Another Look at Skidder Ride Vibration. . .

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ABSTRACT

Whole-body vibration data from recent tests on several log skidders are presented. Weighted RMS (root mean square) and RMQ (root mean quad) values are compared to FERIC results of 1987. The PSD (power spectral density) of the vertical motion of the seat base is compared with the PSDs currently defined in ISO and SAE Standards for the evaluation of seat suspensions on earthmoving equipment and agricultural tractors.

This paper suggests the adoption of a seat performance test such as those contained in ISO 7096/SAE J1385 for earthmoving equipment, or in ISO 5007/SAE J1386 for agricultural tractors. Seat suspensions capable of meeting the requirements of such a test will attenuate the vibration at the frequencies most prevalent on skidders, and should significantly improve the ride on these machines.

Keywords: *Whole body vibration, skidders, seat suspensions.*

INTRODUCTION

The log skidder has a reputation as being perhaps the roughest riding of the forestry machines. Published reports have documented the vibration levels on skidders, and such data indicates the existence of a problem. However it does not provide much help in defining the isolation characteristics required of a skidder seat suspension, or in defining tests to ensure that the suspension is capable of a reasonable level of vibration attenuation in the skidder application.

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RIDE VIBRATION ANALYSIS

RMS Calculations

People like to have "one number" when making comparisons. For ride vibration analysis, the RMS value of the signal is that "one number" that we usually calculate and report. To obtain the RMS value, the signal is squared, then the average or mean taken of this squared signal. The square root of the mean is the RMS value. However, simplifying the results to one number can hide a lot of information that is desirable when trying to develop or understand the performance of a seat. This should be more apparent later.

Frequency Weighting

The human body is not equally sensitive to all frequencies of vibration. A vibration level of 1.0 m/s^2 at a frequency of 10 hz would be much less bothersome than the same level at a frequency of 1 hz. In an attempt to account for this variation in the sensitivity of a human being with respect to the frequency of the vibration, the International Standards Organization (ISO) has adopted a standard, ISO 2631 [4], which defines the manner in which the vibration at various frequencies should be weighted in order to more closely approximate human sensitivity.

In ISO 2631, two such weighting curves are defined. One is for the vertical direction, and one for the fore-aft and lateral directions. Plotted on a log axis, these functions are straight lines.

For the vertical vibration, the human is most sensitive in the 4-8 hz range, and the weighting function has a value of 1.0 in that region. As we move away from this 4-8 hz region, the human becomes less sensitive, and so the weighting decreases.

In the lateral or fore-aft direction, the human is most sensitive in the 0-2 hz range, and the weighting function has a value of 1.0 in that region. As the frequency increases above 2 hz, the human becomes less sensitive, and the weighting decreases. Revised versions of these weighting curves have been included in a number of other standards. Fig. 1 compares these weighting functions for vertical whole-body vibration as defined in the original ISO 2631, in ISO 8041 [8], and in BS 6841 [1]. It appears that revisions to ISO 2631 [5], now in the draft stage, will make its weighting curves the same as in BS 6841.

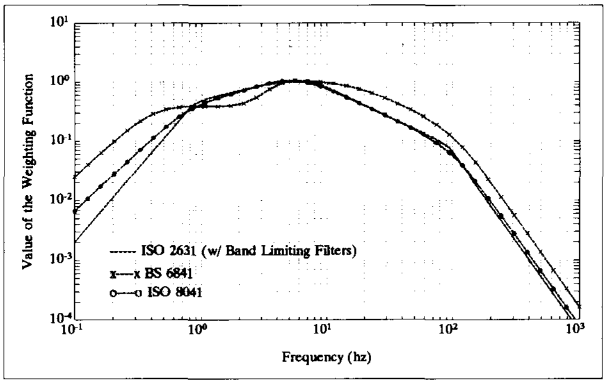

Figure 1: Comparison of vertical weighting functions.

In accordance with the intent of these standards, one way to use these weighting curves is to pass a vibration signal through a filter having the weighting characteristic appropriate for the direction of the vibration. The RMS value of this weighted signal can then be determined as noted above, and results in what is known as the Weighted RMS value. This is, again, a one-number characterization of the vibration which includes the overall effects of the human sensitivity.

The Weighted RMS can also be determined from the spectrum of the vibration signal. To apply the weighting in this manner, a *power* spectrum of the vibration is computed via the FFT process. At each frequency in the spectrum, the amplitude of the vibration is multiplied by the magnitude squared of the weighting function at that same frequency. These weighted amplitudes are summed to get the total weighted power, and the square root of that sum is the Weighted RMS.

Crest Factor

The Crest Factor is defined as the ratio of the maximum value of the signal in some time interval to the RMS value of the signal over that same interval. The Crest Factor is a measure of how much the peak value deviates from the normal value of the signal. Within the context of ISO 2631, BS 6841, and similar documents, the use of the Weighted Crest Factor is to suggest when the use of the Weighted RMS may or may not be appropriate.

LABORATORY SEAT TESTING

For agricultural and earthmoving equipment, the use of a laboratory test stand has all but replaced on-vehicle testing for the certification and homologation of seats and seat suspensions. The use of a test stand as a means of certifying or qualifying a seat is far less expensive than developing and maintaining a special track and eliminates a myriad of variables such as vehicle weight, wheelbase, travel speed, and tire definition (type, size, pressure, etc.). Essentially, the use of the test stand greatly improves the repeatability and reliability of the test.

In the off-road environment, vertical vibration tends to be most severe, and standards have been developed to evaluate the dynamic performance of a seat and seat suspension in this direction. Both ISO and SAE have equivalent standards toward this end [6, 7, 10, 11]. For both agricultural and earthmoving machines the intent was to group machines having similar vibration spectra into the same class. Unfortunately, none of the standards include skidders in any of the defined classes.

The standards take a common approach in that they define the RMS value, the weighted RMS value, and the power spectral density of the vibration signal that is to be inputed to the seat of the base on the test stand. In order to "pass" the test, the Weighted RMS level at the man-seat interface must be less than 1.25 m/s^2 using the input for the class of vehicle to which the seat will be applied. These input spectra were generated by comparing the measured spectra for a number of machines in each class and then defining the characteristics of a physically realizable analog filter whose output would closely approximate the measured spectra when the input to the filter is random noise.

The seat and seat suspension must pass the test with both heavy (98 kg) and light (55 kg) operators.

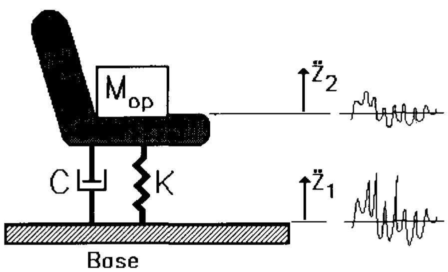

Figure 2: Simplified model of a seat, seat suspension, and operator.

FACTORS AFFECTING SEAT PERFORMANCE

Figure 2 shows a model of a seat and seat suspension. It consists of the seat base, the uppers, the damper and linkage mechanism.

The seat base is attached to the floor of the cab or operator station. The "uppers" consists of the seat cushion, back, armrests, etc. In this simple model, it also includes the weight of the operator. The uppers are supported from the base through a spring and a damper. The spring could be either a coil spring or air bladder mechanism. The damper is usually a shock absorber of some sort, and generally operates by trying to force oil through a small orifice, the size of which may be fixed or adjustable. There is also some linkage between the base and the uppers to restrict the motion of the upper to essentially a vertical path only. Except for friction, the linkage should not introduce any net forces between the base and uppers in the vertical direction.

The vibration of the vehicle gives the base a vertical motion. Some of this motion is transmitted through the spring (K) and damper (C) to the uppers, causing them to move also. One way to characterize the behaviour of a suspension is to determine its FRF, or Frequency Response Function. This again is a function of frequency which tells us at any frequency, how much of the base acceleration is transmitted to the top. Basically, it is measured by dividing the spectrum for the acceleration on the uppers by the spectrum for the acceleration on the base.

SKIDDER FIELD TESTS

In the fall of 1991, ride evaluations on four grapple skidders were conducted during operations on well-separated sites in Alabama and Georgia. The skidders tested were a Caterpillar 518, two John Deere 648E's (two different machines at two different sites), and a Timberjack 450B. The order in which the machines are identified here is arbitrary and does not correspond to subsequent references herein as machines A, B, C, and D.

All the machines were less than one year old and were operated by logging contractors in their normal operations. All data were acquired using the contractor's operators during these normal operations. The machines were of similar size and were similarly equipped, except for much larger tires on one of the units. All of the machines had suspension seats, although the seat suspension in machine D exhibited considerable looseness in the lateral direction because of excessive wear in the suspension tracks and rollers.

Three of the machines were tested using a single operator for that machine. With machine A, data were obtained for four operators doing essentially the same operation.

In general, the operations were similar. All machines were skidding a mixture of 90% pine and 10% hardwood. Typical loads consisted of 6 - 10 tree-length logs. The terrain was somewhat hilly, more so for machines C and D. All machines worked in conjunction with feller-bunchers. After pulling their loads in from the felling site, machines A, B, and D pushed their loads through a delimbing gate and then pulled them to the loading site, dropping them within reach of a grapple loader. On the machine C site, a separate delimbing machine was used, and machine C had to pull its load to within reach of the boom of the delimber before dropping its load and returning to the felling site.

DATA ACQUISITION

The vibration data were acquired as time histories of all the active accelerometer channels. The data acquisition system was a Deere developed system, consisting of an integrated package with signal conditioning for the transducers, anti-aliasing filters, an analog-to-digital converter, mass memory for data storage during acquisition, a 3.5" floppy for down loading the mass memory, and a microcomputer to control it all.

For all the tests, the vibration levels at the man-seat interface were measured in the fore-aft, lateral and vertical directions. With only these 3 channels active, we could record continuously for 30 minutes before the mass memory was filled. However, for

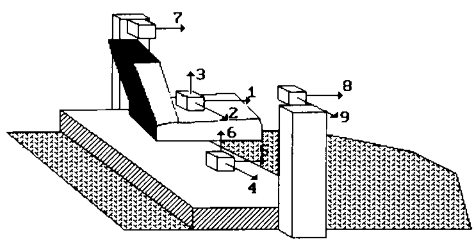

Figure 3: Typical installation of accelerometers on cab and seat.

most tests, signals from another 6 accelerometers mounted on the cab were also recorded. Three of these were in a triaxial configuration as close to the base of the seat as possible; the remaining 3 were mounted at 2 other locations as shown in Figure 3. With this configuration, the fore-aft, lateral vertical, pitch, roll, and yaw motion of any point on the cab could be determined, assuming the structure is a rigid body (which is usually a good assumption up to at least 10 hz). However, in this 9-channel mode, only 10 minutes of data acquisition was possible. This corresponded to roughly 2 cycles of skidder operations.

DATA ANALYSIS

All of the data analysis was done digitally. Much of the processing was performed using a commercial software package called MATLAB386. This software allows one to generate the filters for the frequency weighting of the signals, and makes it easy to calculate spectra, as well as the RMS, Weighted RMS, and other parameters.

TEST RESULTS

Golsse and Hope [2] have measured skidder vibration levels on a number of skidders, and we wanted to compare our results to theirs where possible. They reported their results for segments of the work cycle and presented data on the percentage of time spent in each of the work segments. To make their results more comparable to our results (which were taken over a mix of functions), the expected composite level based on the level and time in each segment was calculated. These calculated results are also shown in Figures 4-6 and are identified as follows:

- G,G, wS - Golsse, Grapple skidder, with Suspension seat
- G,C,woS - Golsse, Cable skidder, without Suspension seat
- G,C,wS - Golsse, Cable skidder, with Suspension seat

Cushion-Weighted RMS Levels

Figures 4-6 show the averages and the ranges of the Weighted RMS levels at the cushion in the fore-aft, lateral and vertical direction (respectively) for the four machines we tested, and for the three classes of machines in the Golsse-Hope study. For our data, the number of runs upon which the statistics are based are shown. For the Golsse-Hope study this number is a range, since they did not report equal numbers of tests for each segment of skidder operation.

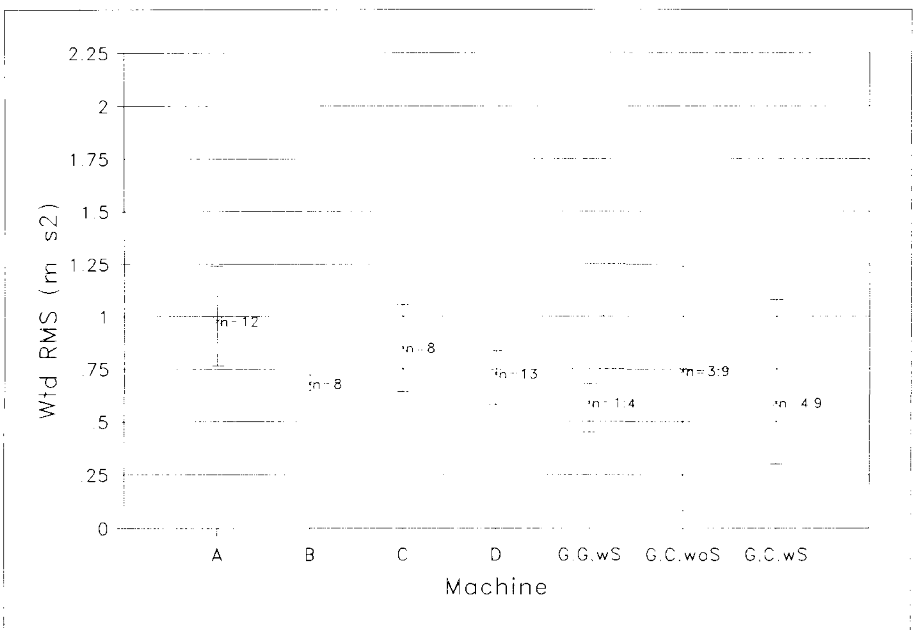

Figure 4: Weighted RMS values, fore-aft on cushion (weighted per ISO 2631).

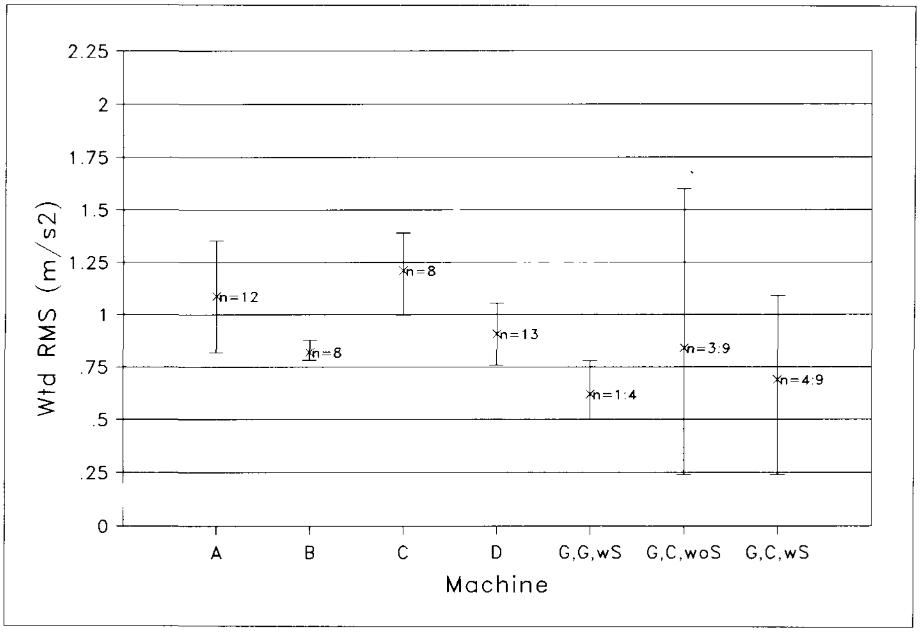

Figure 5: Weighted RMS values, lateral on cushion (weighted per ISO 2631).

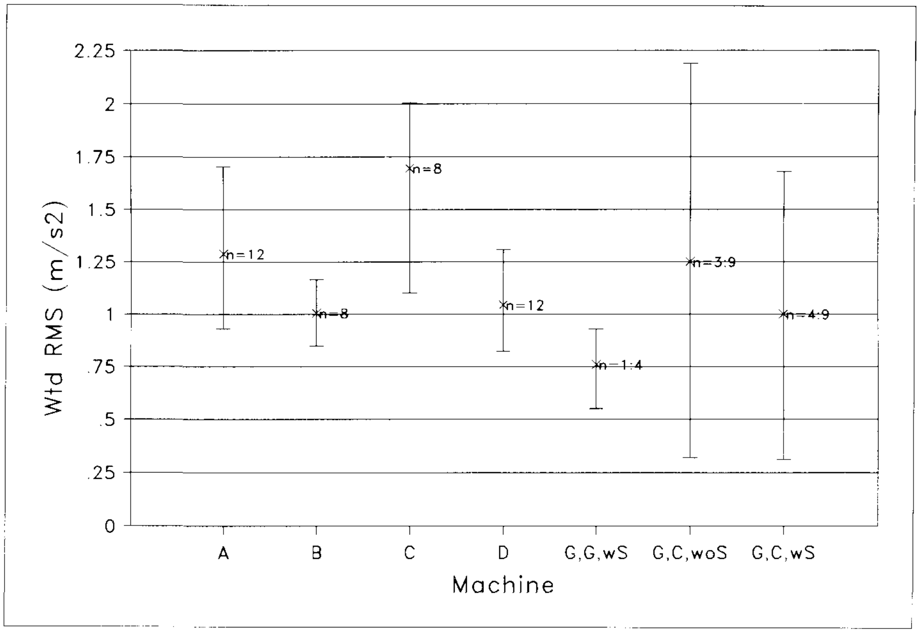

Figure 6: Weighted RMS values, vertical on cushion (weighted per ISO 2631).

As shown in these figures, our *average* levels tended to be higher than the Golsse-Hope data. The range of our data was less than the range they measured on cable skidders and as a percent of the mean was similar to the range of their grapple skidder data.

In the lateral and vertical directions, machine C had the highest levels of the four machines tested. This is believed to be a result of a somewhat more aggressive operator than on machines B and D, a slightly different cycle (little or no time at reduced speeds in the delimbing area), and larger tires operated at relatively high pressures resulting in a stiff vehicle suspension with less "enveloping" of stumps and logs by the tires.

Crest Factors

The Weighted Crest Factor is determined by dividing the maximum peak value of a weighted acceleration signal over some interval by the RMS value of that weighted signal over the same time interval. Initially, ISO 2631 suggested that when the Weighted Crest Factor exceeds 3.0, it might not be appropriate to use the Weighted RMS values to assess the affect of the vibrations on humans. This value was later increased to 6.0.

Table 1 shows the range of Weighted Crest Factors measured in our system, along with the values reported in the Golsse-Hope study. The range of our data is consistent with the Golsse-Hope data.

One of the problems with the Crest Factor is that its significance tends to be viewed incorrectly. Golsse et al. stated that in their study the "CF's measured exceeded this limit (3.0) which suggests that the vibration levels are extremely severe." However, Crest Factors are merely a ratio, and high values of that ratio result whenever the peak value is high relative to the RMS value. A moderate peak value combined with a very low RMS value can have a high crest factor, a moderate peak value combined with a moderate RMS value would have a lower Crest Factor, but the latter is probably less desirable or tolerable. Another problem with the Crest Factor is that the time interval over which it is calculated is not specified. A one-time large peak in the midst of a long and otherwise reasonable vibration signal results in a high Crest Factor being assigned to the whole interval.

Effect of Operator on Observed Levels

As noted earlier, we were able to acquire data for four operators on machine A. When taking these data, notes were made about the aggressiveness of the operator. Based on these observations the operators were ranked from least aggressive (=1) to most aggressive (=4). The vibration levels at the cushion in 3 directions are shown in Figure 7 as a function of that subjective ranking. The vibration levels at the cushion in the vertical direction were 65% higher for the most aggressive operator as compared to the least aggressive operator. This clearly shows the major effect the individual operator

Table 1. Weighted Crest Factors.

Machine	Fore-Aft Direction	Lateral Direction	Vertical Direction
A	5.88 - 9.76	6.11 -12.25	5.95 -15.31
B	6.26 - 10.01	6.26 - 12.31	7.15 - 14.9
C	5.61 - 10.78	5.18 - 11.86	6.62 - 10.77
D	5.87 - 11.19	5.77 - 9.30	8.07 - 20.48
Grapple (*Golsse-Hope*)	6.35	7.46	7.18
Cable w/o Suspension (*Golsse-Hope*)	7.73	6.95	5.05
Cable /w Suspension (*Golsse-Hope*)	8.33	8.79	7.43

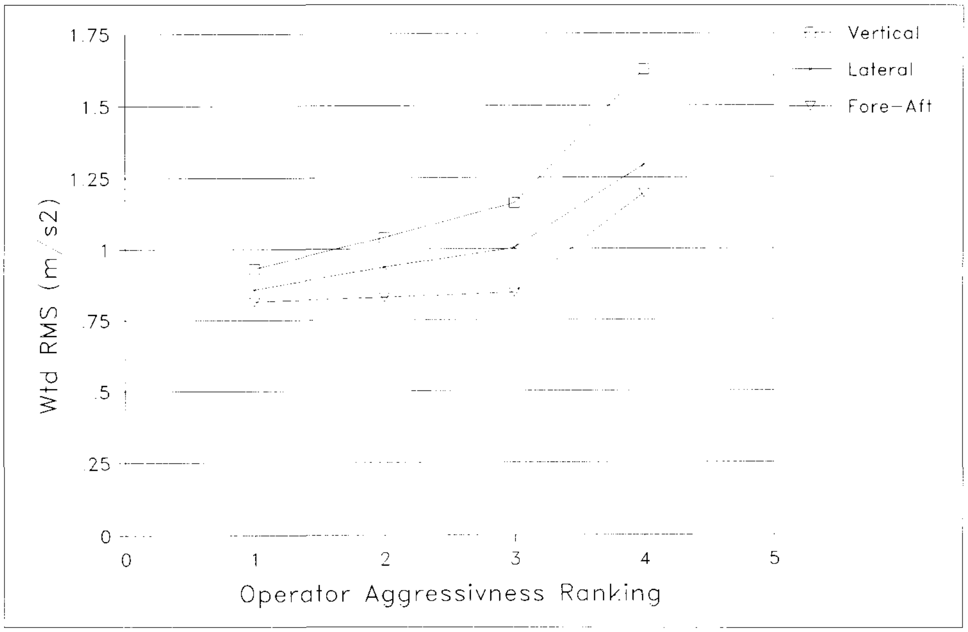

Figure 7. Cushion vibration vs. operator ranking (levels adjusted for equal run times)

can have on his/her own vibration exposure. It also underscores the difficulty the machine manufacturer would face in any attempt to have them certify the whole-body vibration levels experienced on their machines under realistic working conditions.

Effect of Multiple Inputs

In our analysis, we determined the vibrations at the seat base and at an imaginary point affixed to the cab but located at the mean position of the man-seat interface (these calculations are possible using these data from the six accelerometers in the cab). Typical of what we find on other off-road machines, the levels to which the operator is exposed in the fore-aft and lateral directions at the man-seat interface are greater than those measured at the seat base.

It was found that the fore-aft levels increase as one moves higher due to the pitch motion of the vehicle. Thus, the fore-aft vibration of the cab at the height of the cushion more correctly represents the vehicle input to the cushion, and is greater than that measured at the seat base. The levels measured on the cushion are still greater than those on the cab at this height, and these differences are generally due to fore-aft deflections of the seat cushion and suspension mechanism. This also occurs for the lateral direction, with the increased vibration due to the rolling motion of the vehicle.

A similar comparison may be made for the vertical direction. If the two points on the cab are directly above one another, the cab roll and pitch rotations do not result in any increase in the vertical direction when moving from base height to seat height. Thus in the vertical direction, there is a temptation to assess the suspension attenuation characteristics by comparing the levels on the base with those measured on the seat cushion.

A number of studies of whole-body vibration in off-road machines have done this, and have observed that suspension seats may amplify, rather than attenuate, the vertical input to the operator. While this is certainly true if the suspension parameters are not correct for the machine and the operator, there may be another significant factor at work here. In off-road operations, the input to the base of the seat is a complex and simultaneous set of linear motions in the fore-aft, lateral, and vertical directions, as well as rotations in the roll, pitch, and yaw directions. As a result, the operator's motion is also complex and "cross-coupled"—strong lateral and/or fore-aft motion of the operator can and will cause vertical motion of the man-seat interface. In such a multi-input environment, incorrect conclusions about the output/input characteristics along a single axis may result from input and output measurements on that axis alone.

Nevertheless, we calculated the Frequency Response Function (FRF) between the seat base vertical

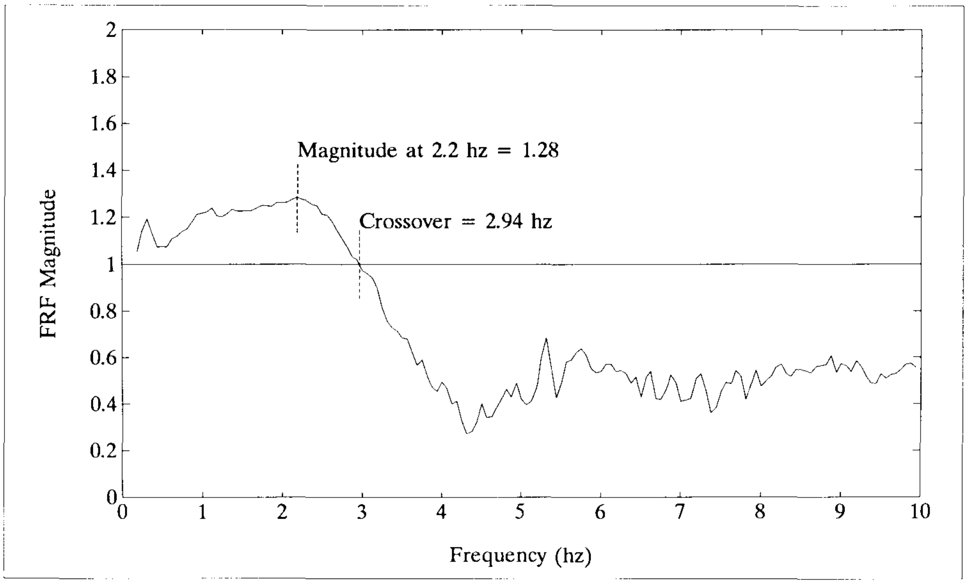

Figure 8: Frequency response of vertical seat suspension, Skidder B.

acceleration and the seat cushion vertical acceleration for each of the test machines. The magnitude of the FRF for machine B is shown in Figure 8, and is typical of the results on all four machines. The crossover frequency was found to be about 2.94 hz, which is too high to obtain good vibration attenuation in the skidder application.

Effect of Weighting Function

There are some differences between the shapes of the various functions used to determine the Weighted RMS (see Figure 1). The differences in these functions are most pronounced for the vertical direction. BS 6841 provides for a lower value of the weighting function below 4 hz and a higher value above 8 hz than does ISO 2631 or ISO 8041. This is significant when dealing with the skidder vibration signals, since the spectrum for the vertical vibration shows the vibration energy in this direction is concentrated in the 1.5 to 3 hz region. A comparison of weighting curves in this frequency range suggests that Weighted RMS values determined using the weighting in BS 6841 will be about 30% lower than with the ISO 8041 or the old ISO 2631 weighting.

RMQ vs. RMS

The intent of the Weighted Crest Factor is to suggest when an alternate method of assessing the effect of the vibration on humans may be desirable. BS 6841 and other standards recommend the use of the RMQ (Root Mean Quad) value instead of the RMS (Root Mean Square) value when the Weighted Crest Factor is high. The RMS and RMQ values are defined as follows:

$$\mathrm{RMS} = \left[\frac{1}{T}\int_{O}^{T} a(t)^2\,dt\right]^{\frac{1}{2}} \tag{1}$$

$$\mathrm{RMQ} = \left[\frac{1}{T}\int_{O}^{T} a(t)^4\,dt\right]^{\frac{1}{4}} \tag{2}$$

Because of the 4th power, the RMRMQ value puts greater emphasis on the peak values within the signal. Figure 9 compares the RMQ values to the RMS values (both weighted per BS 6841). For the skidder data, the two terms are related by

$$\mathrm{WTD\ RMQ} = 1.88 * \mathrm{WTD\ RMS} \tag{3}$$

A similar high correlation between RMS and RMQ was also noted by Monsees et al. [9].

A proposed revision to ISO 2631 notes that the Crest Factor is an uncertain method of deciding whether RMS acceleration can be used to assess human response to vibration. In case of doubt, the document recommends that the Normalized Vibration Dose Value (NVDV) calculated from the Weighted RMS be compared to the NVDV calculated from the weighted RMQ. If the difference between these two values is less than 50%, the RMS-based procedure can be used.

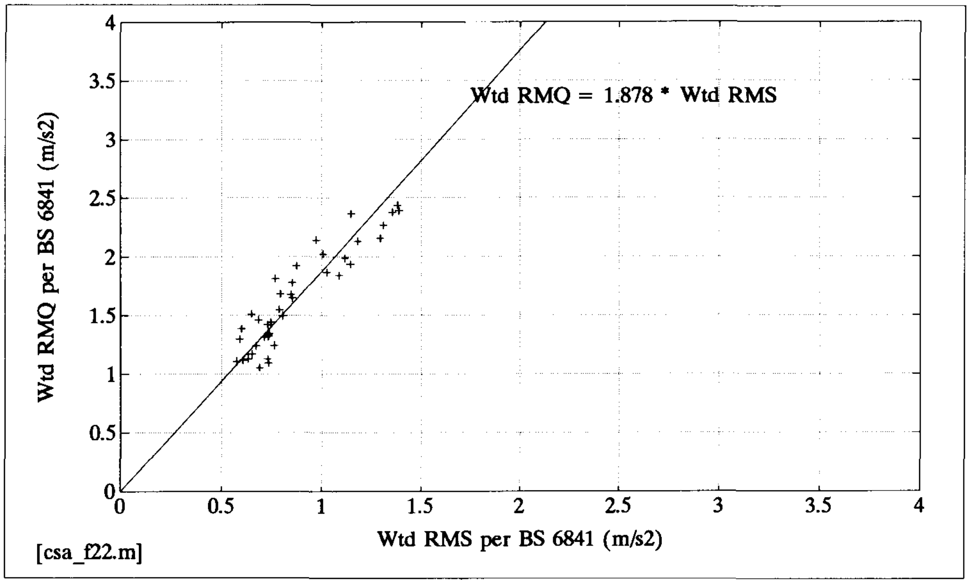

Figure 9: Weighted RMQ and RMS values for vertical direction, BS 6841 weighting.

Using the RMS procedure,

$$NVDV_{rms} = 1.4 * WRMS * \left(\frac{T}{T_8}\right)^{\frac{1}{4}} \quad (4)$$

Using the RMQ procedure,

$$NVDV_{rmq} = WRMQ * \left(\frac{T}{T_8}\right)^{\frac{1}{4}} \quad (5)$$

where

WRMS = Frequency Weighted RMS acceleration
WRMQ = Frequency Weighted RMQ acceleration
T = exposure duration in seconds
T_8 = 28800 seconds (8 hours)

The value of 1.4 is an empirical constant, presumably selected so that the two VDV values are equal for some assumed "typical" vibration signals. For our data, the more correct values appeared to be about 1.9 (see Eqn. 3 and Figure 9).

The ratio of the two Normalized Vibration Dose values is then

$$\frac{NVDV_{rms}}{NVDV_{rmq}} = \frac{1.44\ WRMS}{WRMQ} = \frac{1.4 * WRMS}{1.88 * WRMS} = \frac{1.4}{1.88} = 0.745 \quad (6)$$

Since the values are only 25% different, RMS-based procedures should be acceptable using the guidelines in the proposed revision to ISO 2631.

Seat Test Spectra

As described previously, the seat test standards (ISO 7096, SAE J1385, ISO 5007, and SAE J1386) define the vertical input to the seat in terms of the PSD (Power Spectral Density) of that signal. If a procedure similar to that of these standards is to be used to certify skidder seats, a PSD that is representative of the skidder vertical vibration environment is needed.

Towards this end, the average PSD of the seat base vertical acceleration was computed for each of the five machine and operator combinations tested. The "envelope" PSD (which is mainly the spectrum for machine C) and the overall average PSD based on equal weighting of the PSD's from each of the five machine and operator combinations was also found. These PSD's were compared with the PSD functions prescribed for the 4 classes of earthmoving machines in ISO 7096/SAE J1385, and the 3 classes of agricultural tractors in ISO 5007/SAE J1386. It was found that the Class 3 ISO 5007 Ag tractor not only offers the best match to the skidder data, it offers a very good match (see Figure 10). Passing this test with both a heavy and light operator would require the seat suspension to attenuate the input levels at least 4% (from 1.3 m/s^2 to less than 1.25 m/s^2). Obtaining

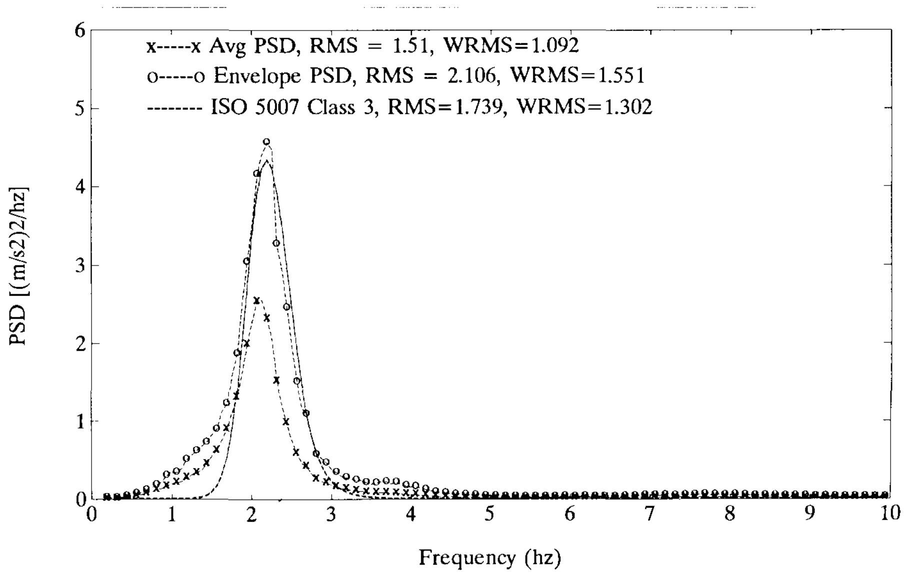

Figure 10: PSD for Seat Base Vertical Accelerations—Machines A1, A4, B, C, and D.

a 4% reduction instead of the apparent 28% amplification as seen in Figure 8 would result in an overall reduction in excess of 30%.

CAN WE REACH ACCEPTABLE LEVELS?

For off-road machines like the skidder, the two main concerns regarding whole-body vibration have been with respect to the effect of that vibration on operator fatigue and on operator health.

ISO 2631 offers a "fatigue-decreased proficiency boundary" and goes on to state that the frequency-dependence and time-dependence of that boundary are "commonly observed," and that the data comes mainly from studies of aircraft pilots and drivers. Yet, as Griffin [3] points out, no published scientific basis for the boundary is given. Griffin goes on to say that "the concept of fatigue-decreased proficiency is not sufficiently well defined to be useful and that the occurrence of time dependent changes induced by vibration are not yet well understood." ISO 2631 also defines an "exposure limit," which, for the vertical direction, varies from 54.6 m/s^2 for durations of 1 to 4 minutes to 0.28 m/s^2 for a 24-hour duration. Griffin notes that the short duration value is very severe, while the 24-hour limit is commonly experienced in public transportation without apparent harm. In effect, he is saying that the slope of the time-dependency relationship in ISO 2631 is too steep, and suggests that dose values, calculated per BS 6841, are a better approach.

Appendix A (section A.6) of BS 6841 states "there is currently no consensus of opinion on the precise relationship between vibration dose value and the risk of injury. It is known that vibration magnitudes and durations which produce vibration dose values in the region of 15 $m/s^{-1.75}$ will usually cause severe discomfort."

There does not appear to be any data to use to define a vibration dose value specifically for skidder operators. Assuming that a dose value of 15 $m/s^{-1.75}$ might be reasonable, what does that value mean in terms of an ISO 2631 Weighted RMS value for skidder vibration? Assuming a VDV of 15 $m/s^{-1.75}$ for the "shift equivalent" time of 6.8 hours used by Golsse results in a Weighted (BS 6841) value of 1.2 m/s^2 (VDV = Weighted RMQ * $t^{1/4}$). This in turn corresponds to a Weighted (BS 6841) RMS value of 1.2 $m/s^2/1.88$ = 0.64 m/s^2 for the skidder data. Finally, we note that the Weighted RMS value is about 70% of the Weighted (ISO 2631) RMS because of the differences in the weighting functions. Thus the Weighted (ISO 2631) RMS value which corresponds to the 15 $m/s^{-1.75}$ dose value guideline is 0.64/0.70 = 0.91 m/s^2.

If the seat suspension can achieve a small reduction (which it must to pass the ISO 5007 Class 3 test) instead of the apparent amplification that now occurs, then a significant improvement in the vibration levels will have occurred, and the resultant average levels should be near those indicated by the suggested guideline in BS 6841.

But as Griffin points out, " it has been common for standards to concentrate on the formulation of limits—often at the expense of providing satisfactory measurement procedures for evaluating vibration with respect of the limits. Useful limits for human exposure to vibration cannot be provided by a committee which is not responsible for either the system causing the exposure or the individuals exposed to the vibration."

CONCLUSION

1. The tests performed were representative of a wide range of skidder operations. In terms of Weighted RMS and Crest Factor values, these test results tended to be somewhat higher than the Golsse-Hope data.

2. Vibration levels are lowest in the fore-aft direction and highest in the vertical direction.

3. Fore-aft and lateral accelerations are higher on the cushion than at the seat base, which is to be expected because of the roll and pitch of the vehicle. If the seat base accelerometers are directly under the seat pad accelerometers, these rotations do not cause any difference between the base and cushion values for the vertical direction. However, this is a multi-input system in which the lateral and fore-aft motions of the operator may contribute to the observed vertical levels on the seat cushion.

4. The operator can have a pronounced effect on the whole-body vibration levels that he experiences, making it very difficult for the vehicle or seat manufacturer to control those levels.

5. The data show that the skidder seat suspension must be capable of reducing the vertical vibrations for which the spectrum peaks at about 2.2 hz. The data also indicate that the suspensions used on the test vehicles were probably too stiff to achieve the necessary attenuation. A significant improvement in skidder ride should result from requiring that seats for the skidder application be tested in a manner similar to that developed and used for testing the seat suspensions for earthmoving equipment and agricultural tractors (ISO 7096 and ISO 5007), respectively.

6. Very good agreement was found between the PSD in the vertical direction at the base of the seat, and the spectrum specified for approval of a Class 3 Agricultural Tractor seat defined in ISO 5007.

7. A modest amount of attenuation by the seat suspension should bring the operator's levels in line with the Vibration Dose Values suggested in BS 6841.

8. Considerable confusion can occur in the analysis and reporting of skidder vibration data because of the differences in the various standards, in particular with respect to questionable empirical constants and the effect of different weighting functions.

REFERENCES

[1] BS 6841. 1987. British standard guide to measurement and evaluation of human exposure to whole-body mechanical vibration and repeated shock," British Standards Institution.

[2] Golsse, J.M. and P.A. Hope. Analysis of whole-body vibration levels during skidding," Technical Report TR-77, Forest Engineering Research Institute of Canada.

[3] Griffin, M.J. International Standard 2631 and British Standard 6841: a comparison of two guides to the measurement and evaluation of human exposure to whole-body vibration and repeated shock. Proceedings of the Conference on the Effects of Vibration on Man, Institut National de Recherche et de Securité, Vandoeuvre-Les-Nancy, France, September 26-28, 1988.

[4] ISO 2631. 1978. Guide for the evaluation of human exposure to whole-body vibration.

[5] ISO/TC 108/SC4. June 1991. Third Committee Draft on ISO/CD 2631—Mechanical vibration and shock—guide to the evaluation of human exposure to whole-body vibration; revision of ISO 2631.

[6] ISO 5007. 1990. Agricultural wheeled tractors - operator's seat—laboratory measurement of transmitted vibration.

[7] ISO 7096. 1982. Earth-moving machinery—operator seat—transmitted vibration.

[8] ISO 8041. 1990. Human response to vibration—measuring instrumentation.

[9] Monsees, et al. Relationship between subjective assessment and objective measurements of tractor ride vibration (RMS and RMQ). Proceedings of the Conference on the Effects of Vibration on Man. Institut National de Recherche et de Securite, Vandoeuvre-Les-Nancy, France, September 26-28, 1988.

[10] SAE J1385. June 1983. Classification of earthmoving machines for vibration tests of operator seats.

[11] SAE J1386. 1986. Classification of agricultural wheeled tractors for vibration tests of operator seats.