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# Design and Evaluation of a Suspension Seat to Reduce Vibration Exposure of Subway Operators: A Case Study

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**Abstract: Subway operators have complained about discomfort caused by whole-body vibration. To address this problem, a suspension seat with extensive ergonomic features has been adapted to the confined space of the subway operator cab. The suspension was modified from an existing suspension in order to reduce the dominant frequency of the subway vertical vibration (2.4 Hz). The suspension seat has been extensively tested on a vertical hydraulic shaker. These tests have shown that the *SEAT* value was lower for a higher vibration level, for higher subject weight, and for the suspension adjusted at median height. The seat also produces a lower *SEAT* value when there was a predominance of the 6 Hz vibration component. The horizontal seat adjustments had no influence on the suspension *SEAT* value. Removing the suspension damper also decreases the *SEAT* value for all the tested configurations. The final version of the suspension seat prototype was validated during normal subway operation with 19 different operators having weight in the 5th, 50th and 95th percentile of the operator population. Accelerations were measured with triaxial accelerometers at the seat cushion, above the suspension and on the floor. In addition to the vibration measurements, each operator was asked about his perceived discomfort from vibration exposure. Globally, the suspension seat attenuated the vertical vibration (*SEAT* values from 0.86 to 0.99), but discomfort due to amplification of the 2.4 Hz component occurred when the suspension height was adjusted at the minimum, even when the global weighted acceleration was lower (*SEAT* value < 1). These results suggest that in order to reduce the discomfort caused by whole-body vibration, the transmissibility of the seat should also be considered, in particular when there is a dominant frequency in the vibration spectra.**

**Key words:** Suspension seat, Whole-body vibration, Discomfort, Dominant frequency, Subway

## Introduction

Subway operators have complained about discomfort caused by whole-body vibration. To address this issue, an extensive study on vibration exposure of Montréal subway operators has been previously realized, where the vibration levels were measured on the different subway lines and for different type of motor cars[1]). Depending of the motor car type (MR-63 and MR-73), the average A(8) values were between 0.39 and 0.58 m/s$^2$ weighted[2]), for an average daily exposure of 5.3 hours. The problem involves mainly the MR-73 motor cars, as the MR-63 motor cars should be replaced in the following years. This MR-73 motor car is mounted on rubber tires, and another study has shown that the running rack (supporting the tires) were very flat (International Roughness Index in average lower than 1.5 m/km)[3]). Thus, the source of vibration appeared to be mainly from internal dynamic of the motor car, excited by out of balance masses located in the subway wheels and tires[1]). In addition, the operator current seat is amplifying the vertical vibration, with a *SEAT* value of 1.05. Thus, the study recommended the use of a suspension seat to reduce the vibration exposure of subway operators. The recommended dynamic characteristics for the suspension seat were a natural frequency smaller than 1.7 Hz and a damping ratio of about 0.45, in order to attenuate the 2.4 Hz dominant frequency of the MR-73 motor car[1]).

Vehicle seat comfort depends upon several aspects: static, dynamic as well as time factors[4]). In absence of vibration, seat comfort depends only on static and time factors. The static seat comfort is mainly dictated by the seat cushion, and can be increased by avoiding high contact pressure areas at the human seat interface. Time factors take into account the discomfort that appears after prolonged sitting due to fatigue. Dynamic factors represent the discomfort caused by exposure to vibration, and increase with the vibration magnitude. Dynamic comfort can be improved by using a seat that attenuates whole-body vibrations transmitted to the operators.

Suspension seats are widely used to reduce whole-body vibration exposure of seated operators. They attenuate vibration mainly in the vertical z-axis, and are used in a wide variety of off-road and on-road vehicles. Conventional seats (with a cushion only) allow only for a small relative displacement between the floor and the seated subject. For that reason,

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they are not effective in attenuating low frequency vibrations, where the associated floor displacements are larger. They have typically a resonance frequency around 4 Hz, and thus are effective to reduce whole-body vibration exposure above about 6 Hz[4]. On the other hand, suspension seats allow greater relative displacement between the floor and the seated subject, resulting in vibration attenuation at lower frequencies.

Suspension seat performances are mainly evaluated by the *SEAT* value, representing the overall attenuation of the vibration by the seat for a given excitation spectra. For a given direction (mainly vertical $z_h$-axis), the *SEAT* value is defined as:

$$SEAT = \frac{Vibration\ level\ on\ the\ seat}{Vibration\ level\ at\ the\ base\ of\ the\ seat} \qquad (1)$$

The vibration level on the seat or at the base of the seat is mainly computed using the ISO 2631-1[2] weighted root mean square (rms) acceleration ($a_w$), or the vibration dose value (*VDV*). The *SEAT* values can be used to predict the dynamic comfort of the seat for a certain user population, as some researchers have found a significant correlation ($R^2$=0.94 and $R^2$=0.97) between the *SEAT* values and the subjective comfort rating, when the results were averaged over 6 subjects[5, 6]. Dynamic seat testing can be performed in the field by measuring the vibration level on the floor and on the seat during normal vehicle operation, or in a laboratory environment under controlled conditions. The ISO 10326-1 standard[7] gives the basic requirements to perform seat testing in a laboratory environment. The tests have to be performed with at least two subjects, having body weight at the 5th and 95th percentile of the body weight of the vehicle users' population, for which the seat is intended. The seat is mounted on a vibrating platform and vibration is applied to the seat while the subject is seated in a standardized posture. The vibration is applied such that the acceleration power spectral density (*PSD*) at the base of the seat is representative of the vehicle for which the seat is tested. Three trials have to be performed for each subject, while the suspension is adjusted at mid-travel. Then, for each subject, the *SEAT* values are estimated from eq. 1, using the weighted rms accelerations averaged over the three trials.

Suspension seats design has to be optimized in order to attenuate continuous vibrations as well as shock-type vibrations. Attenuating both type of vibration pose contradictory design requirements[8]. Continuous or steady-state vibrations require a soft suspension to provide lower vibration transmission, while shock-type vibrations require a more rigid suspension in order to avoid end-stop impacts. The dynamic behaviour of suspension seats, based on the *VDV SEAT* value, has been described by Wu and Griffin[9] for different excitation frequencies and vibration magnitudes in the resonance region of the seat and in the attenuation region (above the resonance region). According to the authors[9], the suspension *VDV SEAT* values (the ratio of the *VDV* on top of the suspension to the *VDV* beneath the suspension) comprise five stages, depending on the vibration magnitude and excitation frequency. For low level vibration (stage 1), the suspension is locked due to friction and behaves like a rigid body, thus the *SEAT* value is unity. As the vibration level increases (stage 2), the suspension starts to move with respect to its base in a highly non-linear manner, and some attenuation or amplification of the base vibration occurs depending whether the excitation frequency is in the attenuation or resonance region. In stage 3, the *SEAT* value reaches a constant value of attenuation or amplification as the suspension is working in its linear range. However, as the vibration level increases further, the suspension starts to impact the end-stop buffers (stage 4), thus the *SEAT* value begins to increase rapidly (vibration amplification). Finally, in stage 5, the suspension hits the end-stop buffers in every cycle, resulting in maximum amplification of the suspension base vibration.

In order to reduce the operators' vibration exposure, a suspension seat with extensive ergonomic features and adjustments has been adapted for the confined space of the operator cab. This paper will present the test and validation of the suspension seat in the laboratory, as well as during normal subway operations.

## Method

*Overview of the seat design approach*

The suspension seat design has been realized in collaboration with several partners: a research team, composed of two specialists in vibration and two specialists in ergonomics; a seat manufacturer; and several employees from the *Société de Transport de Montréal* (STM), composed of subway operators, and representatives of the engineering, maintenance, operations, occupational health and safety and supply departments. These different partners have been part of the prototype development, insuring that the prototype was adapted to the problematic of the MR-73 motor car. The development of the prototype can be resumed roughly in these different steps:

1) Ergonomic and vibration transmissibility evaluation of several candidates of suspension seats (cushions and suspensions evaluated independently for vibration transmissibility).
2) First version of the prototype that includes all the ergonomic adjustments and a pneumatic suspension. The prototype was tested, in a driver cab mock-up, by several operators having different anthropometries.
3) Second version of the prototype, including several changes on the seat adjustments, and displacement of the seat to center the subject weight over the suspension, in order to avoid excessive pitching of the seat. This version of the suspension seat is extensively tested in the lab on a vibration simulator. The suspension hydraulic damper is removed. Preliminary trials in a subway cab in operation.
4) Third refined version of the suspension seat. This version of the prototype is extensively tested in a subway cab during normal operations.

This paper is focusing on steps 3 and 4 of the prototype development and validation.

*Description of the seat*

To reduce the complexity of the seat in order to minimize seat maintenance, a self-adjusted suspension regarding the operator weight could not have been considered. Moreover, it was not optimal to use an independent height adjustment for the seat, since on certain lines the operator has to change train

**Fig. 1. Suspension seat installed in the operator cab.**

**Fig. 2. Subway operator on suspension seat.**

every 20 min. Thus the number of steps required to adjust the seat need to be low, so that the seat adjustment is fast. Two pictures of the suspension seat prototype (third version), with its adjustment controls, are shown in Figs. 1 and 2. A lever allows for the simultaneous adjustment of the seat rotation (40° to the right and 40° to the left, by increment of 8°), backward and forward translation (16 cm total) as well as lateral translation (18 cm total). A switch allows for a total height adjustment of 15 cm, by adding or removing air from the suspension bladder. However, by leaving 3 cm at each end to avoid end-stop impacts, a remaining 9 cm is usable as total height adjustment. The selection of the cushion and suspension have been previously optimized to minimize the *SEAT* value for the MR-73 motor car vibration spectra[10]. Indeed, the cushion has a resonance around 6 Hz, amplifying by more than a factor of 2 the vertical vibration. The suspension is attenuating that same frequency component, resulting in an optimum cushion/suspension combination[10]. To allow the suspension implementation in the small space of the operator cab, the suspension has been rotated by 90°, fitting the suspension largest dimension in the lateral direction of the cab, or in the y-axis direction during laboratory testing.

*Laboratory seat testing*

The second version of the suspension seat prototype has been extensively tested on a vibration simulator with 11 subway operators. These tests were realized to characterize the influence on the *SEAT* value of the operator weight, vibration level, vibration spectra (MR-63 and MR-73), hydraulic damper, as well as horizontal and height adjustment of the seat. The operators were chosen according to body weight that would represent the 5th, 50th and 95th percentile of the operator population weight. Three height adjustments of the suspension were considered. These adjustments correspond to a suspension height of 3 cm from the low end stop, (minimum height); the suspension seat adjusted in the middle of its adjustment range (median height); and the suspension adjusted at 3 cm from the maximum height achieved with maximum pressure in the pneumatic bladder. The minimum and maximum heights were decided in order to avoid end-stop impacts, while providing some vibration attenuation (*SEAT* < 1). The seat has been also tested for different horizontal seat adjustments that are expected to be encountered during subway operation: maximum displacement to the right (9 cm) with clockwise rotation of 24 degrees and maximum displacement to the left (9 cm) with no rotation. The vibration spectra have been defined after extensive measurement in the Montreal subway[1]. Table 1 gives a summary of the different conditions that have been tested in the laboratory.

The PSD of the three vibration spectra (MR-63 severe, MR-73 severe, MR-73 average) used in the laboratory testing are shown in Fig. 3. The MR-73 average spectrum corresponds to the mean vibration spectra of vertical vibration measured on the different subway lines, while the MR-73 severe spectrum represent more severe vibration that are reached for short amount of time (5 s) during subway operation. The MR-73 severe spectrum is equivalent to the MR-73 average spectrum multiplied by a factor of 1.6. The MR-63 severe spectrum was also considered to evaluate the performance of the suspension seat in the eventuality that it would be used in the MR-63 motor cars. The rms weighted ($w_k$) accelerations are 0.6 m/s$^2$ for the MR-63 severe and MR-73 severe spectra and 0.38 m/s$^2$ for the MR-73 average spectrum. The MR-73 spectra clearly have a dominant frequency around 2.4 Hz, while the MR-63 spectrum has a dominant frequency around 6 Hz. The tests were carried out on the IRSST hydraulic platform (see Fig. 4a). The subjects were seated in a straight posture with hands in lap. Acceleration was measured in the

**Table 1. Summary of the conditions tested on the laboratory simulator**

| Subject number (body weight percentile) | Hydraulic damper | Spectra | Suspension height | Horizontal adjustment |
|---|---|---|---|---|
| 1 (5th), 2 (50th), 3 (95th) | Without | MR-73 severe | Minimum<br>Median<br>Maximum | Seat centered with no rotation |
| | With | MR-73 severe | Median | |
| 4 (5th), 5 (50th), 6 (95th) | Without | MR-63 severe<br>MR-73 severe<br>MR-73 average | Median | |
| | With | MR-73 severe | | |
| 7 (5th), 8 (50th), 9 (95th) | Without | MR-73 severe | Median | Maximum displacement to the right (9 cm) with 24° clockwise rotation<br>Maximum displacement to the left (9 cm) with no rotation<br>Seat centered with no rotation |
| | With | | | Seat centered with no rotation |
| 10, 11 (different body weights) | Without | MR-73 severe | Minimum | Maximum displacement to the right (9 cm) with 24° clockwise rotation<br>Maximum displacement to the left (9 cm) with no rotation |
| | | | Maximum | Maximum displacement to the right (9 cm) with 24° clockwise rotation<br>Maximum displacement to the left (9 cm) with no rotation |

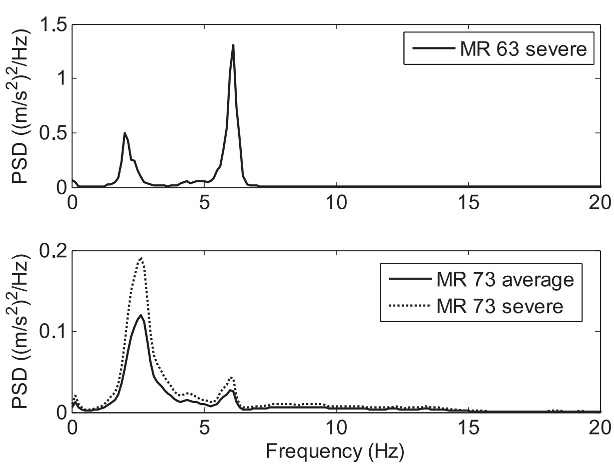


**Fig. 3. Acceleration (unweighted) power spectral density of the different vibration spectra.**

z-axis at three locations: at the base of the seat (floor), on the seat frame above the suspension and on the seat cushion with a seat pad (see Fig. 4b). Each test was performed over three trials of 180 s each. The *SEAT* values were calculated according to eq. 1 using the weighted rms accelerations, and by taking the mean of the three trials.

To get a better understanding of the influence of the suspension height adjustment on the suspension dynamics, the seat transmissibility for the three different height adjustments were calculated using the test with the median weight operator (83 kg) and MR-73 severe spectrum, by taking the square root of the ratio of acceleration auto-spectra:

$$TF = \sqrt{\frac{G_{bb}}{G_{aa}}} \tag{2}$$

Where $G_{bb}$ is the auto-spectrum of the acceleration measured on the seat cushion (vertical direction) and $G_{aa}$ is the auto-spectrum of the acceleration measured at the floor under the seat.

*Testing during normal subway operation*

The suspension seat prototype was validated in the subway during normal operation with 6 different operators having weight around the 5th, 50th and 95th percentile of the operator population weight. The measurements have been performed on the Montreal subway yellow line, between the *Longueuil-Université-de-Sherbrooke* and *Berri-Uqam* stations, where whole-body vibration levels were shown to be higher[1]. Vibrations on the floor were measured near the operator with a triaxial accelerometer (*PCB 356B41*, without the seat pad), while the vibrations on the seat cushion were measured with an accelerometer seat pad (*B&K 4322* with three charge amplifiers *B&K 2635*). At both locations, the vibrations in the three axes were measured according to the basicentric axes[2], as shown in Fig. 5. The time data were recorded and analyzed using a *B&K Pulse* acquisition system. For each operator, the measurements were performed over two round trips of about 20 min each. The operators were asked to adjust the suspension height to their preferences at the beginning of the measurement. However, it was requested to keep a minimum of 3 cm from the limit stops to avoid end-stop impacts. The data acquisitions were halted when the subway was stopped in the stations. Then, the acceleration levels were weighted according to the ISO 2631-1 standard[2]. In addition to the vibration measurements, each operator was asked if he had


**Fig. 4. Laboratory set-up for testing the suspension seat.**

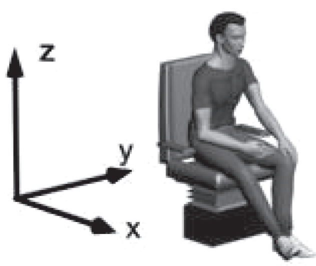


**Fig. 5. Basicentric system of coordinates (ISO 2631-1).**

experienced discomfort related to vibration during the previous interstation (i.e. the track section between two adjacent stations). Pictures of the old seat (with no suspension), and of the operator cab with the cab dimensions are shown in Fig. 6.

Two additional set of tests have been realized on the subway orange line (between the stations *Montmorency* and *Côte-Vertu*): the first one with 13 operators using the new suspension seat, and the second one with 3 operators using the current old seat. For these additional tests, each operator performed a one-way displacement of about 47 min. The vibration accelerations were not recorded during these additional tests. As for the tests on the yellow line, each operator was asked if he had experienced discomfort related to vibration during the previous interstation.

*Analysis of the suspension dynamics*

In order to get a better understanding of the influence of the suspension height adjustment on the suspension performance in attenuating the vertical subway vibrations, a simple one degree of freedom (*1DOF*) mathematical model has been used to simulate the suspension response to vibration. The model is shown in Fig. 7, with the suspension stiffness $K$ (N/m), the viscous damping coefficient $C$ (Ns/m) and the rigid mass $M$ (kg). The rigid mass includes the weight of the operator. Such a model allows estimation of the suspension natural frequency and damping ratio. It can be shown that the suspension transmissibility or acceleration ratio of the seat acceleration ($\ddot{Z}_2$) over floor acceleration ($\ddot{Z}_1$) is given in the frequency domain by:

$$\frac{\ddot{Z}_2(j\omega)}{\ddot{Z}_1(j\omega)} = \frac{Z_2(j\omega)}{Z_1(j\omega)} = \frac{j\omega C + K}{-\omega^2 M + j\omega C + K} \tag{3}$$

Where $\omega$ is the angular frequency and $j = \sqrt{-1}$. For such a *1DOF* system, the natural frequency $f_n$ is given by:

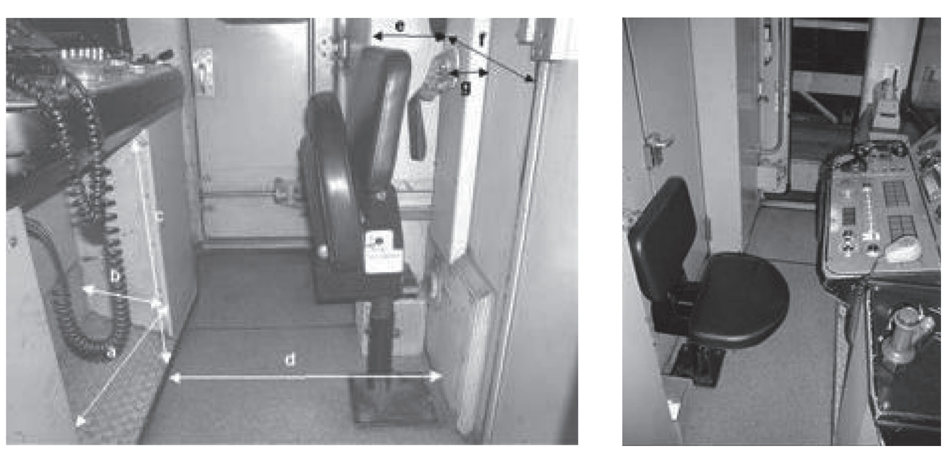


**Fig. 6. Operator cab with dimensions and old seat (MR-73 motorcar).**
a=60 cm; b=24 cm; c=60 cm; d=72 cm; e=20 cm; f=56 cm; g=8 cm.

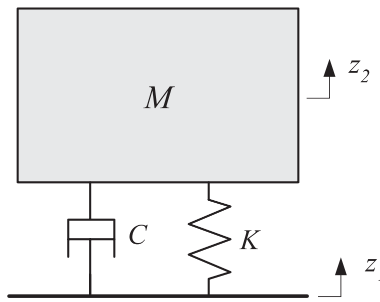


**Fig. 7. One degree of freedom (*1DOF*) model of suspension including operator weight.**

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \tag{4}$$

While the damping ratio $\xi$ is given by:

$$\xi = \frac{C}{2\sqrt{KM}} \tag{5}$$

Then, the parameters $f_n$ and $\xi$ of the suspension only (without the cushion) have been estimated using the $H1$ frequency response estimator. The cushion has not been included in this model to keep it as a simple *1DOF* system. Moreover, the cushion has low dynamic contribution on the 2.4 Hz component. The $H1$ frequency response is calculated using the following:

$$H1 = \frac{G_{ab}}{G_{aa}} \tag{6}$$

Where $G_{ab}$ is the cross-spectrum of the acceleration at the floor (z-axis) with the acceleration under the cushion (above the suspension), while $G_{aa}$ is the auto-spectrum of the acceleration at the floor (z-axis). The $H1$ response has been calculated from the measured seat response using the hydraulic shaker reproducing the subway MR-73 severe spectrum. The measurements have been performed with an 83 kg subway operator seated on the suspension seat, for the three selected suspension height adjustments (minimum, median and maximum). The damper was removed from the suspension (no damper condition). From the estimated $H1$ response, the parameters $f_n$ and $\xi$ have been calculated using the Matlab invfreqs function, assuming a two poles and one zero dynamic system (see eq. 3).

**Table 2. *SEAT* values for different suspension heights**

| | MR73 severe spectra, position centered, rotation of 0° | | |
|---|---|---|---|
| | Subject #1 58 kg | Subject #2 83 kg | Subject #3 107 kg |
| Minimum height, without damper | 0.85 | 0.80 | 0.74 |
| Median height, without damper | 0.74 | 0.73 | 0.70 |
| Median height, with damper | 0.94 | 0.92 | 0.86 |
| Maximum height, without damper | 1.01 | 0.92 | 0.94 |

**Table 3. *SEAT* values for different vibration spectra**

| | Median height, position centered, rotation of 0° | | |
|---|---|---|---|
| | Subject #4 55 kg | Subject #5 83 kg | Subject #6 113 kg |
| MR-73 average, without damper | 0.93 | 0.81 | 0.78 |
| MR-73 severe, without damper | 0.74 | 0.68 | 0.69 |
| MR-73 severe, with damper | 0.96 | 0.86 | 0.84 |
| MR-63 severe, without damper | 0.67 | 0.58 | 0.58 |

**Table 4. *SEAT* values different horizontal adjustments**

| | MR-73 severe, median height | | |
|---|---|---|---|
| | Subject #7 56 kg | Subject #8 85 kg | Subject #9 107 kg |
| Maximum right, rotation of 24°, without damper | 0.75 | 0.71 | 0.70 |
| Maximum left, rotation of 0°, without damper | 0.72 | 0.68 | 0.67 |
| Centered, rotation of 0°, without damper | 0.72 | 0.69 | 0.70 |
| Centered, rotation of 0°, with damper | 0.93 | 0.84 | 0.81 |

## Results

*Laboratory seat testing*

The suspension *SEAT* values for the different test conditions enumerated in Table 1, are shown in Tables 2, 3, 4 and 5 for a total of 11 subjects, 3 suspension heights (Table 2), 3 vibration spectra (Table 3), as well as different horizontal and height adjustments (Tables 4 and 5). In order to evaluate the effect of the damper on the *SEAT* value, the Tables (except Table 5) also report the *SEAT* values for a case with the suspension seat original hydraulic damper installed on the suspension (with damper).

The results show that the *SEAT* value is lower for a higher vibration level (MR-73 severe versus MR-73 average), for higher subject weight, and for the suspension adjusted at median height. The suspension also produces a lower *SEAT* value for the MR-63 spectrum, due to the predominance of the 6 Hz component. The seat horizontal adjustments have no influence on the suspension *SEAT* value, as shown in Tables 4 and 5. Removing the suspension damper also decreases the *SEAT* values for all the tested configurations.

The seat transmissibilities, as defined by eq. 2, are present-

**Table 5. *SEAT* values for horizontal and vertical adjustments**

| | MR-73 severe, without damper | |
|---|---|---|
| | Subject #10 70 kg | Subject #11 130 kg |
| Maximum right, 24°, minimum height | 0.81 | 0.74 |
| Maximum right, 24°, maximum height | 0.89 | 0.90 |
| Maximum left, 0°, minimum height | 0.83 | 0.72 |
| Maximum left, 0°, maximum height | 0.90 | 0.86 |

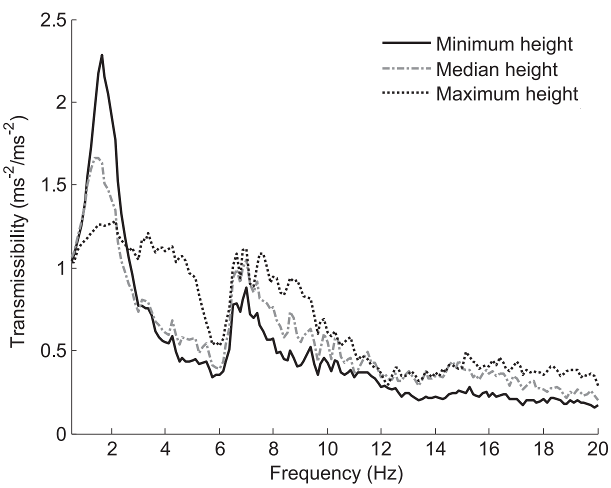


**Fig. 8. Seat transmissibility for different suspension height adjustments.**
83 kg subject, MR-73 severe spectrum.

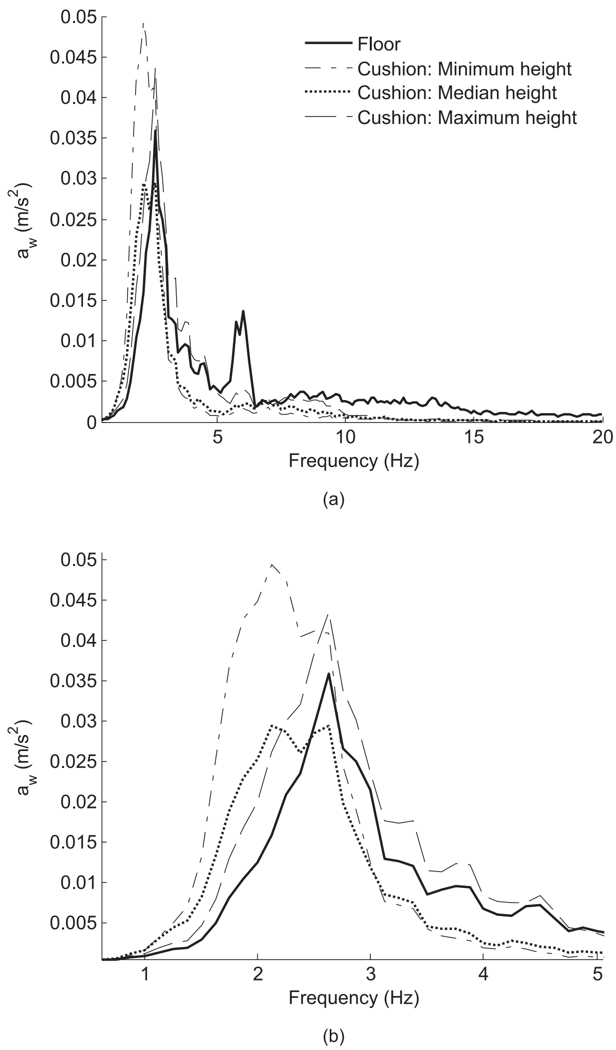


**Fig. 9. (a) Weighted vertical acceleration at the floor and on the seat cushion as a function of suspension height (83 kg subject); (b) Zoom from 0.5 to 5 Hz.**

ed in Fig. 8 for the three different suspension height adjustments, a median weight operator (83 kg), and the MR-73 severe spectrum. When compared to the median height, the minimum height contributes in sharpening the resonance due to a diminution of the suspension damping, while the maximum height has the opposite effect. Fig. 9 shows the weighted vibration acceleration spectra at the floor and at the cushion seat for the three different suspension height adjustments. It is shown that the suspension seat is reducing the vibration level of the 2.4 Hz dominant frequency component for the median height adjustment, while the other adjustments are increasing the 2.4 Hz component. Maximum amplification of the 2.4 Hz component occurs for the minimum height adjustment. Fig. 9 also shows that the 6 Hz component is totally attenuated by the suspension seat, independently of the suspension height adjustment.

*Testing during normal subway operation*

The *SEAT* values, averaged for each operator over the two round trips on the yellow line, for the three orthogonal axes of the basicentric system are presented in Table 6. The suspension height adjustment is also reported in that Table. The lower height adjustment corresponds to an intermediate height between the minimum and median height adjustments, while the higher height adjustment is located between the median and maximum height adjustments. As shown in the Table, the vertical *SEAT* values are between 0.86 and 0.99, lower than the actual rigid seat (1.05). As for the lab tests, the *SEAT* values are lower when the suspension is vertically centered (median height). The seat is amplifying the vibration levels in the $x$ and $y$ directions. However, even after amplification, the weighted vibration levels in the $y$ and $x$ directions are respectively 2 and 4 times lower than the weighted acceleration in the z direction, as shown in Table 7. To illustrate the variation of the seat performance in respect to the vibration level, Fig. 10 shows an example of the vertical *SEAT* value and weighted vertical acceleration level during a run between two stations, with both quantities averaged over 5 s periods. At the beginning of the run (first 5 s), the subway leaves the

**Table 6. *SEAT* values for the three axes**

| | *SEAT* value | | | |
|---|---|---|---|---|
| | x axis | y axis | z axis | Suspension height |
| Operator 1 | 2.400 | 1.220 | 0.991 | Minimum |
| Operator 2 | 2.064 | 1.284 | 0.973 | Higher |
| Operator 3 | 1.585 | 1.290 | 0.946 | Maximum |
| Operator 4 | 1.763 | 1.182 | 0.860 | Median |
| Operator 5 | 1.902 | 1.115 | 0.880 | Median |
| Operator 6 | 1.783 | 1.293 | 0.995 | Higher |
| Average | 1.916 | 1.231 | 0.941 | |

**Table 7. Weighted acceleration at the seat**

| | $a_w$ seat cushion (m/s$^2$, weighted) | | |
|---|---|---|---|
| | x axis | y axis | z axis |
| Operator 1 | 0.151 | 0.228 | 0.462 |
| Operator 2 | 0.129 | 0.232 | 0.457 |
| Operator 3 | 0.101 | 0.231 | 0.443 |
| Operator 4 | 0.114 | 0.226 | 0.402 |
| Operator 5 | 0.124 | 0.212 | 0.409 |
| Operator 6 | 0.117 | 0.238 | 0.486 |
| Average | 0.123 | 0.228 | 0.443 |

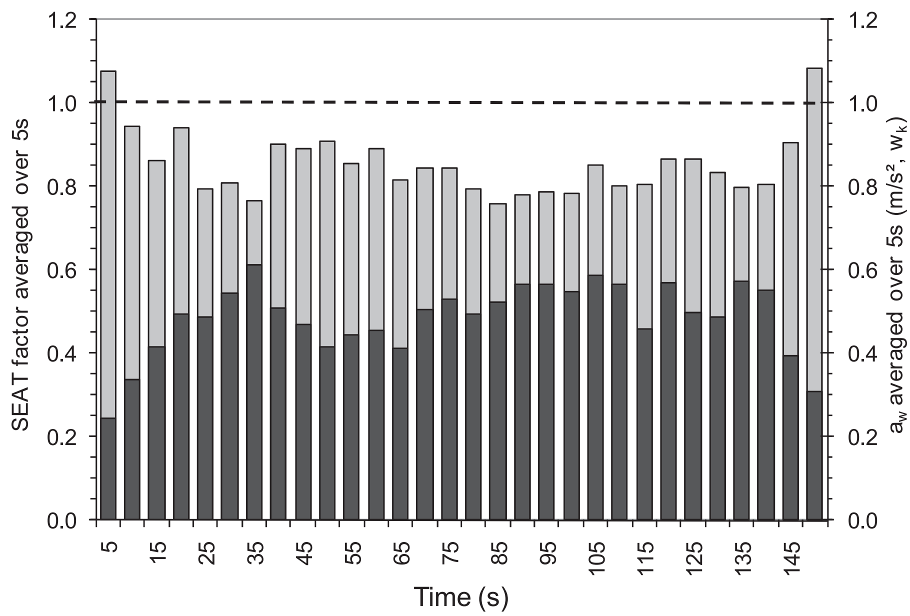


**Fig. 10. Vertical *SEAT* value and weighted acceleration at the floor as a function of time: ( *SEAT* value; weighted acceleration).**

**Table 8. Perceived vibration discomfort during normal subway operation**

| Suspension height adjustment | | Minimum | Lower | Median | Higher | Maximum | Total |
|---|---|---|---|---|---|---|---|
| Number of operators disturbed by the vibration (As a function of the % of interstation the operator was disturbed) | 0% | 2 | 1 | 6 | 2 | 3 | 14 |
| | 15% | | | 1 | | | 1 |
| | 35% | 3 | | | | | 3 |
| | 55% | | 1 | | | | 1 |
| Total | | 5 | 2 | 7 | 2 | 3 | 19 |

station, the speed is low and so is the vibration level, thus the *SEAT* value is greater than 1. However, as the subway increases its speed, the vibration level increases and the *SEAT* value decreases. It becomes evident that the low vibration levels are associated to no or low attenuation, reducing the suspension global *SEAT* values. This is in agreement with Wu and Griffin study[9], where this phenomenon has been attributed to the lock-up and non-linearities of the suspension at low vibration magnitudes.

Table 8 shows whether or not the operators have perceived vibration discomfort after each interstation. These results are for a total of 19 operators: 6 operators testing the suspension seat on the yellow line and 13 operators testing the suspension seat on the orange line (not including the 3 operators testing the old seat). The Table shows the percentage of interstations that operators have reported vibration discomfort, as a function of the suspension height adjustment. As an example, for the median height adjustment, 6 operators reported no vibration discomfort at all, while 1 operator reported discomfort 15% of the time. Over the 19 operators participating in the real test during normal subway operation, 14 did not perceive discomfort from vibration, while 5 had perceived some discomfort from vibration at some interstations. Of these 5 operators, 3 had their suspension adjusted at the minimum height, 1 at the lower height between the minimum and median height, and 1 had is suspension adjusted at the median height. Overall, when asked about their general appreciation of the new suspension seat when compared to the actual rigid seat (n=19 for the suspension seat), 18 operators found it much better, while 1 operator found it a bit better.

**Table 9. Influence of suspension height on suspension natural frequency ($f_n$) and damping ratio ($\xi$): 83 kg subject, for simulated subway vertical vibration**

| Suspension height | $f_n$ (Hz) | $\xi$ |
|---|---|---|
| Minimum | 1.73 | 0.27 |
| Median | 1.49 | 0.44 |
| Maximum | 1.74 | 1.10 |

## Discussion

The laboratory tests have shown that removing the suspension damper decreased the *SEAT* value for all the tested condition. These results are in agreement with a study[8] suggesting that for lower vibration levels and for the cases where there are no impacts or transients to attenuate, lower suspension viscous damping coefficient is associated to better attenuations. Such impacts and transients are not encountered in the Montréal subway, probably because the running racks are very flat[3].

In the laboratory tests, it was shown that in some cases, the 2.4 Hz dominant frequency was amplified by the suspension (more importantly for low adjustment heights of the suspension), even though the global *SEAT* values were smaller than 1. These amplifications of the 2.4 Hz component could be associated to vibration discomforts in some operators. Indeed, Table 8 shows that 4 operators have been disturbed by vibration (35% or more of the time), when the suspension was adjusted at the minimum or lower height. No operators reported discomforts when the suspension was adjusted to the higher or maximum height. These results, although not statistically significant, suggests that those discomforts are related to the amplification of the 2.4 Hz component. This amplification was shown to be more important for the minimum height adjustment than for the maximum height adjustment (see Fig. 9). This is in agreement with a previous study[11], where it was shown that some individuals had different sensitivity to sinusoidal vibration. There were no differences between sinusoidal and broadband vibration perceptions when the sensitivities were averaged over several subjects. Other studies reported similar observations, where the *SEAT* values were well correlated to the perceived dynamic comfort only when averaged over 6 subjects[5, 6]. Thus, when there is a dominant frequency in the vibration spectrum, the frequency-dependent seat transmissibility should be considered in addition to the overall *SEAT* value. That additional consideration could help in minimizing whole-body vibration discomfort for users that are more sensitive to sinusoidal vibrations.

From a previous study on the Montréal subway[1], it was determined that in order to attenuate the 2.4 Hz vibration component, it is necessary that the natural frequency of the suspension be smaller than 1.7 Hz, and that the suspension damping ratio be around 0.45. To verify if these initial design parameters are satisfied by the new suspension seat, an analysis of the natural frequency and damping ratio of the suspension has been performed from the data acquired during the laboratory tests.

The estimated parameters $f_n$ and $\xi$, obtained from eq. 4 and eq. 5, respectively, are shown in Table 9 for the three different suspension height adjustments. It shows that the natural frequency of the suspension is lower than the design criteria (1.7 Hz) only when the suspension is centered (median height, $f_n$ = 1.5 Hz), reducing the 2.4 Hz dominant frequency. In addition, the suspension damping ratio is closed to the design criteria (0.45) for the median height only. The minimum height leads to an under damped suspension while the maximum height leads to an over damped suspension. Thus, the design criteria are met only when the suspension is centered at the median height, leading to a decrease of the 2.4 Hz dominant frequency.

## Conclusion

A suspension seat prototype with extensive ergonomic features has been adapted to the confined space of the operator cab and the vibration environment of the MR-73 motorcar. Laboratory tests on a vertical hydraulic shaker have shown that subject weight, suspension height adjustment, and vibration level were the main factors contributing to the *SEAT* values.

The tests during normal subway operation further allowed the assessment of the suspension seat in the lateral directions, tests that were not possible in the lab on the vertical hydraulic shaker. It was shown that in the y and x axis, vibration levels were amplified by factors of 1.2 and 1.9 respectively, although levels were still 4 and 2 times lower than the vertical vibration in the z-axis. However, since the dominant axis of vibration has been reduced compared to the actual rigid seat, most surveyed operators preferred the new suspension seat for vibration comfort.

Some operators perceived vibration discomfort when the suspension was adjusted to the minimum or lower height, and thus seemed to be more sensitive to the amplification of the 2.4 Hz dominant frequency, even if the global *SEAT* values were lower than 1. These results suggest that in order to reduce discomfort caused by whole-body vibration, the seat transmissibility should be considered, in particular when there is a dominant frequency in the vibration spectra.

The new suspension was also very appreciated by the surveyed operator for the overall comfort and adjustments. Following this study, two prototypes have been installed in two operator cabs. These were installed in MR-73 motorcars for long term assessment of the suspension seat prototype.

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