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### LABORATORY INVESTIGATION OF SEAT SUSPENSION PERFORMANCE DURING VIBRATION TESTING

Alan G. Mayton

Joseph P. DuCarme

Christopher C. Jobes

Timothy J. Matty

National Institute for Occupational Safety and Health, Pittsburgh Research Laboratory  
626 Cochran's Mill Road, Pittsburgh, Pennsylvania 15236  
Telephone: (412) 386-4657; fax: (412) 386-6764; e-mail: [AMayton@cdc.gov](mailto:AMayton@cdc.gov)

#### ABSTRACT

Mining injury statistics show that a significant number of back, neck, and head injuries are linked to exposure from vehicle vibration. Use of a suspension seat is a common way to isolate the vehicle operator from the adverse effects of vibration exposure. Thus, researchers at the National Institute for Occupational Safety and Health<sup>1</sup> – Pittsburgh Research Laboratory (NIOSH – PRL) performed laboratory studies on four passive and two semi-active seat suspension designs. These are typical of seat suspensions commonly found on large off-road heavy surface mining, construction and agricultural vehicles as either replacement or original equipment manufacturer (OEM) systems. One included a pneumatic (air bladder) spring mechanism. The fifth and sixth suspensions were a NIOSH magnetorheological (MR) semi-active damper design based on the pneumatic (air bladder) and one of the coil spring suspensions above. These suspensions were modified with a commercially available MR damper substituted for the OEM damper.

These six seat suspension systems were tested and analyzed, for vertical vibration only, using the ISO 5007 Standard [1]. This paper describes the laboratory vibration tests using a MTS<sup>®</sup> shaker table and discusses the results obtained for the different suspension designs and highlights the rheonetic technology studied. Implications of the seat suspension designs relative to their capabilities for isolating vehicle operators from vibration exposure are discussed.

Results for suspensions 1 through 3 showed frequencies of isolation from 2.1 to 3.0 Hz using the 40-kg (88-lb) mass and from 1.65 Hz to 1.8 Hz using the 80-kg (176-lb) mass. Suspension #4, in tests with only the 80-kg (176-lb) mass, showed an isolation frequency of 3.7 Hz. With the MR

damper added to seat suspension #4, the peak transmissibility was lowered from 1.3 to 0.95 and showed a corresponding downward shift in frequency from 2.25 Hz to 1.4 Hz. In fact, the results for suspension #5 (the MR damper added to seat suspension #4), using test #3 conditions of the programmed control algorithm, showed isolation (attenuation of transmitted vibration) throughout the test frequency range from 1.0 to 6.0 Hz.

#### INTRODUCTION

NIOSH mining vehicle seat and vibration research is dedicated to reducing the risk of injuries to the back, neck, or head for vehicle operators through improved mining vehicle seat designs. In this regard, injury statistics for mobile mining equipment operators from the Mine Safety and Health Administration (MSHA) showed incidences of exposure to whole-body vibration (WBV) and mechanical shock (vehicle jarring or jolting). These injuries can be described as acute and chronic musculoskeletal disorders affecting the back, neck, and head. For example at surface mining operations, Wiehagen et al. [2] reported that vertical and lateral vehicle jarring or jolting were the most frequent cause of injury to bulldozer operators while operating those vehicles. Moreover, the total MSHA reported injuries involving the back, neck, and head for 1999 through 2003 resulted in an average of 57% per year for injuries related to vehicle jarring or jolting.

One way to lessen the adverse effects of vehicle vibration to the operator is through the use of a seat suspension system. Suspension seats are included in most industrial vehicles and are passive in nature in that they consist of a damper and some form of a spring. Moreover, during operation, these vehicles are subjected to significant energy in the 2 to 4 Hz region where conventional seats tend to amplify vertical vibration. A common method used to reduce vehicle operators' exposure to vehicle vibration is to design the natural frequency of the seat suspension so that it is considerably lower than the typical operating frequency generated by the vehicle. However, a large decrease in seat stiffness ("softer" or lower spring rate) is required to achieve this. Considering equation (1):

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$$\omega = \sqrt{k/m} \quad (1)$$

where,  $\omega$  is the natural frequency,  
 $k$  is the spring constant, and  
 $m$  is the mass of the seat and occupant.

A four-fold decrease in seat stiffness is necessary to reduce the natural frequency by 50 % [3]. Doing this however, now increases the likelihood that the suspension system will “bottom out.”

Moreover, a high vehicle torque requirement also contributes to vibration at the operator deck. Passive seat suspension design includes selecting an appropriate natural frequency and optimization of damping according to a set stiffness. The consequence is a trade-off between isolation of vibration peak amplitudes at resonance and isolation of higher frequency vibrations. The potential benefit of a semi-active system is the removal of this trade-off since the system is designed to adjust the damping in real time.

## METHOD

NIOSH-PRL researchers conducted laboratory evaluations on different seat suspension designs, including two based on a magnetorheological (MR) semi-active design. Four of the seat suspensions were replacement or original equipment manufacturer (OEM) systems commonly found on large heavy vehicles. The different suspensions were evaluated to determine how effective they would be if used to improve seat performance on large heavy vehicles used at mining operations.

A single-axis, MTS® hydraulic shaker was used to conduct vibration experiments. The shaker is capable of producing accelerations in excess of 10 g's at frequencies below 1 Hz up to 15 Hz with a maximum travel of 152 mm (6 in). Seat suspensions were tested using the ISO 5007 Standard. The test protocol in ISO 5007 Section 10.1 requires a sinusoidal vibration of amplitude  $\pm 15$  mm (0.59 in) and a frequency range from 0.5 to 2 Hz at 0.05 Hz intervals. Test weights or masses of 40 and 80 kg (88 and 176 lbs) are specified by ISO 5007 to simulate the upper and lower ends (5<sup>th</sup> percentile female and 95<sup>th</sup> percentile male) for the range of seated vehicle operators. In addition to the test mass, extra mass was added to duplicate seat hardware that was removed, such as the seat cushion and backrest. A test frequency range from 2 to 8 Hz at 0.25 Hz intervals was added to measure the transmissibility characteristics for each seat suspension system in the range most sensitive to the human body overall. The acceleration of the shaker and the seat pan was measured and used to calculate the ratio of the seat pan RMS acceleration to the shaker table RMS acceleration (i.e., transmissibility ratio). This ratio is commonly used as a measure of how well the seat suspension is isolating the occupant from vehicle vibration. To aid in the analysis of the data, graphs showing the transmissibility of each suspension were created.

## TESTING PROCEDURES

The test procedures included the following steps:

1. Each channel of an 8-channel Sony® PC208AX digital recorder was calibrated to ensure input.
2. On a data-acquisition log sheet, recorder channel assignments with accelerometer mounting location,

accelerometer axes, signal conditioners, and recorder settings, along with a recording log, were noted.

3. Preliminary shaker testing determined the 8-channel recorder voltage range (-5vdc to +5vdc volts) to capture desired events without clipping signals or being within noise levels (weak signals) of the system.
4. The 8-channel recorder sampling frequency was set to the normal setting – 5 kHz per channel.
5. Accelerometer amplifiers were set for the desired accelerometer charge sensitivity, gain, and a low-pass filter with a cut-off frequency of 100 Hz to reduce unwanted noise.
6. Two B&K® 4370 accelerometers, with magnetic mountings, were attached to the suspension and mounting base plate installed on the shaker; the small-diameter, accelerometer cables were secured to minimize noise induced from unrestrained movement during testing.
  - a. One accelerometer was placed in the center of the rigid portion of the shaker, as close as possible to the seat suspension to record the shaker input accelerations.
  - b. A second accelerometer was placed at the center of the seat suspension (at the seat pan location) to record vibration transmitted through the seat suspension.
7. To identify each test frequency change during data collection and aid in analyzing the data, a 5-Vdc pulse was used as a marker on one of the recorder channels.
8. A completed data-acquisition log sheet described the logging of recorder channel assignments, scaling, amplifier settings, accelerometer charge sensitivity, and recorder test logging.
9. Before testing, the seat cushion and backrest of the seat suspension system were removed and weighed in order to add their weights to the overall test weight.
10. With the appropriate test weight attached, the seat suspension was adjusted to its mid point of travel.
11. With the loaded seat on the shaker, a test with a duration of more than 40 minutes or  $\geq 5000$  cycles was conducted, in the amplitude control mode, with a displacement of  $\pm 15$ mm (0.59 in) at 2 Hz. The seat test load consisted of 40 kg (88 lbs) plus extra weight equal to the removed seat cushion and backrest. This procedure constituted the “break-in” period of the seat suspension.
12. After the break-in period, testing was again performed with amplitude control at a displacement of  $\pm 15$ mm (0.59 in) loaded with 40 kg (88 lbs) plus extra weight for the seat cushion and backrest. The tests continued from 0.5 Hz to 2 Hz in 0.05 Hz intervals and from 2 Hz to 8 Hz in 0.25 Hz intervals; each interval was recorded for 15 s.
13. The same test setup in step number 12 was run by reversing the frequency order.
14. The seat test was conducted in amplitude control mode with a displacement of  $\pm 15$ mm (0.59 in) loaded with 80 kg (176 lbs) plus extra weight equal to the removed seat cushion and backrest. The tests ran

from 0.5 Hz to 2 Hz in 0.05 Hz intervals and continuing from 2 Hz to 8 Hz in 0.25 Hz intervals recording each interval for 15 seconds.

15. Testing was again repeated in a fashion similar to step 14 but with descending frequency order.

In contrast to testing of the replacement and OEM suspension systems, a modified version of the ISO 5007 Standard was used to test the MR fluid damped suspension systems. Modifications included testing solely with the 80-kg (176-lb) weight, using a frequency range of 1 to 6 Hz, and using a displacement of 18 mm (0.7 in). Frequencies below 1 Hz were not tested in that brief testing showed that the inherent frictional resistance of the suspension components caused suspension system transmissibilities to be essentially unity in this region, where no movement of the suspension occurred. A displacement of 18 mm (0.7 in) was selected, instead of 15 mm (0.59 in) specified by ISO 5007, in order to apply enough force to operate the suspensions in the frequency range of interest. Frequencies above 6 Hz were not tested due to safety concerns from high accelerations when operating the shaker at a displacement of 18 mm (0.7 in) in lieu of 15 mm (0.59 in). Moreover, a linear displacement transducer was used to record displacement data. In turn, this data was used to compute accelerations and to verify results computed using the acceleration data.

### DESCRIPTION OF RHEONETIC SUSPENSION TESTING

Rheonetic technology offers the ability to adjust suspension damper characteristics in real time. The fluid used in current semi-active dampers is light oil that contains micron-sized iron particles. When exposed to a magnetic field, the viscosity of the fluid changes proportionally to the strength of the magnetic field. Viscosity changes occur very rapidly, on the order of several milliseconds. The magnetic field is induced by current flow through a coil installed in the piston of the damper. Figure 1 depicts a typical rheonetic damper [4].

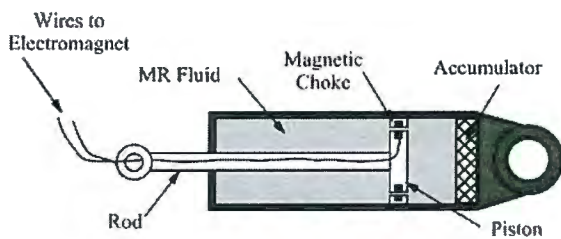


Fig 1: Typical rheonetic damper.

Two commercially available seat suspensions were modified using the rheonetic system. These suspensions are shown in Figures 2 and 3. Both were low travel and low cost designs. The designs were similar, with one using an air-bladder type spring and the other using a standard steel coil spring. The second design also used rolling elements and was subject to much less column friction than the first which used sliding elements in its linkage.

The “brain” of the rheonetic system is a micro-controller that monitors seat position, calculates seat velocity, and provides a control signal to the rheonetic damper. A prototype controller was developed for this project and the hardware

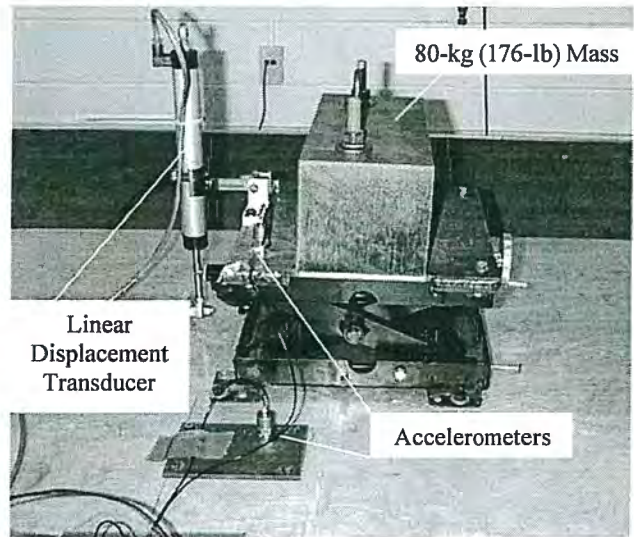


Fig. 2: Suspension #4 mounted to shaker table.

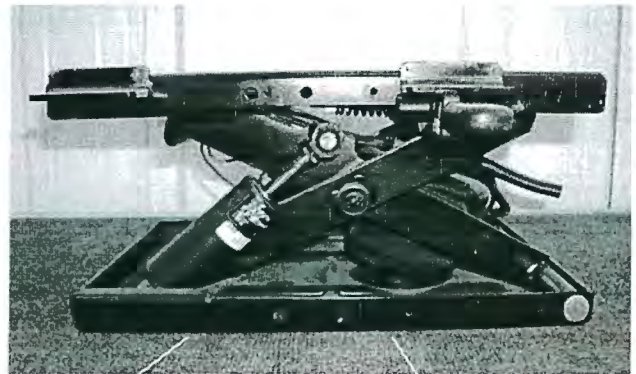


Fig. 3: Seat suspension #5.

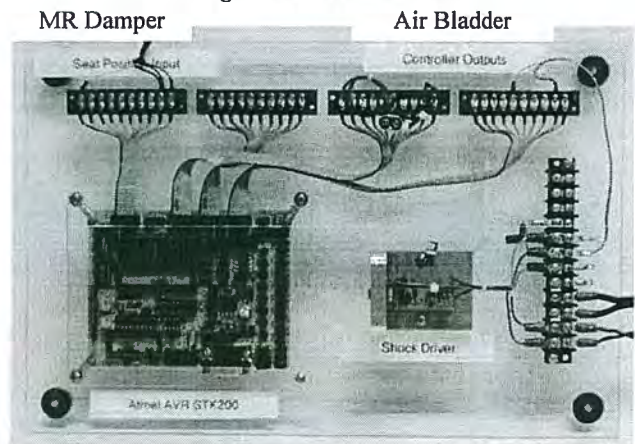


Fig. 4: Prototype rheonetic controller.

assembly is shown in Figure 4. It consists of a Kanda Systems STK200 Micro-controller Development Board, low-pass filter, voltage-to-current converter, and various input-output

terminations. The integrated circuit within the micro-controller was an Atmel AT90LS4434. The micro-computer board also contained a communications port to a personal computer that was used for software development and system tuning.

Numerous control algorithms for semi-active suspension systems have been published in literature. Most of them sense or measure the relative travel and velocity between the suspension system and the sprung mass. Two of the most common algorithms are named Skyhook control and Relative control. Referring to Figure 5, they are normally defined as follows:

**Skyhook Control**

$$\begin{aligned} \text{Set } F_{\text{damper}} &= C \cdot X' \quad \text{if } X' \cdot (X' - Y') > 0 \\ F_{\text{damper}} &= 0 \quad \text{if } X' \cdot (X' - Y') < 0 \end{aligned}$$

**Relative Control**

$$\begin{aligned} \text{Set } F_{\text{damper}} &= C \cdot (X' - Y') \quad \text{if } (X - Y) \cdot (X' - Y') < 0 \\ F_{\text{damper}} &= 0 \quad \text{if } (X - Y) \cdot (X' - Y') > 0 \end{aligned}$$

To implement Skyhook control, the absolute velocity ( $X'$ ) of the seat is required as an input to the controller. This is difficult to obtain in a vehicle because of the lack of an absolute reference frame. For Relative control, only one position sensor, mounted between the vehicle floor and the seat pan, is required. The relative velocities can be obtained by differentiation of the displacement signal. Thus the quantities  $(X' - Y')$  and  $(X - Y)$  can be replaced with the relative measurement  $Z'$  and  $Z$ , respectively.

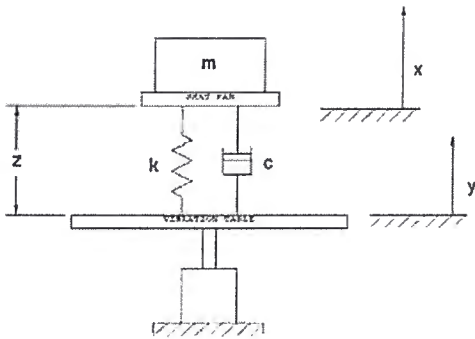


Fig. 5: Basic parameters for considering control algorithm.

**TEST RESULTS**

**Seat Suspension #1 (two linear extension springs)**

Suspension #1 is used in a commercially available seat that has a vehicle operator weight adjustment of 50 to 120 kg (110 to 265 lbs). The suspension included two linear extension springs in the upright position, mounted in parallel with a damper, and with a 10° layback and 8.6-cm (3.4-in) displacement. Figure 6 shows the seat and suspension; whereas, figure 7 shows the suspension alone mounted to the shaker table. The results from testing seat suspension #1 appear in Figure 8. Transmissibility (RMS output

acceleration/RMS input acceleration) versus frequency is plotted for the 40 and 80 kg (88 and 176 lbs) masses.



Fig. 6: Seat with suspension.

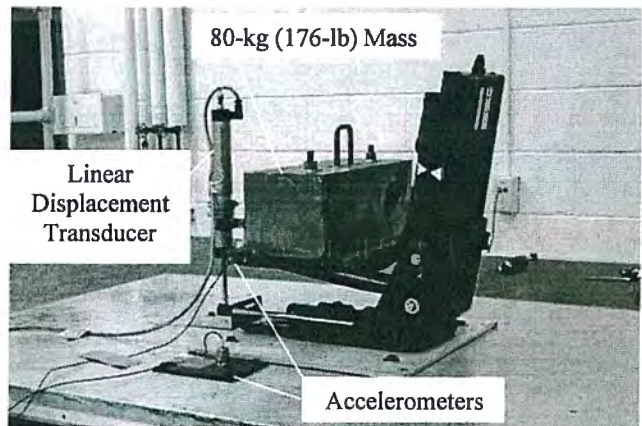


Fig. 7: Seat suspension #1 mounted to shaker.

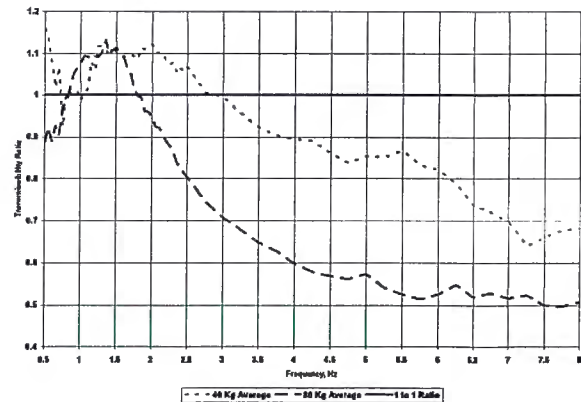


Fig. 8: Seat suspension #1 average results of transmissibility for 40- and 80-kg (88- and 176-lb) tests. Copyright © 2006 by ASME

are presented and discussed in the context of rheonetic design tests.

### Seat Suspension #2 (two linear extension springs)

The seat from which suspension #2 comes is adjustable for vehicle operators weighing 60 to 120 kg (130 to 265 lbs). The mechanical geometry of suspension #'s 1 and 2 are identical. The differences lie in the spring and damping rates. These two seat suspensions are popular replacement seating systems on heavy vehicles weighing 17,499 kg (7,938 lbs) or less. Figure 9 shows the average results for tests with the 40- and 80-kg (88- and 176-lb) masses.

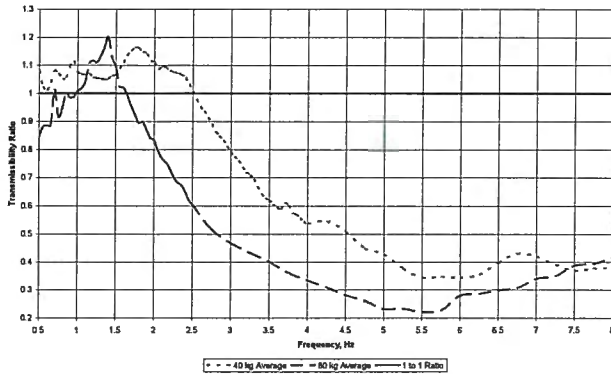


Fig. 9: Seat suspension #2 average results of transmissibility for 40- and 80-kg (88- and 176-lb) tests.

### Seat Suspension #3 (air spring)

Suspension #3 uses an air spring with a standard damper. The mechanical design of the seat is a cross-buck design with the pneumatic spring located in the center with a suspension stroke of 8.9 cm (3.5 in). The average results for the 40- and 80-kg (88- and 176-lb) masses appear in Figure 10.

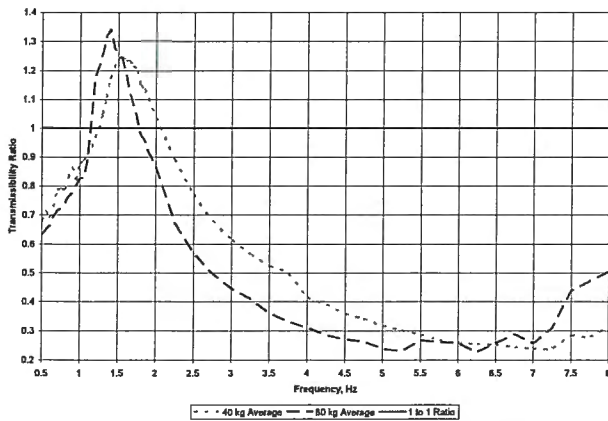


Fig. 10: Seat suspension #3 average results of transmissibility for 40- and 80-kg (88- and 176-lb) tests.

### Seat Suspension #4 (single coil spring)

A fourth seat suspension was a cross-buck damper, coil spring system. The results of tests with this suspension system

### Rheonetic Design Compared with Commercially Available Seat Suspensions

Our research showed that using rheonetic dampers in a semi-active seat suspension system can provide modest improvements in transmissibility. Figures 11 and 12 show transmissibility curves of the rheonetic, OEM mechanical (suspension #4), and air suspensions (suspension #3).

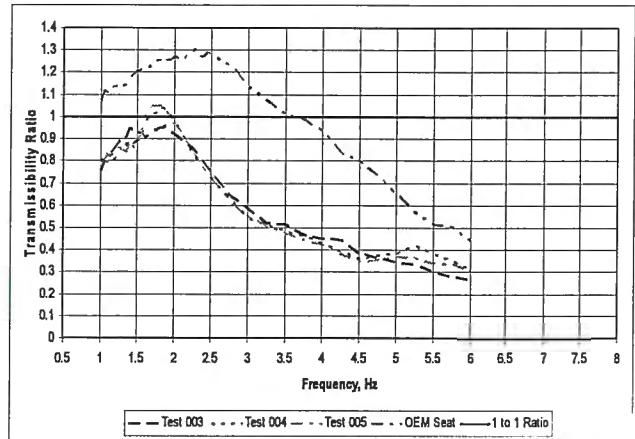


Fig. 11: Transmissibility curves for suspension #4 (OEM mechanical) versus suspension #5 (rheonetic damper with suspension #4) using several test variations of the programmed control algorithm.

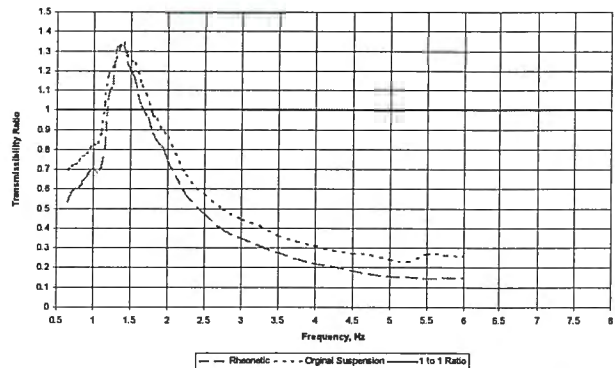


Fig. 12: Transmissibility curves for suspension #3 (air) versus suspension #6 (rheonetic damper with suspension #3).

### DISCUSSION

All spring-mass-damper systems exhibit a resonant frequency. A seat suspension is amplifying the input vibration when the transmissibility is greater than one. For seat suspension systems, it is desirable to have the resonant or natural frequency as low as possible without adversely affecting the performance at higher frequencies. This is due to the sensitivity of a human occupant to vibrations. Typically,

vibrations at frequencies less than 2 Hz need to be of significant amplitude before becoming objectionable. Because of this, higher transmissibility below 2 Hz should not necessarily be viewed as poorer performance. As shown in the graphs, resonant frequencies (the peak of the graph) for the six seat suspensions tested were similar, in that they were at or near 1.5 Hz.

The 40- and 80-kg (88- and 176-lb) mass curves for suspension #1 show good correlation up to 5.5 Hz. After 5.5 Hz there is more variation, particularly with the curve showing decreasing Hz. Intuitively speaking, more mass should provide better attenuation and thus lower transmissibility. This is supported by the 80-kg (176-lb) curve, which shows a greater and more rapid reduction in transmissibility compared to the 40-kg (88-lb) curve, which shows a more gradual reduction in transmissibility.

As shown by a transmissibility less than one, isolation for suspension #2 occurs at about 1.6 Hz with the 80-kg (176-lb) mass. This compares with the similar isolation and lower transmissibility at 2.5 Hz with the 40-kg (88-lb) mass. As noted above, the larger mass shows a greater and more rapid drop in transmissibility as shown by the steepness of the curves for the 80-kg (176-lb) mass in the 1.4 to 2.5 Hz frequency range. Reductions in transmissibility for suspension #2 parallel those for suspension #1 at 1.4 and 2 Hz, respectively, for the 40- and 80-kg (88- and 176-lb) masses. As noted by equation (1) above, the natural frequency of a seat depends on its mass. Thus, a shift in natural frequency with a change in mass is certainly expected.

The isolation frequency for suspension #3 and the 40-kg (88-lb) weight is lower at 2.1 Hz compared to the same for suspension #'s 1 and 2. In general, the results shown for the air-ride suspension #3 are similar to suspension #2.

Common spring-damper systems, such as suspensions #1 through #4, are called passive suspension systems. Their reactions to vibrations are determined by their design parameters. Variable rate springs and velocity proportional dampers can be used to improve ride characteristics over fixed rate components, but as with any mechanical or pneumatic system, the results are a collection of design compromises.

Active systems dissipate energy from the suspension system by forcing extension or retraction in response to measured and anticipated motion. In these suspensions, the dampers are replaced with hydraulic cylinders that are pressurized by a hydraulic pump. The major drawback for this type of system is that it is complicated, costly, and needs a power supply. Some safety concerns also apply.

Semi-active systems, such as suspensions #5 and #6, change the suspension response using rheonetic technology to adjust suspension damper characteristics in real time.

The OEM mechanical system, suspension #4, has significant friction in its design due to sliding components. The rheonetic damper (suspension #5) was tested using the relative control algorithm programmed into the controller. Several parameters (e.g., damping coefficient, velocity, displacement) were varied for tests. These variations appeared to have little effect on the performance in the frequency range tested. As the frequency of the input vibration increased, it appears that any advantage the rheonetic technology could provide in vibration isolation becomes less. This is

anticipated outcome since damping effects are greatest near resonance.

Suspension #6 (addition of the rheonetic damper and controller to suspension #3 - air), did not provide as significant an improvement in vibration isolation as it did in the OEM mechanical suspension (#5). This is most likely due to the design of the air-ride suspension being a superior design. It was a more expensive and robust design and included rolling elements to reduce friction. Since it incorporated an air spring, the rate could be somewhat adjusted for the load, unlike suspension #4, the cross-buck damper, coil-spring system with its fixed rate spring. The advantage of the rheonetic damper was not as great, although the system was only marginally investigated. Nevertheless, it did provide a measurable improvement. Moreover, the rheonetic technology is available from LORD Corporation (<http://www.lord.com>) as a retrofit kit to several popular seat suspension systems.

Likely applications for any of the above seat suspensions are large surface mining vehicles such as haulage trucks, bulldozers, pan scrapers, front-end loaders, and graders. Using the above suspensions with improved seat padding, such as that reported by Mayton et al. for underground coal mine haulage vehicles [5], shows the potential for improved isolation of vehicle operators from WBV and terrain-induced vehicle jars and jolts that can increase the risk of operator injuries.

## CONCLUSIONS

Seat suspensions 1 through 3 showed peak transmissibilities, and amplification of transmitted vibration, occurred at 1.6 Hz to 2.0 Hz for the 40-kg (88-lb) mass and 1.3 to 1.4 Hz for the 80-kg (176-lb) mass. Similarly, peak transmissibilities for suspension #'s 4 and 5 showed amplification of vibration at 2.5 Hz and 1.4 Hz with only the 80-kg (176-lb) mass.

For suspension #5 (the MR damper added to seat suspension #4), the peak transmissibility was lowered from 1.3 at 2.25 Hz to 0.95 at 1.4 Hz. Moreover, peak transmissibilities for several variations of the programmed control algorithm for suspension #6 showed amplification at 1.75 Hz and attenuation at 1.4 Hz with only the 80-kg (176-lb) mass.

Isolation frequencies for seat suspensions 1 through 3 occurred from 2.1 Hz to 3.0 Hz using the 40-kg (88-lb) mass and from 1.65 Hz to 1.8 Hz using the 80-kg (176-lb) mass. Suspension #4, in tests with only the 80-kg (176-lb) mass, showed an isolation frequency of 3.7 Hz.

Furthermore, seat suspension #5 for test #3 setting of the programmed control algorithm attenuated the transmitted vibration throughout the test frequency range from 1.0 to 6.0 Hz. These results suggest that the application of rheonetic technology, in particular, should provide heavy vehicle operators with improved isolation from seat transmitted vibration and a subsequent reduction in back and neck strain injuries.

## REFERENCES

- [1] ISO, 1990, "ISO 5007 – Agricultural Wheeled Tractors – Laboratory Measurement of Transmitted

Vibration. International Organization for Standardization, Geneva, Switzerland.

[2] Wiehagen WJ, Mayton AG, Jaspal JS, Turin FC [2001]. An analysis of serious injuries to dozer operators in the U.S. mining industry. Pittsburgh, PA: U.S. Department of Health and Human Services, Public Health Service, Centers for Disease Control and Prevention, National Institute for Occupational Safety and Health, DHHS (NIOSH) Publication No. 2001-126, IC 9455, 23 pp.

[3] Griffin, M. J., 1990, *Handbook of Human Vibration*. Academic Press Ltd., NY, pp 399-401.

[4] Spencer BF, Dyke SJ, Sain MK, Carlson JD. [1996]. Phenomenological Model of a Magnetorheological Damper. ASCE Journal of Engineering Mechanics, 23 pp.

[5] Mayton AG, Amirouche F, and CC Jobes (2005). Comparison of Seat Designs for Underground Mine Haulage Vehicles Using the Absorbed Power and ISO 2631-1(1985)-Based ACGIH Threshold Limit Methods. *International Journal of Heavy Vehicle Systems*, vol. 12, no. 3, pp. 225-238.