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INFLUENCE OF SEAT CUSHIONS ON VIBRATIONS IN EARTH MOVING VEHICLES

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INTRODUCTION

Vibration amplification by vehicle seat systems has been reported previously [1,2]. To account for this and establish seat, or cushion, parameters for optimum vibration isolation, it is necessary to have a realistic model for the man-seat system. Although there is evidence to show that these arrangements behave like 2-DOF systems [1], it appears that the only representations employed previously have been based on 1-DOF mass-spring-dashpot models. Here, we describe the initial results of our studies of the modelling of vibration behaviour at the man-cushion interface. The model employed is simplistic and does not account for all the processes operating in these systems.

The seats, from JCB '3CX' earth moving vehicles, consisted of injection moulded polypropylene structural foam shells with fabric/flexible PVC covered full depth high resiliency polyurethane cushions. Two load-bearing seat cushions, A ('old') and B ('new') having approximate dimension 45 cm x 50 cm x 11 cm were investigated.

Experimental

Quasi-static F-D diagrams for whole cushion were measured using an Instron in conjunction with the profiled plate used for the measurement of vibrations at the buttock-cushion interface. The F-D behaviour for cushion B at a deflection rate of 0.5 cm/min is shown in Fig. 1. The mechanical behaviour of the cushions was characterised by increasing F to 100 kg and then cycling between 30 kg - 100 kg. It is seen that the F-D diagram was highly non-linear and there was considerable short-term fatigue. For modelling purposes, the hysteresis was expressed in terms of the ratio of energy dissipated/energy stored in one cycle. This was equated to the damping factor $d(\omega)$ of the system. Measurements were made at deformation rates of 0.5 cm/min - 20 cm/min. The upper limit corresponded to a frequency of about 0.08 Hz. The F-D behaviours of A and B were much the same except that for A, $d(\omega) = 0.2$ and B, $d(\omega) = 0.34$. No rate effects were observed.

DYNAMIC MODEL REPRESENTATIONS

1-DOF Model

So far, our studies have been confined to z-axis response and we have ignored friction damping between the man and the back cushion and the interaction between x-y and z axis excitations. A simple 1-DOF system is shown in Fig. 1 in which the person is represented by mass $M_1 = 75$kg and the seat by a complex stiffness $S_1^*(\omega) = S_1'(\omega)\,\{1 + d_1(\omega)\}$. Both $S_1'(\omega)$ and $d_1(\omega)$ were assumed to be independent of ω. The vibration amplification for sinusoidal excitation is given by

$$T^* = \tilde{Z}_1^*/\tilde{Z} = (1 + jd_1)/\{(1 - \omega^2 M_1/S_1') + jd_1\}$$

Non-linear mechanical response was incorporated into the model by estimating the values of S_1' and d_1 from the gradient and hysteresis of the F-D diagram at the quiescent (or Q) point, Fig. 1. S_1' was obtained by using a shape factor, K = loaded area of buttocks and thighs/loaded area of indentor. For cushions A and B, S_1' at the Q point was approximately 15×10^3 N m^{-1}. The behaviour predicted by this model for cushion A ($d_1 \simeq 0.2$), curve (a), is compared in Fig. 3 with that of a parallel spring-dashpot model with $C/C_c \simeq 0.1$, curve (b). Although this model cannot account for the multiple peaks observed in the current, and previous, work, one interesting result is that for 40 kg < M_1 < 100 kg it predicts that the resonance frequency f_n of the man-seat system is essentially independent of M_1, as shown in Fig. 4.

3-DOF Model

Two multi DOF human body representations have been investigated; the 4-DOF Payne-Band (P-B) model [2] and the model described in ISO 5982-1981(E). The 3-DOF hysteresis damped model, given in Fig. 5, is based on the P-B model. The mass of head, arms and shoulders have been lumped together as M_1 and the resilient members are represented as complex stiffness $S_i^* = S_i'(1 + d_i)$, i = 1,2, where the parameters are independent of frequency. The cushion is represented by S_3^*. Values of d_i (i = 1,2) were estimated for each element by fitting the response of the elements of the P-B model to that of a hysteresis element with the same S_i and M_i. In Fig. 6, seat vibration amplification in a JCB '3CX' (old) vehicle travelling over the JCB wave motion track at 8 mph is compared with that predicted by the model.

Discussion

It is seen that there is fair agreement between the predicted and the observed behaviour. The values of seat parameters used differ from those measured under static conditions. This was not unexpected since it is known that at these frequencies pneumatic processes influence both the stiffness and the damping of the open-cell foams used in seat systems [3].

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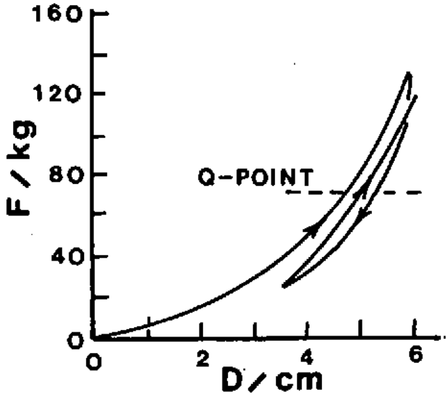


FIG. 1. FORCE-DEFLECTION DIAGRAM FOR A JCB '3CX' CUSHION

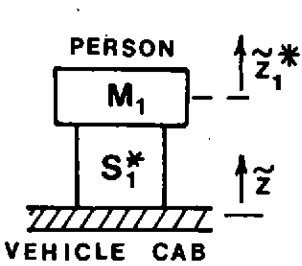


FIG. 2. 1 DOF LUMPED PARAMETER MODEL REPRESENTATION

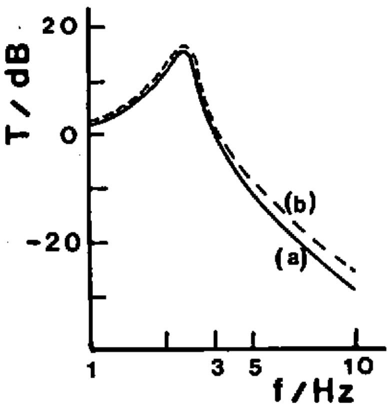


FIG. 3. PREDICTED TRANSMISSIBILITY OF A JCB '3CX' SEAT CUSHION BASED ON 1-DOF MODELS.

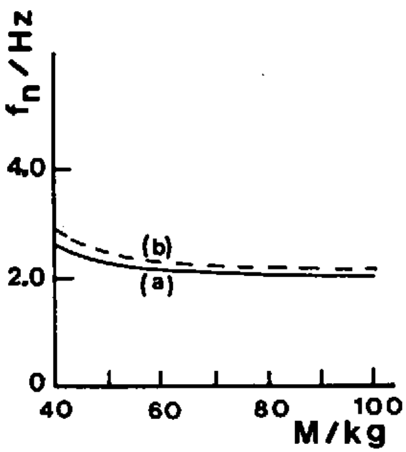
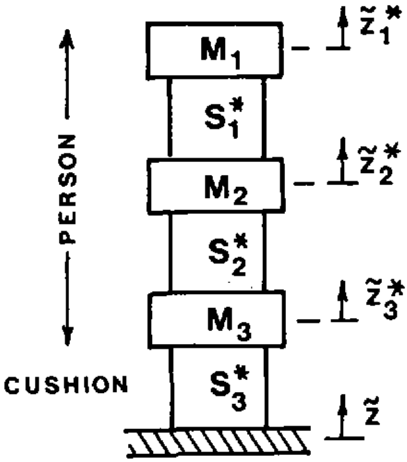


FIG. 4. PREDICTED RESONANCE FREQUENCY AS A FUNCTION OF BODY MASS FOR (a) CUSHION A, (b) CUSHION B.



$M_1 = 34$ kg

$S_1' = 118 \times 10^3$ N m^{-1}

$d_1 = 0.22$

$f_1 = 9.4$ Hz

$M_2 = 29$ kg

$S_2' = 65 \times 10^{-3}$ N m^{-1}

$d_2 = 0.45$

$f_2 = 7.6$ Hz

$M_3 = 24$ kg

$S_3' = 15 \times 10^3$ N m^{-1}

$d_3 = 0.2$

$f_3 = 4.0$ Hz

FIG. 5. 3-DOF REPRESENTATION OF A SEATED PERSON

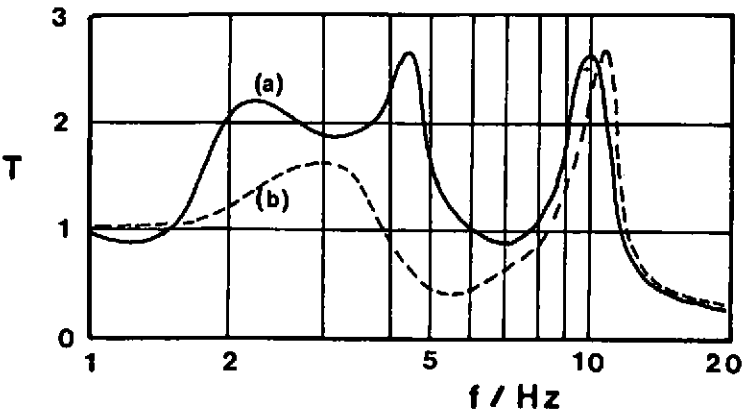


FIG. 6. MEASURED CAB/SEAT TRANSMISSIBILITY IN A JCB 3CX, (CURVE (a)) COMPARED WITH THAT PREDICTED BY THE LINEAR 3-DOF MODEL OF FIG. 5, WITH $S_3' = 60 \times 10^3$ N m^{-1} AND $d_3 = 0.4$ (CURVE (b))