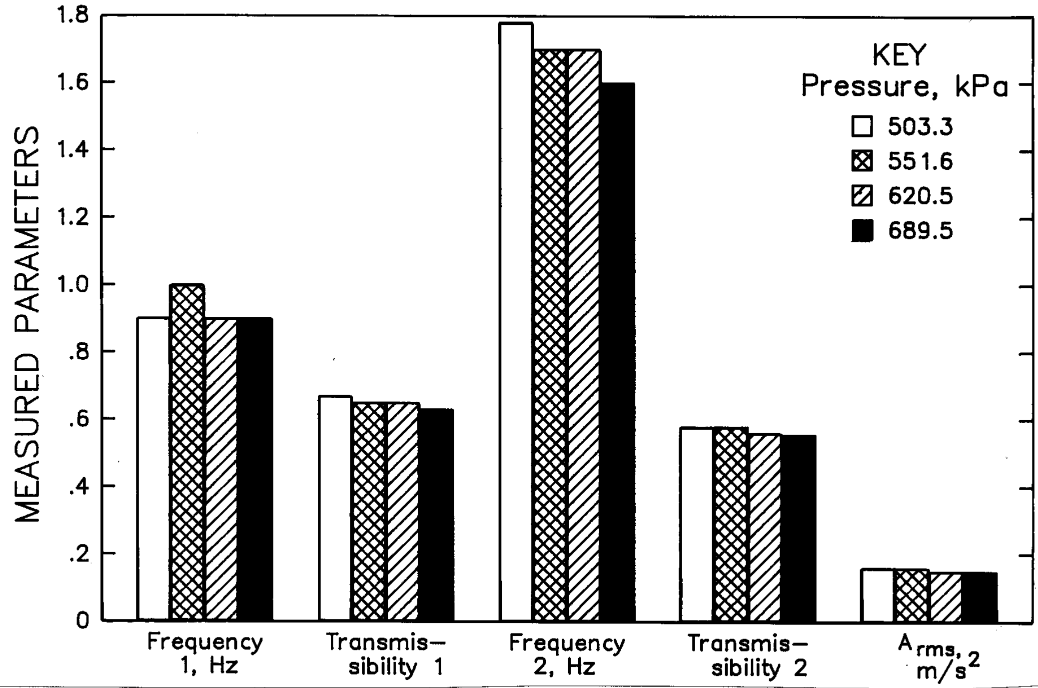

Figure 23.—Transmissibility peaks and frequency of seat-base and overall RMS acceleration with differing air pressures for new 6500/577 air suspension, 68.0-kg mass.

In figure 20, for an air pressure of 393 kPa (57 psi), there exist two extremely large transmissibility peaks. Similar peaks exist in figure 20 for the seat pressurized to 448.2 kPa (65 psi). Although the levels of transmissibility have significantly decreased, they are still considered extremely large. Figures 22 and 23 show the major trends for the effect of pressure. The first and second transmissibility peaks occur at decreasing frequency and are of smaller magnitude for increasing pressure. For the 6500/577 tested at pressures greater than 503.3 kPa (73 psi), the transmissibility never exceeds a value of 1. The overall acceleration RMS level for pressures over 503.3 kPa (73 psi) is nearly a whole order of magnitude less than for 393.0 kPa (57 psi). From figure 21, the seat behaves similarly for pressures between 503.3 kPa (73 psi) and 690 kPa (100 psi). The gain in seat performance by increasing the air pressure above 503.3 kPa (73 psi) is modest; however, higher pressures may be required for actual field usage.

Effect of Cushions

To test the effect of seat cushions on the dynamics of the air suspension 6500/575 and 6500/577 seats, a fixed air pressure of 503.3 kPa (73 psi) and test mass of 68.0 kg (150 lb) was employed.

The transmissibility plots for the air suspension seats are shown in figure 24. The peak transmissibilities, their corresponding frequencies, and overall acceleration RMS's are given in table 9.

Table 9.—6500/575 and 6500/577 peak transmissibility, frequency, and overall acceleration RMS for various cushions

Seat and cushion	Frequency peak, Hz	Transmis-sibility peak	A_{rms}, m/s^2
6500/575:			
OEM	1.0	0.75	0.16
Sun-Mate ...	1.0	.55	.13
	1.5	.53	
Pudgee	1.0	.58	.12
	1.4	.54	
6500/577:			
OEM	.9	.67	.16
	1.8	.58	

A_{rms} Acceleration root mean square.
OEM Original equipment manufacturer.

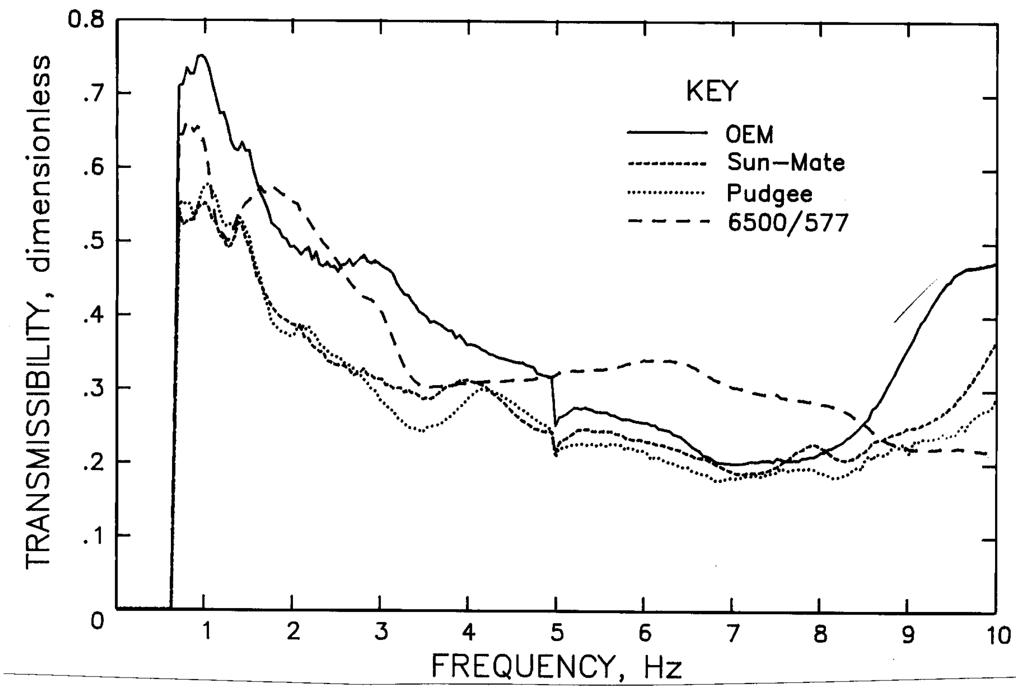

Figure 24.—Transmissibility of seat-base with differing seat cushions for new 6500/575 and 6500/577 air suspension. OEM, Sun-Mate, and Pudgee cushions were tested on 6500/575; 6500/577 was tested using OEM cushion.

The transmissibility peak never attained a value greater than 1 for any air spring suspension and seat cushion combination. In addition, all peak transmissibilities occurred below 1.8 Hz. All seats exhibited a peak transmissibility near 1.0 Hz. The 6500/575 with Sun-Mate and Pudgee cushions and the 6500/577 also had a second peak transmissibility between 1.4 and 1.8 Hz. Seat cushions with increasing densities produced lower overall acceleration RMS values.

CONCLUSIONS

Four off-road vehicle seats were vibration tested using an acceleration test spectrum. The tests were conducted by linearly sweeping the acceleration spectrum in time. The levels of the tests and the measured responses on the seats did not exceed ISO 2631's 4-h, FDP time. The seats tested employed a scissor linkage. Two seats employed a mechanical-coil-spring suspension and two seats employed an air spring suspension. All seats included an oil hydraulic shock absorber. The two mechanical seats tested were of the same model, differing only in length of service. One seat was rebuilt after extensive use in the field, while the other seat was new and sent directly from the factory. The 6500/575 and 6500/577 air suspension seats employed vertical suspensions that were similar in design. The 6500/577 air suspension seat also contained a longitudinal suspension and air chamber seat cushion that was absent for the 6500/575.

MECHANICAL SEATS

For the new mechanical suspension seat with OEM's seat cushion and backrest, the typical response is characterized with three peak transmissibilities. The three transmissibility peaks typically occur in the bands 0.8 to

1.1 Hz, 1.6 to 3.2 Hz, and 5.8 to 6.9 Hz. The mechanical seat dynamic response is dependent on the PMR or operator's weight adjustment. As the PMR decreases from 19.6 to 7.1 N/kg (2 to 0.72 lbf/lbm), the vibration characteristics of the seat improve. For decreasing PMR, the RMS of the seat's peak acceleration (A_{rms}) decreased from 1.03 to 0.48 m/s^2 (3.37 to 1.57 ft/s^2). For decreasing PMR, the peak transmissibility levels are also decreased. The new and rebuilt mechanical suspension seats were tested using three different seat cushions. The rebuilt seat was tested with a PMR equal to 13.1 N/kg (1.33 lbf/lbm). The new mechanical suspension seat was tested with a fixed PMR of 13.1 and 9.8 N/kg (1.33 and 1 lbf/lbm). The seat cushions tested for both the new and rebuilt seats were (1) OEM, (2) Sun-Mate, and (3) Pudgee. The densities of the seat cushions in ascending order are OEM, Sun-Mate, and Pudgee. For a PMR equal to 13.1, the new and rebuilt seats compared similarly for peak acceleration RMS levels and peak transmissibility levels in the 1.6- to 3.2-Hz frequency band. For ascending cushion density, the A_{rms} value for the rebuilt mechanical seat decreased from 0.72 to 0.65 m/s^2 (2.36 to 2.13 ft/s^2). For the new mechanical seat the OEM cushion had lower A_{rms} than the Sun-Mate cushion but greater than the Pudgee cushion. The difference in transmissibility and A_{rms} values between the rebuilt and the new seat may be due to differences in preload setting, suspension spring stiffness, and shock absorber damping. The peak transmissibilities for all mechanical seats and cushions were greater than 1; hence, vibration was not generally attenuated for peak transmissibilities. Moderate attenuation was achieved at higher frequencies ($f > 8.0$ Hz). For the new mechanical seat and PMR equal to 9.8, the OEM cushion had the lowest overall RMS acceleration level of 0.56 m/s^2 (1.84 ft/s^2), while the Sun-Mate cushion had the largest with a value of 0.68 m/s^2 (2.23 ft/s^2). The Sun-Mate and Pudgee cushions had lower transmissibilities than the OEM cushion for frequencies less than 3.5 Hz. The mechanical seat suspension exhibited greater transmissibility than expected. The seats were tested at acceleration levels below the ISO 2631's FDP, 4-h exposure time. Hence, if an operator was exposed to the test vibrations levels while working an 8-h shift, he or she could complete work with little degradation of work performance due to vibration exposure. Because of the nonlinear nature of the response of the chair, the transmissibility could change due to differing vibration levels. Hence, the results appear to give the trends in transmissibility of the seat at low vibration levels. The transmissibility of the seat was higher than expected; possible contributing factors are as follows: (1) at the acceleration levels tested, the legs of a person may limit the vertical motion of the seat, prevent rocking, and provide damping, while an inert mass provides none of these; and (2) the suspension may have been constructed to provide attenuation at acceleration levels that can cause serious health risks. Hence, at the acceleration levels tested, which were generally low, the suspension system may operate differently than when acceleration levels are present that can cause harmful affects to the operator. The seat should be adjusted to a preload setting of the operator's weight. This setting would position the chair at its mid-ride position. A chair positioned at the mid-ride position reduces the operator's risk of receiving high acceleration levels due to the seat's hitting the limit stops during large shocks.

AIR SUSPENSION SEATS

The 6500/577 air suspension seat was initially tested with varying levels of air pressure from 393 to 690 kPa (57 to 100 psi). For the pressure range of 393 to 503 kPa (57 to 73 psi), the response of the air suspension seats appears to be extremely sensitive to air pressures. For increasing air pressures of 393 to 503.3 kPa (57 to 73 psi), A_{rms} decreased nearly an order of magnitude from 0.61 to 0.16 m/s^2 (2.0 to 0.52 ft/s^2). In addition, the seats pressurized at 393 and 448.2 kPa (57 and 65 psi) exhibited two large transmissibility peaks in the frequency bands of 1.2 to 1.5 Hz and 2.3 to 3.1 Hz. For seats pressurized from 503.3 to 690 kPa (73 to 100 psi), the transmissibility never attained a value greater than 1. Hence, the 6500/577 pressurized greater than 503.3 kPa (73 psi) attenuated vibrations for all frequencies. There was little difference in overall acceleration RMS level due to increasing air pressure from 503.3 to 690 kPa (73 to 100 psi).

As with the mechanical seats, the 6500/575 and 6500/577 air suspension seats were tested with different seat cushions. For the cushion comparison tests, all seats were pressurized to a level of 503.3 kPa (73 psi). The conducted tests were 6500/575 with OEM, Sun-Mate, and Pudgee cushion and backrest, and 6500/577 with OEM air chamber seat cushion and longitudinal suspension. For all cushion combinations tested the maximum transmissibility occurred for frequencies between 1.0 and 2.0 Hz. For cushions of increasing density, the peak transmissibility and A_{rms} level decreased.

The performance of the 6500/577 seat quickly deteriorates as the air pressure decreases. An air suspension seat inflated to pressures greater than 503.3 kPa (73 psi) provides a stiffer ride, hence decreasing the chances of the seat's hitting limit stops when installed for actual use in an off-road vehicle. The response of a simple single degree of freedom linear spring mass system exhibits a peak transmissibility greater than one at the system's natural frequency. Above the natural frequency, the transmissibility decreases. The actual air suspension measurements at pressures greater than 503.3 kPa (73 psi) have peak transmissibilities that are less than 1. These results appear to be fairly consistent with the results obtained using a person, dummy, and inert mass as the load on the seat. At the vibration levels tested, the air suspension seat pressurized above 503.3 kPa (73 psi) provided better vibration attenuation than the mechanical suspension.

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