MINIMIZATION OF HUMAN BODY RESPONSES TO LOW FREQUENCY VIBRATION: APPLICATION TO TRACTORS AND TRUCKS

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Abstract—Experimental studies of the vibrations transmitted to and tolerated by tractor and truck drivers indicate that the drivers are subjected to extremely uncomfortable levels of vertical and pitch vibrations in the frequency range of 0.5 to 11 Hz. In this study, an occupant-tractor system is modeled as a lumped parameter system. The composite model is simulated for vertical and pitch vibrational response to ground reaction by steady-state sinusoidal forcing function inputs and for transient responses by trapezoidal type of inputs. A relaxation type seat suspension located in the plane of the center of gravity of the chassis of a tractor is introduced, and the parameters of the seat suspension are determined for minimized human body responses. When these responses are compared with those of an optimized relaxation seat suspension located behind the center of gravity at the conventional location of a tractor and also with the experimental results of other investigators, it is found that the recommended location is the best among those considered. It reduces all the human body-segment responses significantly and the acceleration level to much below the eight-hour "exposure limit" tolerance curve, and thus improving riding comfort.

INTRODUCTION

"A survey by orthopedic surgeons in the United States definitely establishes that riding on trucks or tractors either causes or aggravates a number of disorders of the spine and supporting structures of drivers."[1]. High incidences of osteoarthritis, traumatic fibrositis, herniated disks, coccygodynia, traumatic lumbosacral pain, abdominal pain, and intestinal disorders have been observed among drivers of trucks, tractors, and other vehicles or machinery, which produce appreciable vibrations and jolts[2]. Experimental studies of the vibrations transmitted to and tolerated by tractor occupants indicate that drivers especially are subjected to extremely uncomfortable levels of both vertical and angular (pitch) vibrations in the frequency range of 0.5 to 11 Hz[1]. The solution for the problem lies in isolating the occupant from basic vehicle vibration, both vertical and pitch, by means of a suitable suspension. In this article, a method of reducing the intensity of harmful vibrations is recommended. This method requires the provision of a relaxation seat sus-
pension in the plane of the center of gravity of the tractor and a suitable selection of parameters.

Vibration intensity is characterized by the amplitude ratio, acceleration level, relative amplitude between adjacent body parts, and pitch of the tractor. Any isolation of vibration that can be achieved by providing a suspension should reduce all these characteristics. The peak acceleration levels in conventional tractors are on the order of 0.5 to 1.5 g in the frequency range of 2 to 7 Hz\(\text{[1]}\), and standard seats of different suspension parameters give rise to amplitude ratios of 2.5 to 4.5. These vibration acceleration levels are of a much higher intensity than the one-minute "exposure limits" proposed by the International Standard Organization (ISO)[3]. Therefore, it is recommended that the acceleration levels be reduced much below the eight-hour "exposure limit" proposed by ISO by providing a seat suspension in the plane of the center of gravity of the tractor and suitably selecting its parameters.

Earlier works[4, 5] contained designs for tractor suspension systems that would isolate vertical and pitch vibrations. Such suspension systems would not be very effective because they were designed only on the measurement of the transmissibilities of seat vibration. Mathews[6] found that just the measurement of vibration on the suspended seat alone does not truly reflect vibration levels to which human body parts are exposed. Hence, designing just the seat suspension without taking into account the combined effect of the vehicle and the occupant does not yield satisfactory results. Work at the United Kingdom's National Institute of Agricultural Engineering[7] has shown that it is necessary to simulate mechanically the human body characteristics together with the seat. Therefore, in this study, a model of both the occupant and tractor is developed in the form of a lumped mass system interconnected by springs and dashpots. Mathews[8] also found that, "for the best vehicle ride the man should be as near as possible to the center of gravity of the tractor." Therefore, in this article it is recommended that the seat be moved forward from its conventional position behind the tractor's center of gravity to the plane of the center of gravity.

During the research for this study, a composite model consisting of a human body, a tractor, and its relaxation seat suspension located in the plane of the tractor's center of gravity was subjected to sinusoidal, idealized field or road profile, vibrations at the tyre contact points. The resulting transient and steady state responses of each body part found by computer simulation were studied to select the parameters of the relaxation seat suspension so that the occupant vibration intensity, as characterized by the human body acceleration levels, amplitude ratios, and relative displacements, was reduced to a minimum in the 0.5 to 11 Hz frequency range.

**ANALYSIS OF A TRACTOR-OCUPPANT'S VIBRATION RESPONSE**

*Human occupant model*

The tractor occupant acts as a lumped parameter model at low frequencies of from 0.5 to 100 Hz[1, 9]. This model was idealized as a seven degrees of freedom, nonlinear, lumped parameter[10]. As shown in Fig. 1, the lumped masses of head, back, torso, thorax, diaphragm, abdomen, and pelvis were connected by springs and dashpots, which represent the elastic and damping properties of the connective tissue between the segments. The model proposed by Mucksian and Nash[10] was modified to include the damping and elasticity of the buttocks. The values of the tissue, spring, and dashpot parameters were obtained from studies of the characteristics of specific subsystems[10, 11]. These values are listed in Table 1. The validity of this model was established by the fact there was good agreement between its response and that recorded experimentally by other investigators.
Fig. 1. Occupant tractor model with relaxation suspension to seat in the plane of center of gravity of a tractor.

**Tractor model**

The tractor was idealized, as shown in Figure 1, by having the seat, chassis, and tyre masses lumped together and interconnected by springs and dashpots to the seat suspension system. The tyres were represented by linear vertical springs in parallel with velocity dependent dampers. The parameters obtained from Mathews[4] for the tractor and the optimum parameters of the relaxation seat suspension found by computer simulation of the composite model are listed in Table 2. The relaxation seat suspension[12] consists of a spring (of constant $K_s$) in parallel with a system of springs (of constant $K_{sx}$) in series with a dashpot (of constant $C_s$). The relaxation seat suspension was characterized by two
Table 1. Parameter values of the occupant model.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass (M)</td>
<td>M₁ = 5.45, M₂ = 6.82, M₃ = 32.762, M₄ = 1.362, M₅ = 0.455, M₆ = 5.921, M₇ = 27.23</td>
</tr>
<tr>
<td>Damping constant (C)</td>
<td>C₁ = 3.580, C₂ = 3.58, C₃ = 0.292, C₄ = 0.292, C₅ = 0.292, C₆ = 0.292, C₇ = 0.371</td>
</tr>
<tr>
<td>Spring constant (K)</td>
<td>K₁ = 52.6, K₂ = 52.6, K₃ = 0.877, K₄ = 0.877, K₅ = 0.877, K₆ = K₇ = 0.877, K₈ = 25.5</td>
</tr>
</tbody>
</table>

* The units of damping constants giving rise to linear and nonlinear forces respectively are: kN/m/sec and kN/(m/sec).†
† The units of spring constants giving rise to linear and nonlinear forces respectively are: kN/m and kN/m².

Table 2. Parameter values of the tractor and relaxation suspension.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Distance of back wheel from center of gravity (a)</td>
<td>0.847 m</td>
</tr>
<tr>
<td>Distance of front wheel from center of gravity (b)</td>
<td>1.185 m</td>
</tr>
<tr>
<td>Distance of seat from center of gravity (c)</td>
<td>0 m</td>
</tr>
<tr>
<td>Wavelength of road/field irregularity (l)</td>
<td>4.57 m</td>
</tr>
<tr>
<td>Radius of gyration of tractor (p)</td>
<td>1.0224 m</td>
</tr>
<tr>
<td>Magnitude of impressed vibration (field depression or elevation) (A)</td>
<td>0.05 m</td>
</tr>
<tr>
<td>Phase angle between the front and back tyre inputs (α)</td>
<td>160 degrees</td>
</tr>
<tr>
<td>Mass (M)</td>
<td>M₉ = 4.537, M₁₀ = 2667.24</td>
</tr>
<tr>
<td>Damping constant (C)</td>
<td>C₉ = 0.1848, C₁₀ = 2.374</td>
</tr>
<tr>
<td>Spring constant (K)</td>
<td>K₉ = 2.943, K₁₀ = 553.280</td>
</tr>
<tr>
<td>Relaxation suspension parameters (dimensionless)</td>
<td>γ = 4, β = 10</td>
</tr>
</tbody>
</table>

* Represents the parameter value for two (front or back as the case may be) tyres.

parameters known as \( γ^* = \sqrt{K_{xx}/K_x} \) and \( β = K_{xx}/C_x \sqrt{M_x/K_x} \), in which \( M_x \) is the mass of the seat.

**Occupant-tractor composite model**

The composite model of the occupant-tractor moving on irregular terrain is shown in Fig. 1. The Continuous Systems Modeling Program (CSMP)** was used to simulate the composite model with a computer for 1) steady state responses such as amplitude ratios, acceleration levels of body parts, and seat and pitch response of the chassis, to sinusoidal inputs applied at the tractor tyres and 2) transient vertical vibration responses of the body parts and seat to trapezoidal pulse inputs applied at the tractor tyres.

* \( γ \) is related to what is known as the "relaxation factor," \( h \), by the relationship \( γ = \sqrt{8h}[12] \) in which \( h = K_{xx}/K_{xx} \), and \( K_{xx} \) is the critical value of the relaxation spring.
** See Appendix for CSMP description.
Steady state analysis

In deriving the dynamic model of the tractor-occupant for simulation and analysis, the following simplifying assumptions were made:

1. The road or field profile was considered to be sinusoidal and 0.05 m in amplitude[1].
2. The vehicle was considered to move only in the longitudinal plane that passes through the center of gravity. The wheels were combined with the chassis mass.
3. The forces and couples caused by the rotating wheels and draught forces were ignored.
4. The rotational (pitch) vibrations of the body parts of the occupant were considered to be the same as those of the tractor chassis.
5. The displacements were considered to be sufficiently small for the tractor’s tyres and spring motions to be always within their linear range and to allow the sine of angles to be replaced in the equations of motion by the angles in radians, i.e., \( \sin \Theta = \Theta \).

The composite model of tractor-occupant was thus subjected to the sinusoidal vibrations caused by the ground reaction forces the tractor is subjected to in its speed range while traversing the terrain. While deriving the governing equations of motion, both the pitch and vertical motion of the tractor were included. The stiffness and damping characteristics of the torso, thorax, diaphragm, and abdomen were represented by nonlinear springs and nonlinear dashpots[10]. The equation of motion for each mass consisted of both the inertial term and the forces exerted on the mass by the springs and dashpots occasioned by the relative motion of the connected masses. The governing second order, coupled, nonlinear, ordinary, differential equations of the various masses of the composite model are given below.

\[
M_h \ddot{y}_1 + C_h(\dot{y}_1 - \dot{y}_2) + K_h(y_1 - y_2) = 0,
\]

\[
M_b \ddot{y}_2 + C_h(\dot{y}_2 - \dot{y}_1) + C_b(\dot{y}_2 - \dot{y}_7) + C_{tb}(\dot{y}_2 - \dot{y}_3) + C_{tb}(\dot{y}_2 - \dot{y}_3)^3 + K_h(y_2 - y_1) + K_{tb}(y_2 - y_3) + K_{tb}(y_2 - y_3)^3 + K_b(y_2 - y_7) = 0,
\]

\[
M_t \ddot{y}_3 + C_{tb}(\dot{y}_3 - \dot{y}_2) + C_{tb}(\dot{y}_3 - \dot{y}_2)^3 + C_t(\dot{y}_3 - \dot{y}_4) + C_t(\dot{y}_3 - \dot{y}_4)^3 + K_{tb}(y_3 - y_2) + K_{tb}(y_3 - y_2)^3 + K_t(y_3 - y_4) + K_t(y_3 - y_4)^3 = 0,
\]

\[
M_{th} \ddot{y}_4 + C_t(\dot{y}_4 - \dot{y}_3) + C_t(\dot{y}_4 - \dot{y}_3)^3 + C_{th}(\dot{y}_4 - \dot{y}_5) + C_{th}(\dot{y}_4 - \dot{y}_5)^3 + K_t(y_4 - y_3) + K_t(y_4 - y_3)^3 + K_{th}(y_4 - y_5) + K_{th}(y_4 - y_5)^3 = 0,
\]

\[
M_d \ddot{y}_5 + C_{th}(\dot{y}_5 - \dot{y}_4) + C_{th}(\dot{y}_5 - \dot{y}_4)^3 + C_d(\dot{y}_5 - \dot{y}_6) + C_d(\dot{y}_5 - \dot{y}_6)^3 + K_{th}(y_5 - y_4) + K_{th}(y_5 - y_4)^3 + K_d(y_5 - y_6) + K_d(y_5 - y_6)^3 = 0,
\]

\[
M_a \ddot{y}_6 + C_d(\dot{y}_6 - \dot{y}_5) + C_d(\dot{y}_6 - \dot{y}_5)^3 + C_a(\dot{y}_6 - \dot{y}_7) + C_a(\dot{y}_6 - \dot{y}_7)^3 + K_d(y_6 - y_5) + K_d(y_6 - y_5)^3 + K_a(y_6 - y_7) + K_a(y_6 - y_7)^3 = 0,
\]

\[
M_p \ddot{y}_7 + C_a(\dot{y}_7 - \dot{y}_6) + C_a(\dot{y}_7 - \dot{y}_6)^3 + C_b(\dot{y}_7 - \dot{y}_2) + C_b(\dot{y}_7 - \dot{y}_2)^3 + K_a(y_7 - y_6) + K_a(y_7 - y_6)^3 + K_b(y_7 - y_2) + K_b(y_7 - y_2) = 0,
\]
\[ M_s \ddot{y}_8 + C_p(\dot{y}_8 - \dot{y}_7) + K_s(y_8 - y_7) + K_p(y_8 - y_7) + K_{st}(y_8 - y_9) = 0, \quad (8) \]
\[ K_{st}(y_9 - y_s) + C_s(\dot{y}_9 - \dot{y}_{10}) = 0, \quad (9) \]
\[ M_{ct} \ddot{y}_{10} + C_s(\dot{y}_{10} - \dot{y}_9) + K_s(y_{10} - y_8) + C_{gf}(\dot{y}_{10} + b\dot{\Theta}) + K_{st}(y_{10} - a\dot{\Theta}) + C_{st}(\dot{y}_{10} - a\dot{\Theta}) = 0, \quad (10) \]
\[ M_{ct}(y_{10} + b\dot{\Theta}) - aC_{st}(y_{10} - a\dot{\Theta}) + bC_{gf}A \sin \omega t + C_{sf}A \cos(\omega t - a) + K_{gf}A \sin \omega t + K_{st}A \sin(\omega t - a), \quad (11) \]

In these equations, \( \dot{y}_1, \dot{y}_7, \) and \( \dot{y}_9 \) represent the corresponding accelerations, velocities, and displacement from the equilibrium position of the respective masses, and \( \dot{\Theta} \) represents the rotation of the chassis. The above coupled, nonlinear, differential equations were solved by using a CSMP simulation on an IBM 370/155 computer to obtain the \( \dot{y}_i, \dot{y}_t, \) and \( \dot{\Theta} \) responses to steady state, sinusoidal, forcing functional inputs at the tyres at different vibration frequencies. The amplitude ratios of the various body parts and seat were computed by dividing the amplitude responses of the body parts by the input amplitude of vibration \( (A, \) at the tractor tyres). The parameter variation for the relaxation seat suspension placed in the plane of the tractor’s center of gravity was set so that the responses of the body parts would be minimized in the 0.5 to 11 Hz frequency range. The parameters of the suspension that give the minimum vibration responses for the body parts are listed in Table 2.
Transient analysis

The purpose of this study was to choose the proper design parameters for seat suspension so that body parts are not damaged by sudden, high amplitude, relative displacements at the onset of vibrations, when a tractor encounters sudden obstructions for short time intervals. This is supported by von Gierke\textsuperscript{13}, who stated, ‘‘it is not the pressure per se, but the resulting relative displacement of adjacent tissue that leads to the stimulation of various receptors as well as to ultimate injury.’’ The obstructions or ground irregularities are shown in Fig. 2 as idealized trapezoidal inputs with a maximum amplitude of 0.5 m. The two inputs in sequence represent the front and back tyre displacements respectively. These inputs represent two obstacles or surface irregularities that are 0.05 m high, 0.3 m wide, and separated by 1.58 m. When the front tyre is on the peak of the first obstacle, the second obstacle is located 0.45 m \((a + b - 1.58 = 0.45 \text{ m})\) ahead of the back tyre. The front tyre is subjected to the first input irregularity and the back tyre to the second after a time lag of 0.09 seconds. This characterizes the time taken by the tractor’s rear tyre to reach the irregularity when moving at 18 km/hour. The distances between obstacles and the tractor’s speed were chosen to represent the most adverse type of condition to which tractor tyres could be subjected.

The vibration inputs at the front and back tyres are mathematically represented by \(x_1\) and \(x_2\) respectively in the following equations:

\[
x_1 = \frac{10A}{T} [u(t) - u(t - 0.1T)] + A[u(t - 0.1T) - u(t - 0.9T)]
+ 10A \left(1 - \frac{t}{T}\right) [u(t - 0.9T) - u(t - T)].
\]

\[
x_2 = \frac{10A(t - 1.5T)}{T} [u(1.5T) - u(t - 1.6T)] + A[u(t - 1.6T) - u(t - 2.4T)]
+ \frac{10A}{T} [2.5T - t] [u(t - 2.4T) - u(t - 2.5T)].
\]

In these equations, \(u(t)\) represents the unit step function defined as

\[
u(t) = 0 \quad \text{for} \quad t < 0
= 1 \quad \text{for} \quad t \geq 0
\]

and \(A = 0.05 \text{ m}, \ T = 0.06 \text{ seconds}.

The governing vibration equations of the composite model for the trapezoidal displacement inputs at the tractor tyres are the same as Eqs. (1) to (9) for the body parts, tractor seat, and relaxation suspension damper piston. The equations for the chassis were modified as follows:

\[
M_{cf} \ddot{y}_{10} + C_s(\dot{y}_{10} - \dot{y}_9) + K_s(y_{10} - y_8) + C_{cf}(\dot{y}_{10} + b\dot{\Theta})
+ K_{cf}(y_{10} + b\Theta) + C_{gr}(\dot{y}_{10} - a\dot{\Theta}) + K_{gr}(y_{10} - a\Theta)
= C_{cf} \dot{x}_1 + C_{gr} \dot{x}_2 + K_{cf} x_1 + K_{gr} x_2.
\]

\[
M_{cr} \ddot{\Theta} + bC_{cf}(\dot{y}_{10} + b\Theta) - aC_{cf}(\dot{y}_{10} - a\Theta) + bK_{cf}(y_{10} + b\Theta) - aK_{cf}(y_{10} - a\Theta)
= bC_{cf} \dot{x}_1 - aC_{gr} \dot{x}_2 + bK_{cf} x_1 - aK_{gr} x_2.
\]

Here, \(\dot{x}_i\) and \(x_i\) represent corresponding velocities and displacements at the tractor tyres.
CSMP simulation was used for programming Eqs. (14) and (15) along with Eqs. (1) through (9) on the computer to give $y_i$, the transient amplitude responses of the body parts and seat, and the relative displacements between adjacent body parts. The parameters of the relaxation seat suspension were such that the relative displacements between body parts were minimized in the 0.5 to 11 Hz frequency range. The minimum response parameters of the seat suspension in the transient vibration analysis were found to be the same as those presented in the steady state vibration analysis and listed in Table 2.

RESULTS

Validation of the model

Figure 3 shows the calculated head-to-pelvis acceleration ratio as a function of frequency. Superimposed thereon are the experimental values given by Goldman and von Gierke[14] and Pradko et al. [15, 16], for sinusoidal inputs. The good agreement between the model calculations and the experimental values provides a measure of confidence in the parametric values of the model.

Table 3 shows the predicted acceleration ratios for the head, back, torso, thorax, diaphragm, and abdomen. The first resonant peak for each of the body parts occurs at approximately 3 Hz. This is in general agreement with the results of Coermann et al. [17] and Roberts et al. [18]. The model can be validated further by correlating it with the subjective response, which is characterized by the back pains of live subjects at 4 to 5 Hz, to sinusoidal longitudinal vibrations, as indicated by the data given by Magid et al. [19, 20].

Steady state vibration responses for a relaxation seat suspension located in the plane of a tractor's center of gravity

The validated composite model with its minimum response parameters is then used to find the responses of the body parts to sinusoidal vertical vibrations within the frequency range of 0.5 to 11 Hz. The results representing the responses of some of the body parts subjected to maximum vibrations are then compared with the results of other research workers and the ISO recommendations[3].

Fig. 3. Head-to-pelvis acceleration ratio.
Table 3. Variations of acceleration ratios of the body parts with frequency.

<table>
<thead>
<tr>
<th>Frequency of Vibration (Hz)</th>
<th>Head/ Pelvis</th>
<th>Back/ Torso/ Thorax/ Pelvis</th>
<th>Thorax/ Pelvis</th>
<th>Diaphragm/ Pelvis</th>
<th>Abdomen/ Pelvis</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>1.0</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>1.0</td>
<td>1.01</td>
<td>1.055</td>
<td>1.09</td>
<td>1.07</td>
<td>1.07</td>
</tr>
<tr>
<td>2.0</td>
<td>1.08</td>
<td>1.075</td>
<td>1.185</td>
<td>1.11</td>
<td>1.11</td>
</tr>
<tr>
<td>3.0</td>
<td>1.12</td>
<td>1.11</td>
<td>1.2</td>
<td>1.18</td>
<td>1.14</td>
</tr>
<tr>
<td>4.0</td>
<td>1.13</td>
<td>1.09</td>
<td>1.18</td>
<td>1.05</td>
<td>1.055</td>
</tr>
<tr>
<td>5.0</td>
<td>1.12</td>
<td>1.07</td>
<td>1.14</td>
<td>1.025</td>
<td>0.960</td>
</tr>
<tr>
<td>6.0</td>
<td>1.07</td>
<td>0.960</td>
<td>1.02</td>
<td>0.955</td>
<td>0.89</td>
</tr>
<tr>
<td>7.0</td>
<td>1.0</td>
<td>0.885</td>
<td>0.921</td>
<td>0.86</td>
<td>0.80</td>
</tr>
<tr>
<td>8.0</td>
<td>0.91</td>
<td>0.830</td>
<td>0.88</td>
<td>0.773</td>
<td>0.725</td>
</tr>
<tr>
<td>9.0</td>
<td>0.9</td>
<td>0.787</td>
<td>0.82</td>
<td>0.707</td>
<td>0.662</td>
</tr>
<tr>
<td>10.0</td>
<td>0.92</td>
<td>0.99</td>
<td>1.025</td>
<td>0.757</td>
<td>0.735</td>
</tr>
<tr>
<td>11.0</td>
<td>0.90</td>
<td>0.825</td>
<td>0.7</td>
<td>0.620</td>
<td>0.615</td>
</tr>
</tbody>
</table>

As shown in Fig. 4, the head and tractor seat have maximum amplitude ratios of 0.618 and 0.522, respectively at 1 Hz. This indicates that the head is subjected to a greater amplitude than the seat at lower frequencies. The results at higher frequencies indicate that the head undergoes greater attenuation of vibration than the seat. Superimposed thereon is the amplitude ratio responses of the head for a tractor equipped with a suspension seat. These responses were obtained from Radke’s experiment[1] and from Matthews experiment[4]. By comparing the model’s computed maximum response of the head with that of Radke’s[1], it was found that an 83.3% reduction in response occurs at low frequencies and a 99.3% reduction at high frequencies. By comparing the model-computed seat response with the seat response obtained in Mathews experiment[4] with a standard type of front axle suspension, it was found that the relaxation seat suspension located in the plane of the center of gravity of the tractor reduces the amplitude ratio of the seat from 6 to 0.522.

Of all the body parts, the thorax undergoes the greatest amplitude ratio response. Figure 5 shows the responses of the back, torso, thorax, and diaphragm, the maximum responses of which are equal to 0.616, 0.63, 0.635, and 0.633 respectively at 1 Hz.

The model-computed acceleration responses of the body parts and seat in the frequency range of 0.5 to 11 Hz for a sinusoidal type of input at the tractor tyres are plotted in Figs. 6 and 7. Figure 6(a) shows the curves for the steady state acceleration responses of the head and tractor seat. The maximum response of the head is on the order of 1.197 m/sec^2 at 1 Hz. Comparison of the model-computed acceleration response of head to that of Dupuis et al.[21] at 2.58 Hz shows that the relaxation type of a seat suspension in the plane of the center of gravity of a tractor reduces the acceleration response of the head by 96.1%. As shown in Fig. 6(b), the maximum acceleration intensities of back and torso are found to be equal to 1.193 m/sec^2 and 1.22 m/sec^2 at 1 Hz, respectively. Both the curves in their responses subsequently show a decreasing trend up to 11 Hz.

Of all the body parts, the thorax responds the most to acceleration. Figure 7 shows the responses of the thorax, diaphragm, abdomen, and pelvis. The maximum acceleration responses are 1.229 m/sec^2, 1.225 m/sec^2, 1.217 m/sec^2, and 1.157 m/sec^2 respectively at 1 Hz. The eight-hour “exposure limit” curve prescribed by the ISO is superimposed on these curves. It can be seen that the maximum acceleration responses in the frequency range of 0.5 to 11 Hz of the body parts (especially the thorax) fall below the eight-hour “exposure limit” tolerance curve, thereby indicating that riding comfort is improved.

The model-computed pitch response of a chassis in the frequency range of 0.5 to 11 Hz is represented in Fig. 8, which shows that the maximum pitch response of the chassis
Fig. 4. Comparison of our model amplitude ratio responses of head and seat for relaxation suspension to seat in the plane of center of gravity of a tractor and the experimental responses of Radke[1] and Mathew[4] for conventional seat suspension located behind C.G. (center of gravity) of tractor (refer to Fig. 11 for the conventional location of a seat suspension).
Fig. 5. Amplitude ratio responses of (a) torso and back, and (b) diaphragm and thorax for relaxation seat suspension located at the plane of C.G. of tractor.
Fig. 6. Model acceleration responses of (a) head and seat, and (b) back and torso for relaxation seat suspension at the plane of C.G. of tractor.
The principal objective of the study was to choose those design parameters for a seat relaxation suspension that would prevent the body parts of an operator from being damaged by sudden high amplitude relative displacements at the onset of vibrations when a tractor encounters sudden obstructions for short time periods. (These obstructions are idealized by the trapezoidal type of pulse inputs shown in Fig. 2.) The group of body parts that experience maximum relative displacement from one another are selected here for the sole purpose of representing their responses. Figures 9(a) and 9(b) represent the tran-
Fig. 8. Chassis pitch response for relaxation suspension to seat, at the plane of C.G. of tractor.
Fig. 9. Transient responses of (a) pelvis and back, and (b) pelvis and seat for relaxation seat suspension at the plane of C.G. of tractor.

Table 4. Comparison of the maximum responses when the relaxation seat suspension is located behind the center of gravity and in the plane of center of gravity of a tractor.

<table>
<thead>
<tr>
<th>Sl. no.</th>
<th>Vibration characteristics steady/transient</th>
<th>Relaxation type of seat suspension behind C.G. of the tractor</th>
<th>Maximum magnitude</th>
<th>Body/tractor parts involved</th>
<th>Relaxation type of seat suspension at the plane of C.G. of the tractor</th>
<th>Maximum magnitude</th>
<th>Body/tractor parts involved</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Amplitude ratio (steady)</td>
<td>2.399 Thorax</td>
<td>0.635 Thorax</td>
<td>16 minutes Pelvis</td>
<td>8 hours Thorax</td>
<td>0.642 Chassis</td>
<td>8 hours Thorax</td>
</tr>
<tr>
<td>2.</td>
<td>Acceleration (m/sec²)</td>
<td>5.62 Pelvis (at 11 Hz)</td>
<td>1.229 Thorax</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.</td>
<td>Absolute amplitude (mm) transient</td>
<td>13.73 Thorax</td>
<td>12.87 Thorax</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.</td>
<td>Relative amplitude between adjacent parts (mm: transient)</td>
<td>2.55 Abdomen and pelvis</td>
<td>1.48 Abdomen and pelvis</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.</td>
<td>Pitch of chassis (degrees/cm of input amplitude) (steady)</td>
<td>0.659 Chassis</td>
<td>0.642 Chassis</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>6.</td>
<td>Exposure limit (steady)</td>
<td>16 minutes Pelvis</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Fig. 10. Transient responses of (a) diaphragm and abdomen, and (b) pelvis and abdomen for relaxation seat suspension at the plane of C.G. of tractor.

Transient responses of the pelvis-back and pelvis-seat combinations respectively. The response of the seat is seen to be lower than that of the pelvis. All the responses subside to zero at the end of 4.56 seconds. The maximum relative displacement between the pelvis and back is seen to be on the order of 0.54 mm at 0.144 seconds.

Figures 10(a) and 10(b) show the transient responses of the diaphragm-abdomen and pelvis-abdomen, respectively. It is seen that of all the body parts, the maximum relative displacement takes place between the pelvis and abdomen and that it is on the order of 1.48 mm. By taking the length between these parts as 170 mm [22], the strain value can be computed as 0.871%. This is much less than the breaking index* (42%) indicated in von Gierke [11] for the same body parts.

Steady state and transient vibration responses for a relaxation seat suspension located behind the center of gravity of a tractor

Equations similar to those presented in the Analysis Section were used to calculate the steady state and transient vibration responses of the body parts and chassis for a

* Breaking Index = Breaking Strength/Young’s Modulus = Percentage of increase in length required to break [11].
Fig. 11. Occupant-tractor model with relaxation type seat suspension located (at the conventional place) behind the C.G. of the tractor.
Fig. 12. Amplitude ratio responses of (a) back and torso, and (b) thorax and diaphragm for relaxation seat suspension located behind C.G. of tractor.
Human body vibration responses

condition in which the optimized (or minimum response parameter) relaxation seat suspension is located 0.768 m behind the center of gravity of a conventional tractor (Fig. 11). The results are shown in Figs. 12 to 14. In Fig. 12, it is seen that the back, torso, diaphragm, and thorax have maximum amplitude ratios of 2.32, 2.37, 2.393, and 2.399 respectively at 1 Hz. Of all the body parts, the thorax has the maximum amplitude ratio. Figure 13 shows how the acceleration responses vary with the frequency of the abdomen, thorax, diaphragm, and pelvis. The maximum values for the abdomen, thorax, and diaphragm are 4.83, 4.86, and 4.86 m/sec² at 1 Hz, whereas the maximum value for the pelvis is 5.62 m/sec² at 11 Hz. It is seen that, of all the body parts, the pelvis has the maximum acceleration response.

As shown in Fig. 14(a), the maximum amplitude responses of the diaphragm and abdomen have a phase difference of 0.024 seconds. Figure 14(b) represents the transient vibration responses of the abdomen and pelvis for which the maximum amplitude responses are on the order of 13.53 and 12.92 mm respectively, with a phase difference of 0.024 seconds. Computer results indicate that the maximum relative displacement between these two body parts is 0.168 seconds, and its value is on the order of 2.55 mm.

Comparison of the maximum responses of a relaxation seat suspension conventionally located 0.768 m behind a tractor’s center of gravity and a seat located in the plane of the center of gravity can be made with the help of the responses shown in Figs. 4 through 10 and 12 through 14 and Table 4. When a relaxation seat suspension is positioned in the plane of the center of gravity instead of in the conventional location behind the center of gravity, it reduces the maximum amplitude ratio by 73.5%, the acceleration intensity by

![Graph showing acceleration responses of abdomen, pelvis, diaphragm, and thorax with 16 min. exposure limit from ISO, ref. 3 superimposed on it for relaxation seat suspension located behind C.G. of tractor.](image-url)
Fig. 14. Transient responses of (a) diaphragm and abdomen, and (b) abdomen and pelvis for relaxation seat suspension located behind C.G. of tractor.

78.13%, the transient absolute amplitude of the body parts by 6.25%, the relative amplitude between the adjacent body parts by 42%, and the pitch response of the chassis by 2.58%. The exposure limit is increased from 16 minutes to 8 hours, thereby, increasing the riding comfort occasioned by vibration to a considerable extent.

CONCLUSIONS

From the responses presented in Fig. 4 through 10 and 12 through 14 and Table 4, the following conclusions can be made:

1. As indicated by the responses of steady state vibration to sinusoidal input at the tyres, body parts experience high responses at lower frequencies and lower responses at higher frequencies compared to the responses of the seat.

2. The transient vibration responses of the body parts, when a trapezoidal type of pulse input is applied at the tyres, are higher than those of the seat. The above conclusions indicate that the choice of the suspension parameters should be based on the principle
of minimizing the amplitude ratios of the body parts rather than minimizing the response of only the tractor seat.

3. Shifting the optimized relaxation seat suspension from its conventional place behind the center of gravity of a tractor to the plane of the center of gravity considerably reduces the occupant's different body responses.

4. From the response characteristics of the steady state body parts, it has been found that locating the optimized relaxation seat suspension in the plane of the center of gravity of a tractor reduces the maximum of the amplitude ratio response of the body parts to 0.635 and of the acceleration of the body parts to $1.229 \text{m/sec}^2$ at 1 Hz, thereby increasing the vertical vibration "exposure limit" to 8 hours and the pitch response of the chassis to 0.642 degrees/cm of input amplitude. This considerably improves the riding comfort.

5. From the response characteristics of the transient body parts, it has been found that the maximum relative displacements between the body parts are on the order of 1.48 mm. This indicates that the vibration isolation characteristics of the relaxation seat suspension located in the plane of the center of gravity of a tractor are very effective.

REFERENCES

APPENDIX

CSMP description

The Continuous System Modeling Program (CSMP) is a problem-oriented program designed to facilitate the digital simulation of continuous processes on large-scale digital machines. The program provides an application-oriented language that allows these problems to be prepared directly and simply from a set of ordinary differential equations. The program includes a basic set of functional blocks with which the components of a continuous system may be represented and accepts application-oriented statements for defining the connections between these functional blocks. CSMP also accepts FORTRAN statements, thereby allowing the user to handle nonlinear and time-variant problems of considerable complexity readily. Input and output are simplified by means of user oriented control statements.

A fixed format is provided for printing (tabular format) and print plotting (graphic format) at selected increments of the independent variable. Through these features, CSMP permits the user to concentrate upon the phenomenon being simulated, rather than upon the mechanism for implementing the simulation. For details, the IBM publications system/360 Continuous System Modeling Program Application Description (H-20-0340-1) and user's manual (H-20-0367-2) may be referred.