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Static and Dynamic Characteristics of Scissors-Type Guiding Mechanism Work Suspension Seat

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***Abstract*-The International Organization for Standard ISO 2631-1 (1997), EU Directive 2002/44/EU, British Standard BS6841:87 and other standards of international repute have specified the acceptable levels of whole-body vibration (WBV) exposure for operators of mobile equipment and work vehicles. Compliance with these standards has not guaranteed absolute eradication or prevention of epidemiological disorders or injuries associated with low frequency vibrations, shocks, bumps, jolts and other irregularities. Therefore, effective and economic vibration isolation based on appropriate suspension seats is still necessary. This paper presents the findings of a practical study of the frequency response of a loaded X-type mechanism work suspension seat that was subjected to low frequency vertical sinusoidal vibrations in the range of 1-10Hz at three levels of amplitude on a test rig. The resulting transmissibilities of the seat due to the input excitations were analysed. The scissors mechanism work suspension seat had a static stiffness of 5.4kN/m, while the maximum travel attained was 82mm. Resonance occured mostly between 1.0 and 3.0Hz. However, the overall performance of the seat over the entire low frequency range provided an acceptable comfort level for the work vehicle operator, according to ISO standards.**

Keywords*- *Whole-Body Vibrations; Low Frequency Vibrations; Frequency Response; Transmissibility; Resonance

I. INTRODUCTION

The causes of numerous accidents involving commercial heavy vehicles (tankers, trucks, semi-trailers, passenger service vehicles, buses and coaches) on highways are endless. This class of vehicles are typically used to carry heavy loads, thereby requiring stiff suspensions that make dampening of low frequency vibrations challenging. Transmitted vibration is a major cause of discomfort to the operator, especially low frequency vibrations resulting mostly from the random undulation of the track terrain, and sometimes from the improper damping of vibrations of the engine. Such vibrations have been known to injure the spine and other parts of the operator's body. It is easy to show that the sensibility and alertness of drivers are, at most times, eroded due to fatigue, tension and weariness induced by inconvenience, lack of comfort and excessively long journeys on rough terrain [1-3].

Vibration responses due to mass, suspension and the tires of vehicles as they move along rough terrain is largely complex but is typically simplified in literature using single degree of freedom or two degree of freedom lumped parameter mass-spring-damper models. In the course of driving the vehicle, the subjects are exposed to continuous whole body vibration in the low frequency range. These are transmitted through the static seat of the vehicle (this is built into the sprung mass shown schematically in Fig. 1).

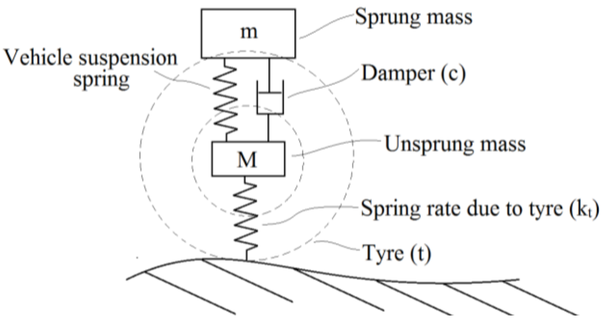

Fig. 1 Suspension of a Work Vehicle on a Terrain with Sinusoidal Cross-Section

Work vehicles are generally designed to be rugged and, in past decades, all such vehicles were fitted with static seating. This design has little consideration for vibration isolation, thereby exposing seated occupants to vertical vibrations with exciting frequencies around 4 - 8Hz [1-10]. The case for replacing static seats with suspension seats in work vehicles is now so well established that it makes sound commercial sense to install them as standard equipment [11]. Moreover, passive seat suspension has fixed spring rate therefore fixed natural frequency. The choice of its damping characteristics is therefore the only available parameter to prevent resonance or determine frequency isolation [12].

According to Gunston, wheeled off road vehicle suspension can be expected to exhibit a narrow band of vibration as low as 2 Hz on the vehicle floor, with the vehicle behaving like a lightly damped system as the vehicle mass rides on large pneumatic tire. A foam cushion seat will not provide any vibration isolation in such vehicles at such low frequencies, it might even act to amplify it. Adding a seat suspension with a sufficiently low first resonance frequency, typically around 1-2 Hz, will reduce these low frequency vehicle vibrations [13].

While resonance occurs at around 4 Hz in the common foam and spring type of seat, lower resonance frequencies of approximately 2 Hz occurs in suspension seats [14]. Thus, the capacity of the seat to function as a good vibration isolator depends more on the static and dynamic performance of its suspension mechanism than on the inherent elastic properties of its cushion. Ebe and Griffin reported that conventional solid foam or an unsprung foam seat typically has resonance at around 5 Hz induced vibration when loaded by a man [15].

Sweatman and McFarlene showed that the exposure level for a heavy transport truck puts its occupants in the ISO 2631 caution zone with approximately 0.42 to 2 m/s^2 vibration acceleration in the vertical axis [16-18]. Vertical vibrations between 2.5 and 5 Hz generates strong resonance in the neck vertebra and lumbar region with amplification reaching about 240%. Amplification up to 200% is set up due to 4 to 6 Hz resonances in the trunk region while 20 to 30 Hz vibrations could set up resonances with up to 350% amplification in the head and shoulder region [8, 19, 20]. Meanwhile, exposure to whole-body vibrations (WBV) is associated with several musculo-skeletal disorders of the spine [5, 6, 21-23].

Having realized the necessity of suspension seats in work vehicles, their economic viability becomes a bigger issue when one considers the need for commercial production. At the moment, considerable research effort is being committed to developing appropriate controlled vehicle suspension system concurrently with the research in vehicle seat suspension. It is not clear yet if it will be economically viable to build separate controlled suspension systems for the vehicle wheels and seat. Some researchers are already advocating an integrated approach, where the seat suspension is modelled as an integral part of a larger human-seat-vehicle model [24-26]. This system is unreasonable and complicated to handle with at least five inter-related subsystems: driver, seat, vehicle, road excitations and dynamics, and control systems. Analysis of these systems have so far, only been done by numerical experimentation.

The scissors-type guiding mechanism suspension seat is popular among the commercially available suspension seats designed for work vehicles. The mechanism is easy to construct and install moreover it does not require much to operate and unlike the parallelogram-type end-stops are easily adapted to it. It has also been employed in many laboratory analyses [1, 7, 10, 13, 21], but most of the works on suspension seats employed pneumatic, semi-active and active suspensions [1, 7, 9, 13, 26]. The cost of installing controlled suspensions for both seat and vehicle suspension has to be cleared. In view of this, this work is focussed on the use of the classical suspension elements, helical spring and viscous dampers, in conjunction with the scissors suspension element in the frame.

II. TEST SUSPENSION SEAT CONFIGURATION

The seat mechanism analysed in this study is similar in configuration to those commercially available. Its suspension mechanism employed a scissors-type guide mechanism fitted with a telescopic viscous damper and spring. A pre-twisted torsion bar spring is sometimes preferred because of the large magnitude of pre-load it can sustain and the smaller space it requires compared to an equivalent coil spring. For ergonomics considerations, the seat mechanism allows for adjustments (not shown in the figure) to cope with the various physiques of work vehicle operators or drivers. Fig. 2 presents sample pictures of some commercially available scissors-type work vehicle suspension seats commercially available, they are however not the ones used in this work.

The driver on the seat constitutes a mass supported by a spring as shown in Fig. 3. In spring - damper - mass systems, the response to an input force varies with frequency, and the force occurring close to the resonant frequency of the mass on the seat can be magnified considerably, resulting in unpleasant vertical vibrations of the driver. This condition is worsened if the vehicle is unladen.

It is a well-established fact that the transmissibility equations of linear systems are only a rough approximation of the response of real seat models due to non-linearities exhibited by the cushion and suspension; therefore, the actual transmissibility of such seats must be determined experimentally using laboratory procedures. It also remains true that the best way to evaluate a suspension seat is to examine its response in the true environment of its operation. While laboratory testing simulated with sine waves and random signals are approximations, the result of such tests give extremely valuable guidelines and benchmarks for real life experimental tests in actual environments.

Sample Scissors Mechanism Seat with Cushion Installed

Sample Scissors Seat Mechanism with Damper, Pneumatic Suspension and Spring

Sample Scissors Seat Mechanism without Damper and Spring

Fig. 2 Scissors Mechanism Suspension Seat used in Work Vehicles [27, 28]

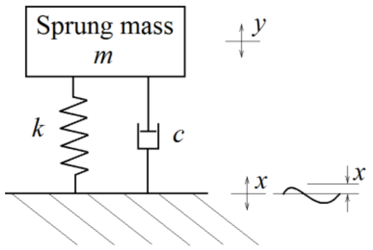

Fig. 3 Simplified Man/Seat Model Subjected to Vibrations

III. VIBRATION ISOLATION WITH A SUSPENSION SEATING

The main function of the suspension design of a work seat is to reduce and attenuate the vibrations and shocks reaching the driver. It achieves this by acting as a mechanical filter whose behavior can be predicted by the vibration theory of the one degree of freedom system shown Fig. 3. The effectiveness (or transmissibility) of such an isolator (suspension seat) is given as the ratio of the maximum force transmitted to the impressed force or the output amplitude to the input amplitude. Thus, transmissibility,

$$T_R = \frac{F_1}{F_0} = \left|\frac{\omega^2 Y}{\omega^2 X}\right| = \left|\frac{Y}{X}\right| \tag{1}$$

where, F_0 is the maximum impressed (input) force, F_1 is the maximum transmitted force to the driver's body, X is the maximum input amplitude and Y is the maximum output amplitude.

In an actual situation, the driver (operator) on a suspension work seat is the idealized model in Fig. 1 with two degrees of freedom. However, for the suspension seat mounted on a stationary rigid laboratory rig, as was the case in this investigation, the situation depicted a single degree of freedom case consisting of a rigid mass coupled to a base through a viscous damper and a spring of linear rate, as shown in Fig. 3. The equation of motion can be expressed as:

$$m(\ddot{y} - \ddot{x}) = -c(\dot{y} - \dot{x}) - k(y - x) \tag{2}$$

by letting $x = Xe^{i\omega t}$ and $y = Ye^{i(\omega t - \phi)} = Ye^{i\omega t}e^{-i\phi}$, Eq. (2) transforms to:

$$(m\omega^2 + i\omega c + k)Ye^{-i\phi} = (k - i\omega c)X \tag{3}$$

From which the amplitude ratio is obtained as

$$\left|\frac{Ye^{i\phi}}{X}\right| = \left|\frac{k + i\omega c}{(k - m\omega^2) + i\omega c}\right| \tag{4}$$

Hence the transmissibility function for the steady state excitation of the seat can be expressed as

$$T_R = \left|\frac{\omega^2 Y}{\omega^2 X}\right| = \left|\frac{Y}{X}\right| = \sqrt{\frac{1+(2\xi r)^2}{(1-r^2)^2+(2\xi r)^2}} \tag{5}$$

where k is the spring rate (that is, the linear stiffness) of the seat mechanism, m is the lumped mass of the driver and seat mechanism, ϕ is the phase angle, $r = \frac{\omega}{\omega_n}$ is the frequency ratio (that is, the ratio of the excitation frequency to the natural frequency), and $\xi = \frac{c}{c_r}$ is the damping factor (that is, the ratio of actual damping to critical damping which is given by $c_r = 2m\omega^2$).

Fig. 4 represents the theoretical frequency response applicable to the one degree of freedom man/seat model given in Fig. 3 at three assumed levels of damping (that ξ =0,0.25, 0.5, 0.75 and 1.0) using Eq. (4).

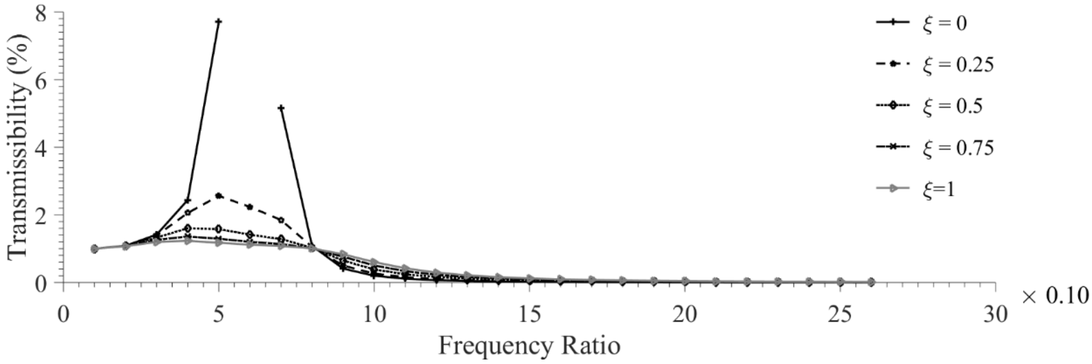

Fig. 4 Theoretical Frequency Response of a Man/Seat Model for Five Levels of Damping Ratios

In Fig. 4 it is shown that transmissibility reduction is achieved as the damping ratio increased. Theoretically, resonance occurs when the frequency ratio is in the region of 1. Beyond the resonance region the presence of damping is less advantageous as shown in Fig. 4. Damping is therefore essential to prevent resonance especially in vehicle seats exposed to low frequency vibrations.

IV. INVESTIGATION METHODOLOGY

With the damper and cushion fitted the suspension seat mechanism was subjected to a static load consisting of a tin mannequin, a static weight and sprung weights of seats, which totalled 70kg. The road terrain was assumed to be sinusoidal in the cross-section, as shown in Fig. 3. This was stimulated in the test using a seat test rig supplied with a movable platform, which was vibrated in the vertical direction by means of a hydraulic vibrator using a sinusoidal input.

The tests were performed in conformity with International Standards Organisation (ISO) standards, as stipulated in ISO2631 [17, 18]. The investigation to determine the dynamic response of the seat mechanism was carried out in the frequency range of 1 to 10Hz using three different input amplitudes: X = 3.12mm, 6.25mm and 12.5mm. Each test was carried out with and without the damper being connected to the seat in order to investigate the level of necessity of damping as well as, the effect of damping on the dynamic responses. The amplitude of the output vibration of the loaded suspension seat at each frequency and input amplitude were measured with a linear potentiometer calibrated and used as a displacement transducer.

V. RESULTS AND DISCUSSION

Fig. 5 shows the load–displacement curves for the evaluated suspension seat mechanism. The seat was evaluated with and without the damper fitted. The curves show the combined hysteretic characteristic of the entire suspension media that make up the seat mechanism. A maximum travel of about 82mm was achieved with an overall load of 104.2kg when the damper was not fitted, and 79.98mm with the same overall load when the damper was fitted.

The results also show that without the telescopic viscous damper fitted, material damping in the metal structures contributes substantially to the overall damping characteristics of the mechanism. Using the idealised load-deflection curve derived from Fig. 5, where the upper and lower parts of the hysteresis have been ignored because they are typically caused by end-stops not normally reached during operation, the average static stiffness estimated for the suspension mechanism was 5400Nm^{-1}, and natural frequency value was 1.35Hz (at zero viscous and frictional damping).

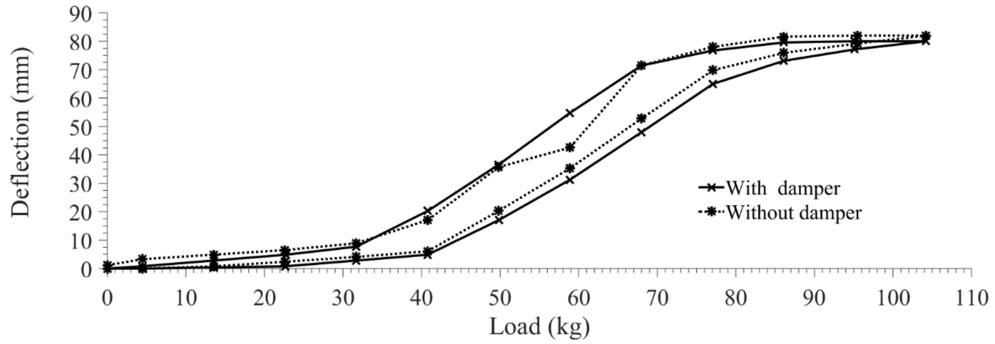

Fig. 5 Load – Displacement Curves for the Suspension Seat Mechanisms

Experimental (Actual) Frequency Response

Figs. 6, 7 and 8 present the actual frequency (dynamic) responses of the suspension seat at input amplitudes of X=3.12mm, 6.25mm and 12.5mm and under a mass of 70kg evaluated with and without the viscous damper and seat cushion fitted.

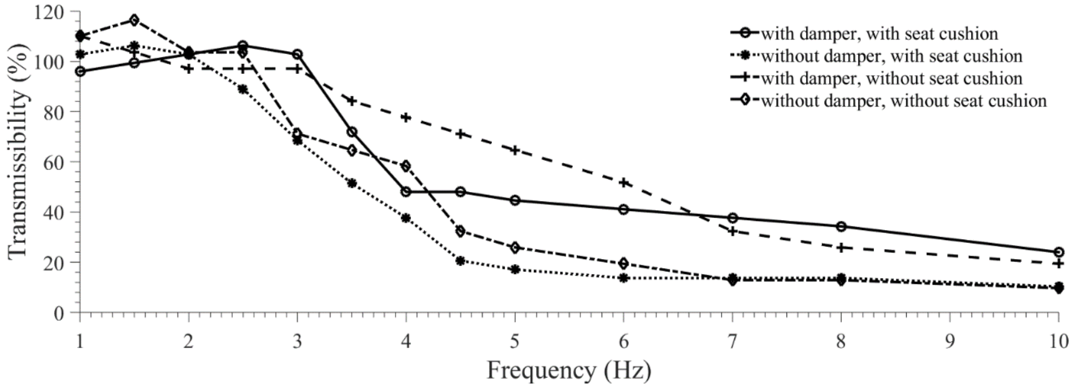

Fig. 6 Transmissibility of Seat at Input Amplitude X = 3.12mm

In Fig. 6 where the input amplitude was 3.12mm, there were no prominent peaks but the suspension seat experienced resonance around 1.5Hz when it was not fitted with damper but was 2.3Hz when the damper and the seat cushion was fitted. Beyond this resonance point the vibration progressively attenuated. Except for when the damper and seat cushion were not fitted. The resonance peaks were also marginally above 100%.

Fig. 7 shows that improved responses were achieved at the higher input amplitude of 6.25mm though the peaks are less prominent. The transmissibility values were also lower when the damper was not fitted. Beyond the resonance points, the trend for input amplitudes 3.12 and 6.25mm are similar.

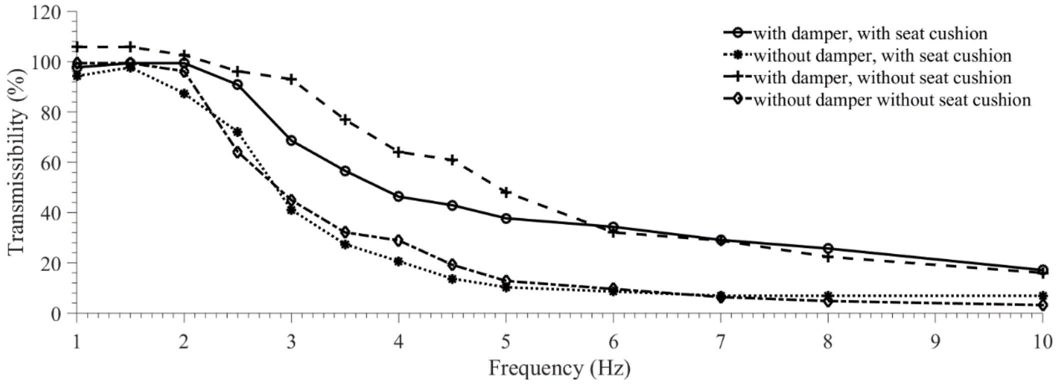

Fig. 7 Transmissibility of Seat at Input Amplitude X = 6.25mm

The necessity for the damping element becomes obvious when the input amplitude reaches 12.5mm as shown in Fig. 8. Without the damper fitted, resonance occurred at 1.5Hz when the seat cushion was fitted. The transmissibility reached 420% around 1.8Hz when the cushion was removed. Figs. 6, 7 and 8 indicate the level of discomfort that the vehicle operator could experience on terrains with uniform amplitudes of 3.12mm, 6.25mm and 12.5mm. This variation in the amplitude of excitation does not however seem to have significant effect on the dynamic response of the seat in the resonant region except in the case of 12.5mm amplitude “without damper”.

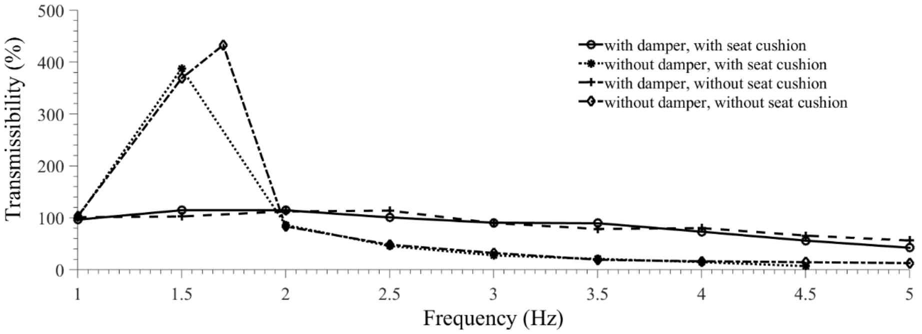

Fig. 8 Transmissibility of Seat at Input Amplitude X = 12.5mm

From the perspectives of a design engineer, damping is an important tool in reducing the output amplitude of vibrations and subsequently transmissibility at frequency values that are close to the resonant frequencies. This study has shown that when the input frequency is fairly small (< 6.25mm) frictional damping in the mechanism is sufficient in preventing resonance peaks. When the input frequency approaches 12.5mm additional damping becomes essential.

As the level (or extent) of discomfort is a function of both the transmissibility value and the duration of exposure to the excitation, the larger the value of each of these two parameters, the greater the discomfort. Normally, the desirable dynamic response or performance of the suspension seat is that the transmissibility at low frequency values should be less than unity (that is, 100%), or in ideal situations, negligibly small. In reality, this is rarely the case.

VI. CONCLUSION

The factors that determine the level of transmission of vibrations to the work vehicle operator (driver) are the input disturbances, seat transmissibility and vehicle dynamics. The effects of the first two factors have been investigated in this study. The results showed that the X-linkage configuration of the suspension seat coupled with the presence of damping caused a reduction in the peak (resonance) transmissibility values for the different input amplitudes of vibration to values within 100 and 120% at frequency values between 0.8 and 3Hz. This indicates a much better comfort level for the same frequency range. In addition, these results indicate a better comfort level for the work vehicle operator within the low frequency range of vibration that is normally encountered on roads and rough tracks. There is potential for further development of this configuration of suspension seat to achieve less than 100% transmissibility values at resonance.

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