Pergamon

Journal of Terramechanics, Vol. 32, No. 4, pp. 205–219, 1995
Elsevier Science Ltd

0022–4898/95 $9.50+0.00

0022–4898(95)00017–8

TRACTOR RIDE VIBRATION—A REVIEW

NIRANJAN PRASAD*, V. K. TEWARI† and RAJVIR YADAV*

Summary—All human occupants vibrate to some degree in any moving vehicle, whether it is a car, truck, train, aeroplane or boat. The reviews on the effect of vibration on human health have shown serious evidence of operator ill health that may be attributed to tractor drivers. The frequency range of 2–6 Hz has been observed to be the most harmful for the human operator because resonance occurs within this frequency range. Considerable effort has been made to establish the optimum design parameters for tractor seats. Further reductions in the level of ride vibrations experienced by tractor operators appear to be necessary and some possible methods of achieving significant improvements have been outlined. These methods have led to the evolution of several seat suspension systems and emphasise the use of the principles of ergonomics in the operator's seat design. In this paper, a review of the work done in this area, particularly with respect to tractor operators, is presented and discussed.

INTRODUCTION

Vibration occurs in all moving bodies and tractors are no exception. It results from the interaction of the vehicle with the rough terrain and from the vehicle's power source. The level of ride vibration on tractors during normal operation is frequently in excess of internationally accepted levels [1]. A considerable amount of time and energy is spent in minimizing this vibration problem. Suspended seats fitted to most tractors reduce the vertical component of vibration, but the levels are still undesirably high and there is little potential for further improvement using this technique. A further reduction in the vibration level may be obtained by introducing wheel suspension, but it makes the system complicated and costly. Cab suspension is another method of reducing ride vibration which is almost as expensive as wheel suspension, although useful. The objective of this review is to collect information regarding tractor vibration and research work done to date to reduce it at the operator's level.

TYPES OF TRACTOR VIBRATION

Vibrations primarily are of two types: sinusoidal and random [2]. Sinusoidal vibration is regular in nature and predictable. Random vibration is irregular and unpredictable. The mechanical vibrations that affect the human body can be grouped into two directional vibrations, i.e. rectilinear and rotational (or angular).

The rectilinear vibrations (vertical, longitudinal and transverse) are transmitted to the human body along appropriate directions in an orthogonal coordinate system, the origin of which coincides with the location of the heart (Fig. 1.). Accordingly, three

*Research Scholar and †Assistant Professor Agricultural and Food Engineering Department, I.I.T., Kharagpur, India 721302

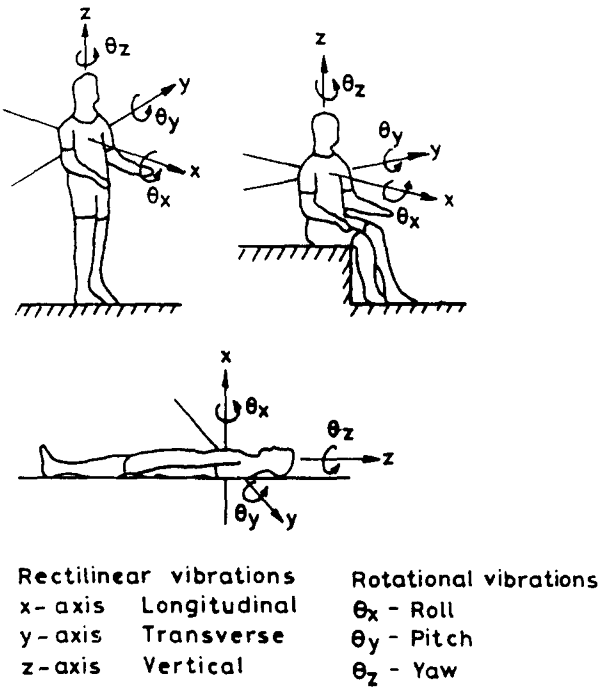

Fig. 1. Directions of basicentric coordinate system for mechanical vibrations influencing the human body [1].

components of vibrations, a_z, a_x and a_y, are considered. Accelerations in the foot or buttocks to the head along a vertical axis are designated as a_z; accelerations in the fore and aft (chest to back) axis as a_x; and in the lateral (right to left side) axis as a_y [1].

Studies by Janeway [3] show that the predominant vibrational motion of a wheel-tractor is vertical and that the operator is most sensitive to this vibration. The effect of longitudinal and transverse modes depends on the type of task being performed by the tractor [4].

The angular vibration that affects the human body can be resolved into yaw, pitch and roll (Fig. 1). The rotational mode of vibration does not usually cause much discomfort [5]. However, in some instances, such as tractors going over rough terrain, the pitching or rolling motions of the seat may be more disturbing than rectilinear vibrations.

EFFECTS OF VIBRATION

Physiological effect

The seat plays an important role in improving the man–machine interface in the tractor system. The main reason for physiological damage is the transmission of excessively large forces to the body. (It has been recognized that pressure is one of the major causes of skin ulcers.) Pressure exerted over a long period of time can cut off blood supply to the tissue and cause mechanical damage. A good cushion on the seat pan distributes the pressure more uniformly over the skin area. Thus it decreases the incidence of ulcers and lengthens the tolerable time period in a given body position [6].

In sitting for extended time periods, individuals can relieve stresses that are potentially damaging to their tissues by periodic shifting of position. The common regions of pressure-induced damage are the tissues near the ischial tuberosites and sacrum region where high stresses are sustained by the tissue during sitting.

There is an inverse relationship between the tolerable pressure limits and the duration of such pressure. The time–pressure relationship depends on many other factors, such as the general health of the subject, his or her diet, skin cleanliness, etc. It is also known that skin and tissue can tolerate much higher cyclic pressure than constant pressure.

The stress analysis of a seated buttock is a complex contact problem involving large deformation and non-linear material properties. Chow [7] made the first systematic study of the buttock–cushion interaction problem. An axisymmetric experimental model of a buttock made of a PVC gel was used to conduct studies on various surfaces. The PVC gel material was cast into a hemisphere around a wooden core and embedded with lead pellets and rods so that the strains in the model could be measured experimentally from X-ray photographs before and after seating on various surfaces. Assuming a linear infinitesimal strain theory, Chow [7] calculated strain distributions in the gel and particularly examined how interface shear strain was influenced by the mechanical properties of the seat cushion and its covering material.

Chow and Odell [6] extended the work by devising an axisymmetric finite element model of a buttock with which normal and shear stress distributions could be calculated throughout as functions of prescribed load or displacement conditions for surface nodes. They investigated stresses and strains in the model when supported by water, by mercury, by a flat, frictionless and rigid surface, by a modified cosine pressure distribution, and by a foam cushion, with and without interface friction. Attempts were not made in this work to evaluate the range of cushion materials in terms of stress and strain distributions arising in the buttock or to take account of the material properties of the gel which were known to be non-linear.

In spite of the above theoretical and experimental investigations of the problem, there continues to be a need for a practical method whereby an objective judgement can be made as to the relative performance of tractor seat pan and back rest cushion materials with respect to reducing tissue stresses and stress gradients. It is therefore desirable to have a relatively simple model that can be used to assess the relative performances of cushion, seat pan and back rest profiles in terms of the stress pattern during seated conditions.

Several researchers have measured the pressures at the human–seat interface [8–10]. However, the visco-elastic behaviour at the interface is completely altered by the sensors used. Furthermore, many of these sensors cannot be used under dynamic loading conditions.

Glance [11] developed a finite element model of the seat pan and foam cushion. Solid elements were utilized to represent the three-dimensional contour of the occupant sitting in the seat. The same analysis technique was utilized to represent pressure contours of the seat back. Pressure distributions derived from experimentally measured comfort levels were utilized as a guide to arrive at the desired computer-generated theoretical pressure distribution.

Health effect of whole body vibrations

A number of subjective studies concerning the health and safety of highway and off-highway vehicle drivers have concluded that prolonged exposure to low-frequency, large-amplitude and whole-body vibrations either aggravates or causes degenerative physical symptoms and degrades the driver's response to certain stimuli (which may affect the driver health and safety). Probably the most comprehensive survey of the health effects of tractor driving is that by Rossegger and Rossegger [12] which covers a sample of 371 tractor drivers, extended over a number of years and includes a record of the subject's professional and medical history, initial and subsequent medical examination with X-ray, blood pressure, pulse frequency and other tests. Two forms of health problems are clearly demonstrated to result from long periods of tractor operation: stomach complaints and disorders of the spine.

Dupuis and Christ [13] found that, over a period of five years, the percentage of tractor drivers with spinal deformations increased from 72.5 to 78.9%. They also found a marked increase in the severity of the abnormalities over the same period whereby the percentage of people with definite abnormalities increased from 50.2 to 68.7%, and the limited abnormalities decreased from 22.3 to 10.2%. This indicates that prolonged exposure greatly increases the risk of spinal abnormalities. Drivers of vehicles subjected to low-frequency bounce vibrations risk developing motion sickness [14].

Performance effects of vibration

Tractor drivers experience discomfort when exposed to excessive low-frequency vibration during many farming tasks. This results in impaired performance of tractor drivers, leading to under-utilization of the available power of the tractor [15]. Matthews [16] also investigated the effect of acceleration levels (in the range of 0.0–0.35 g at 2.5 and 3.5 Hz) on compensatory tracking, ability to maintain constant foot pressure and deterioration of visual acuity. Performing tasks that require steadiness or precision of muscular control is likely to show decrement from vibration. Yet tasks that primarily measure central neural processes, such as reaction time, monitoring and pattern recognition, appear to be resistant to degradation during vibration. In fact, Poulton [17] points out that vibration between 3.5 and 6.0 Hz can have an alerting effect on subjects engaged in boring vigilance tasks. Within this range, tensing the trunk muscles attenuates the amplitude of shoulder vibration. Tensing the muscles is a good method for maintaining alertness. Outside of the 3.5–6.0 Hz range the subject can attenuate shoulder vibration more by relaxing the trunk muscles.

SUBJECTIVE RESPONSE TO WHOLE BODY VIBRATION

The subjective response most often assessed in vibration studies is comfort. Comfort, of course, is a state of feeling and so depends in part on the subject experiencing the situation, defined as 'not having discomfort'. Generally, feelings quantified as uncomfortable, annoying, very uncomfortable, or alarming are used in almost all studies. Investigators [18–21] have also tried to link the physical characteristics of vibration, most notably frequency and acceleration, to the subjective evaluations of comfort. Their studies usually result in equal comfort contours (Fig. 2) for combinations of frequency and acceleration. The resulting contours differ widely

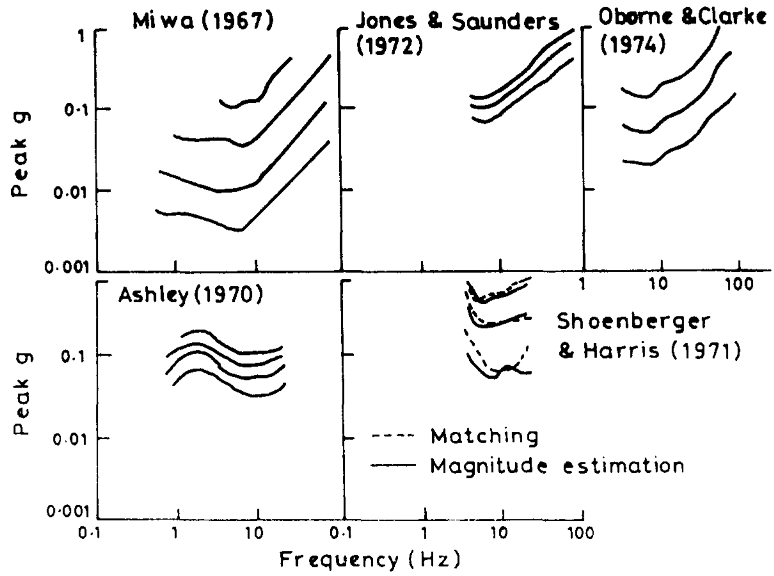

Fig. 2. Equal sensation contours produced in the laboratory by different investigators.

from study to study [22]. This discrepancy is due to different methodologies, subject populations, vibration environments and semantic comfort descriptors. It appears that individuals react in markedly different ways to whole-body vibration, and it is difficult to determine the causes of these differences [23]. An individual's response to vibration can be quite stable over a period of time [24]. This issue of individual differences in response to vibration provides a challenge for researchers and designers alike. Oborne *et al.* [23], for example, determined equal sensation contours for 100 subjects. They found that 70% of the subjects did respond in the same way as depicted by the average contour.

The research on ride comfort (or discomfort) has been conducted at the NASA Langley Research Centre, with over 2000 test subjects [25]. To avoid semantic confusion, NASA developed a ratio scale of discomfort, measured in DISC units. The scale is anchored (1 DISC) at the discomfort threshold (i.e. where 50% of the passengers feel uncomfortable). A vibration rated 2 DISC is considered twice as uncomfortable as a vibration rated 1 DISC. The percentage of people feeling uncomfortable rises rapidly with each unit of DISC.

Leatherwood *et al.* [25] point out that the NASA model is useful for estimating the average level of discomfort experienced by a group of passengers in a combined noise and vibration environment and not for predicting the discomfort of an individual passenger.

It is not easy to identify the bodily sensations that form the basis for comfort or discomfort judgments. Whitham and Griffin [26] indicated which specific body locations were uncomfortable at various vibration frequencies (acceleration was held constant at 1.0 m/s^2 rms). In general, most responses of seated subjects indicated pain in the lower abdomen at 2 Hz, moving up the body at 4 and 8 Hz, with most responses indicating the head region at 16 Hz. At 32 Hz the responses were divided between the head and the lower abdomen, while at 64 Hz they were mostly located near the principal vibration input site.

VIBRATION EVALUATION

There are four physical factors of primary importance in determining the human response to vibration, namely, the intensity, the frequency, the direction and the duration (exposure time) of the vibration.

The intensity of vibration is generally described by acceleration, which is normally expressed in terms of metres per second squared (m/s^2). The magnitude of vibration, that is, acceleration, is expressed as a root mean square (rms) value. When the peak values are measured, these are converted as appropriate to rms values before comparing with the limits given in ISO 2631 [1].

In the practical evaluation of any vibration of which a physical description can be given in terms of those factors, three main human criteria can be distinguished. These are given as:

(i) the preservation of working efficiency (fatigue-decreased proficiency boundary),
(ii) the preservation of health or safety (exposure limit), and
(iii) the preservation of comfort (reduced comfort boundary).

Fatigue-decreased proficiency

The fatigue-decreased proficiency (fdp) is a function of exposure time and frequency. Figure 3 shows the vertical acceleration limits and Fig. 4 shows the longitudinal and transverse limits, respectively, as a function of frequency and response times from 1 min to 25 h. The boundary specifies a limit beyond which exposure to vibration can be regarded as carrying a significant risk of impairing working efficiency in many kinds of tasks, particularly those in which time-dependent effects (fatigue) are known to worsen performance. It should be noted that for man, the most sensitive frequency ranges are 4–8 Hz for vertical (Fig. 3) and below 2 Hz for longitudinal and transverse vibrations (Fig. 4), and that human tolerance for vibration decreases with increased exposure time. It is evident on comparing Figs 3 and 4 that the tolerance for transverse vibration is lower than that for longitudinal vibration at very low frequencies, and that the reverse is true for higher frequencies.

Exposure limit

The exposure limit as a function of frequency and exposure time is of the same general form as the fatigue-decreased proficiency boundary, but the corresponding levels are raised by a factor of two (6 dB higher). In other words, maximum safe exposure is determined for any condition of frequency, duration and direction by doubling the values allowed according to the criterion of fatigue-decreased proficiency (Figs 3 and 4). Exceeding the exposure limit is not recommended without special justification and precautions, even if no task is to be performed by the exposed individual.

Reduced comfort boundary

The reduced comfort boundary (which is derived from various studies conducted in the transport industry) is assumed to lie at approximately one-third of the corresponding levels of the fatigue-decreased proficiency boundary in ISO 2631. Moreover, it is assumed to follow the similar functional relationship with time and frequency. Values for the reduced comfort boundary are thus obtained from the corresponding values of the fatigue-decreased proficiency boundary by a reduction of 10 dB (Figs 3 and 4). In

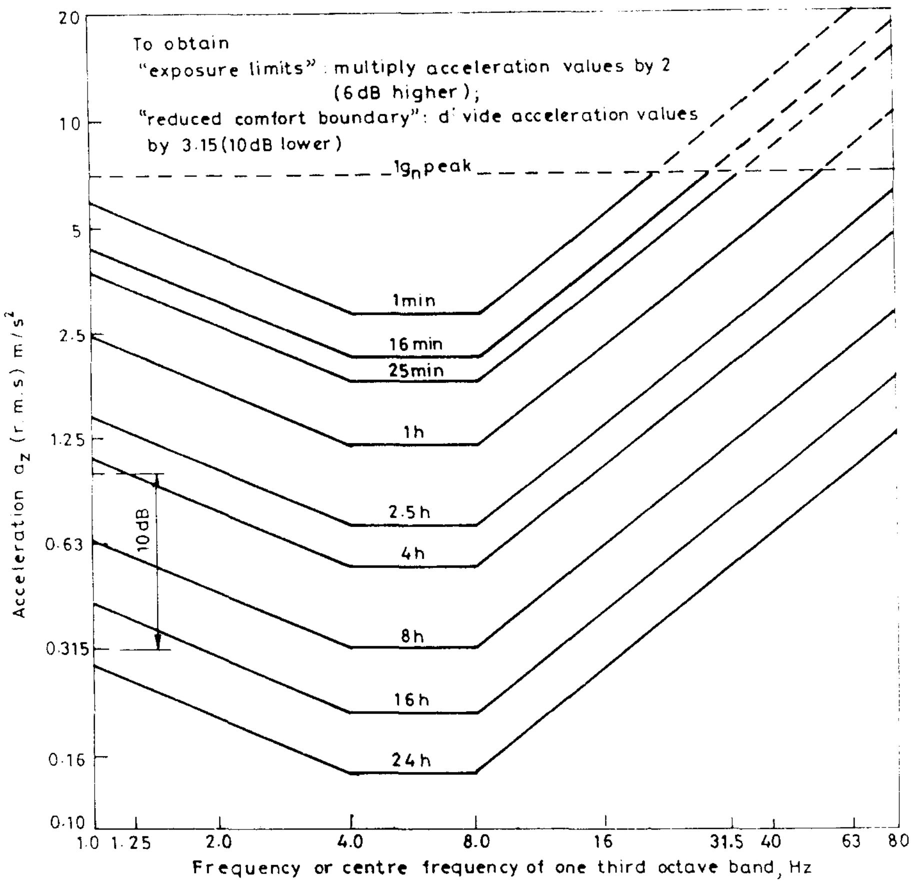

Fig. 3. Vertical (a_z) acceleration limits as a function of frequency and exposure time: fatigue-decreased proficiency boundary [1].

the transport situation the reduced comfort boundary is related to difficulties of carrying out such operations as eating, reading and writing.

PREDICTION OF TRACTOR RIDE VIBRATION

Several studies have simulated the ride vibration characteristics of unsuspended tractors. However, in very few cases simulations have been tested against measured results. Dale [27] has described a linear frequency domain simulation based on a rigid body tractor with a pivoted front axle and linear tyre characteristics. Vibration predictions made for one particular tractor using this model were similar to the vibration measured on that tractor. Crolla *et al.* [28] showed that root mean square (rms) accelerations in excess of 40% are not unusual, and that, for some tractors, trends in the predicted rms levels with changing vehicle speed are quite different from those which occur in practice.

It is widely accepted that poor tyre description is the main source of error in the vibration prediction [4, 29–31]. The tyre model used by Dale [27] and most other researchers describes each tyre as three mutually perpendicular Voigt–Kelvin units

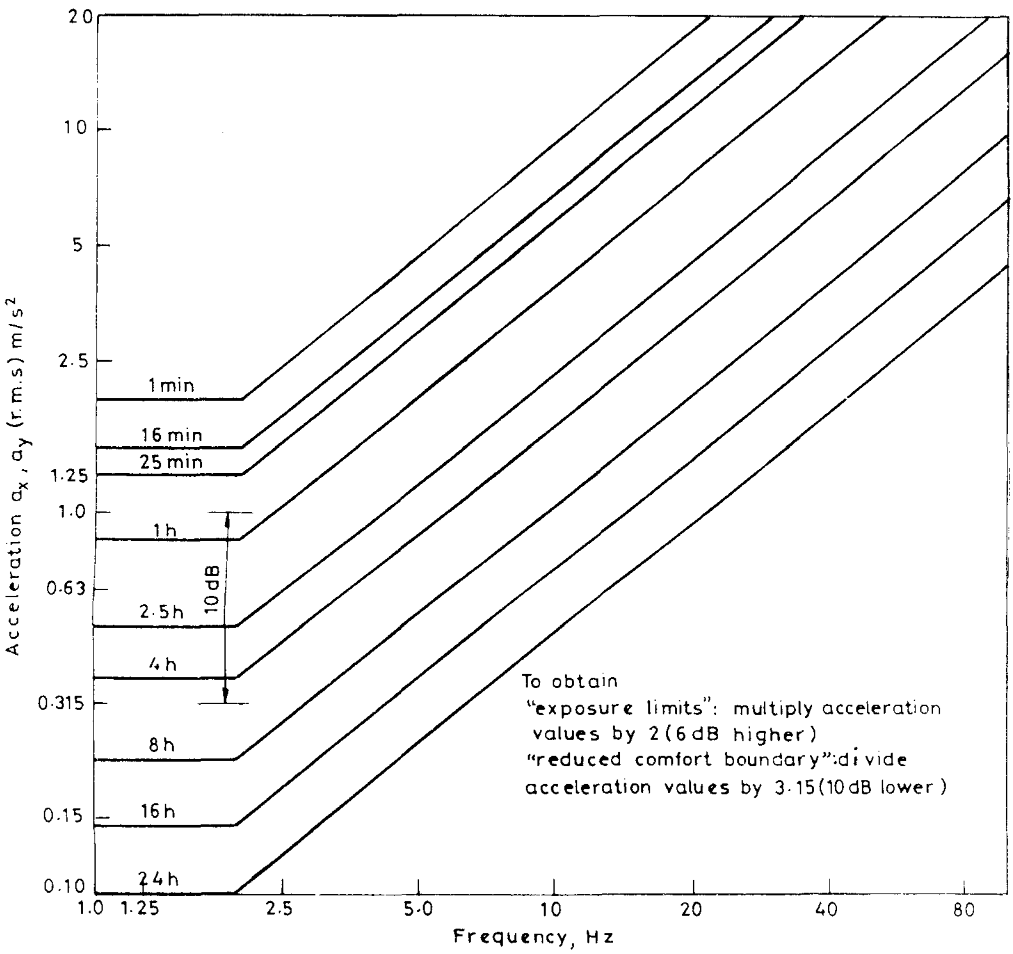

Fig. 4. Transverse (a_x, a_y) acceleration limits as a function of frequency and exposure time: fatigue-decreased proficiency boundary [1].

(linear spring and viscous damper in parallel) joining the tractor to the ground in the vertical (or radial), longitudinal (or tangential) and lateral (or axial) directions (Fig. 5). Until recently there has been almost no reliable information available concerning the suspension characteristics of rolling agricultural tyres. Much of the published simulation work has relied on the measured characteristics of stationary tyres, although these were known to differ from those of rolling tyres.

Measurements of the radial suspension characteristics of rolling agricultural tractor tyres have been made recently by Kising and Gohlich [32], and Lines and Murphy [33]. These measurements have indicated that for most purposes, a Voigt–Kelvin element is a suitable representation of the radial characteristics of agricultural tyres. However, it seems unlikely that this simple concept is valid for the lateral and longitudinal characteristics. Lines [34] has shown that a more appropriate description of these horizontal characteristics of rolling tyres might be obtained using a spring and viscous damper in series. This concept has been used by Crolla *et al.* [27] to produce a substantial improvement in the accuracy of ride vibration prediction (Fig. 6). Lines and Peachey [35] modelled a single degree of freedom system consisting of a tyre

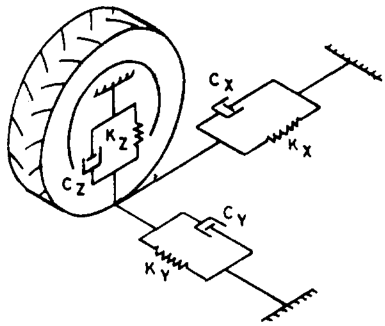

Fig. 5. Tyre model consisting of Voigt–Kelvin units (linear spring and damper in parallel).

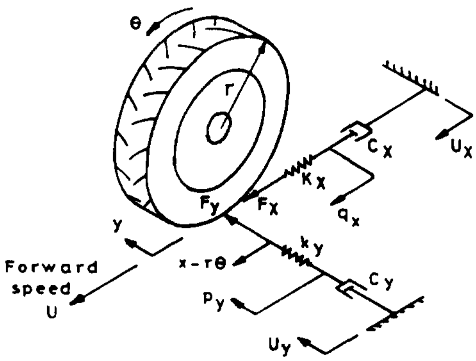

Fig. 6. Tyre model consisting of spring and damper in series.

rolling over a rough profile while supporting a mass using tyre data obtained from rolling tyres. All the measures of simulation accuracy which have been applied show improvement using these data. Most noticeable is the prediction of mode frequency, where errors were less than 3%. The rms acceleration level was reliably predicted to better than 30% and, in half of the cases examined, to be better than 10%.

Later, Lines *et al.* [36] modelled the vertical and pitch vibration of a four-wheel agricultural tractor. The accuracy of this model was assessed by comparison with measurements of tractor vibration. Improvements in modelling accuracy were obtained by the use of radial tyre suspension characteristics measured on rolling tyres rather than those obtained from stationary tyres. They stressed the need for longitudinal characteristic measurements for reliable prediction of levels of vibration.

Lines and Peachey [35] modelled a simple one-wheel vehicle (Fig. 7) as it rolled over a rough surface using the suspension characteristics of rolling agricultural tyres. The comparison with vibrations measured on such a vehicle showed that the natural frequencies of vibration were consistently predicted to within ±3% and that the rms acceleration levels were predicted to within ±30% with a 0.97 probability. Predictions made using the suspension characteristics of rolling tyres are more accurate than those made using stationary tyre characteristics. The greatest limitation on the accuracy of the model is the error in the definition of the profile of the rough surface.

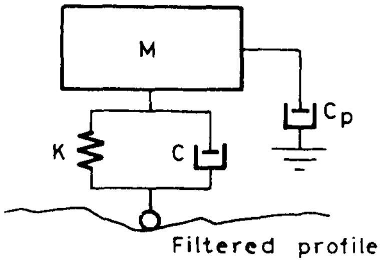

Fig. 7. Schematic model of the vibrating wheel carriage.

IMPROVING OPERATOR COMFORT

Conventional agricultural tractors are not equipped with special suspension systems between the axles and the chassis. Some amount of suspension between the road excitation and the tractor is provided by the tractor tyres and the reaction of the vibration behaviour of the tractor itself. Because of the low damping rate of the tyres, resonance occurs when the tractor is excited by the other frequencies in the range of its natural frequency. Based on this fact, the tractor has to be considered as an underdamped vibration system. Usually it is equipped with either suspended seats or suspended cabs to improve operator comfort.

Suspended seat

Suspension type seats are usually fitted to high speed rubber tyred agricultural machines. Suspension seats are cushion-mounted to a resilient spring, damper and linkage mechanism called the seat suspension. The purpose of the suspension is to reduce the level of machine vibrations transmitted to the operator, especially in the low-frequency range (2–6 Hz).

Seat suspension model

Several seat suspension models have been developed by various researchers. Verma [36] developed a two degrees of freedom system model for the seat–operator system in which he used a spring and damper for suspension and two masses, one spring and one damper for the operator (Fig. 8). In the operator model, the pelvis mass was lumped with the seat mass to yield a total mass of m_1. The thorax plus head masses (m_2) were lumped together and connected with the pelvis mass by a spring and damper, which were analogous to spine stiffness and muscle damping, respectively.

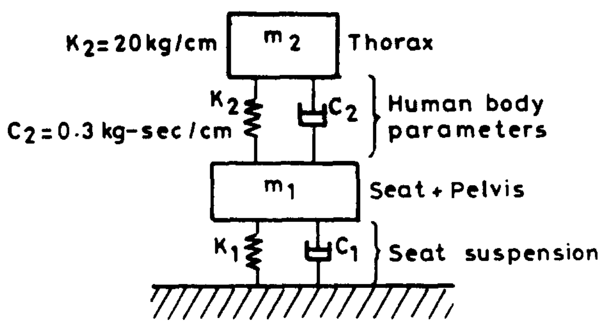

Fig. 8. Mechanical model of seat with the operator as a single degree damped spring mass system.

The ratio between thorax plus head and pelvis masses was considered to be 5:4. The cushion parameters were omitted in the model.

Gouw *et al.* [37] modelled suspension seats as a two degrees of freedom system (Fig. 9). Incorporating the driver mass on the seat (m_0), cushion characteristics (spring constant K_c and damping coefficient C_c), suspension mass m_s, suspension spring rate K_s, damping coefficient of shock absorber C_s, forces due to Coulomb friction F_f and elastic limit stops F_s (due to spring rate K_{st}). The contribution of driver mass to the seat was represented by inert mass equal to $\frac{5}{7}$ of the total driver mass.

Shankar and Afonso [38] studied the concept of a lateral seat suspension (Fig. 10) as a means of improving the ride comfort in off-road vehicles. The ride performance of such a suspension was investigated through computer modelling and simulation. A computer parametric study showed that a lateral seat suspension with a dynamic vibration absorber can significantly improve ride comfort by 75% and reduce peak relative displacement by 7%.

Rakheja and Shanker [39] developed a combined seat isolator for bounce, pitch and roll accelerations (Fig. 11). This isolator showed excellent attenuation of tractor vibration with satisfactory relative motion responses. However, the seat height was increased due to the additional frames for lateral suspension, which necessitated modification in the design of the controls around the operator.

Active seat suspension is another development to improve the sitting comfort of the tractor operator. Stikeleather and Suggs [40] have developed such a system, which consists of a vibration-sensing vibrometer, phase compensating network, power amplifier, feedback system, servo valve, hydraulic power supply and a linear actuator to drive the seat up and down which makes it possible to maintain the seat location constant irrespective of the vertical motion of the chassis. Performance of the active suspension system can definitely be justified, however, for many applications, economic justification may be difficult because of the cost of good servo-valves.

Suspended cab

Numerous cab suspensions have been developed for highway vehicles over the past decade, namely Ford's CL-9000, Easy Rider-I, Easy Rider-II, etc. Easy Rider-I, a completely pneumatic cab suspension for highway tractor–trailer vehicles, provides low and variable natural frequency along with static position compensation [41].

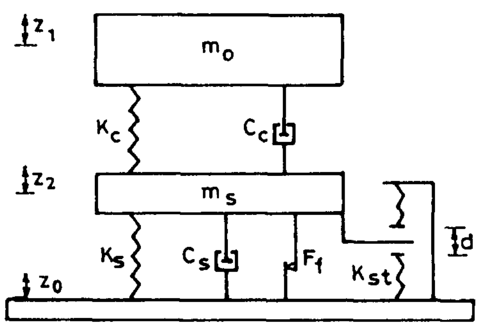

Fig. 9. Two degrees of freedom model of a suspension seat.

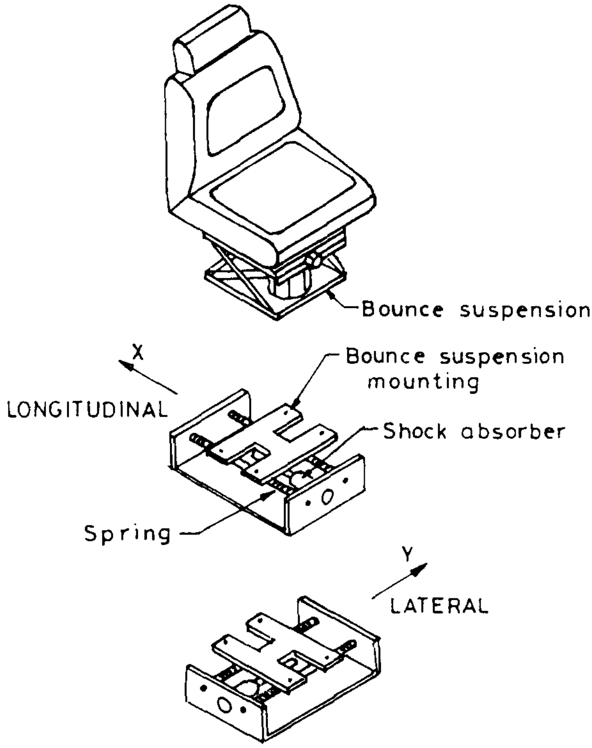

Fig. 10. Schematic of a multi-mode seat suspension.

Crosby and Allen [42] presented the highway vehicle cab analysis associated with the centre of gravity of cab geometry and the ratio of sprung to unsprung mass. The analysis showed that such factors influence the ride significantly.

Although exhaustive efforts have been made to develop effective cab suspension for highway vehicles, only limited studies have been conducted to develop cab suspensions for off-road vehicles. A scale model for a one degree of freedom tractor had been simulated by Suggs and Huang [43]. The suspension system consisted of two torsion springs and the natural frequency was observed to be 2.5 Hz. The scaled model offered improved ride at high frequencies alone. Hilton and Moran [44] conducted experiments on tractor cab suspension with natural frequencies of 0.8 and 1 Hz in bounce mode, 0.6, 0.85 and 1.2 Hz in pitch mode, and 0.5 Hz in roll mode. They found that bounce, pitch and roll vibrations could be effectively attenuated with natural suspension frequencies of 0.8, 0.6 and 0.5 Hz, respectively. Using computer models for a linear four degrees of freedom agricultural tractor cab, performance characteristics of cab suspension have been evaluated for passive, active and semi-active suspension elements [45]. Rakheja and Shanker [46] developed a five degrees of freedom cab suspension model selecting optimal suspension parameters to minimize roll and lateral vibrations of the tractor. A bounce suspension seat mounted on a sprung cab was modelled (Fig. 12) and the corresponding ride performance was assessed with regard to ISO proposed fatigue decreased proficiency limits. They found that a sprung cab can provide an excellent ride in the longitudinal and pitch modes. Moreover, the cab suspension was capable of isolating the driver not only from

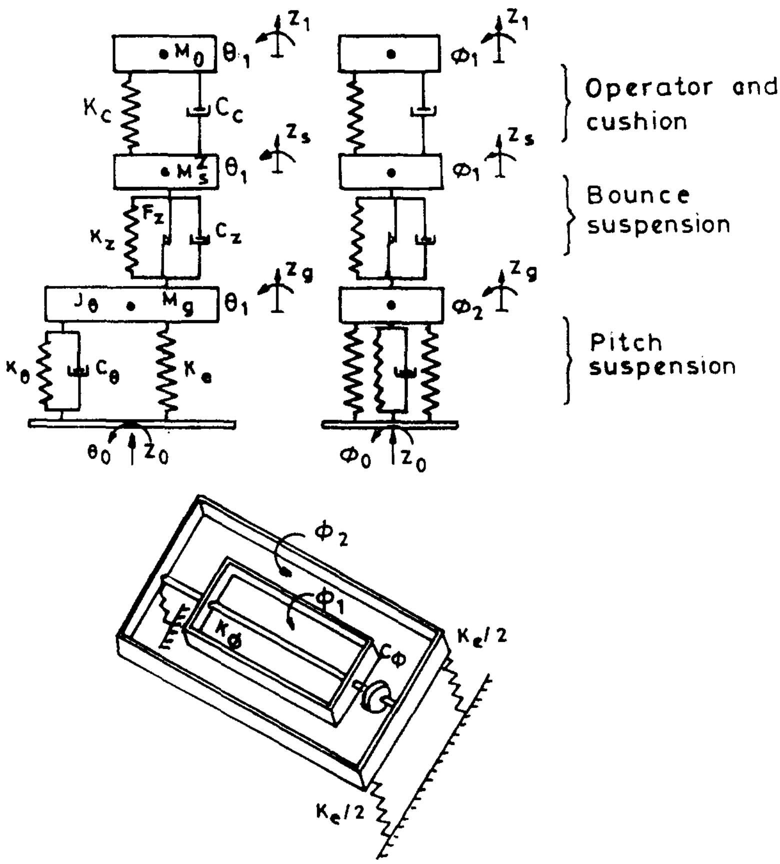

Fig. 11. Mathematical model of the combined seat isolator.

terrain-induced vibrations but also from forces introduced due to implement loads. The lateral ride of the tractor improved significantly by lowering the c.g. of the cab.

The need for suspended cabs requires changes in tractor design which will ultimately lead to increased cost, which renders it unfeasible for developing countries.

CONCLUDING REMARKS

The review on the effect of vibration has shown serious evidence of ill health and impaired performance ability of tractor drivers leading to an under-utilization of available power from the tractor. The frequency range of 2–6 Hz is the most harmful. Therefore, the vibration level at the operator's seat needs to be attenuated within acceptable limits in this frequency range. There continues to be the need for a practical method to judge objectively the relative performance of tractor seat pan and back-rest cushion materials with respect to reducing tissue stresses and stress gradients. Therefore, a simple model is needed to assess the relative performance of cushion, seat pan and back-rest profiles. Considerable efforts have been made to develop an optimum design for tractor seats for European and American operators. However, no such systematic effort is available for Asian subjects and hence there is a need for more study. Further reductions in the level of ride vibrations experienced

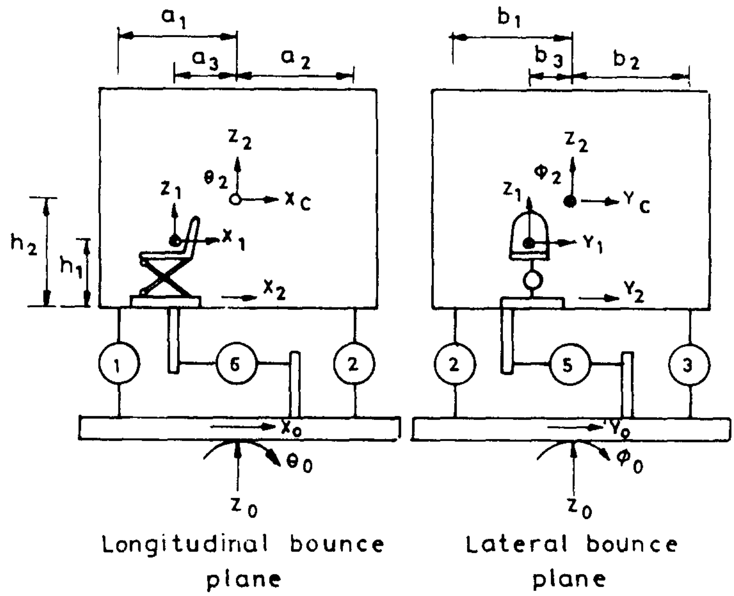

Fig. 12. Plane model representation of the cab and bounce seat suspension.

by tractor operators appear to be necessary and some possible methods of achieving significant improvements have been outlined, while effective methods of transverse vibration reduction may well emerge from subsequent research.

The attenuation of ride vibration to a considerable extent could by achieved by employing an appropriate seat suspension system, but the adequate attenuation of low-frequency vibration is not possible to achieve by seat suspension only. Extensive research is needed on suspended cab designs, which are very expensive but can provide an excellent ride in the longitudinal and pitch modes.

REFERENCES

[1] ISO, Guide for the evaluation of human exposure to whole body vibration. Standard 2631, International Organization for Standardization (1985).

[2] M. S. Sanders and E. J. McCormick, *Human factors in Engineering and Design* (6th Edition). McGraw-Hill Book Company, New York (1987).

[3] R. N. Janeway, Human vibration tolerance criteria and application to ride evaluation. SAE Paper 751066 (1975).

[4] R. M. Stayner, T. S. Collins and J. A. Lines, Tractor ride vibration simulation as an aid to design. *J. Agric. Engng Res.* **29**, 354–355 (1984).

[5] M. J. Griffin, E. M. Whithm and K. C. Parsons, Vibration and comfort—I. Translational seat vibration. *Ergonomics* **25** (7), 603–630 (1982).

[6] W. W. Chow and E. I. Odell, Deformations and stresses in soft body tissues of a sitting person. *J. Biomechanical Engng* **100**, 79–87 (1978).

[7] W. W. Chow, Mechanical properties of gels and other materials with respect to their use in pads transmitting forces to the human body. Ph. D. dissertation, University of Michigan, Ann Arbor (1974).

[8] D. S. Drummond *et al.*, A study of pressure distributions measured during balanced and unbalanced sitting. *J. Bone Joint Surgery* **64A**, 1032–1039 (1982).

[9] L. K. Holley *et al.*, A new pressure measuring system for cushions and beds—with a review of the literature. *Paraplegia* **17**, 461–474 (1979).

[10] L. A. Jaros *et al.*, The spiral pressure monitor. *Int. Conf. of the Assoc. for the Advancement of Rehab. Tech.*, Montreal. pp. 308–309 (1988).

[11] P. Glance, Computer-aided engineering of seat structure. *SAE Trans.*, Sec. 5, 396–402 (1987).

[12] R. Rossegger and S. Rossegger, Health effects of tractor driving. *J. Agric. Engng Res.* **5** (3), 241–274 (1960).
[13] H. Dupuis and W. Christ, Study of the risk of spinal damage to tractor drivers. Report, Max Planck Institut für Landarbeiter, landtech., Bad Kreuznach (1966).
[14] J. Reason, Motion sickness: some theoretical and practical considerations. *Applied Ergonomics* **9** (3), 167 (1978).
[15] J. Matthews, The measurement of tractor ride comfort. SAE Paper 730795 (1973).
[16] J. Matthews, Ride comfort for tractor operator I: Review of existing information. *J. Agric. Engng Res.* **9** (1), 3-31 (1964).
[17] E. Poulton, Increased vigilance with vertical vibration at 5 Hz: an alerting mechanism. *Applied Ergonomics* **9**, 73–76 (1978).
[18] C. Ashley, Equal annoyance contours for the effect of sinusoidal vibration on man. *Shock Vibration Bull.* **41** (2) (1970).
[19] R. W. Shoenberger and C. Harris, Psychophysical assessment of whole-body vibration. *Human Factors* **13**, 41–50 (1971).
[20] A. J. Jones and D. J. Saunders, A scale of human reaction to whole body, vertical, sinusoidal vibration. *J. Sound Vibration* **23**, 1–4 (1972).
[21] D. J. Oborne and M. J. Clarke, The determination of equal comfort zones for whole body vibration. *Ergonomics* **17**, 769–782 (1974).
[22] D. J. Oborne, A critical assessment of studies relating whole-body vibration to passenger comfort. *Ergonomics* **19**, 751–774 (1976).
[23] D. J. Oborne, T. Heath and P. Boarer, Vibration in human response to whole-body vibration. *Ergonomics* **24**, 301–313 (1981).
[24] D. J. Oborne, The stability of equal sensation contours for whole-body vibration. *Ergonomics* **21**, 651–658 (1978).
[25] J. Leatherwood, T. Dempsey and S. Clevenson, A design tool for estimating passenger ride discomfort within complex ride environments. *Human Factors* **22**, 291–312 (1980).
[26] E. M. Whitham and M. J. Griffin, Measuring vibration on soft seats. SAE Paper 770253 (1977).
[27] A. K. Dale, The theoretical prediction of tractor ride vibration. *Proc. Inst. Measurement and Control Conf.* London (1978).
[28] D. A. Crolla, D. N. L. Horton and R. M. Stayner, Effect of tyre modeling on tractor ride vibration predictions. *J. Agric. Engng. Res.* **47**, 55–77 (1990).
[29] J. Matthews, The ergonomics of tractors. *ARC Res. Rev.* **3** (3), 59–65 (1977).
[30] D. A. Crolla, Off road vehicle dynamics. *Vehicle System Dynamics* **10**, 253–266 (1981).
[31] D. A. Crolla and E. B. Maclaurin, Theoretical and practical aspects of the ride vibration dynamics of off-road vehicles—Part I. *J. Terramechanics* **22** (1), 17–25 (1985).
[32] A. Kising and H. Gohlich, Dynamic characteristics of large tyres. Paper no. 88357 presented to AG ENG 88, *Agricultural Engineering Conf.*, Paris (1988).
[33] J. A. Lines, and K. Murphy, The stiffness of agricultural tyres. *J. Terramechanics* **28** (1), 49–64 (1991).
[34] J. A. Lines, Ride vibration of agricultural tractors: transfer functions between the ground and tractor body. *J. Agric. Engng Res.* **37**, 81–91 (1987).
[35] J. A. Lines and R. O. Peachey, Predicting the ride vibration of a simple unsuspended vehicle. *J. Terramechanics* **29** (2), 207–221 (1992).
[36] J. A. Lines, R. O. Peachey and T. S. Collins, Predicting the ride vibration on an unsuspended tractor using the dynamic characteristics of rolling tyres. *J. Terramechanics* **29** (3), 307–315 (1992).
[37] G. J. Gouw, S. Rakheja, S. Shankar and Y. Fework, Increased comfort and safety of drivers of off-highway vehicles using optimal seat suspension. *SAE Trans.*, Sec. 2, 541–548 (1990).
[38] S. Sankar and M. Afonso, Design and testing of lateral seat suspension for off road vehicles. *J. Terramechanics* **30** (5), 371–393 (1993).
[39] S. Rakheja and S. Sankar, Suspension design to improve tractor ride: I Passive seat suspension. *SAE Trans.*, Sec. 4, 1096–1104 (1984).
[40] L. F. Stikeleather and C. W. Suggs, An active seat suspension system for off road vehicles. *Trans. ASAE* **13** (1), 99–106 (1970).
[41] A. W. Foster, Truck cab suspension. SAE Paper no. 780408 (1978).
[42] M. J. Crosby and R. E. Allen, Cab isolation and ride quality. SAE Paper no. 740294 (1974).
[43] C. W. Suggs and B. K. Huang, Tractor cab suspension design and scale model simulation. *Trans. ASAE* **12** (3), 283–285 (1969).
[44] D. J. Hilton and P. Moran, Experiments in improving tractor cab suspension. *J. Agric. Engng Res.* **20**, 433–448 (1975).
[45] D. G. Roley, Tractor cab suspension performance modelling. Ph.D. Thesis, Univ. of California, Davis (1975).
[46] S. Rakheja and S. Sanker, Suspension design to improve tractor ride: II Passive cab suspension. *SAE Trans.*, Sec. 4, 1105–1112 (1984).