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A SEMI-ACTIVE CONTROL POLICY TO REDUCE THE OCCURRENCE AND SEVERITY OF END-STOP IMPACTS IN A SUSPENSION SEAT WITH AN ELECTRORHEOLOGICAL FLUID DAMPER

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A semi-active on-off control policy has been developed to reduce the severity of suspension seat end-stop impacts caused by shocks or high magnitude vibration. An electrorheological fluid damper was used to realize the required two-state damping. The effects of the free travel (i.e., the relative displacement within which the suspension damper has low damping) and the on-state damping on end-stop impacts were investigated with a sinusoidal input motion. It was found that both a shorter free travel and higher on-state damping reduced both the occurrence of end-stop impacts and their severity. The control policy was also tested with a random signal at different input magnitudes. The on-off control policy improved the performance of the seat suspension when end-stop impacts would otherwise occur with high magnitude inputs, without causing poor vibration isolation with low magnitude inputs. It is concluded that a successful compromise can be achieved between steady vibration isolation and end-stop impact reduction.

1. INTRODUCTION

Many vehicles used for industrial, agricultural, forestry and other transport purposes have significant vertical vibration in the frequency range 1–12 Hz, especially in the range 3–5 Hz. Because conventional seat cushions cannot isolate such vibration, suspension seats, with a suspension mechanism below the seat cushion, were developed. The suspension mechanism is designed to isolate vibration, with the seat cushion primarily used to provide a reasonable body pressure distribution. Usually, the suspension mechanism has a low stiffness for good vibration isolation performance, but limited travel so as not to impede a driver's interaction with controls. Suspensions with low stiffness have a large dynamic deflection when exposed to either shocks or low frequency vibration of high magnitude. A large deflection may cause the suspension to reach the limits of its travel and produce an impact (i.e., an "end-stop impact"). Many suspension seats use rubber buffers to minimize the severity of these impacts.

End-stop impacts can be a problem with suspension seats in vehicles. Stiles *et al*. [1] reported the results of a field survey of the vibration experienced by drivers of agricultural tractors. They found that 45% of suspension seats increased the acceleration levels and suggested that end-stop impacts degraded the performance of the suspension seats. Wu and Griffin [2] presented a method of testing suspension seats for end-stop impacts, classified their vibration responses into five stages according to input magnitude, and

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analyzed the influential factors in the different stages. They suggested that their end-stop impact test could supplement existing tests so as to evaluate the overall performance of suspension seats.

A parametric sensitivity study has also been conducted by means of computer simulation [3]. It was found that the end-stop impact performance of suspension seats with only bottom buffers could be improved by the use of both top and bottom buffers, and by selecting an appropriate stiffness for the buffers. It was suggested that a damper with an instantly adjustable damping force could further improve the dynamic performance of a seat suspension by providing low damping to isolate low magnitude vibration and high damping when end-stop impacts are more likely to occur. Lewis [4] described a test, based on a single-degree-of-freedom model, for predicting the occurrence of end-stop impacts. He concluded that this simple model could be used to predict the occurrence of end-stop impacts with a given combination of suspension seat and vehicle motion.

Passive seat suspensions are widely used as relatively inexpensive, simple and reliable devices to isolate vibration. However, the inherent performance limitations of passive suspensions, especially those arising from fixed damping, have encouraged researchers to develop active suspensions. Active suspensions can greatly improve vibration isolation performance [5, 6]. However, active suspensions require an energy source (such as a compressor or pump), sensors, controllers, actuators, servo-valves, switching devices and a computer control in order to achieve superior vibration isolation. In consequence, they are more expensive, more complex and less reliable, and so the implementation of active shock and vibration isolation systems has been limited [7]. To achieve the required performance benefits and overcome limitations of active systems, Karnopp *et al*. [8] developed the concept of a semi-active vibration isolation system. Semi-active isolation systems are similar to passive systems in that all suspension elements generate their respective forces passively. However, the damping force generated by the damper can be varied according to a control policy so as to achieve the best vibration isolation performance.

Most semi-active control policies have been developed to reduce the r.m.s. acceleration of a sprung mass, not the instantaneous acceleration or instantaneous relative displacement, which is important when preventing end-stop impacts. In a non-linear system with "on-off" damping, an improved steady acceleration isolation performance does not necessarily mean a reduction in the relative displacement or, therefore, a reduction in either the occurrence or the severity of end-stop impacts.

In this paper, a semi-active on-off control policy is described, by means of which one can isolate low magnitude vibration but actively control the damper when end-stop impacts would otherwise occur. An electrorheological fluid damper was used to realize the required two state damping. An experimental study has been conducted to assess the effectiveness of this control policy.

2. REVIEW OF SEMI-ACTIVE CONTROL POLICIES AND HARDWARE IMPLEMENTATION WITH AN ACTIVE DAMPER

The initial semi-active system was based on skyhook semi-active control [8, 9]. The control policy was designed to modulate the force generated by a passive device to approximate the force that would be generated by a damper connected to ground (skyhook damper). The passive device could only absorb vibration energy. Whenever it was required to supply energy to the system, the best the device could do was to supply no force at all. Otherwise the device would provide a force proportional to the absolute velocity of the mass. The switching of the device could be controlled by the term $\dot{x}(\dot{x} - \dot{x}_0)$. If the product of the absolute velocity, $\dot{x}$, of the mass and the relative velocity, $(\dot{x} - \dot{x}_0)$, between the mass and the base was positive, the damper was switched ''on'', so a force was generated to reduce the acceleration of the mass. If this term was negative, the damper was switched ''off'' so that no force was generated. The control policy can be expressed as:

$$Fd = \begin{Bmatrix} c\dot{x}, & \dot{x}(\dot{x} - \dot{x}_0) > 0 \\ 0, & \dot{x}(\dot{x} - \dot{x}_0) < 0 \end{Bmatrix}. \tag{1}$$

It was shown that skyhook semi-active control can provide a performance very similar to that of an active system. However, the cost and complexity of skyhook control may be prohibitive for general use. In order to simplify the hardware implementation and reduce the cost, a number of simplified on-off control policies have been developed.

Krasnicki [10] analyzed the performance of a simplified on-off control scheme. The on-off damper operates between the sprung mass and the base as a conventional passive damper during the vibration attenuation portion of the vibration cycle, but a zero damping coefficient is assumed when a passive damper would normally accelerate the mass. The damper force generated is proportional to the relative velocity between the mass and the base, instead of the absolute velocity of the sprung mass. The control policy can be expressed by the following equation

$$Fd = \begin{Bmatrix} c(\dot{x} - \dot{x}_0), & \dot{x}(\dot{x} - \dot{x}_0) > 0 \\ 0, & \dot{x}(\dot{x} - \dot{x}_0) < 0 \end{Bmatrix}. \tag{2}$$

This control policy requires a measure of the absolute velocity as well as the relative velocity. The accurate measurement of the absolute velocity may be difficult to achieve. Integrating the signal from an accelerometer to obtain velocity does not always yield sufficiently accurate results, particularly at low frequencies.

Because damper force causes an increase in the magnitude of the mass acceleration whenever forces due to the spring and the damper have the same sign, and the signs of the damper and the spring force are the same as the signs of relative velocity and relative displacement, respectively, Rakheja and Sankar [11] developed their on-off control scheme:

$$Fd = \begin{Bmatrix} c(\dot{x} - \dot{x}_0), & (x - x_0)(\dot{x} - \dot{x}_0) < 0 \\ 0, & (x - x_0)(\dot{x} - \dot{x}_0) > 0 \end{Bmatrix}. \tag{3}$$

That is, whenever the damper force may accelerate the mass, the damper is switched ''off''; whenever the damper force can react with the spring force so as to decelerate the mass, the damper is switched ''on''. This policy is easy to implement, as only the relative displacement and the relative velocity have to be measured. This can be achieved by using simple position and velocity transducers.

However, this control policy has potential for improvement. During the on-state of the damper, the instantaneous damper force is seldom exactly equal in magnitude to the instantaneous spring force. In consequence, the surplus force will still accelerate the mass. Alanoly and Sankar [12] proposed a continuous control policy, which can be considered as a further development of the preceding policy. In this control policy, the damping coefficient is continuously variable, depending on the relative displacement and the relative velocity:

$$c = \begin{cases} -k(x - x_0)/(\dot{x} - \dot{x}_0), & (x - x_0)(\dot{x} - \dot{x}_0) < 0 \\ c_{off}, & (x - x_0)(\dot{x} - \dot{x}_0) > 0 \end{cases}. \tag{4}$$

This control policy indicates that if the spring force and the damping force exerted on the sprung mass are in the same direction, to reduce the sprung mass acceleration, the damping force should be a minimum value c_{off} (because actual dampers cannot provide zero damping force even if they are switched "off"). On the other hand, if the spring force and the damping force are in opposite directions, then the damping force should be adjusted in such a way that it will be equal to the spring force in magnitude so as to produce zero acceleration of the sprung mass.

Wu *et al*. [13] pointed out that in this control policy the desired damping force may be beyond the range that the damper can supply. In this case, the maximum possible damping c_{on} should be applied. Furthermore, they developed a new control policy based on the preceding scheme. Instead of continuously adjusting the damping force, the damping is set at either the maximum value, c_{on}, or the minimum value, c_{off}, depending upon the operating conditions:

$$c = \begin{cases} c_{on}, & -k(x - x_0)/(\dot{x} - \dot{x}_0) > c_{th} \\ c_{off}, & -k(x - x_0)/(\dot{x} - \dot{x}_0) < c_{th} \end{cases}. \tag{5}$$

Here c_{th} is referred to as the threshold damping coefficient. They suggested that c_{th} might be taken as 30% of the critical damping coefficient for the spring-mass system.

Decker and Schramm [14] proposed a control policy such that if the absolute velocity of the spring mass is within a given band ($|\dot{x}| < v_{th}$), then the required damper force is low and the damper is switched off, irrespective of whether the absolute velocity is positive or negative. If the absolute velocity of the sprung mass is beyond the given band, there are two cases: (1) if $\dot{x} > v_{th}$, the damper should be switched on during the rebound stage ($\dot{x} - \dot{x}_0 > 0$) but not during the compression stage ($\dot{x} - \dot{x}_0 < 0$); (2) if $\dot{x} < v_{th}$, the damper should be switched on during the compression stage ($\dot{x} - \dot{x}_0 < 0$) but not during the rebound stage ($\dot{x} - \dot{x}_0 > 0$). This control policy can be expressed as

$$c = \begin{cases} c_{on}, & \dot{x} > v_{th} \quad \text{and} \quad \dot{x}(\dot{x} - \dot{x}_0) > 0 \\ c_{off}, & \dot{x} < -v_{th} \quad \text{and} \quad \dot{x}(\dot{x} - \dot{x}_0) < 0 \\ c_{off}, & |\dot{x}| < v_{th} \end{cases}. \tag{6}$$

Generally, the acceleration response of an on-off damper exhibits discontinuities at the time of switching, thus a significant jerk may be experienced by the system mass. An on-off scheme has been devised to minimize the jerk experienced by the mass [15]. The switching term of the damper is the product of the relative velocity and the relative acceleration $(\dot{x} - \dot{x}_0)(\ddot{x} - \ddot{x}_0)$. Miller and Nobles [16] presented and discussed a similar problem in both on-off and continuously variable semi-active suspensions. They suggested solutions involving modifications to the basic control policies or system hardware.

Currently, there are two kinds of active damper hardware implementation. The first kind of damper changes the damping coefficients by the modulation of the orifice area through which the damper fluid flows. The second kind of damper changes the damping coefficients by controlling the viscosity of the damper liquid (i.e., electrorheological fluid, ERF).

Krasnicki [10] used a damper consisting of an hydraulic actuator used in conjunction with an electro-hydraulic servo-valve modulating the controlling orifice area. In the off-state, the full command voltage was applied to the valve, while zero voltage was applied to the valve in the on-state. Nell and Steyn [17] used a two-state controllable damper consisting of a cylinder, valve ports and a conventional valve configuration. Wong *et al*. [18] and Wu *et al*. [13] used an ERF fluid damper in their study. In this study, an ERF damper is also used. This kind of damper is discussed in more detail below. Miller [19] studied the effect of hardware limitations on an on-off semi-active suspension, including the effect of non-zero off-state damping, valve dynamics, and digital filter dynamics. These effects were investigated by examining the frequency response for a single-degree-of-freedom model. It was found that these factors can degrade the performance of an on-off suspension.

3. ON-OFF CONTROL POLICY FOR REDUCING END-STOP IMPACTS

For a seat suspension, high damping will suppress the transmissibility at resonance, but worsen the vibration isolation of high frequency components. Low damping, will improve the isolation of the high frequency components, but the transmissibility at resonance may be high. The relative displacement between the sprung part of a seat-suspension and the seat base may be large enough to cause end-stop impacts. The lower the damping, the higher the probability of end-stop impacts and the greater the severity of end-stop impacts [2].

Usually, seat suspensions are designed such that their resonance frequencies are well below the dominant frequency of the input vibration. If the input vibration does not cause the sprung part of a suspension mechanism to hit the end-stop buffers, a lower suspension damping may provide greater vibration isolation. If the input vibration is so high that severe end-stop impacts occur, greater suspension damping will reduce both the occurrence and the severity of end-stop impacts. In order to get optimum performance, an adjustable damper, which can be switched manually or automatically between a hard mode and a soft mode according to the driver's estimate of the surface roughness or the recent average input motion at the seat base, might be used. This type of damper control is defined as hard-soft control. However, the input vibration may not be statistically stationary. Even when a vehicle operates on a rough field, some periods of the input vibration may not cause end-stop impacts. Consequently, hard mode damping may unnecessarily increase the vibration exposure of the driver due to poor isolation of the high frequency components of the input vibration.

If the damper is generally set to soft mode so as to provide good isolation of the high frequency components, and adjusted automatically to the hard mode only when end-stop impacts are likely to occur (i.e., when hard mode is necessary), the optimum performance might be achieved. The damper switching must then depend on the instantaneous motion instead of the recent average vibration. To distinguish damper switching based on the instantaneous motion from that based on the recent average vibration, hard mode and soft mode damping are called "on-state" and "off-state" damping, respectively, in the following text.

End-stop impacts will occur whenever the relative displacement between the sprung part of a seat suspension and the seat base exceeds half of the suspension travel. If the damper is switched "on" whenever the relative displacement exceeds a pre-set displacement threshold, d_{th} (see equation 6), severe end-stop impacts might be prevented. Twice the value of this threshold defines the free travel of the suspension. Therefore, a free travel of ± 0 mm means that the damper is always "on" (i.e., hard mode) and a free travel of $+\infty$ mm means that the damper is always "off" (i.e., soft mode):

$$c = \begin{Bmatrix} c_{on}, & |d| > d_{th} \\ c_{off}, & |d| < d_{th} \end{Bmatrix}. \tag{6}$$

The free travel of the suspension determines the dynamic performance of the seat suspension. The shorter the free travel, the more similar the performance will be to that with hard mode damping, not isolating well the high frequency components; the longer the free travel, the more similar the performance to that of soft mode damping, with less prevention of end-stop impacts. The optimum free travel, which depends on the input motion as well as the suspension mechanism, must be determined. The characteristics of the on-state damping also needs to be studied: the input vibration may not be so large as to need the maximum on-state damping to reduce end-stop impacts.

The relative displacement of a seat suspension can be either predicted from the instantaneous input vibration or measured with a displacement transducer. Lewis [4] predicted the occurrence of suspension seat end-stop impacts by predicting the relative displacement of a seat suspension with a single-degree-of-freedom linear model. The acceleration time history at the seat base was passed through a digital filter, which had the characteristics of the relative displacement transmissibility of the single-degree-of-freedom seat suspension linear model (the relative displacement divided by the acceleration of the seat base, i.e., equation 7). For a particular combination of suspension seat and vehicle motion, a comparison of predicted and measured values leads to the conclusion that a single-degree-of-freedom linear model could be used to predict the relative displacement of the seat, and thus the probability of the occurrence of end-stop impacts, when the relative displacements are large. Thus one can consider the equation

$$H(s) = \frac{-1}{(s^2 + 2\zeta\omega_n s + \omega_n^2)}\ , \tag{7}$$

where s is the Laplace operator, ω_n is the natural radial frequency and ζ is the damping ratio of the suspension.

The advantage of this method is that only an accelerometer is needed, with the digital filter representing the response of the seat included in the control program. However, the method may introduce some error in the prediction for the periods immediately following the switching-on of the damper or the occurrence of an end-stop impact: these non-linearities will impair the application of the linear model. If the relative displacement repeatedly exceeds the threshold, the prediction may not be accurate

In this study the relative displacement was measured directly with a displacement transducer. Two accelerometers were used to measure the accelerations on top of the suspension and at the base of the seat. This made it possible to evaluate the severity of the end-stop impacts.

4. ELECTRORHEOLOGICAL DAMPER AND DESCRIPTION OF EXPERIMENT

An ERF fluid is a mixture of an insulating oil and a semi-conducting, solid particulate. The viscosity of the fluid increases with increased electric field applied across the fluid. This causes increased resistance to the fluid flow and so variable damping can be achieved by varying the electrical field. It has been reported that ERF fluids have sufficiently fast response times (a few milliseconds) to be suitable for continuously adjustable dampers [18]. The voltages required to activate the phase change in ERF fluids are typically in the order of 4 kV/mm of fluid thickness with a current in the order of 10 μA. So the total power required to trigger the phenomenon is quite low [20]. The ERF dampers used by Wong *et al*. [18] provided a wide range of damping ratios, between 0·1 to 1·5, but their electrical and mechanical properties were highly dependent on the operating temperature. At sub-zero temperatures, the electrorheological effect was not significant. Wu *et al*. [13] found that the damping characteristics of the ERF damper used in their study depended not only on the electrical field strength but also on the frequency of excitation. For the ERF damper used, the equivalent damping ratio decreased significantly with an increase in the excitation frequency. They believed that this was primarily because the shear ratio of the fluid they used decreased with an increase in the shear rate. At high frequencies of excitation, the duration of the applied voltage could be too short for the mechanical properties of the fluid to change sufficiently; the semi-active system would then behave similarly to a passive system. At low frequencies of excitation, the control voltage applied to the ERF will last for a longer time and will be more likely to realize the intended change in the mechanical properties of the ERF.

In this study, an ERF damper was used to realize the intended on-state and off-state damping. The damper was a double-tube shock absorber with a fixed fluid area. The maximum stroke of the damper was 40 mm. When fitted to the seat suspension, the damper had an inclination angle of approximately 45° when the seat suspension was at its mid-position. It was used at room temperature and with low frequencies of excitation. In preliminary tests, the electrorheological effect was found to be significant and controllable. The energy consumptions at 500 V, 1000 V, and 1500 V were 0·4 W, 3·2 W, and 8·9 W, respectively.

Supporting a rigid mass of 55 kg, the seat suspension tested had a resonance frequency of 1·7 Hz and a linear travel of 60 mm. It had a pair of bottom rubber buffers with a thickness of 35 mm but no top buffers. The seat was designed for use in trucks and tractors. A computer program was written in the *HVLab* software package to drive a vertical vibrator, on which the suspension seat was attached, and simultaneously acquire the relative displacement of the suspension, the acceleration at the seat base, and the acceleration on top of the seat suspension. The software also controlled the damper according to the on-off control policy. Piezo-resistive accelerometers (Entran model EGCSY-240D-10) and an LVDT displacement transducer were used. A DC voltage amplifier linearly amplified the control signal from the computer (0 to 5 V) to the voltage applied to the ER fluid (0 to 1500 V). The experiment arrangement is illustrated in Figure 1.

Two types of excitation signal were used in different tests. The first was a 5 s sinusoidal signal at the resonance frequency of the seat suspension. The second was a 60 s random signal with the spectrum specified in ISO/DIS 5007 [21] for class III tractors. The centre frequency of the spectrum was 2·2 Hz, close to the resonance frequency of the seat suspension and so likely to induce end-stop impacts. In the tests, the input magnitudes of the two signals were increased step by step so as to investigate the suspension performance with different input magnitudes. The on-state voltage applied to the damper was 1500 V (except where stated), and the off-state voltage was 0 V. Suspension vibration dose value (VDV) ratios were used to evaluate the dynamic performance of the seat suspension, as suggested by Wu and Griffin [2]:

$$\text{Suspension VDV ratio} = \frac{\text{VDF on top of suspension}}{\text{VDV at base}}, \tag{8}$$

where the VDV was calculated with the frequency weighting W_b [22].

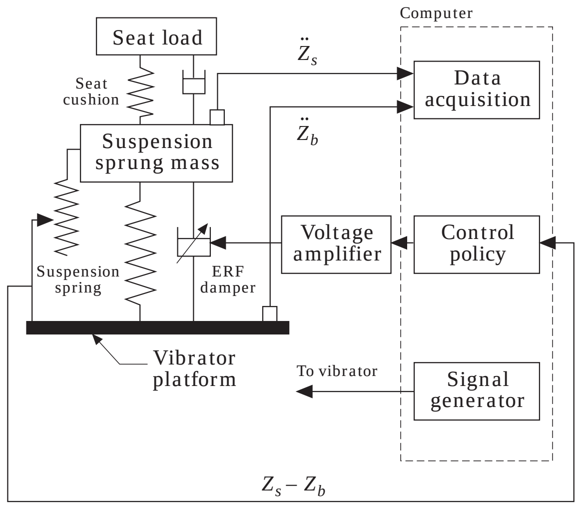

Figure 1. Schematic diagram of the experimental arrangement.

Figure 2 shows the time histories of the acceleration, on top of the suspension, and the suspension relative displacement with 0 V and 1500 V applied to the damper, respectively (i.e., the damper with soft mode and hard mode damping) while exposed to the random

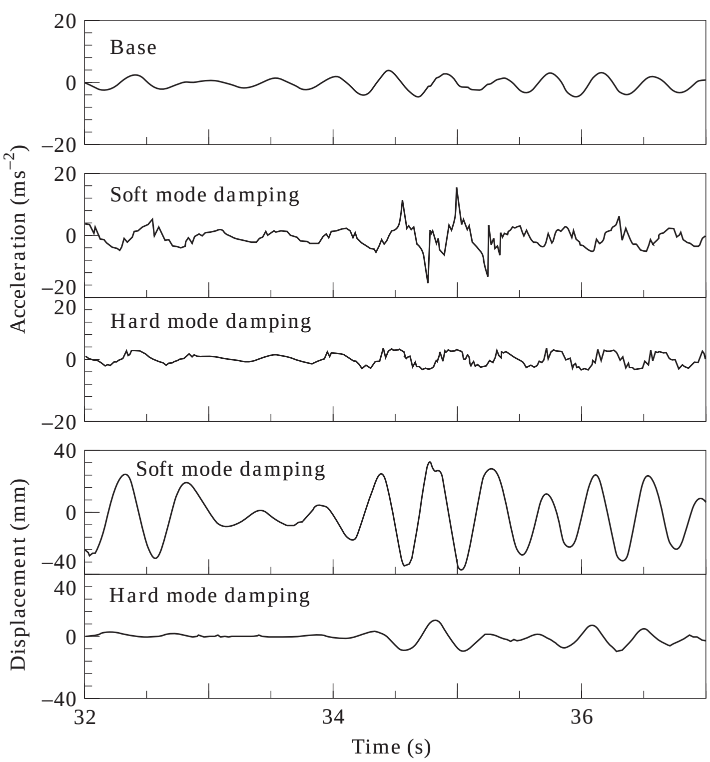

Figure 2. Acceleration on top of the suspension, and relative displacement of the suspension with soft mode and hard mode damping.

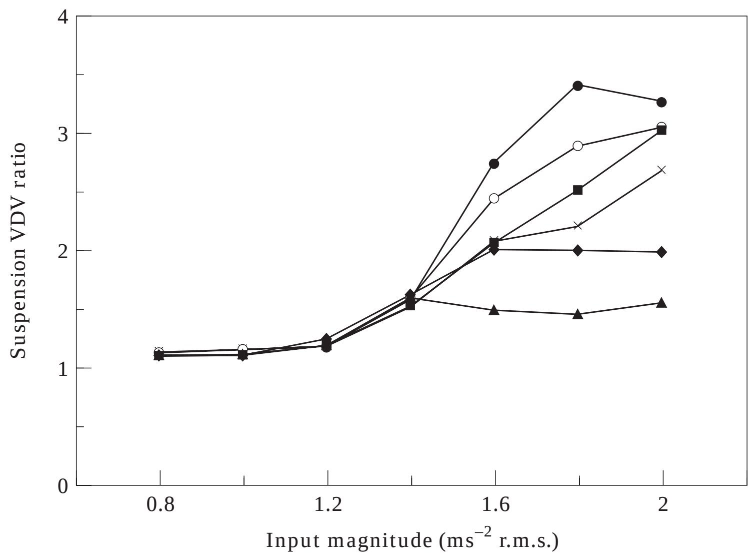

Figure 3. Suspension VDV ratio with various lengths of free travel (from ± 5 mm to infinity) and different input magnitudes of sinusoidal motion. —●—, Soft mode; —○—, ± 25 mm; —■—, ± 20 mm; ⋇, ± 15 mm; —◆—, ± 10 mm; —▲—, ± 5 mm.

vibration with the input magnitude of the base excitation at 1·56 ms^{-2} r.m.s. For this input magnitude, with soft mode damping, the suspension relative displacement exceeded the linear travel of the seat suspension (i.e., ± 30 mm) and severe end-stop impacts occurred. With this input and hard mode damping, the suspension was locked up, except for the largest excitations, and so the suspension mechanism seldom worked during the 60 s excitation. The significant difference between the two responses indicates that the ERF damper was capable of providing a wide range of damping. This suggests it would be suitable for testing the benefits of the on-off control policy.

5. EFFECTIVENESS OF ON-OFF DAMPER IN REDUCING END-STOP IMPACTS

5.1. EFFECT OF DIFFERENT FREE TRAVEL

Excited with the sinusoidal signal, the suspension seat was tested with different amounts of free travel. For soft mode damping (i.e., a free travel of $+\infty$ mm) and for conditions with the free travel at ± 25, ± 20, ± 15, ± 10 and ± 5 mm, the relation between the input magnitude and the measured suspension VDV ratio is shown in Figure 3. When the input magnitude was sufficiently high to cause severe end-stop impacts, a short free travel reduced the severity of the impacts. The shorter the free travel, the more significant the reduction in the severity of impacts. However, a very short free travel will be expected to introduce unnecessarily high damping and so degrade the isolation of low magnitude vibration, as discussed above. The severity of end-stop impacts was not reduced significantly when the free travel was too long (for example ± 25 mm): the damping force did not have sufficient time to do work and dissipate the vibration energy of the sprung mass, and so end-stop impacts occurred. The response delay of the control system made the actual free travel longer than the pre-set threshold. Figure 3 suggests that a free travel ± 10 mm or ± 15 mm may be suitable.

5.2. EFFECT OF ON-STATE DAMPING

A second test was conducted in which the free travel was constant ± 15 mm, but the on-state control voltage varied from 300 to 1500 V. The results are shown in Figure 4. A higher on-state control voltage, which yielded a higher on-state damping, reduced the severity of the end-stop impacts. However, this may not have been found if the damper had provided extremely high on-state damping: the switching of the damper could cause impacts similar to those caused by end-stop buffers. Miller and Nobles [16] presented and discussed a similar problem in both on-off and continuously variable semi-active suspensions.

5.3. TEST RESULTS USING RANDOM VIBRATION INPUT

The suspension seat was also tested with the random signal. The on-state control voltage was 1500 V. The acceleration time histories on top of the seat suspension, and the suspension relative displacement without the on-off control (i.e., soft mode damping: a free travel of $+\infty$ mm) and with on-off control (a free travel of ± 10 mm) are shown in Figure 5 with the same input magnitude as the conditions in Figure 2. With on-off control, when the instantaneous base acceleration was low, the suspension vibration was almost the same as that without on-off control. When the instantaneous base acceleration was high, the peak acceleration decreased significantly compared with that without on-off control. This suggests that the on-off control policy was effective in reducing the severity of end-stop impacts, without worsening the steady vibration isolation. By comparing Figure 5 with Figure 2, it can be seen that with the on-off control a compromise was reached between steady vibration isolation and reduction of end-stop impact severity, while with merely hard-soft control the reduction of end-stop impact severity impaired the isolation of lower magnitude vibration.

Different input magnitudes were used to investigate the suspension performance: the lowest input magnitude made the suspension mechanism work in its linear operation range

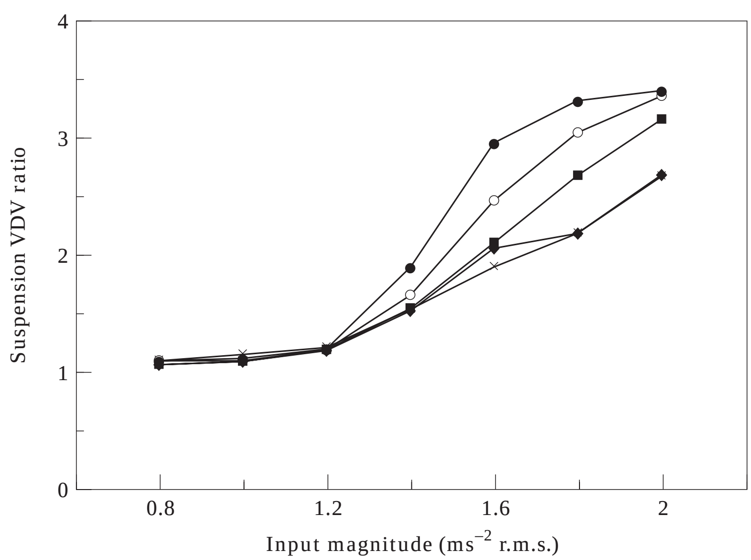

Figure 4. Suspension VDV ratio with various different on-state voltages applied to electrorheological fluid damper with different input magnitudes of sinusoidal motion. —●—, 300 V; —○—, 600 V; —■—, 900 V; ⁎, 1200 V; —◆—, 1500 V.

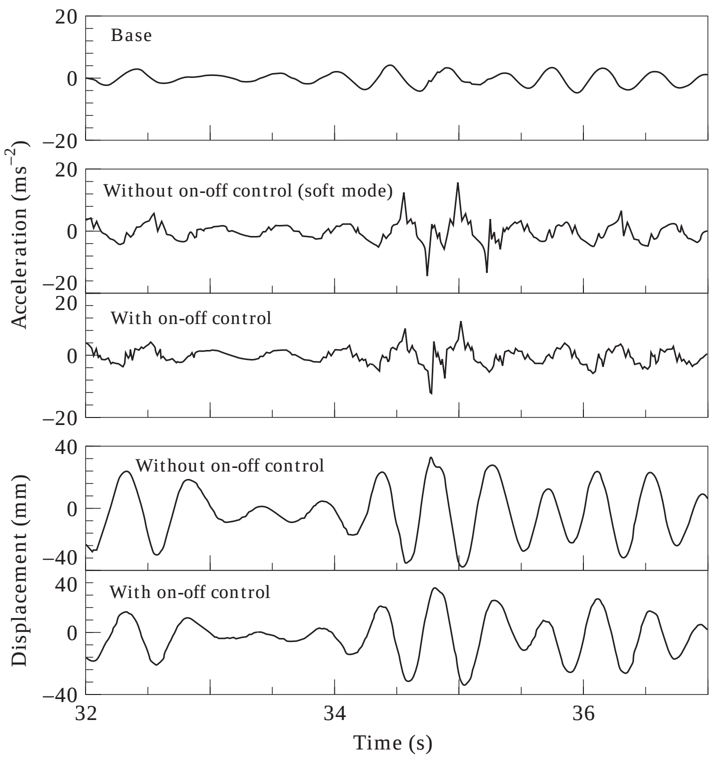

Figure 5. Acceleration on top of the suspension, and the relative displacement of the suspension without on-off control and with on-off control.

while the highest input magnitudes caused severe impacts when using soft mode damping. Figure 6 shows the measured relation between the input magnitude and the VDV ratio.

With high input magnitudes, on-off control successfully reduced the severity of end-stop impacts when severe end-stop impacts occurred with soft mode damping. With high input magnitudes, the performance with on-off control was close to that with hard mode damping and superior to that with soft mode ($\pm\infty$ mm free travel). With low input magnitudes, the performance with on-off control was similar to that with soft mode damping and superior to that with hard mode (± 0 mm free travel). Therefore, the on-off control policy enabled the seat suspension to achieve a better dynamic performance with all magnitudes of the input vibration and so reach a successful compromise between vibration isolation and end-stop impact reduction.

6. DISCUSSIONS AND CALCULATIONS

Conventional suspension seats use rubber buffers to reduce the severity of end-stop impacts. An impact between a relatively soft rubber buffer and the sprung mass is less severe than that between two stiff steel surfaces and, therefore, the severity of end-stop impacts can be reduced to some degree by appropriate selection of the stiffness and thickness of rubber buffers [3]. However, current rubber materials only act as springs to store vibration energy: due to their low damping they cannot absorb much vibration energy. The ability of conventional rubber buffers to reduce the severity of end-stop impacts is therefore limited.

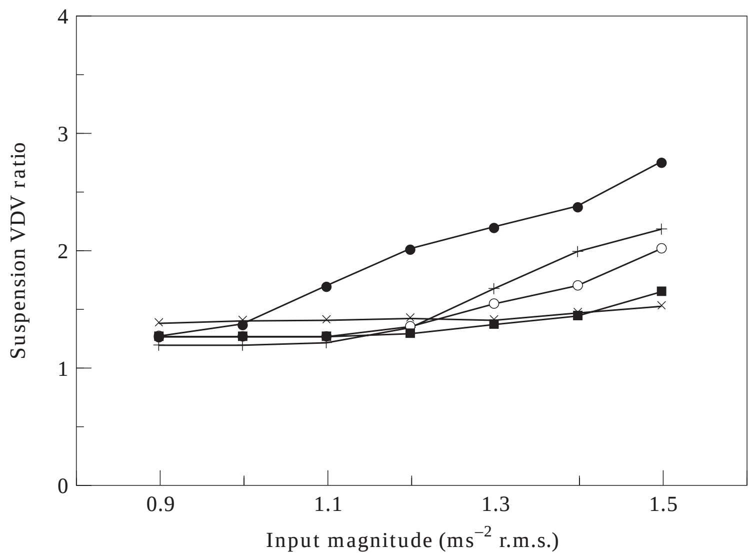

Figure 6. Suspension VDV ratio with different free travels (from ± 0 mm to infinity) and different input magnitudes of random motion. —●—, Soft mode; +, $\pm 15{\cdot}0$ mm; —○—, $\pm 10{\cdot}0$ mm; —■—, ± 5.0 mm; ⋇, hard mode.

After reviewing different semi-active control policies, a new on-off control policy was developed which enabled the seat suspension to isolate passively low magnitude vibration, and control actively end-stop impacts when such impacts would otherwise occur. An electrorheological fluid damper was used to provide off-state damping (i.e., low damping) and on-state damping (high damping or "end-stop" damping).

The effects of free travel and "on" state damping were investigated with sinusoidal input motions. It was found that a shorter free travel, as well as a higher "on" state damping helped to prevent end-stop impacts from occurring, or reduced the severity of impacts that did occur.

With a random signal at various magnitudes, the performance of the seat suspension with the on-off control policy was improved when severe end-stop impacts otherwise occurred with soft mode damping (low damping), without causing poor vibration isolation with low magnitude inputs. The comparison of on-off control with hard-soft control indicated that the on-off control was superior to hard-soft control with random excitation: the on-off control always enabled the seat suspension to achieve a better dynamic performance, irrespective of vibration magnitude, and so reach a successful compromise between vibration isolation and end-stop impact reduction.

The superiority of on-off control over hard-soft control may be expected to show even more clearly with real vehicle vibration, since this usually contains a wider distribution of magnitudes and frequencies than that specified in ISO/DIS 5007 [21].

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