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EVALUATION OF VIBRATION AND SHOCK ATTENUATION PERFORMANCE OF A SUSPENSION SEAT WITH A SEMI-ACTIVE MAGNETORHEOLOGICAL FLUID DAMPER

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The potential benefits of a semi-active magnetorheological (MR) damper in reducing the incidence and severity of end-stop impacts of a low natural frequency suspension seat are investigated. The MR damper considered is a commercially developed product, referred to as "*Motion Master semi-active damping system*" and manufactured by Lord Corporation. The end-stop impact and vibration attenuation performance of a seat equipped with such a damper are evaluated and compared with those of the same seat incorporating a conventional damper. The evaluation is performed on a servo-hydraulic vibration exciter by subjecting the seat-damper combinations to a transient excitation with dominant frequency close to that of the seat and continuous random excitation class EM1 applicable to earth-moving machinery, and a more severe excitation realized by amplifying the EM1 excitation by 150%. Tests are performed for medium and firm settings of the MR damper and for seat height positions corresponding to mid-ride and $\pm 2{\cdot}54$ and $\pm 5{\cdot}08$ cm relative to mid-ride. The results indicate that significantly higher levels of transient excitation are necessary to induce end-stop impacts for the seat equipped with the MR damper, particularly when set for firm damping, the difference with the conventional damper being more pronounced for seat positions closer to the end-stops. Under the EM1 excitation, the results indicate that under conditions which would otherwise favour the occurrence of end-stop impacts for a seat equipped with a conventional damper, the use of the MR damper can result in considerably less severe impacts and correspondingly lower vibration exposure levels, particularly when positioned closer to its compression or rebound limit stop.

1. INTRODUCTION

Vertical suspension seats are widely used in heavy road and off-road vehicles to isolate the drivers from road-induced whole-body vibration and shock. The performance characteristics of suspension seats, comprising either mechanical or pneumatic springs and passive dampers, have been extensively investigated under different types of excitations via field or laboratory assessments and analyses of coupled suspension seat–occupant models [1–3]. Based on the work reported by Wu and Griffin [4], it is possible to describe the vibration attenuation performance of such seats in three different categories based upon magnitude and predominant frequency of vehicular ride vibration. Under low-level floor vibration, the suspension mechanism may exhibit lock-up behaviour due to inherent Coulomb friction. The sprung part of the seat, including the occupant, thus moves in phase with the floor resulting in either little or no vibration attenuation. The effect of such low-level vibration on the occupant, however, may be regarded as insignificant.

The suspension performance within the second category relates to relatively higher levels of continuous vibration, such as those encountered in heavy highway vehicles traversing relatively rough roads and off-road vehicles. Such vibration excitations cause considerable relative motion across the suspension components, mostly within the permissible suspension travel such that the suspension bottoming or topping does not occur. The suspension seat thus results in amplification and/or attenuation of vibration depending on the properties of its components and predominant frequencies of excitation. The performance assessments of suspension seats under these types of vibration have been mostly emphasized in the reported studies and current standards [1, 5]. The suspension performance in the third category relates to higher magnitudes of vehicular vibration that may cause the suspension to exceed its free travel limit and result in repetitive impacts with the end-stops. The suspension seats employed in freight and resource sector vehicles are mostly designed to provide free travel ranging from 100 to 150 mm, and elastic elements are used to limit the travel beyond this range. Owing to the progressive hardening properties of elastic end-stops [6], severe impacts against the end-stops arising from excessive relative motion of the seat can transmit high-magnitude shock motions to the occupant. The suspension performance under such excitations is thus mostly determined from its shock attenuation performance, often expressed in terms of the ratio of vibration dose values (VDV) measured on the seat relative to that at the base [4, 7].

The ride vibration environment of vehicles employed in highway transportation of freight and passengers, and in some of the off-road sectors, comprises low-to-medium levels of vibration of continuous nature and occasional shock motions. The suspension performance for these vehicles can thus be considered to fall within the first two categories. The ride vibration of many heavy road vehicles, specifically those employed in urban public transport sector, often includes shock motions arising from interactions of tyres with extreme road irregularities, such as pot holes, rut formations and drain covers [8]. The shock motions transmitted to operators of urban buses are known to be more frequent and more severe. This is attributed to two major factors related to vehicle design and operation. The primary suspension employed in modern buses comprises relatively soft air springs and hydraulic shock absorbers that result in lightly damped bounce mode oscillations of the sprung mass with natural frequency in the 1·25–1·5 Hz range. The suspension for such vehicles is invariably designed to provide low natural frequency within this range in order to maximize passenger comfort and reduce the road damage [9]. The interactions of vehicle tyres with severe road irregularities causes high-amplitude bounce motion of the sprung mass that predominates in the vicinity of the sprung mass natural frequency. Since the air suspensions employed at the driver's seat also exhibit their natural frequency within the same range, the suspension seat may experience excessive relative motions and severe repetitive impacts against the end-stops whenever extreme road irregularities are encountered. The suspension seats employed in many off-road vehicles are also believed to be subjected to end-stop impacts while operating on relatively rough terrains. The isolation of drivers from the shock and vibration environment of such vehicles thus necessitates the design of suspension seats that can minimize both the transmission of continuous vibration and shock motions to the operator.

The attenuation performance under steady state vibration and prevention of end-stop impacts under high-magnitude vibration and shock, however, poses contradictory design requirements for the suspension seats [10, 11]. A lightly damped and soft suspension is considered desirable for effective attenuation of continuous vibration of low-to-medium levels, provided that the excitation occurs at frequencies well above the seat's natural frequency. The attenuation of high-magnitude vibration and shock, on the other hand, requires suspension designs with higher damping and stiffness to prevent end-stop impacts from occurring. In environments involving combinations of low, medium and high levels of continuous vibration and shocks, means of achieving variable damping are thus desirable to adapt the seat attenuation performance accordingly. This can be achieved through the incorporation of active or semi-active damping within the suspension. A limited number of active suspension concepts that can dissipate as well as provide energy in response to variations in excitations have been explored to achieve improved shock and vibration attenuation performance [12, 13]. The application of an actively controlled suspension at the seat, however, is most likely to be prohibitive in a majority of vehicles due to the associated high cost and requirement of power for the servo-actuator system.

Alternatively, semi-active dampers with only minimal power requirement could be applied to achieve variable damping to enhance suspension performance under complex vibration and shock environments. Systems such as a semi-actively controlled electrorheological (ER) fluid damper have been realized [14], which have shown promising performance potential to achieve control of end-stop impacts. Other semi-active dampers based upon magnetorheological (MR) fluids have also been developed for primary and secondary vehicle suspensions [15], where modulation of the damping force is achieved through variations in fluid viscosity caused by the applied current. These dampers can offer considerable potential to realize variable damping with minimal time delay and minimal power consumption.

In this study, the potential benefits of using an adaptively controlled MR damper for reducing the incidence and severity of suspension seat end-stop impacts are investigated through laboratory experiments. The vibration and shock attenuation performance of an air-suspension seat equipped with an MR damper are measured under different excitations and for different seat height positions and damper settings. The results are discussed in terms of measures defined for assessment of shock and vibration exposure characteristics and seat attenuation performance, and compared with those of a seat with conventional hydraulic damper. Based on the comparison, an assessment is made of the MR damper's overall ability to reduce the incidence and severity of end-stop impacts, and to limit the vibration exposure levels for the seat occupant.

2. DESCRIPTION OF MAGNETORHEOLOGICAL DAMPER

2.1. COMPONENTS AND OPERATING PRINCIPLE

The semi-active MR fluid damper considered as part of this study is a commercially developed product, referred to as "*Motion Master semi-active damping system*" and manufactured by Lord Corporation. The *Motion Master* damping system consists of a controllable damper filled with MR fluid, a controller with an integrated sensor arm, a three-position ride mode switch offering light, medium and firm damping, and a microprocessor. Figure 1 illustrates a pictorial view of its components. The damper design is

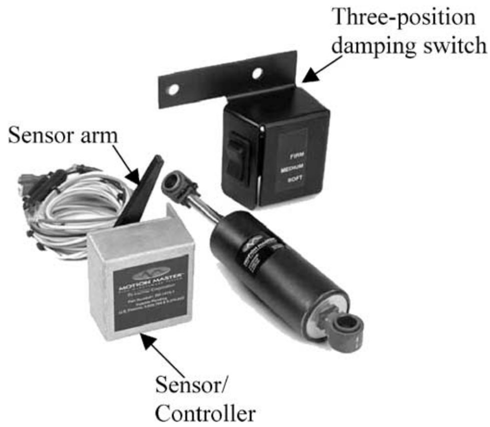

Figure 1. A pictorial view of the *Motion Master* system components.

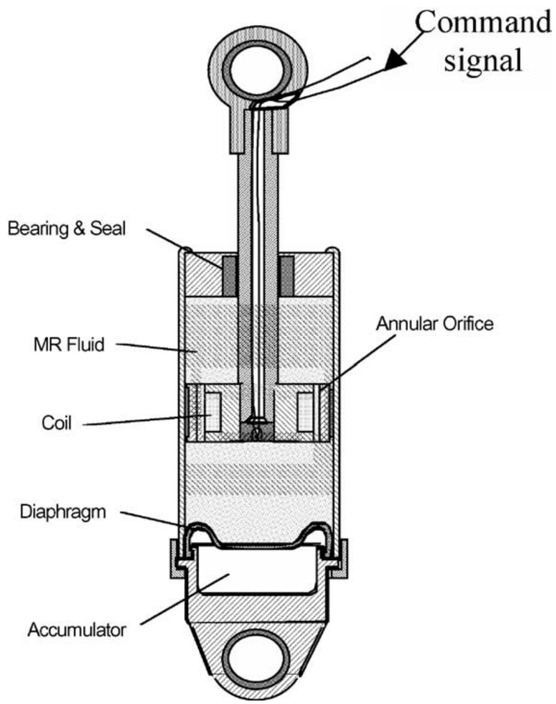

Figure 2. Schematic of the *Motion Master* MR fluid damper.

similar to a conventional single-tube damper, which consists of a piston with orifices and a gas chamber. Electromagnetic coils are located within the piston around the orifices, as shown in Figure 2.

When the MR damper is incorporated within a seat suspension, the relative position of the suspension and its rate of change serve as the primary feedback variables for the adaptively controlled suspension damping. The mid-ride position of the sprung pan is defined as the reference position of the suspension, and the instantaneous position of the suspension with respect to the predefined reference is sensed using a rotary potentiometer. The rotary potentiometer, contained within the controller unit, is attached to the

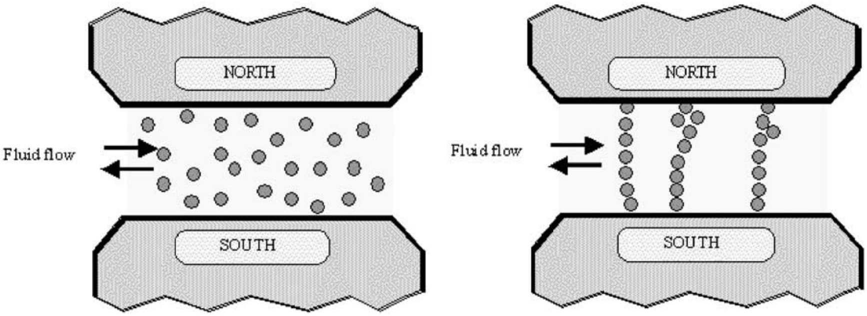

Figure 3. Response of MR fluid to the applied magnetic field.

suspension linkage through a sensor arm shown in Figure 1. The controller modulates the control current to the electromagnetic coils using the feedback signal from the sensor and the *Motion Master* adaptive control algorithm.

The controller consists of a microprocessor powered by a 12-V power source and contains the control algorithm. The feedback signal is manipulated to generate the command current to the electromagnetic coils to automatically adjust the damping characteristics in response to the feedback signal of the relative position of the sensor. The command current sets up a magnetic field created by the coil, which causes the iron particles in the MR fluid to form a chain in the direction perpendicular to the flow of fluid through the orifice, increasing the fluid's apparent viscosity. This effect is illustrated in Figure 3. The MR fluid is capable of changing from a free-flowing liquid to a near-solid state in under 10 ms, enabling real-time control of damping force.

2.2. DAMPING CHARACTERISTICS

The instantaneous damping force developed by the MR damper is directly related to the command current generated by the controller. Figure 4 illustrates the damper force–velocity characteristics under different values of constant current, ranging from 0 to 1·5 A. The results clearly show considerable variations in damping force with applied current signal. The algorithm is formulated to provide light damping under low magnitudes of relative deflections. While operating on relatively smooth roads, the suspension damper operates in a passive mode with either little or no current transmitted to the coil. When the suspension encounters relatively large motions, the algorithm detects the relative position of the seat suspension and its time rate of change, and generates an appropriate command current signal for the magnetic coils to increase the damping force to prevent collision with the end-stops. A three-position ride mode switch is also integrated within the design that allows the driver to adjust the firmness of the overall ride by setting the damping to one of three settings: soft, medium, and firm. Although all the ride modes provide modulation of damping to prevent end-stop impacts, the magnitude of damping force developed to prevent end-stop impact increases with the firmness of the ride setting.

The curves presented in Figure 4 reveal nearly symmetric force–velocity characteristics in compression and rebound. Under low-to-medium levels of control current, the force–velocity curve can be considered to provide two-stage damping: a high damping coefficient at low velocities and a lower coefficient at higher velocity. The transition from high-to-low damping coefficient occurs at a preset velocity. Under higher levels of control current, the force–velocity curve can be considered to provide three sequential stages defining a

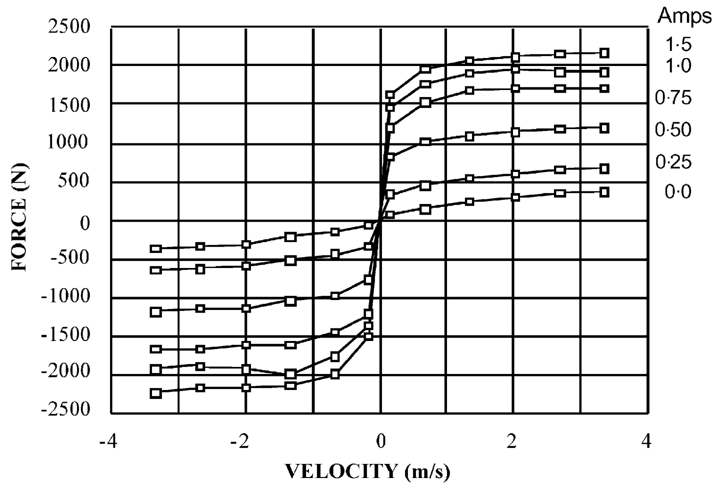

Figure 4. Force–velocity characteristics of an MR fluid damper as a function of the coil current.

maximum damping coefficient at velocities less than 0·02 m/s, an intermediate value at velocities from 0·02 to 0·06 m/s and a low coefficient at higher velocities. The results further show that only minimal power is required to achieve considerable variations in damping force. The controller current in the current design of MR fluid damper was limited to 2 A.

3. TEST METHODS AND ANALYSIS

3.1. SUSPENSION SEAT DESCRIPTION AND LOAD

A commercial vertical suspension seat comprising an air spring and an inclined conventional hydraulic damper was considered for the study. This particular seat, often used in heavy road vehicles, had suspension components supported within a pair of nearly parallel, but curved links connecting the seat base with the sprung pan. The relative motion of the suspension along the compression direction was limited by an elastic stop mounted on the suspension base. The limiting of suspension motion in the rebound direction was achieved by contact of the upper curved link with the seat pan. The curved geometry of the linkage, however, reduced the severity of the end-stop impact in the rebound mode. The free travel of the suspension, measured between the two end-stops, was 165 mm.

For the purpose of conducting the tests, the seat was loaded with a rigid mass of 67 kg configured using steel plates and lead-shot to achieve low centre of gravity height of the load. The rigid load was secured to the seat cushion using a seat belt restraint.

3.2. TEST MATRIX

Laboratory experiments were conducted to evaluate relative performance characteristics of the candidate seat equipped with a conventional and a *Motion Master* MR fluid damper. Identical tests, described below, were performed on three different suspension–damper combinations: (i) suspension with conventional damper; (ii) suspension with MR fluid damper under a medium setting; and (iii) suspension with MR fluid damper under a firm setting. The seat with the selected damper–suspension combination was installed on a vibration platform supported on a servo-hydraulic vibration exciter. The vibration exciter was capable of generating a peak-to-peak displacement of 150 mm. A seat accelerometer was placed at the load–cushion interface to measure the transmitted vertical vibration on the seat, while a single-axis accelerometer was installed on the platform to measure the vertical input vibration. The rotary potentiometer, used as the feedback sensor for the *Motion Master* system was also attached to the suspension linkage to monitor its relative motion. The potentiometer signal was further monitored to detect objectively contacts with the end-stops, when they occurred.

Each suspension seat and damper combination was initially tested by adjusting the seat height to correspond to the mid-ride position. Subsequent tests were conducted for seat heights set at $\pm$ 2·54 and $\pm$ 5·08 cm with respect to mid-ride to introduce some variations in adjustment positions as would be expected for occupants with different anthropometric characteristics.

3.3. VIBRATION STIMULI

In order to establish the natural frequency and estimate the damping characteristics of the various suspension–damper combinations, a series of measurements was conducted under constant displacement (19 mm peak) sinusoidal excitations swept in the 0·5–3 Hz frequency range for seat heights corresponding to the mid-ride position. The base and seat acceleration signals were analyzed to determine the vibration transmissibility response of the suspension.

Subsequently, the vibration attenuation and end-stop impact performance of the various suspension seat–damper combinations were investigated under three different excitations. These included a transient or shock excitation, continuous vibration excitation class EM1 defined in ISO-7096 [5], and EM1 excitation amplified by 150%. The transient excitation was synthesized from the measured vertical acceleration response of a bus under a typical pot-hole input. The displacement time history of the synthesized excitation derived from the acceleration response measured at the bus-floor is shown in Figure 5. The time history

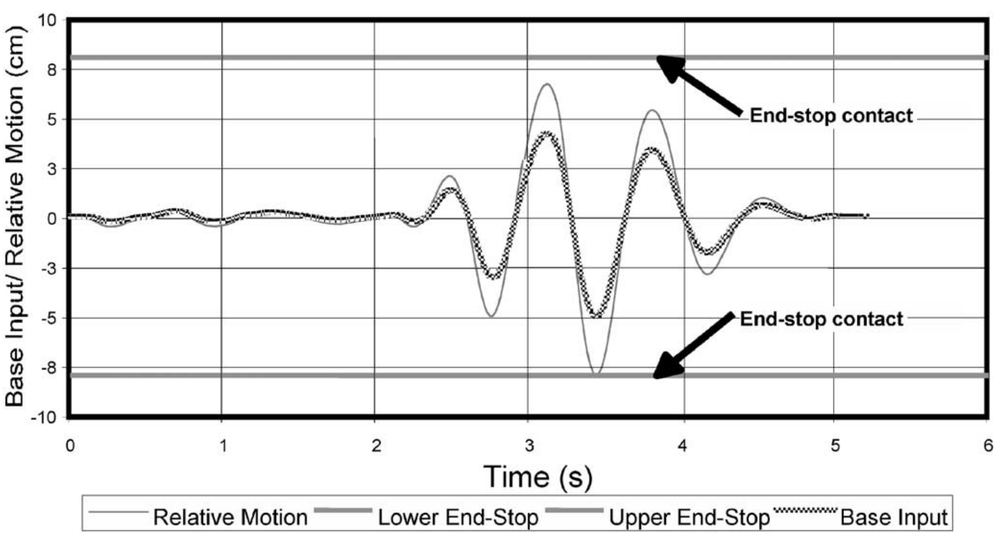

Figure 5. Time histories of transient displacement excitation and relative displacement response, and free travel limits of suspension.

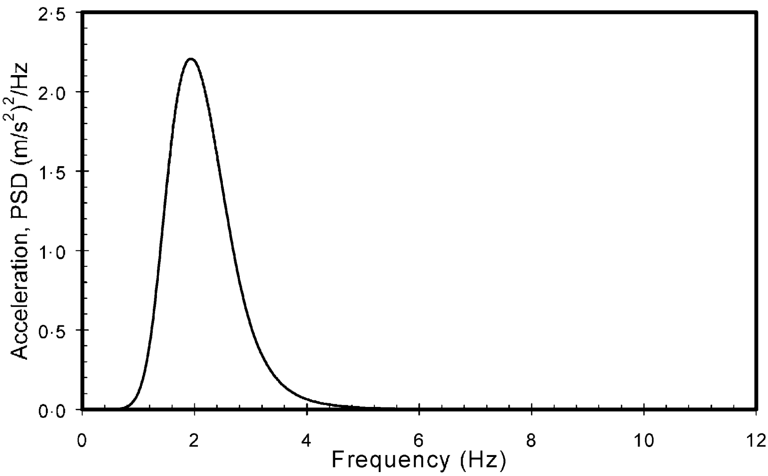

Figure 6. Power spectral density (PSD) of base acceleration corresponding to EM1 excitation [5].

reveals a dominant frequency of 1·5 Hz, which is most likely attributed to the vertical mode resonance of the sprung mass of the bus. This excitation is expected to yield excessive relative motion of the suspension seat, since its predominant frequency occurs in the vicinity of the natural frequencies of the seat–damper combinations considered in the study. The experiments under this excitation were performed to evaluate the amount of vibration energy required by the suspension to cause end-stop impacts. The magnitude of the transient excitation was, therefore, gradually increased until an end-stop impact in the compression or rebound directions could be detected. The acceleration time trace due to base excitation at which an end-stop impact occurred was then recorded and analyzed according to the various performance criteria selected. The end-stop impact condition was detected through continuous monitoring of the relative displacement signal in relation to the free travel limits of the suspension, as illustrated in Figure 5.

The vibration attenuation performance and occurrence of end-stop impacts of each damper–suspension combination were also investigated under EM1 and a more severe excitation realized by amplifying the EM1 excitation by 150%. Figure 6 illustrates the PSD of input acceleration corresponding to EM1 excitation, as defined in ISO-7096 [5]. The acceleration signals measured at the seat base and load–cushion interface were analyzed in accordance with the various performance criteria. The relative vibration attenuation and end-stop performance characteristics of the MR fluid damper were then assessed based upon these measures for each of the selected seat height adjustments.

3.4. EVALUATION CRITERIA

In conformity with the parameters defined in ISO 2631-1 [7] for assessing human exposure to vibration and shock and to follow up on procedures already introduced to assess the attenuation performance of suspension seats, the relative evaluation of each of the suspension–seat damper combinations was performed on the basis of frequency-weighted r.m.s. accelerations, a_w, vibration dose value, VDV, seat effective amplitude transmissibility, $SEAT$, VDV ratio, and crest factor, f_c. Frequency-weighting W_k, whose transfer function is defined in ISO 2631-1 within the pass-band of 0·4–100 Hz, was applied to the recorded acceleration time histories, $a(t)$, to compute these various values. On the basis of the resulting frequency-weighted acceleration time histories, $a_w(t)$, the base parameters were computed from:

Frequency-weighted r.m.s. acceleration: $$a_w = \left[\frac{1}{T}\int_0^T a_w^2(t)\,\mathrm{d}t\right]^{1/2}, \tag{1}$$

Vibration dose value: $$VDV = \left[\int_0^T a_w^4(t)\,\mathrm{d}t\right]^{1/4}, \tag{2}$$

Crest factor: $$f_c = \frac{[a_w(t)]_{max}}{a_w}. \tag{3}$$

In the above, T corresponds to the total integration time of acceleration time histories and $[a_w(t)]_{max}$ represents the peak value of frequency-weighted instantaneous acceleration, subsequently denoted by a_{wp}, within the time trace. The duration of the time traces, T, was fixed to 64 s for the tests under transient excitation and 128 s for the tests involving EM1 excitation. The performance criteria defining the vibration energy required to create end-stop impacts under transient excitation were selected as base measured $a_{w,base}$, VDV_{base} and $a_{wp,base}$. As for the assessment of the shock and vibration transmission performance characteristics of the various suspension-damper combinations under EM1 excitation, the following performance criteria were applied: seat measured $a_{w,seat}$, VDV_{seat}, $a_{wp,seat}$, $f_{c,seat}$ and $SEAT$ and VDV ratio defined as

$$SEAT = \frac{a_{w,seat}}{a_{w,base}}, \qquad VDV\ ratio = \frac{VDV_{seat}}{VDV_{base}}. \tag{4, 5}$$

This last performance parameter is considered to take into account more appropriately the short-duration high acceleration likely to arise during impacts with the end-stops [4].

4. RESULTS AND DISCUSSIONS

4.1. VIBRATION TRANSMISSIBILITY CHARACTERISTICS

The measurements were performed under sine sweep excitation to determine the vibration transmissibility characteristics of each suspension–damper combination over the 0·5–3 Hz frequency range. This was realized by setting the seat height at mid-ride and determining the seat frequency transfer function from the ratio of frequency spectrum measured at the seat to that at the base. The measurements performed with the seat equipped with the passive damper revealed a natural frequency near 1·48 Hz and corresponding peak acceleration transmissibility of 1·51. The suspension seat equipped with the MR fluid damper revealed natural frequencies near 1·37 Hz and 1·49 Hz corresponding to medium and firm settings. The corresponding peak acceleration transmissibility values were attained as 1·63 and 1·59. Assuming linear viscous damping, the suspension damping ratios for the suspension with conventional, and medium and firm settings of MR fluid dampers were estimated as 0·46, 0·39 and 0·40 respectively. These results thus suggested that at the optimal mid-ride position, the seat with conventional damper offered higher damping than that provided with the MR damper, for which both medium and firm settings led to almost identical damping under a similar seat adjustment position.

4.2. TRANSIENT ENERGY REQUIRED TO INDUCE END-STOP IMPACTS

The transient or pot-hole excitation was selected for the purpose of evaluating the resistance offered by each suspension–damper combination to the generation of end-stop

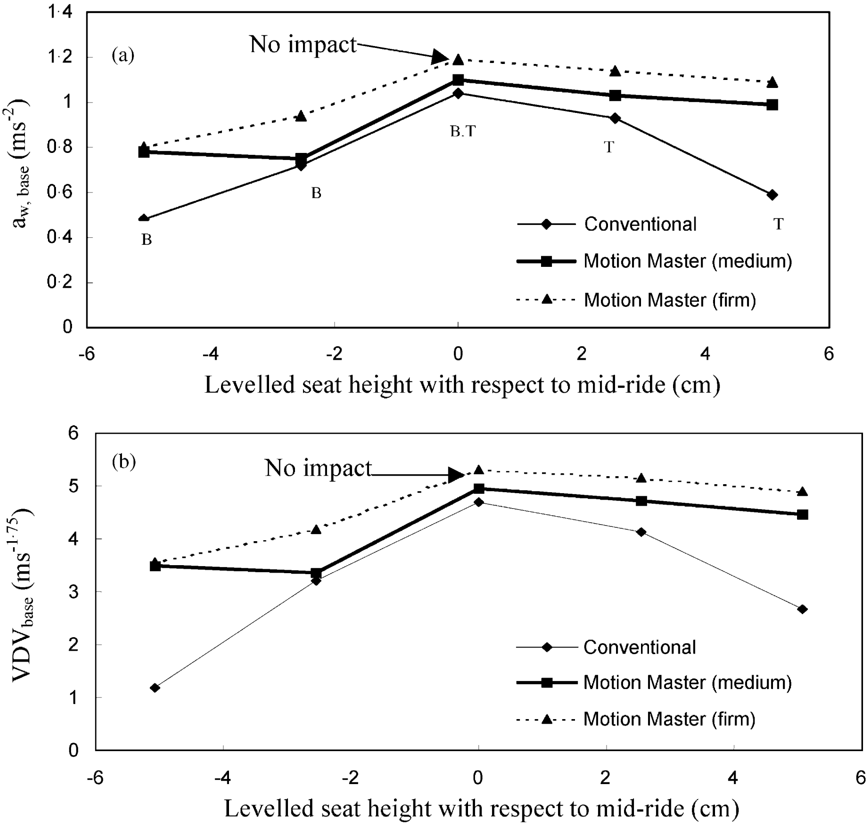

Figure 7. Comparison of the transient base input energy required to induce end-stop impacts for the various seat–damper combinations: (a) frequency-weighted r.m.s. acceleration; (b) vibration dose value (B = bottoming, T = topping).

impacts. The transient excitation had been shown to have a dominant frequency in the proximity of 1·5 Hz, thus close to the natural frequency of each suspension–damper combination. The resistance to impact offered by each seat for each height adjustment setting was determined by increasing the gain of the excitation until end-stop impacts (either topping or bottoming) could be observed. Corresponding values of $a_{w,base}$, VDV_{base} and $a_{wp,base}$ were then computed to provide indications of the vibration energy that would be necessary to induce end-stop impacts.

Figure 7 provides a summary of the results obtained in terms of frequency-weighted r.m.s. acceleration $a_{w,base}$ and VDV_{base} values for different suspension positions ranging from $-5{\cdot}08$ to $+5{\cdot}08$ cm. These results clearly indicate a significantly higher resistance to end-stop impacts for the seat equipped with the *Motion Master* damper when the seat height was adjusted closer to top and bottom end-stops ($\pm 5{\cdot}08$ cm). With both conventional and MR dampers, the energy required to induce bottoming at $-5{\cdot}08$ cm was less than that required for topping at $+5{\cdot}08$ cm, especially when the results were analyzed on the basis of VDV_{base}. At positions closer to the mid-ride, the difference in energies required by the conventional and the MR dampers was significantly less, particularly when the *Motion Master* was set for medium damping. The *Motion Master* damper was thus seen to be most effective as the suspension approached its end limits of travel. Furthermore, firm damper setting was seen to be most beneficial to prevent end-stop impacts under this particular excitation. At $-5{\cdot}08$ cm, however, both firm and medium settings were seen to lead to the same result.

As could be expected, bottoming was more prone to occur for seat height positions set below mid-ride, while topping was more frequent for seat positions above mid-ride. At mid-ride, however, differences in behaviour were observed for the various dampers: a conventional damper resulted in both top and bottom impacts being produced, while the *Motion Master* damper with medium setting involved topping only. With the firm setting of the *Motion Master* damper, no impacts could be induced due to limitations on the actuator stroke. Both performance indices based on VDV_{base} and $a_{w,base}$ led to the same observed trends, although the differences between dampers could be brought out more easily on the basis of the VDV_{base}.

4.3. PERFORMANCE EVALUATION UNDER EM1 EXCITATION

While under EM1 excitation, end-stop impacts could only be induced for seat positions at $\pm 5{\cdot}08$ cm with respect to mid-ride, an amplification of this excitation amplitude to 150% of its nominal value proved to be sufficient to provoke these impacts for most of the other seat positions closer to mid-ride. The results obtained under EM1 and amplified EM1 are summarized in Figures 8 and 9, respectively, where the performance indices reported are $SEAT$ value, VDV ratio and crest factor f_c. Under both levels of excitation, the response measures expressed in terms of VDV ratios and crest factors were found to emphasize the relative differences between conventional and *Motion Master* damping, particularly when end-stop impacts occurred. Overall, the use of the *Motion Master* damper was found to be the most effective, not necessarily to eliminate the occurrence of end-stop impacts, but to reduce the severity of these impacts whenever they occurred.

Under EM1 excitation, the results presented in Figure 8 do not reveal any notable benefit of the MR damper when the suspension height was set at $\pm 2{\cdot}54$ cm with respect to mid-ride. Within this range, no occurrences of end-stop impacts were observed with either of the two types of dampers, and the MR damper led to slightly higher values of $SEAT$, VDV ratio and crest factor for certain suspension heights. This could be expected in view of the lower damping provided by the MR fluid damper in the vicinity of mid-ride and the occurrence of dominant vibration excitation within the amplification-frequency range of the suspension. In contrast, however, the significantly higher damping provided by the MR damper when the levelled seat height was closer to the bottom end-stop, resulted in values of VDV ratio and crest factor which were quite substantially lower than those applicable to the conventional damper. This effect was not as clearly seen on the basis of the $SEAT$ value in view of the impulsive nature of the signals created by the impacts. With a seat position closer to the top end of the free travel of the suspension, the MR damper was successful in eliminating the end-stop impacts which were otherwise produced with the conventional damper. Consequently, the VDV ratio and crest factor were observed to be significantly reduced for that position.

When the excitation level was increased by a factor of 1·5 (i.e., 1·5EM1), the results shown in Figure 9 indicated significant benefits of the MR damper over the entire range of seat positions. With this excitation, the seat with the conventional damper was observed to lead to end-stop impacts at all the positions, thus necessitating the MR damper to provide increased damping either to reduce the severity or prevent the occurrence of impacts. The significant reductions in VDV ratio observed in Figure 9 for the MR damper were also associated with corresponding reductions in the crest factor. In contrast to the reduction in

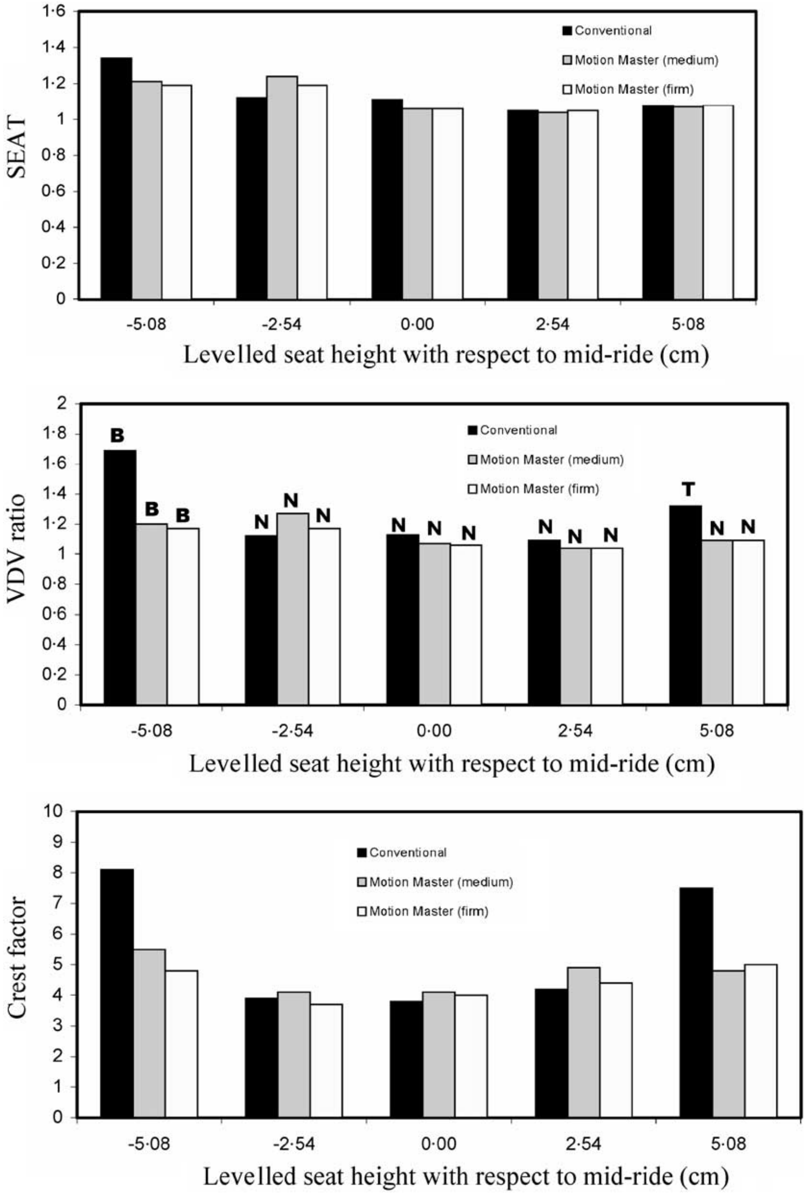

Figure 8. Variation of *SEAT* value, *VDV* ratio and crest factor with levelled seat height for the various seat–damper combinations under EM1 excitation (B = bottoming, T = topping, N = no impact).

VDV ratio, the reduction in crest factor was perhaps not as pronounced for the seat position set at $-5{\cdot}08$ cm owing to the reduced frequency of the impacts brought by the *Motion Master* damper. For all the seat positions considered under 1·5EM1 excitation, a firm damping setting of the MR damper was seen to be most beneficial in reducing the exposure levels, irrespective of whether or not end-stop impacts had occurred. A summary is provided in Figure 10 of the per cent reduction of seat frequency-weighted r.m.s. acceleration a_w, *VDV* and peak frequency-weighted acceleration a_{wp}, brought about by the use of the

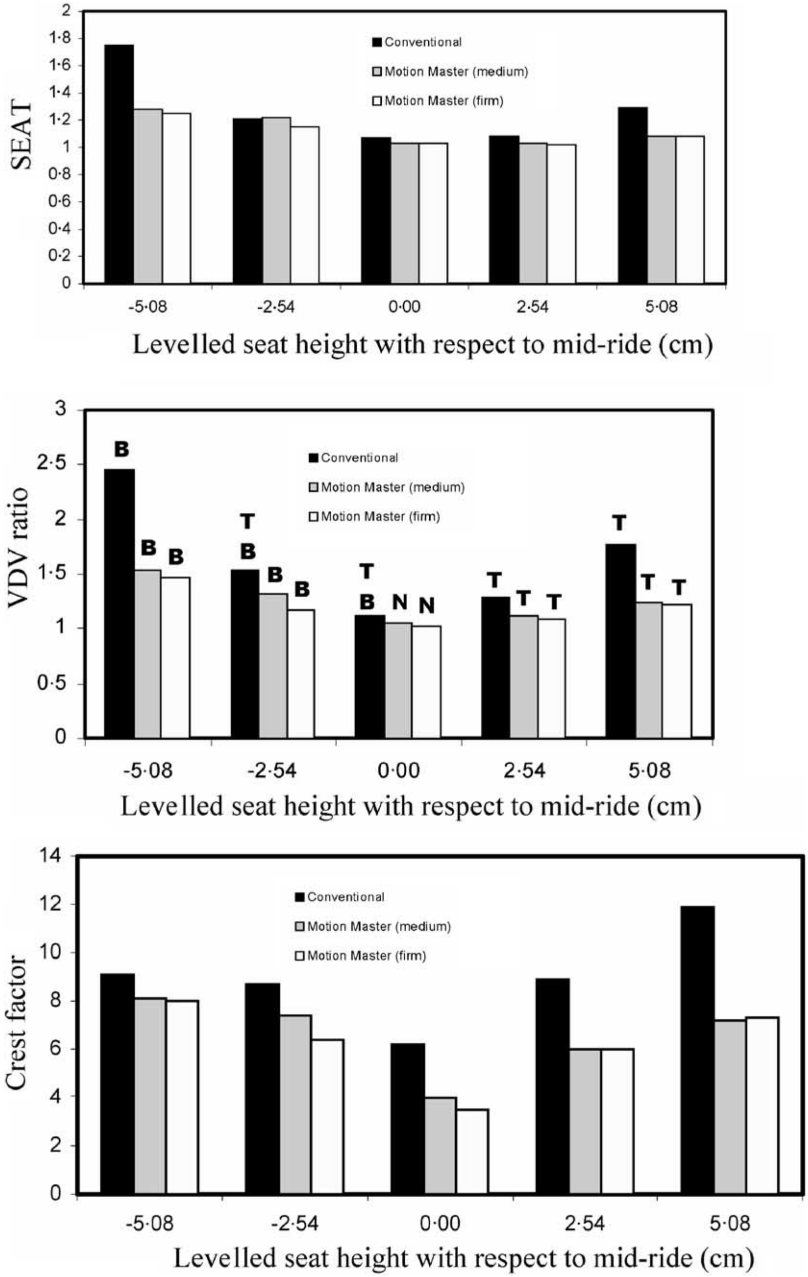

Figure 9. Variation of *SEAT* value, *VDV* ratio and crest factor with levelled seat height for the various seat–damper combinations under 1·5EM1 excitation (B = bottoming, T = topping, N = no impact).

Motion Master damper with firm setting relative to that of the conventional damper under 1·5EM1 excitation. As expected, these results suggest that the most important reductions in a_w and VDV occurred at seat positions closer to the bottom and top buffers; the reduction in VDV value approaching 40% at the position closer to the bottom end-stop. Interestingly, the reduction in peak frequency-weighted instantaneous acceleration brought about by the *Motion Master* was observed to be quite important (i.e., more than 30%) at all

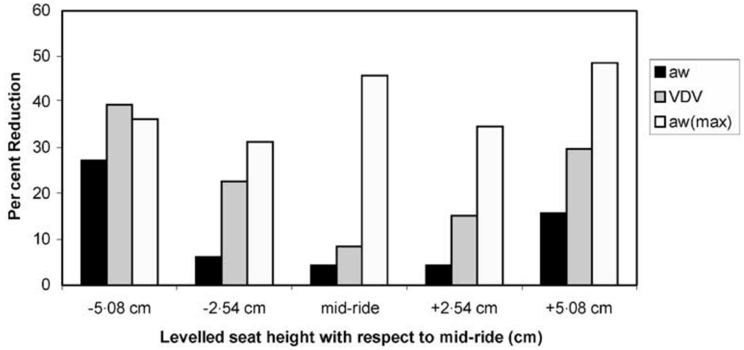

Figure 10. Reduction in vibration exposure characteristics related to the use of the *Motion Master* damper with firm setting under 1·5EM1 excitation.

the seat positions, particularly at mid-ride where end-stop impacts were completely eliminated.

5. CONCLUSIONS

The semi-active magnetorheological (MR) fluid damper, integrated within a suspension seat, provided increased damping when the seat displacements approached the end limits of the free travel of the suspension, and lower damping when set closer to mid-ride. The MR-fluid damper thus offered considerable potential to achieve a better compromise between the shock and vibration attenuation performance of suspension seats. The results clearly showed superior performance of the MR fluid damper in reducing the occurrence and severity of end-stop impacts, specifically under high-magnitude excitations and when the suspension ride height was adjusted closer to the end-stops. The benefical effects of such a damper for reducing whole-body vibration exposure levels expressed in terms of frequency-weighted r.m.s. acceleration and *Vibration dose value* (*VDV*) were seen to be most important in conditions when end-stop impacts would otherwise have been imminent when using a conventional damper. In such cases, reductions in *VDV* approaching 40% were possible when replacing a conventional damper with the MR damper. In conditions, where the end-stop impacts did not occur while using a conventional damper, the benefits of the MR damper were observed to be less important, the overall relative performance varying with the seat height position and MR damping setting.

REFERENCES

1. P.-É. Boileau and S. Rakheja 1990 *International Journal of Industrial Ergonomics* **5**, 275–291. Vibration attenuation performance of suspension seats for off-road forestry vehicles.
2. S. Rakheja, Y. Afework and S. Sankar 1994 *Vehicle System Dynamics* **23**, 501–524. An analytical and experimental investigation of the driver–seat–suspension system.
3. X. Wu, S. Rakheja and P.-É. Boileau 1999 *Proceedings of the SAE International Congress, Detroit, U.S.A., March, Paper No.* 1999-01-1304. Dynamic performance of suspension seats under vehicular vibration and shock excitations.
4. X. Wu and M. J. Griffin 1996 *Journal of Sound and Vibration* **192**, 307–319. Towards the standardization of a testing method for the end-stop impacts of suspension seats.
5. International Organization for Standardization 2000 *International Standard* ISO-7096. Earth moving machinery—Laboratory evaluation of operator seat vibration.

6. J. REBELLE 2000 *Proceedings of the 35th U.K. Group Meeting on Human Responses to Vibration, Southampton, England*, 221–238, *September*. Development of a numerical model of seat suspension to optimize the end-stop buffers.
7. INTERNATIONAL ORGANIZATION FOR STANDARDIZATION 1997 *International Standard* ISO-2631/1. Mechanical vibration and shock—Evaluation of human exposure to whole-body vibration. Part 1: general requirements.
8. P.-É. BOILEAU, S. RAKHEJA and J. BOUTIN 2000 *Proceedings of the 35th U.K. Group Meeting on Human Responses to Vibration, Southampton, England*, 305–316, *September*. Characterization of the vibration environment of urban buses.
9. S. RAKHEJA, A. K. W. AHMED, X. YANG and C. GUERETTE 1999 *Proceedings of the SAE International Truck and Bus Meeting, Detroit*, U.S.A., *November*. Optimal suspension damping for improved driver- and road-friendliness of urban buses.
10. T. GUNSTON 2000 *Proceedings of the 35th U.K. Group Meeting on Human Responses to Vibration, Southampton, England*, 207–219, *September*. An investigation of suspension seat damping using a theoretical model.
11. P.-É. BOILEAU, S. RAKHEJA and P. J. LIU 1997 *International Journal of Vehicle Design, Heavy Vehicle Systems* **4**, 244–265. A combined seat suspension—vehicle driver model for estimating the exposure to whole-body vehicular vibration and shock.
12. L. F. STIKELEATHER and C. W. SUGGS 1970 *Transactions of the American Society of Agricultural Engineers* **13**, 99–106. An active suspension system for off-road vehicles.
13. G. J. STEIN and I. BALLO 1991 *Vehicle System Dynamics* **20**, 57–58. Active vibration control for the driver's seat for off-road vehicles.
14. X. WU and M. J. GRIFFIN 1997 *Journal of Sound and Vibration* **203**, 781–793. A semi-active control policy to reduce the occurrence and severity of end-stop impacts in a suspension seat with an electrorheological fluid damper.
15. Y. K. AHN, M. AHMADIAN and S. MORISHITA 1999 *Vehicle System Dynamics* **32**, 199–216. On the design and development of a magnetorheological mount.