Bull. Tokyo Univ. For., 111, 25–48 (2004)

The Whole-body Vibration Evaluation Criteria of Forestry Machines

Jae-Heun OH*2, Bum-Jin PARK*1, Kazuhiro ARUGA*1, Toshio NITAMI*1, Du-Song CHA*2 and Hiroshi KOBAYASHI*1

Introduction

Forestry operations have become increasingly mechanized. Today's forestry machines can cut down the tree, delimb it, buck it to proper lengths and load it on forwarder or truck without a single worker setting foot on the ground. Consequently, mechanized forestry operations lead to improved productivity. Not all of the mechanized operations, however, have been beneficial. New occupational hazards have been introduced into their work environment by modern forestry machines. Operators in forestry machines are exposed to noise, vibration, exhausting fumes and other hazards in the daily works. These factors also may all contribute to what might be termed the "ride quality" of a forestry vehicle. Among some of the above factors, such as vibration, particularly ride vibration, is one of the least understood stressors in the forestry operations and can be measured objectively. Also, covering cap can decrease harms of exhausting fumes and noise, but ride vibration is serious because most of forestry machines have not been equipped with the suspension system.

Ride vibration is a problem in forestry machines and is often the cause of injury and discomfort. Ride vibrations are generally induced by ground unevenness, moving elements within the machine, or implements. Many researchers have studied ride vibration over the past sixty or more years. In forestry field, previous studies have shown that forestry vehicle operators are exposed to high levels of whole-body vibration (SHISHIUCHI, 1972; GOLSSE and HOPE, 1987; HANSSON, 1990).

Forestry operation like logging operations requires forestry machines to operate productively; a productive forestry machine operator drives as fast as it can and each trip brings in as many trees as possible. Forestry machines travel over rugged forestry terrain that includes rocks, logs, stumps, and abrupt transitions. When traveling fast over the rugged forestry terrain, the operators experience harsh ride vibration. Due to high driving speeds and increased capacity pushed by labor costs, large-scale production, and a competitive environment, vibration levels of forestry machines have also increased.

Neglecting machine dynamics in the design results in reduced or poor rides performance, operator discomfort, health problems and increased noise. Thus, in designing mobile forestry machine with improved ride, machine dynamics must be considered and designed with machines need to evaluate the ride, as it would be driven up to ride limiting speed. To evaluate designs of forestry machines with improved ride, we must consider forestry machines designs under common test condition and at the extreme ride conditions.

Although standards exist for assessing the character of the ride vibration, little information is available on how to use them to test and evaluate the ride vibration at driving limit speed.

Therefore, the objective of this study is to collect information on ride vibration and its

*1 Laboratory of Forest Utilization, Graduate School of Agricultural and Life Sciences, The University of Tokyo.
*2 College of Forest Sciences, Kangwon National University.

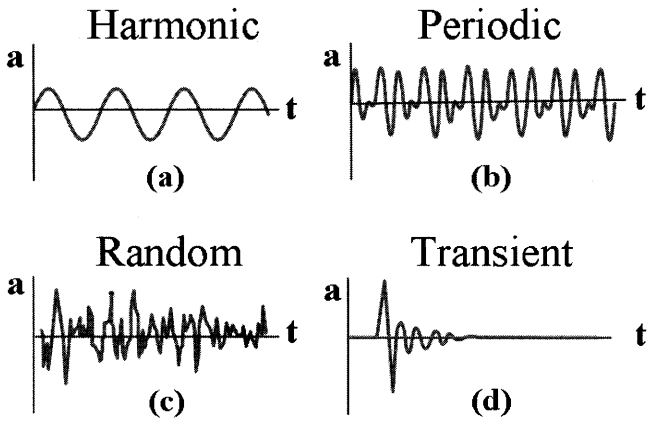


Fig. 1. Types of vibration.

effects on operator and to briefly review the several ride vibration standards and criteria having relevance to off-road vehicles such as forestry machines and to suggest recommendations for the effective use of such criteria in ride evaluation of forestry machines.

Types of Ride Vibration

Vibrations primarily can be divided into the following categories: harmonic and periodic vibration, random vibration and transient vibration. Harmonic and periodic vibration is regular in nature and predictable because that is made up of one or several sinusoidal components and repeats itself in time, for example, vibration caused by out of balance tires on a road vehicle and engine (Fig.1 (a), (b)). Random vibration is irregular and unrepeatable because that does not repeat itself continuously, for example, vibration experienced when driving a vehicle on a bumpy road (Fig. 1 (c)). Transient vibration is caused by mechanical shock in short duration, for example, vibration occurring when a skidder blade striking a stump or a sudden drop into a hole (Fig. 1 (d)). In practice, there will be mostly a combination of harmonic, random and transient vibrations

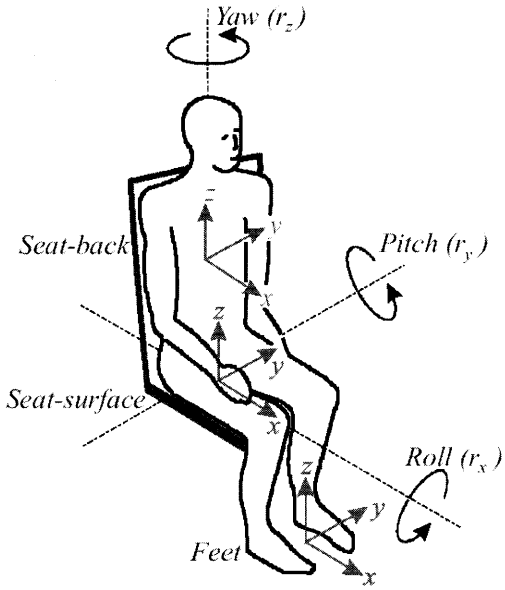


Fig. 2. Direction of basicentric coordinate systems for mechanical vibration influencing humans.

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. 2). Accordingly, three components of vibrations, a_x, a_y and a_z are considered. Accelerations in the foot or buttocks to the head along a vertical axis are designed 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 (ISO 2631, 1997). HUANG *et al.* (1967) studied on the

physiological responses and performance deterioration of human subjects exposed to a range of sinusoidal vibrations in vertical, longitudinal, and transversal directions for the most critical frequencies found in the field operations. Of these three directions, the vertical vibrations had the most significant effect on operators. This result coincided well with the results of the works by GERKE and HOAG (1981). They showed that vibrations in the vertical mode appeared to be severe than those in other modes although the vehicle seats were designed to reduce ride vibrations in the vertical mode.

The angular vibration that effects the human body can be resolved into yaw, pitch and roll (Fig. 2). The rotational mode of vibration does not usually cause much discomfort (GRIFFIN *et al.*, 1982). However, in some instances, such as forestry vehicles going over rough terrain, the pitching or rolling motions of the seat may be more disturbing than rectilinear vibrations.

Characteristics of Ride Vibration

Ride vibrations of off-road vehicles induced by various internal and external excitations such as engine, transmission, implement, ground unevenness, etc. Frequency of the vibrations depends upon the types of excitations. The vibrations induced by the irregular ground surface have frequencies lower than 50 Hz usually while those by the engines exhibit frequencies higher than that generally, over idling frequencies of the engine (LAIB 1977). The vibrations induced by such excitations cause not only loosening or weakening of fasteners but also operator's discomforts including mental and physical stresses which have a detrimental effect on the operator's health, particularly in connection with spinal and stomach disorders, in the long run.

The human body may be considered as a complicated visco-elastic system with many masses, spring and dampers (Fig. 3). Each sub-system has its own resonance frequency band and the interactions between sub-systems are influenced by the body's position. One

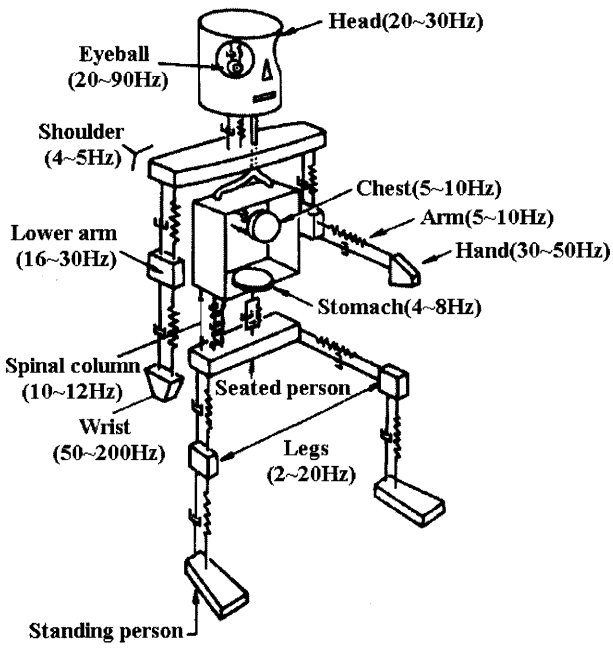


Fig. 3. Simplified mechanical system of the human body (BRÜEL & KJÆR, 1988).

of the most important parts of the system with respect to vibration and shock seems to be the abdominal part with the resonance occurring in the 4–8 Hz ranges. The other main resonant effect is found in the range 20–30 Hz related to the head-neck part. Also, a vibration in the region 20–90 Hz correlates with the eyeball resonance. Above 100 Hz, the model from Fig. 3 is not useful, and other more complex analyses have to be used. Some of the analyses show that the skull itself has a fundamental mode of vibration in the region of 300–400 Hz with resonances for higher mode around 600–900 Hz (BRÜEL & KJÆR, 1988).

SUGGS (1973) reported also that the commonly occurring vertical frequencies for off-road vehicles are coincident with the 4 to 8 Hz natural frequency of human body in the vertical mode, and pointed that such frequency ranges are coincided with natural frequency of stomach. Generally, horizontal modes of vibration in off-road vehicles have been observed below 2 Hz. The range of frequencies most often associated with effects of whole body vibration on health, activities and comfort is approximately 0.5hz to 100 Hz. At very low frequency frequencies (less than about 0.5 Hz) the principal effect of oscillation is a type of motion sickness (GRIFFIN 1990).

1. Effects of ride vibration to human

Exposure of the human body to vibrations can result in biological, mechanical, physiological, and psychological effects. Ride vibration intensities are normally positively correlated with ground speed and often become intolerable as speed is increased (LIJEDAHL *et al.* 1996). GRIFFIN(1990) showed that whole-body vibration generally concerns frequencies between about 0.5 and 100 Hz and acceleration magnitudes between about 0.01 and 10 m/s^2 (peak). HUANG *et al.* (1967) studied on the physiological responses and performance deterioration of human subjects exposed to a range of sinusoidal vibration in vertical, longitudinal, and transversal directions for the most critical frequencies found in the field operations. From the review of the literature, ride vibration indicates two main effects

- reduced operator comfort and a possible health risk to the operators of such vehicles
- reduced ability of the operator to execute manual dexterity task as well as reduced visual acuity.

1.1 Effects on health

From a health risk point of view, the frequently cited work of ROSSEGGER and ROSSEGGER (1960) indicated that the incidence of spine deformation and stomach disorders among tractor drivers was significantly higher than that of any other trade of the same age in the survey (Table 1).

Measurements on tractor drivers showed abnormal radiographic changes of the spine in 61% of the cases of drivers with more than 700 tractor driving hours per year. For more

Table 1. Pathological spine deformations in different occupations as determined by X-ray examinations.

Occupation	% with spine deformations	Average age (years) of sample
Truck drivers	80.0	—
Tractor drivers	71.3	26
Miners	70.0	51
Bus drivers	43.6	40
Factory workers	43.0	45
Construction workers	37.0	51

Source: R. Rossegger and S. Rossegger. "Health effects of tractor driving." J. Agr. Engr. Research, vol 5, no. 3, 1960.

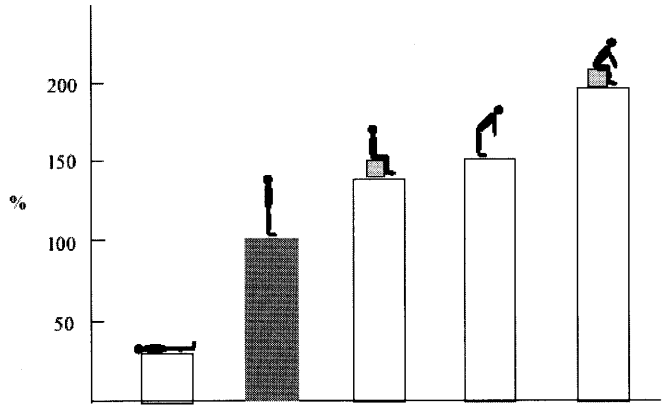


Fig. 4. The relative loads on the third lumbar disk for various body postures.

than 1,200 driving hours per year, abnormal radiographic changes of the spine are detected in 94% of the cases (CHRIST and DUPUIS, 1966). This indicates that prolonged exposure greatly increases the risk of spinal abnormalities. Similar results have been reported for drivers of heavy earth-moving equipment and truck drivers (BOSHUIZEN *et al.*, 1992; POPE and HANSSON, 1992). Thus, lumbago and spinal diseases have mainly shown in long term seated operators like drivers, since sitting posture works have high disc pressure in relation to the other posture works as shown in Fig. 4.

1.2 Performance effects

The performance of operator can be affected unfavorably by exposure to vibrations. The effect of vibrations on sight, as well as performing the operator's ability to carry out driving tasks and other functions associated with operating a vehicle, has been examined extensively. MATHEWS (1964) reported on the effects of ride vibration on compensatory tracking, foot pressure constancy and visual acuity as shown in Fig. 5. It is noted that the reduction in ability is more pronounced as the vibration level increases. Visual performance is generally impaired most by vibration frequencies in the range of 10 to 25 Hz. (SANDERS and MCCORMICK, 1992). MOSELEY and GRIFFIN (1986) found that, for frequencies below 3 Hz, reading time and errors are greater when only the display is vibrated than when only the reader is vibrated. A decreased visual acuity, a vague image and a loss of sight have been proved as consequence of exposure to vibrations. These troubles cause a considerable increase of perceptual load. The greatest problems appear with frequencies between 4 and 8 Hz; tracking errors grow proportionally with the acceleration of an appearing vibration. If an operator is exposed to vibrations for longer periods, he will clearly make more errors and mistakes. Also HANSSON and SUGGS (1973) have studied the effect of low frequency vibrations on the ability of the operator to manipulate levers and pedals. They found that the accuracy of manipulation of controls decreased with vibrations of a frequency higher than 2–2.5 Hz and a peak-to-peak amplitude of 10 to 20 mm. The best results were obtained when the resistance for movements was 0.5–2.6 kp for hand-operated controls, and 2.3 kp for foot-operated controls. These values fit the levels suggested by HANSSON *et al.* (1973).

WUOLIJOKI (1981) expressed similar results in simulated ride vibration of a forest machine using by full-scale mockup of a forwarder cab. He tested twelve subjects in the simulator using sinusoidal and stochastic vibration. Heart rate, flicker fusion frequency, tracking ability, blood pressure, finger skin temperature, head displacement, manipulative

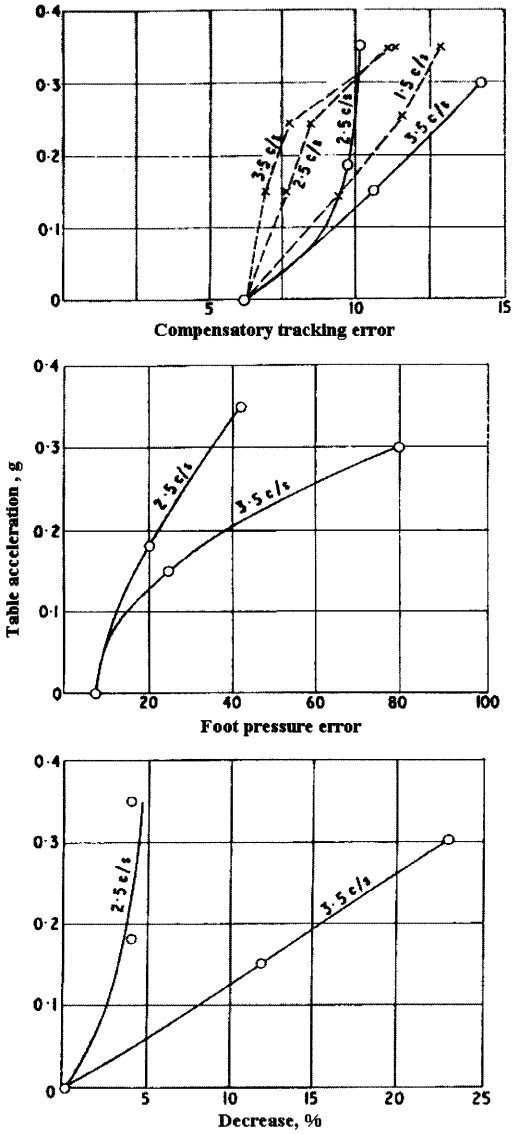


Fig. 5. The effect of vertical (full line) and transverse (broken line) vibration on compensatory tracking (top), foot pressure constancy (center) and visual acuity (bottom). Visual acuity not impaired by transverse vibration (MATHEWS, 1964).

ability, subjective rating of the vibration, and urinalysis samples were measured for each subject. Analysis of the data revealed that there were no statistically significant effects of whole body vibration on tracking ability, heart rate, blood pressure, blood constituents, skin temperature, or flicker fusion frequency. Significant effects were observed in subjective ratings of the vibration, manipulative ability, and head displacement. There was a significant correlation between the subjective ratings and head displacement.

Ride Vibration Studies in Forestry Vehicles

1. Ride vibration studies in forestry operations

There is less data on the ride vibration of forestry vehicles than on that of agricultural tractors, because most cases are fewer in use, and they have been introduced more recently. Early forestry vehicles were very often based on agricultural tractors in any case. Most studies of ride vibration from forest equipment operation have concentrated on measuring vibration exposure levels and comparing them with various exposure guidelines.

Many vibration studies in forest machines have been made for the past 40 years in European country. SJØFLOT (1970) studied the seat-vibration of an tractor skidding logs on an even forest ground, and found the highest rms acceleration in the vertical direction (1.3 m/s^2) in the interval 2–4 Hz.

A more extensive study was made by AHO & KÄTTÖ (1971), who found the highest acceleration value to be in the same frequency interval and direction, but the value was higher (3.8 m/s^2) for a speed of 6 km/h (Fig. 6).

If data is compared with the draft ISO/DIS 2631-recommendation "Guide for the evaluation of human exposure to whole body vibration", it will show that the limit of 2.5 hours daily exposure is exceeded at speeds over 5.8 km/h in the frequency interval of 1–5 Hz. Also the 4-hours limit is exceeded in the especially critical interval of 2–6 Hz. Studies by HANSSON and WIKSTRÖM (1974) on 6 forwarders and 4 processors showed less vibration in the forwarders than did AHO and KÄTTÖ (1971).

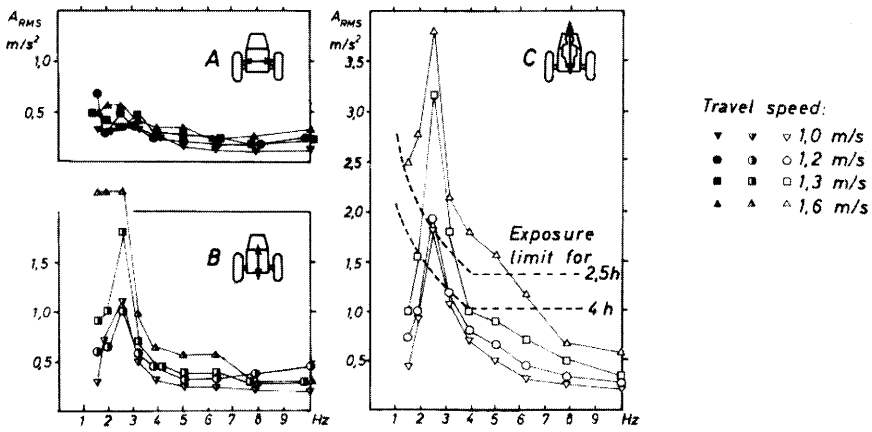


Fig. 6. Vibration level on forestry tractor traveling at different speeds on a test track (AHO and KÄTTÖ, 1971)
A: at the mounting point of the seat in the lateral direction, B: at the mounting point of the seat in the vertical direction, C: on the operator's back in the vertical vibration.

HANSSON and WILSTRÖM (1981) presented data in one-third-octave band form, for comparison with the ISO 2631. The most severe ride was found on skidder and forwarders, with z-axis rms acceleration in the most prominent frequency band exceeding the curve for 0.8 m/s^2. Of more potential interest to the present review is data from the study by WIKSTRÖM *et al.* (1991) aimed at correlating subjective reactions to shock events with various objective measures of the shock acceleration histories.

WASSERMAN *et al.* (1978) have studied heavy equipment operators exposed to whole body vibration including vibration data from a CAT skidder operated by an experienced driver. Vibration levels were measured at the operator-seat interface during a work cycle that included loaded and unloaded travel in the woods at 4–6 mph and road travel at 15 mph. The spectral data was averaged to depict "a typical vibration dose that operators might receive over the work day." In the observed data, the maximum accelerations occurred at 1 Hz in the lateral (x) direction. The average acceleration level of the 1 Hz peak was 1.28 m/s^2.

WEBB and HOPE (1983) investigated ride vibration as part of a larger study of ergonomics in skidder operations. Twenty-one skidders, representing nine different models from five manufacturers, were included in the study. Vibrations levels at the operator-seat interface were recorded with a Human Response Vibration Meter (Brüel & kjær, type 2512) during actual logging operations. A maximum equivalent continuous vibration level of 0.9 m/s^2 was observed in the x-axis. Vertical (z) and horizontal (y) accelerations averaged 0.7 m/s^2. Large peak values were recorded, ranging from 11 to 25 m/s^2.

HANSSON (1976) summarizes a research program that studied vibration of forestry equipment and the physiological effects on the operators. The results of the physiological studies have been noted previously. Fifty-six machines, including skidders, loaders, forwarders, feller benchers, and specialized processors, were tested during regular operation with the normal operators. Based on time study and vibration data from element of a machine's work cycle, the maximum allowable exposure time was calculated by ISO 2631. All of the machines exceeded the limit for fatigue-decreased proficiency during an 8-hour shift. Skidders and loaders had the highest vibration levels occurred during unloaded travel. He also suggests that tire characteristics and travel speed significantly affect

vibration, but no quantitative analysis were performed.

The results of SHISHIUCHI (1980) show that driving and skidding task done by logging tractor drivers in the early 1970's exposed to high levels of vibration. The average rms acceleration levels measured under same running speed, the seat vibrations of crawler tractors were greater than those of wheeled tractors and the seat vibrations of the crawler or wheeled tractors driven on hard surface of the ground such as gravel road were greater than those on soft surface of the ground such as forest land. For about 3 km/h running speed, accelerations of the seat vibration logging tractors driven on the forest land or on skid road were also observed to be below than fatigue-decreased proficiency boundary of the ISO standard (1974).

GOLSSE and HOPE (1987) report measurements for a number of skidders. They report z-axis magnitudes of around 1 m/s^2, with peaks of 5 m/s^2 to 7 m/s^2, and x-axis and y-axis components not much smaller. All these values were frequency weighted, and assuming that the main frequency was about 2 Hz, the unweighted peak z-axis acceleration would have been about 10 m/s^2. the more severe transient accelerations on forestry machines are associated with passage over large boulders and tree stumps. These may involve slower forward speeds than tractors on farm roadways, but they would also involve much larger displacements.

RUMMER (1988) examined forest workers' attitude towards whole body vibration and developed a computer ride simulation model that can be used to improve the design of forest machines. The study surveyed a representative sample of 26 forest equipment operators and rated their subjective assessment of whole body vibration as a source of environmental stress.

In recently, ride vibration measurements related to small sized forestry machines were trequently reported. MARSILI *et al.* (1998) examined a new off-road machine that has been introduced into the European market for forestry use. The results indicated that the level of vibration on the track test was low and should not compromise the health of the driver. However, during transport on uneven ground and at high average speed (12 km/h), vibration at the seat (1.7 m/s^2) was such that a driver's exposure, prudently, should not exceed 2 h per day, if the most stringent criterion of the international Standard is followed.

OH *et al.* (2002) showed that the weighted acceleration levels of average ride vibration in the dominant direction were from 1.92 to 2.93 m/s^2 at forest road. When these results are compared with the health guidance caution zones of ISO 2631-1(1997), if average transporting work time in seat was exceed the one hour per day, continually, they are exposed to potential health risks because mini-forwarder operator appear to inner and excess the 2–4

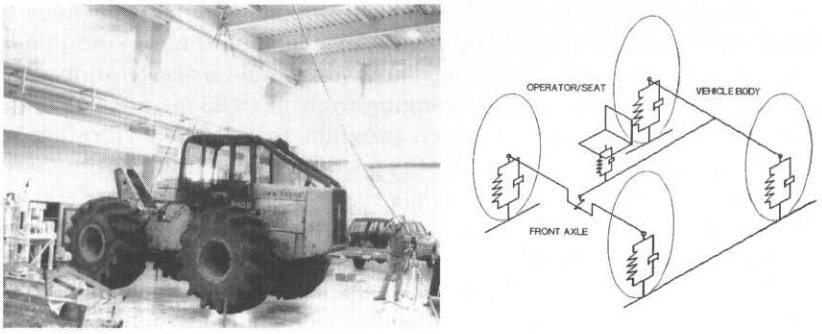


Fig. 7. Measuring machine properties and their dynamic model for whole body vibration analysis (RUMMER, 1986).

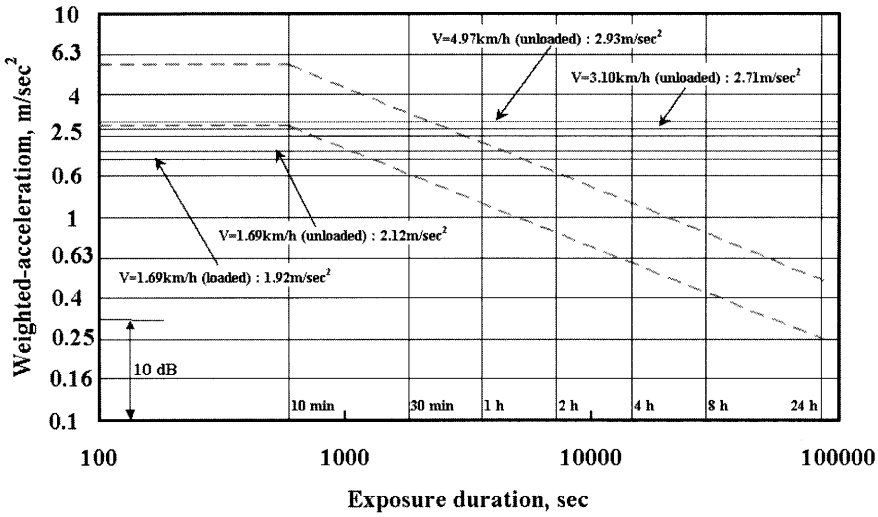


Fig. 8. Health guidance caution zone and vibration exposure levels on rubber-tracked mini-forwarder (OH *et al.*, 2002).

hours zone (Fig. 8).

Although the reviewing of the literature did not reveal evidence of physical injury resulting from long-term ride vibration exposure in forestry operation like logging and transportation, it can be known that forest operators exposed ride vibration to be one of the most discomforting aspects of their work.

2. Driving characteristics of forwarders and skidders in timber harvesting

While operating, forestry machines such as forwarder and skidder can reach the maximum design speed. However, operators of forestry machines don't drive continuously at fast speeds. Off-road travel speeds for skidders and forwarders are in the neighborhood of 2.4–5.6 km/h, with 5.6 km/h considered to be fairly fast (LEGAULT and POWEL, 1975). In a large quantity of data assembled and analyzed in FINLAND by HAARLAA (1975), average speed of log transport equipment under both summer and winter conditions was about 4 km/h with forwarders being slower than skidders. It has been reported that the FMC 200 BG tracked skidder traveled empty at 5.6 km/h off the road, whereas the specifications for that vehicle give the maximum speed in high gear as 24.0 km/h. GOLSSE and HOPE (1987) reported an average travel empty speed of 5.7 km/h and an average travel loaded speed of 4.8 km/h in 8 working skidder.

SAKAI *et al.* (1995) reported that uphill skidding speed of an articulated-frame wheel skidder was about 6.0 km/h where gradient is 6°–13° without log loading, but downhill skidding speed with log loading of less than 4.6 m^3 varied over a range of 2.6–14.0 km/h when gradient became −6°−−13°. These average speed values show that skidders travel much slower than the machine's maximum speed. This difference between the maximum speeds and actual travel speeds indicates that skidders operate under conditions that limit their speed. Therefore, we can know that operators may limit their speed due to ride limiting vibration.

In addition, when collecting and loading the log, forestry machines slow down to maneuver and to move back and forth. Also, when traveling, forestry machines slow down to make turns, to maneuver around or over obstacles, and to avoid damaging standing

trees.

Similar opinions were expressed in a survey of Japanese forest machine operators conducted by imatomi (1988). He reported that operators of forestry vehicles on rough ground like strip road are usually operated in one of the lower gears at part throttle due to their safety and health. He has also shown that the physiological load of a forestry vehicle operator in off-road driving is higher than that in on-road driving through the field experiment.

Skidders and forwarders are recognized as having a rough ride, the roughest of any forest machine (WEGSCHEID, 1994), and reported ride vibration exceed the levels recommended in the ISO-2631 vibration standard (ISO, 1985). Also their operators were affected by vibration more than processor or feller buncher operators (RUMMER, 1986).

Ride Vibration Assessment Criteria

1. Evaluation methods of ride vibration

Ride quality is composed with driver's sensation of the terrain-induced vibration environment of a vehicle, and is generally difficult to assess. Numerous studies have been conducted to establish the ride assessment criteria for preservation of driver comfort, health, safety, and performance.

In general, ride comfort boundaries of vehicle operators are difficult to determine because of the variations in individual sensitivity to vibration and a lack of a generally accepted method of approach to the assessment of human response to vibration. Two different methods are frequently used to evaluate the whole-body vibration: subjective and objective.

1.1 Subjective methods

Subjective methods are often based upon subjective response of vehicle ride comfort on an absolute scale and used to assess relative ride ranking of a group of vehicle, operator tolerance in relation to productivity, vibration interference with normal operator control task, health aspect to vocational exposure, competitive significance and cost/benefit ratio of potential ride improvement (STIKELEATHER, 1976). Subjective methods, however, often lead to misleading information due to a multitude of inconsistencies dependent upon age, preferences, and moods of the subject at the time of experiment.

1.2 Objective methods

Alternatively, objective methods provide an assessment methodology based on direct measure of physical quantities such as velocity, acceleration, absorbed power, and jerk over the frequency range of interest. Intermediate methods also attempts to correlate the response of test subjects in qualitative terms, such as "unpleasant" or "intolerable" with vibration parameters measure at the location where the test subject is situated under actual driving conditions (FINE, 1964; VAN DEUSEN 1968).

2. Review of evaluation criteria for ride vibration

As mentioned above, the assessment of human response to vibration is complex in that the results are influenced by the variations in individual sensitivity, and by the test methods used by different investigators over the years, numerous objective ride comfort criteria have been proposed, however, a generally acceptable criterion is yet to be established. Some of the proposed ride criteria and standards are reviewed as follows.

2.1 Earlier criteria of interest

Earlier studies on human body response to vehicle vibration were primarily carried out by using shake table (REIHER and MEISTER, 1931; JACKLIN and LIDDELL, 1933; MEISTER, 1935; DIECKMANN, 1947). Most of this research pertains to human response to sinusoidal excita-

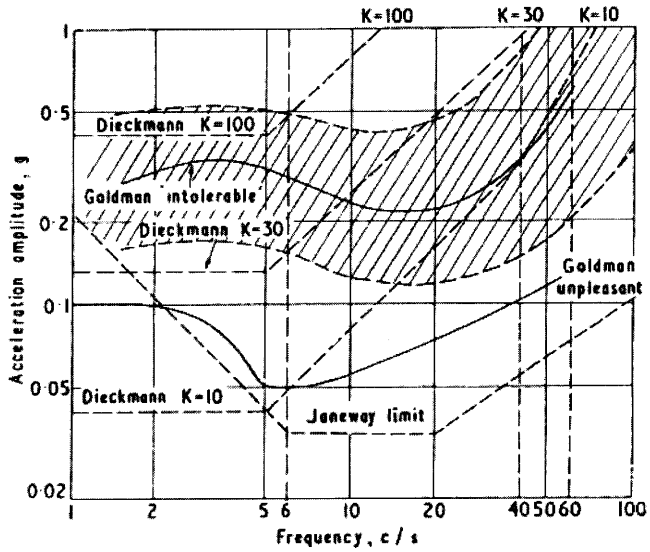


Fig. 9. Comparison of ride comfort criteria.

tion. It is intended to identify zones of comfort (or discomfort) for humans in terms of vibration amplitude, velocity, or acceleration in a given direction over a specific frequency range. Based on physiological, subjective and physical studies, DIECKMANN (1957) proposed constants related to comfort zones and fatigue time limits for passenger car vibrations. A constant, K, is proposed based on levels of constant acceleration up to 5 Hz, constant velocity from 5-40 Hz, and constant displacement above this frequency. The Dieckmann's criterion proposed three different comfort zones based upon the constant values, which are defined as the followings:

- For $K=0.1$ to 1: imperceptible to slightly uncomfortable.
- For $K=1$ to 10: slightly disagreeable to disagreeable with a fatigue time of 1 hour.
- For $K=10$ to 100: very disagreeable to exceedingly disagreeable with a fatigue time of 1 minute.

These K values are not very often used today but they are important in relation to subsequent German standards. They were used as German Standard (Deutsches Institut fur Normung, 1958) DIN 4025.

GOLDMAN (1948) analyzed the vibration data acquired from several sources, and deduced three comfort levels in the vertical mode in terms of acceleration and frequency contents. The comfort levels were referred to as perceptible, unpleasant, and intolerable. The vibration data used by Goldman was obtained from a variety of experiments where the subjective and physical environments varied considerably.

JANEWAY (1948) recommended exposure limits for vertical vibration of passenger cars in terms of maximum jerk in the frequency range of 0–6 Hz, and maximum acceleration for middle frequency range of 6–20 Hz. The safe limits of vibration exposure were proposed based on the survey of subjective tolerance data, which represented an attempt to set a level at which no discomfort is experienced by the most sensitive passenger.

The appropriate limits proposed by DIECKMANN (1957) and JANEWAY (1948) are compared with summarized data by GOLDMAN (1961) based on vibrator tests by various workers, with exposure time 5–20 min. The shaded areas of Fig. 9 about Goldman's limits represent the standard deviations of individual observer reactions about the mean levels. A

comparison of the Dieckmann, Janeway and Goldman ride criteria revealed that the human is most sensitive to vertical vibration below 20 Hz (VAN DEUSEN, 1968). Since the above ride assessment criteria have been established based on sinusoidal vibration at a constant frequency, their application to assess the vehicle's random ride vibration is questionable. VON ELDICK THIEME (1961) and BUTKUNAS (1966), however, have outlined methods for applying existing ride comfort criteria to random vibration environmental of vehicles.

2.2 Current standard methods

2.2.1 VDI 2057

In Germany, the K-value was proposed as a measure of vibration intensity, based on subjective assessment (VDI-2057, 1963). A new version of VDI 2057 (1987) published by German Society of Engineers (VDI) employed the shape of the frequency contours defined in ISO 2631 (1974) while retaining the concept of K-values. It includes the definitions for a seated human body, where the initiation of excitation happens at the seat surface and the hands. It does not apply for initiation of excitation at the feet and at the seat-back of a seated person. The equivalent comfort contours are applicable in the frequency range of 1 to 80 Hz for the seat. The mathematical description of the curve for the seat in z-direction is given as

$$1Hz \leq f \leq 4Hz: KZ = 10 \frac{a_z}{[m/s^2]} \sqrt{\frac{f}{Hz}} [K-Value]$$

$$4Hz \leq f \leq 8Hz: KZ = 20 \frac{a_z}{[m/s^2]} [K-Value]$$

$$8Hz \leq f \leq 80Hz: KZ = 160 \frac{a_z}{[m/s^2]} \frac{|Hz|}{f} [K-Value]$$

where, f represents the frequency and a_z the rms value of the acceleration in z-direction of the seat pan. The frequency weighting $B_{seat,z}(f)$ can be derived by

$$B_{seat,z}(f) = \frac{KZ}{a_z}$$

Fig. 10 depicts the frequency weighting $B_{seat,z}(f)$ of the equivalent comfort contour KZ for the seat. The relation between the equivalent comfort contour KZ and the frequency weighting $|W_k(f)|$ for the acceleration, defined in ISO 2631, is given as

$$KZ = 20|W_k(f)|$$

For initiation of excitation in x- and y-direction the equivalent comfort contours KX

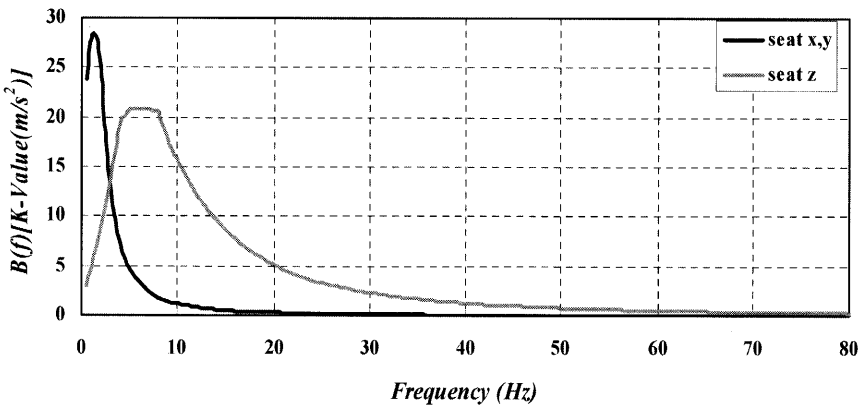


Fig. 10. Frequency weightings $B_i(f)$ for the seat in x-, y- and z- direction derived from the VDI 2057.

and KY are given as follow:

x-direction

$$1Hz \leq f \leq 2Hz: KX = 28 \frac{a_x}{[m/s^2]} [K-Value]$$

$$2Hz \leq f \leq 80Hz: KX = 56 \frac{a_x}{[m/s^2]} \frac{[Hz]}{f} [K-Value]$$

y-direction

$$1Hz \leq f \leq 2Hz: KY = 28 \frac{a_y}{[m/s^2]} [K-Value]$$

$$2Hz \leq f \leq 80Hz: KY = 56 \frac{a_y}{[m/s^2]} \frac{[Hz]}{f} [K-Value]$$

The frequency weighting $B_{seat,x}(f)$ and $B_{seat,y}(f)$ can be derived as

$$B_{seat,x}(f) = \frac{KZ}{a_z}$$

$$B_{seat,y}(f) = \frac{KZ}{a_z}$$

The relation to the frequency weighting $|W_d(f)|$, given in ISO 2631, for these two directions are defined as

$$KX = KY = 28 |W_d(f)|$$

The frequency weightings $B_{seat,x}(f)$ and $B_{seat,y}(f)$ are shown in Fig. 10.

Originally, the K-value was introduced by DIECKMANN (1955) based on physiological, subjective and physical studies. The vibration perception threshold corresponds to $K=0.1$ and, at the other extreme, the limit bearable for only 1 min was given by $K=100$. Up to now the German Society of Engineers (VDI) has modified the definition of the K-value several times. The relation between the perception quantity (K-value) and the subjective perception is shown in Table 2. Because the standard VDI 2057 also applies to vibrations in buildings, the subjective predictions given in Table 2 are not really suited for description of the comfort of vehicles. Thus only the numeric values for the K-value are used for comparing the comfort of the vehicles.

Table 2. Relation between the perception quantity and the subjective perception.

Perception quantity KX, KY, KZ	Description of the subjective perception
<0.1	Not perceptible
$0.1 \sim 0.4$	Just perceptible
$0.4 \sim 1.6$	Easily perceptible
$1.6 \sim 6.3$	Strongly perceptible
$6.3 \sim 100$	Very strongly perceptible
>100	Very strongly perceptible

2.2.2 ISO 2631

A general guide for defining the human tolerance to whole body vibration has been proposed in the International Standard, ISO 2631 (1978). This guide is recommended for the evaluation of vibration environments in transport vehicles as well as in the industry, and defines three distinct limits for whole-body vibration in the frequency range of 1–80 Hz:

(i) Exposure limits are related to the preservation of safety (or health) and should not be exceeded without special justification; (ii) Fatigue or decreased proficiency boundaries related to the preservation of working efficiency, which apply to such tasks as driving a road vehicle or a tractor; and (iii) Reduced comfort boundaries are concerned with the preservation of comfort.

Fig. 11 illustrate the fatigue or decreased proficiency boundaries for exposure to vertical and horizontal vibration, which are define in terms of root mean square values (rms) of acceleration as a function of frequency and exposure time. It can be seen that the

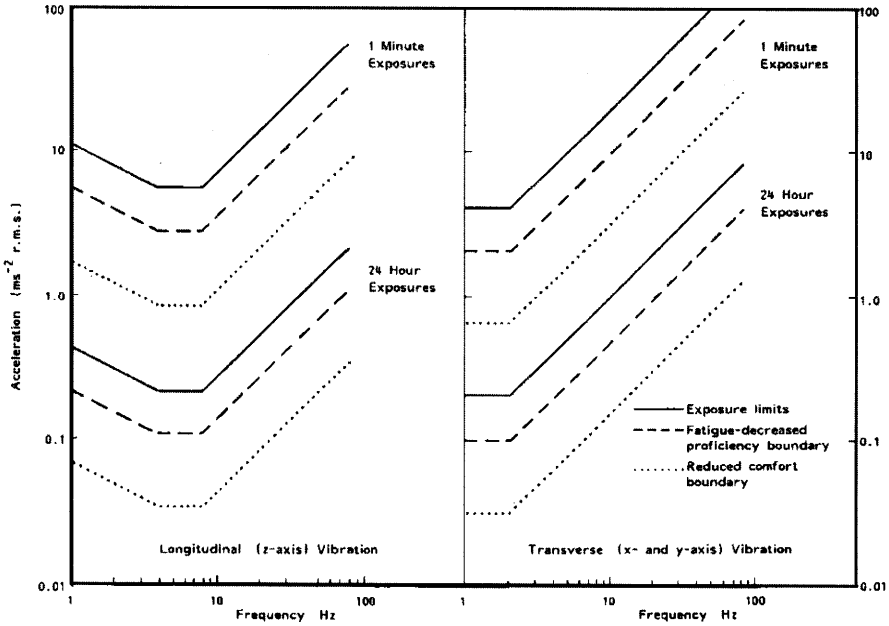


Fig. 11. The exposure limits, fatigue-decreased proficiency boundaries and reduced comfort boundaries for 1 min and 24 h exposures to whole-body vibration as given in ISO 2631.

exposure limits are reduces as the average daily exposure time increases. When vibration takes place in more than one direction simultaneously, corresponding boundaries apply to each vector component in the three axes. The exposure limits for safety (or health) reasons are obtained by raising the fatigue for decreased proficiency boundaries by a factor of two (6 dB higher), whereas the reduce comfort boundaries are obtained by lowering the boundaries by a factor of 3.15 (10 dB lower).

SMITH (1976) proposed a procedure to convert the ISO-2631 (1978) limits to a form usable for direct comparison with ride vibration data presented in power spectral density (PSD) form. The ISO-2631 (1978) ride assessment criterion is most widely accepted, and is applicable to whole-body vibration. However, this standard fails in quantifying ride criteria under rotational modes of vibration. ISO has recently proposed revised weighting factors and methodology in order to quantify the ride quality by a single number along the translational and rotational axes (ISO-2631, 1997). This method of assessing the ride is gaining popularity due to its simplicity. However, there have been many concerns associated with the measurement procedures, and vibration limits proposed in the revised ISO-2631. The methods can be applied for evaluating the exposure to vibration containing occasional high peaks (high crest factors, the ratio of peak value to the overall rms acceleration). For vibration with crest factors below or equal to 9, the basic evaluation method is normally considered adequate. For certain types of vibrations, especially those containing occasional shocks, the basic evaluation method may underestimate the severity with respect to discomfort even when the crest factor is not greater than 9. In case of doubt it is therefore recommended to use and report additional greater more than 9. The first method, referred to as the running rms method, takes into account occasional shocks and transient vibration by use of a short integration time constant. The vibration magnitude is

defined as a maximum transient vibration value (MTVV), $a_w(t_0)$ given by:

$$a_w(t_0)=\left\{\frac{1}{\tau}\int_{t_0-\tau}^{t_0}[a_w(t)]^2dt\right\}^{1/2}$$

where, $a_w(t)$ is the instantaneous frequency-weighted acceleration, t is the integration time for running averaging, t is the time and t_0 is the time of observation. The second method, referred to as the fourth power vibration dose method, is more sensitive to occasional peak vibration than the basic evaluation method by using the fourth power instead of the second power of the acceleration time history as the basis for averaging. The fourth power vibration dose value (VDV) in meters per second to the power 1.75 ($m/s^{1.75}$), or in radians per second to the power 1.75 ($rad/s^{1.75}$), is defined as:

$$VDV=\left\{\int_0^t[a_w(t)]^4dt\right\}^{1/4}$$

where T is the duration of measurement. The "fourth power" criterion has been recently accepted for inclusion in the new draft of ISO-2631 guide (1997).

2.2.3 ISO 5008

ISO 5008 defines a 100 m 'smoother track' and 35 m 'rougher track' that may be used to obtain standard measurements of the weighted vibration magnitudes on the seats of wheeled tractors. Fig. 12 shown the power spectral densities of the profiles defined in ISO 5008. Tracks having similar properties have been constructed from blocks of wood or concrete and are sometimes arranged in a circle so as to allow the recording of longer periods of vibration needed for some types of analysis. Tractors should be driven over the smoother track at 12 km/h and over the rougher track at 5 km/h. The standard is designed to encourage comparable measurements in the three translational axes (x, y, and z) of vibration on the seats of tractors. The procedure is used by the testing institutes of several countries and so provides a standardized comparison of the severity of vibration on different types and makes of agricultural vehicle.

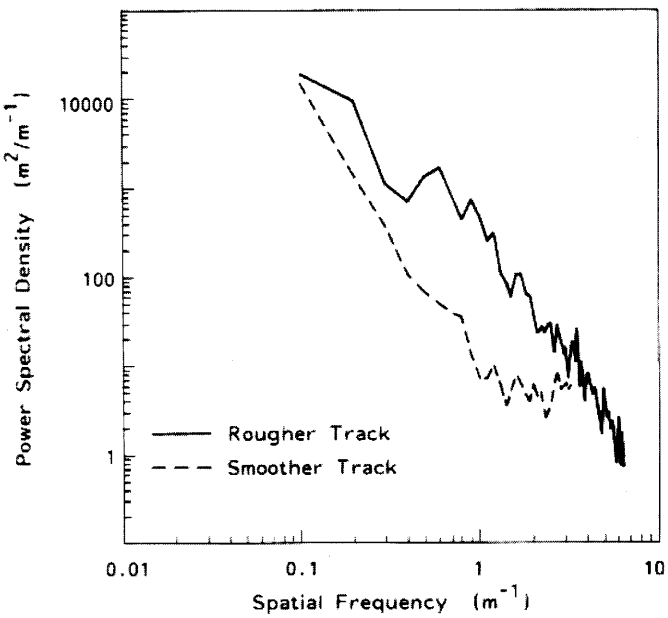


Fig. 12. Power spectral densities of the elevation of the 100m 'smoother' track and the 35 m 'rougher' track defined in ISO 5008(1979).

2.2.4 BS 6841

The British Standard BS 6841 was published in 1987 in response to what was perceived in Britain as the failure by ISO standard 2631 (1985) to appropriately address some of the major issues of human exposure to whole-body vibration. It appears to have encountered the general appreciation of users, with just a few shortcomings having been identified. Although it might certainly be improved, it may represent a good basis of discussion for the drafting of a future standard on this topic.

The BS 6841 recommends that measurements are performed on: three translational and rotational axes between the seat and the ischial tuberosities; three translational axes between the back and the seat backrest; three translational axes beneath the feet. As in the case of ISO 2631-1, assessment of vibration from the viewpoint of implied health risks is based on a limited subset including the three translational axes on the surface supporting the subject and the x-axis acceleration on the backrest (seated subjects only).

Six frequency curves are provided covering most combinations of vibration axes and effects (health, hand control, vision, comfort, perception, motion sickness) as shown in Fig. 13. Only three curves are used in the evaluation of vibration from the point of view of health effects. The most widely used one of these frequency weightings (W_b), which has been designed to mimic human sensitivity to vertical motions, is based mostly on subjective responses with some input provided by transmissibility data. It is unclear to what extent it provides a good representation of human susceptibility to injury.

The basic recommended method makes use of tri-axial rms. acceleration in bs 6841. The use of time-integrated fourth power of tri-axial acceleration (VDV) is supported when the crest factor is above 6. However, when evaluating and assessing vibration for health effects, VDV is the only option given. When only measurements of a_w are available, the vibration dose value may be estimated from a_w (in which case it is called eVDV).

An unique threshold limiting curve is provided, applying to all durations from less

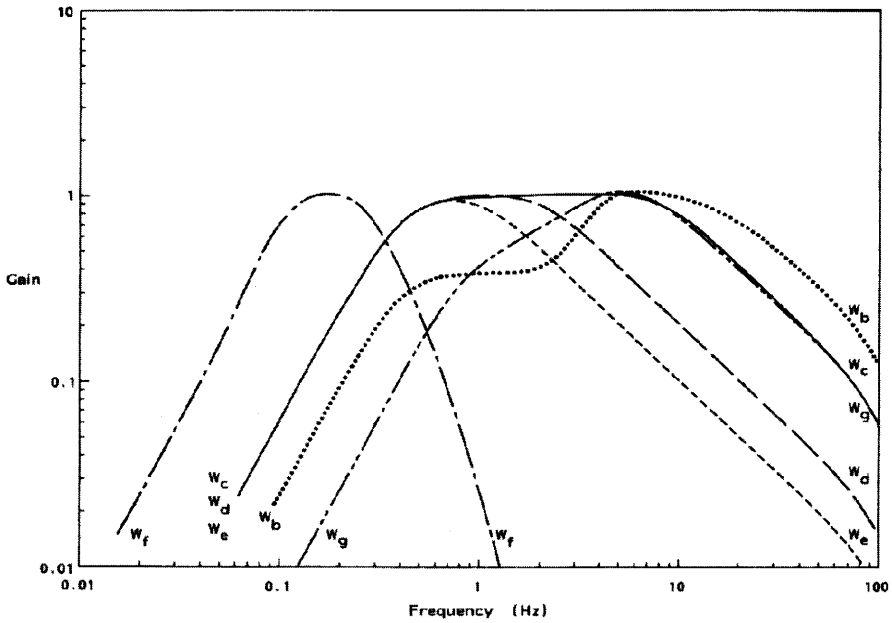


Fig. 13. Acceleration frequency weightings defined in BS 6841.

Table 3. Comparison of frequency weighting for ride vibration.

(Unit: dB)

Frequency (Hz)	Vertical direction		Horizontal direction	
	JIS	ISO	JIS	ISO
1	−6	−6	3	0
2	−3	−3	3	0
4	0	0	−3	−6
6.3	0	0	−7	−10
8	0	0	−9	−12
16	−6	−6	−15	−18
31.5	−12	−12	−21	−24
63	−18	−18	−27	−30
80	—	−20	—	−32
90	−21	—	−30	—

than a minute to 24 hours. The final result is therefore independent of the subdivision of the daily exposure.

2.2.5 Japanese Standard

Among the Japanese Standard, JIS C 1510 (Vibration level meters) and TR Z 0006 (Evaluation of human exposure to whole-body vibration) have often been used as ride vibration evaluation criterion. JIC C 1510 employed the frequency weightings defined in ISO 2631 (1974) and was expressed as follows

$$Acceleration\ level = 20 \log_{10} \frac{a}{a_0}.$$

Where a is the measured acceleration (in m/s^2 rms) and a_0 is a reference magnitude which varies according to the ISO 2631 frequency weightings. However, when it compares with ISO 2631, as shown in Table 3, frequency-weighting values of horizontal direction were greater 3dB than that of ISO 2631.

Also, proposed in 1976, Japanese tolerance limits fall into three categories according to the three vibration sources: factories, construction equipment and roads. For vibration from factories the limiting levels were in the range 60–65 dB for quiet residential areas and in the range 65–70 dB for commercial and industrial residential areas. The ranges were reduced by 5 dB at night. The proposed limiting levels for vibration from construction machinery were either 70 or 75 dB depending on the source and the period of use. Levels in excess of these limits were permissible but only for short periods. For road traffic, the limit was 65 db for quite residential areas and 70 dB for commercial areas and special measures were urged where vibration from the Shinkansen rail express caused vibration in excess of 70 dB. But, recently Japanese Standards Association has been used translated JIS (TR Z 0006, 2000) from ISO Standards (ISO 2631-1, 1997) for human response to vibration.

2.3 Alternative Methods

2.3.1 Absorbed power

The concept of absorbed power was first discussed in the mid-1960s by a number of researchers. The studies of PRADKO *et al.* (1965, 1966), LEE and PRADKO (1968) are largely responsible for the interest in absorbed power as an indicator of human response to whole-body vibration. They achieved results that subjective experience of vibration is related to the amount of vibration energy absorbed by the human body and absorbed power may be used as an indicator of subjective response.

The average absorbed power is determined from the intensity and frequency of the

input vibration as:

$$P=\sum_{i=1}^{n} K_i a_i^2$$

where P is averaged absorbed power, a_i is rms acceleration at a frequency 'i' in m/s^2, K_i is absorbed power constant of the body at frequency 'i', and n is the number of discrete by the scalar sum of absorbed powers associated with each of three translational axes.

Pradko and Lee suggested that the time dependency of fatigue or discomfort might be determined from the absorbed power. A decade later, JANEWAY (1975) strongly advocated these observations and pointed out that the concept of energy absorption should be added in the standard ISO 2631, at least for the occupational comfort criteria.

Since then, very little has appeared in published literature in this area. However, in regard to hand-arm vibration, the concept of absorbed power has been discussed more widely. For instance, LIDSTORM (1977) presented that the prevalence of vibration induced injuries within different occupational groups is related to the amount of absorbed energy.

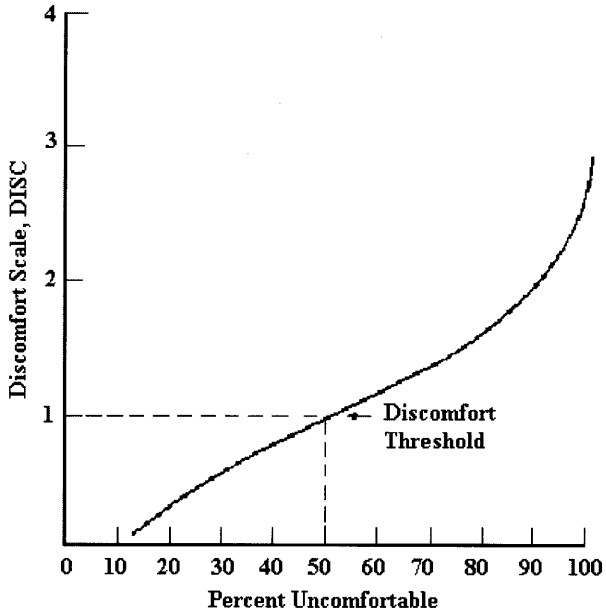
The absorbed power criterion has also been extensively used to assess military vehicle ride due to its simplicity. It provides a single number rating of the ride environment, which is function of the vibratory modes, intensities, frequency contents, body orientation, posture, etc. Average absorbed power in the range of 6–10 W, is considered acceptable for off-road vehicles, and has been extensively used to assess military vehicle ride (MURPHY and AHMAD, 1986).

In recent years, LUNDSTROM *et al.* (1998) have carried out the investigation of whole-body vibration energy absorption during different exponential conditions. The relation between absorbed power and frequency, exposure level, direction, upper-body position, body weight and gender were focused in the study. Their experimental results showed that: i) the absorbed power was strongly related to the frequency of the vibration, with peaks in the range of 4–6 Hz; ii) the amount of absorbed power increases with the frequency up to a peak in the range of 4 to 6 Hz during exposure at a constant acceleration level; iii) absorbed power at each frequency was proportional to the square of the acceleration to the square of the acceleration; and iv) maximal absorption tended to increase and slide towards lower frequencies when the body position was changed from erect to relaxed.

They conclude that absorbed power might be a better quantity for risk assessment than those specified in ISO 2631 (1997) since it also takes the dynamic force applied to the human body into account.

MANSFIELD *et al.* (2000) reported a laboratory study comparing subjective judgments of vibration and shock severity with the absorbed power and with all combinations of standard frequency weighting and analysis methods. In their experiment, 24 human subjects were exposed to 15 vertical vibration stimuli (5 stimulus types) compromising random vibration, repeated shocks and combinations of random vibration and shock at 0.5, 1.0, and 1.5 m/s^2 rms unweighted vibration. For assessment of all stimuli types together, absorbed power gave higher correlation with subjective discomfort than acceleration-based methods. Some o their conclusions are: 1) absorbed power correlates well to subjective discomfort; ii) absorbed power was significantly better than rms or MTVV methods at predicating discomfort; and iii) ISO 2631-1 requires clarification and amendment.

Evidently, the standard ISO 2631-1 only gives the guidelines of the acceleration magnitude on the vibrating surface. However, in many studies especially on hand-arm vibrations, the absorbed power is related to fatigue and injury. As a result, it is logical to follow a similar approach to describe the extend of vibration actually being transmitted to and absorbed by the human body. Energy dissipation is a scalar quantity that makes it easy to add up contributions from all three directions to a single value. Therefore, the



absorbed power in the human body will be a better indicator of fatigue and occupational health hazard. Such an approach can be extended to the driver comfort studies also.

2.3.2 Ride comfort index of NASA

The research on ride comfort has been conducted at the NASA Langley Research Center, with over 2000 test subjects (LEATHERWOOD *et al.* 1980). To avoid semantic confusion, NASA developed a ratio scale of discomfort, measured in DISC units (Fig. 14). 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.* (1980) 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.

This ride comfort index is not as widely recognized as ISO 2631, and lacks the benefit of having a readily interpretable exposure duration based output.

Ride Evaluation Criterion of Forestry Machine at Ride Limiting Condition

As mentioned in introduction, in order to evaluate designs of forestry machines with improved ride, we must consider forestry machines designs under common test condition and at the extreme ride conditions like ride limiting condition. Especially, since forestry machines often travel over rugged forestry terrain that includes rocks, logs, stumps, and abrupt transitions and when traveling fast over it, the operators experience harsh ride vibration. Therefore, it is important to test the ride quality at ride limiting conditions. Also these ride limiting conditions can provide the appropriate driving speed and strategies for comfortable ride.

Although the VDI 2057, ISO 2631, and BS 6841 have been used to assess vibration exposure, it is not desirable to use these standards for determining the ride limiting speeds

of the forestry machines. Because the VDI 2057, ISO 2631, and BS 6841 standards attempt to characterize the human sensation response to vibration their weighting curves and formulation have changed over time to accommodate criticisms. In contrast a biomechanical measure of vibration, like absorbed power, does not change over time and therefore it can be consistently and uniformly used to determine ride limiting speeds of forestry machines. The absorbed power criterion formulates the observation that off-road vehicle operators tend to drive such that they receive no more than 6-watt of absorbed power and does not directly model the operator or their control actions.

When evaluating ride vibration of forestry machines in the field, researchers encounter many confounding factors that make comparisons of machines. Researchers can control the machine related parameters like machine type, tire size, inflation pressure, and seat type. But researchers in the field cannot easily control or characterize the varied terrain conditions and differences between drivers. Thus controlled tests need to be performed like those described by MURPHY (1981).

MURPHY (1981) compares several tracked vehicles using measured and simulated absorbed power. For each of the test vehicles at a variety of test conditions, absorbed power versus machine speed, and the speed at the 6-watt level of absorbed power versus terrain roughness were determined. Thus, for a given set of conditions and using an absorbed power versus speed curve, the speed corresponding to the 6-watt level of absorbed power can be identified. Also using the speed at the 6-watt level of absorbed power versus terrain roughness curve, influence of terrain roughness on travel speed can be identified.

Forestry machines operate over a wide range of speeds and terrains. The absorbed power criterion offers a means to make comparisons between different machine designs under similar and extreme test condition.

Conclusions

This study is carried out to collect information regarding ride vibration and its effects on operator and to review briefly the several ride vibration standards and criteria having relevance to off-road vehicles such as forestry machines and to suggest recommendations for the effective use of such criteria in ride evaluation of forestry machines.

The review on the effects of ride vibration to human has shown serious evidence of ill health and impaired performance ability of driver. Especially, the frequency range of 2-8 Hz is most harmful to operating performance of driver. Therefore, the vibration level at the operator's seat needs to be attenuated within acceptable limits in this frequency range. The review on the ride vibration of forestry machines lead to a clearer definition of the problem of ride vibration in timber harvesting. Also we can know that operators may limit their speed due to ride limiting vibration.

Criteria of ride vibration were reviewed and recommended. Currently, although the useful criteria like VDI 2057, ISO 2631, and BS 6841 have been used to assess vibration exposure, little information is available on how to use them to determine ride limiting speed of off-road vehicle. However, since absorbed power criterion is a crude representation of the operators' vehicle control response to ride vibration, it would be more useful in evaluation of forestry machine designs. Thus, to apply the absorbed power criterion in design evaluations of forestry machines, more information are also needed about driving behavior of the forestry machine operators.

Summary

Ride vibration is a problem in forestry machines that can cause both injuries and discomfort to the operator. In order to evaluate forestry machine design with improved

ride characteristics, it is useful to determine ride limiting operating speeds. Currently, although useful criteria like VDI 2057, ISO 2631, and BS 6841 have been used to assess vibration exposure, little information is available on how to use them in determining the ride limiting speed of off-road vehicles like forestry machines. However, a criterion of a 6 -watt level of absorbed power would be more useful to determine the ride limiting speed because the absorbed power criterion is a crude representation of the operators' vehicle control response to ride vibration. Therefore, although there still are many deficiencies, the absorbed power criterion should be a pertinent criterion for the ride quality evaluation of forestry machines when used with other standards.

Key words: ride vibration, forestry machines, absorbed power, ergonomics

Literature Cited

AHO, K. and KÄTTÖ, J. (1971) Experiment for developing a method how to measure and evaluate the rocking of the forest tractor. VAKOLA. Study report **9**:1–41.

BUTKUNAS, A. A. (1966) Power spectral density and ride evaluation. SAE Transactions paper No. 660138, pp. 681–687.

British Standard 6841 (1987) Measurement and evaluation of human exposure to whole-body mechanical vibration and repeated shock. British Standards Institution.

BOSHUIZEN, H. C., BONGERS, P. M. and HULSHOF, C. T. J. (1992) Self-reported back pain in fork lift truck and freight container tractor drivers exposed to whole-body vibration. Spine **17**(1): 59–65.

BRÜEL and KJÆR (1988) Human vibration. B & K Booklet series. 23 pp.

CHRIST, W. and DUPUIS, H. (1966) Über die Beanspruchung de Wirbelsäule unter dem Einfluss sinusförmiger und stochastischer Schwingungen. Int. Z. Angew. Physiol. Arbeitsphysiol. **22**: 258.

DIECKMANN, D. (1957) Einfluss vertikeler mechanischer Schwingungen auf den Menschen. Internat. Z. angew. Physiol. einschl. Arbeitphysiol. Bd. 16, S.: 519–564.

FINE, R. (1964) Correlation of vertical acceleration and human comfort in a passenger car. SAE Journal, January, 1964.

GERKE, F. G. and Hoag, D. L. (1981) Tractor vibration at the operator's station. Trans. of the ASAE **24** (5): 1131–1134.

GOLDMAN, D. E. (1948) A review of subjective responses to vibratory motion of the human body in the frequency range 1 to 70 cps. Naval Med. Res. Inst., Natl. Naval Med. Ctr., Bethesda, Md., Proj. N. M.-004001, Rpt. 1.

GOLDMAN, D. E. and GIERKE, H. E. (1961) Shock and vibration handbook. Effects of shock and vibration on man. Vol. 3, sec. 44, McGraw-Hill Book Co.

GOLSSE, J. M. and HOPE, P. A. (1987) Analysis of whole-body vibration level during skidding. Technical report: No. TR-77. Quebec, Canada: Forest Engineering Research Institute of Canada (FERIC).

GRIFFIN, M. J., WHITHM, E. M. and PARSONS, K. C. (1982) Vibration and comfort-I. Translational seat vibration. Ergonomics **25**(7): 603–630.

GRIFFIN, M. J. (1990) Handbook of human vibration. Academic press Inc., San Diego, CA 92101.

HAARLAA, R. (1975) The effect of terrain on the output in the forest transportation of timber. Journal of Terramechanics **12**(2): 55–94.

HANSSON, J. E. and SUGGS, C. W. (1973) Lågfrekvens vibrationers effekt på reglagemanövrering (summary: The effect of seat vibration on vehicle operators' lever and pedal control capabilities). Instituionen för skogsteknik. Skogshögskolan. Rapporter och Uppsatser **63**: 1–22.

HANSSON, J. E., KLUSELL, L., PETTERSSN, B. and SVENSSON, A. (1973) Skogsmaskinen som arbetsplats (Summary: Ergonomic evaluation of logging machines). Forskningsstiftelsen Skogsarbeten. Redogörelse **3**: 1–52.

HANSSON, J. E. and WIKSTRÖM, B. O. (1981) Comparison of some technical methods for the evaluation of whole-body vibration. Ergonomics **24**(12) 953–963.

HANSSON, J. E. (1990) Ergonomics design of large forestry machines. International Journal of Industrial Ergonomic 5 (1990) 255–266.

HUANG, B. K. and SUGGS, C. W. (1967) Vibration studies of tractor operators. Trans. of the ASAE **10**(4): 478–482.

IMATOMI, Y. (1997) An ergonomic study on the geometrical design and density of tractor skidding-roads. Bulletin of the Forestry and Forest Products Research Institute No. 373, Ibaraki, Japan.

ISO 2631 (1974) Guide for evaluation of human exposure to whole body vibration. International Organization for standardization.

ISO 2631-1 (1997) Mechanical vibration and shock —Evaluation of human exposure to whole-body vibration—. International Organization for standardization.

ISO 5008 (1979) Agricultural wheeled tractors: Measurement of vibrations transmitted globally to the operator. international Organization for standardization.

International Organization for standardization (1990) Human response to vibration-measuring instrumentation (ISO 8041) Geneva, ISO.

JACKLIN, H. M. and LIDDELL, G. J. (1933) Ride comfort analysis. Engineering bulletin, Research Series No. 44, Purdue Univ.

JANEWAY, R. N. (1948) Vehicle vibration limits to fit the passenger. SAE passenger car and production meeting, Detroit, march 5. 48 pp.

JANEWAY, R. N. (1975) Human vibration tolerance criteria and applications to ride evaluation. SAE paper No. 750166. Warrendale, PA: Society of Automotive Engineers.

JIS C 1510 (1995) Vibration level meters. Japanese Standards Association.

LAIB, L. (1977) Measurement and mathematical analysis of agricultural terrain and road profiles. Journal of Terramechanics **14**(2): 83–97.

LEATHERWOOD, J., DEMPSEY, T. and CLEVENSON, S. (1980) A design tool for estimating passenger ride discomfort within complex ride environments. Human Factors **22**: 291–312.

LEE, R. A., and PRADKO, F. (1968) Analytical analysis of human vibration. SAE paper No. 680091. Warrendale, PA: Society of Automotive Engineers.

LEGAULT, R. and POWELL, L. H. (1975) Evaluation of FMC 200 BG grapple skidder. FERIC Technical Report No. 1.

LIDSTROM, I.-M. (1977) "Vibration injury in rock drillers, chiselers and grinders. Some views on the relationship between the quantity of energy absorbed and the risk of occurrence of vibration injury". *In*: Proceedings of the international Occupational Hand-Arm Vibration Conference, National Institute for Occupational Safety and Health, Cincinnati, OH, USA. DHEW (NIOSH) Publication No. 77–170, pp. 78–83.

LILJEDAHL, J. B., TURNQUIST, P. K. SMITH, D. W. and HOKI, M. (1996) Tractors and their power units 4th ed., The American Society of Agricultural Engineers.

LINES, J. A. (1987) Ride vibration of agricultural tractors: transfer functions between the ground and the tractor body. Journal of Agricultural Engineering Research **37**(2): 81–91.

LUNDSTROM, R., HOLMLUND, P. and LINDBERG, L. (1998) Absorption energy during vertical whole-body vibration exposure. Journal of Biomechanics **31**: 317–326.

MAGNUSSON, M. L. and POPE, M. H. (1998) A Review of the biomechanics and epidemiology of working postures (It isn't always vibration which is to blame). Journal of sound and vibration. **215**(4): 965–976.

MANSFIELD, N. J., HOLMLUND, P. and LUNSTROM, R. (2000) Comparison of subjective responses to vibration and shock with standard analysis methods and absorbed power. Journal of Sound and Vibration **230**(3): 477–491.

MARSILI, A., RAGNI, L. and VASSALINI, G. (1998) Vibration and Noise of a Tracked Forestry Vehicle. Journal of Agricultural Engineering Research **70**: 295–306.

MATTHEWS, J. (1964) Ride comfort for tractor operators. Part I. Review of existing information. Journal of Agricultural Engineering Research **9**(2): 3–31.

MOSELEY, M. and GRIFFIN, M. J. (1986) Effects of display vibration and whole-body vibration on visual performance. Ergonomics **29**: 977–983.

MURPHY JR., N. R. (1981) Armored Combat Vehicle Technology (ACVT) Program Mobility/Agility Findings. Proceedings of the Army Sciences Conference 15–18 Jun., 15 p, Army Engineer Waterways Experiment Station, Vicksburg, MS.

MURPHY JR., N. R. and AHMAD, F. H. (1986) Comparison of measures of vibration affecting occupants of military vehicles. Technical report GL-86-18, NTIS No. AD-A178359. Vicksburg, Mississippi: Department of the Army, Waterways Experiment Station, Corps of Engineers.

OH, J. H., PARK, B. J. ARUGA, K. NITAMI, T. CHA, D. S. and KOBAYASHI, H. (2002) A study on dynamic characteristis of forestry vehicle-vibration characteristics of a tracked mini-forwarder. Proceedings of Annual Meeting of Japanese Forest Society **113**: 738.

POPE, M. H. and HANSSON, T. H. (1992) Vibration of the spine and low back pain. Clinical Orthopaedics and Related Research **279**: 49–59.

PRADKO, F., ORR, T. R. and LEE, R. A. (1965) Human vibration anaysis. Paper 650426. Socirty of Automotive Engineers, Mid-year Meeting, Chicago.

PRADKO, F. and KEE, R. A. (1966) Vibration comfort criteria. Paper 660139. Society of Automotive Engineers.

REIHER, H. and MEISTER, F. J. (1931) Die empfindlichkeit des menschen gegen erschutterungen, Forschung auf dem gebiete des ingenieurswesens, Vol. 2: 381–386.

Richard A. LEE and Fred PRADKO (1968) Analytical analysis of human vibration. SAE Technical paper No. 680091.

ROSEGGER, R. and ROSEGGER, S. (1960) Health effect of tractor driving. Journal of Agricultural Engineering Research **5**(3): 241–250.

RUMMER, R. B. (1986) Modeling the ride dynamics of rubber-tired skidders. In Proc. 1986 Winter Meeting, American Society of Agricultural Engineers, Paper No. 86–1607. Chicago, IL, 16–19 Dec.

RUMMER, R. B. (1988) Whole-body vibration exposure of forest equipment operators in the southern United states. Doctoral dissertation. Auburn, Alabama, Department of industrial Engineering, Auburn University.

SAKAI, H., SAKAMOTO, K. and TAKEI, H. (1995) Actual speed of an articulated-frame wheel-skidder of full-tree skidding. Journal of the Japanese Forest Engineering Association. **10**(3): 233–242.

SANDERS, M. S. and MCCORMICK, E. J. (1992) Human factors in engineering and design, 6th ed. Mcgraw-Hill Book Co., New York.

SHISHIUCHI, M. (1972) Basic study on the vibration of the tractor used in forestry works (I) vibration of the crawler-type tractor. Journal of Japanese Forestry Society **54**(12): 399–407.

SHISHIUCHI, M. (1980) A fundamental Study of analysis of logging tractor performance. Bulletin of the Iwate University Forests **11**: 1–139.

SJØFLOT, L. (1970) Measuring and evaluating low frequency vibrations (0.3–110 Hz) acting on machine operators in agriculture and forestry. Landbruksteknisk Institutt. Forsøksmelding no. 19, 67 pp.

SMITH, C. C. (1976) On using the ISO standard to evaluate the ride quality of broad band vibration spectra in transportation vehicles. Transactions ASME, Journal of Dynamic Systems, Measurements, and Control, pp. 440–443.

STIKELEATHER, L. F. (1976) Review of the ride vibration of standards and tolerance criteria. Trans. of the ASAE **82**: 1460–1467.

SUGGS, C. W. (1973) Agricultural machinery noise and vibration level in comparison to human comfort and safety limits. ASAE Paper 73–524.

VAN DEUSEN, B. D. (1968) Human response to vehicle vibration. SAE technical paper No. 680090.

VDI 2057/1 (1987) Effect of mechanical vibrations on human beings; fundamentals, classification, terms. VDI.

VON ELDIK THIEME, H. C. A. (1961) Passenger riding comfort criteria and methods of analyzing ride and vibration data. SAE paper No. 295A.

WASSERMAN, D. E., DOYLE, T. E. and ASBURRY, W. C. (1978) Whole-body vibration exposure of workers during heavy equipment operation. NIOSH Technical Report.

WEBB, R. D. G. and HOPE, P. A. (1983) Ergonomics and skidder operations in northern Ontario: A preliminary investigation. Canadian Forestry Service Information Report DPC-X-15.

WEGSCHEID, E. (1994) Another look at skidder ride vibration. Journal of Forest Engineering **5**(2): 21–32.

WILSON, J. N., KIRK, T. G., LANG, F. G., TREMBLAY, M. L. and BURTON, R. T. (1986) Whole body vibration

in off road forestry vehicles. ASAE paper 86-1611.
WUOLIJOKI, E. (1981) Effects of simulated tractor vibration on the psycho-physiological and mechanical functions of the driver: comparison of some excitatory frequencies, Work Efficiency Assoc. No. 234.

(Received July 25, 2003)
(Accepted January 9, 2004)

林業機械に対する全身振動評価基準に関する考察

呉 宰憲[*2]・朴 範鎭[*1]・有賀一広[*1]・仁多見俊夫[*1]
車 斗松[*2]・小林洋司[*1]

[*1] 東京大学大学院農学生命科学研究科
[*2] 江原大学校山林科学大学

要 旨

現在，森林作業で車両系林業機械が多く使用されているが，走行時に発生する全身振動はオペレータに障害と不快感を与える原因として考えられている。この林業機械の全身振動問題を評価するためには，乗車限界作業速度を決定することが有用である。その評価法には，VDI 2057, ISO 2631, BS 6841 などの振動暴露の評価基準がよく使用されているが，不整地を走行する林業機械に関する研究は少ない。様々な基準について検討した結果，6W レベルの吸収動力基準が全身振動に対するオペレータの乗車感をそのまま表現することが出来るため，不整地を走行する林業用車両の乗車限界速度の決定に対して有用であると判断された。

キーワード：乗車振動，林業機械，吸収動力，人間工学

Abstract

Building Social Relations Centered on Citizens in Utilizing SATOYAMA —A Case of "ASAZA Project" in Kasumigaura Watershed—; Situations and its Significance

Taeko YOSHIMURA and Mitsuhiro MINOWA

This study aims to examine the social situation in Satoyama preservation focusing on citizens. It is based on a case study of the ASAZA Project. The investigation method involved interviews with the main participants and questionnaires to "ICHINICHI-KIKORI" volunteers. As the result, "Nonprofit Organization Asaza Fund" and "Kasumigaura Brushwood Fascine Association" organized other participating constituents. "ICHINICHI-KIKORI", which is one of the activities of the ASAZA Project, offered opportunities to various citizens for participating in the project. Those facts may suggest the possibility of expanding the project via citizens.

The Whole-body Vibration Evaluation Criteria of Forestry Machines

Jae-Heun OH, Bum-Jin PARK, Kazuhiro ARUGA, Toshio NITAMI, Du-Song CHA and Hiroshi KOBAYASHI

In most forest operations, forestry vehicles have been used. However, when forestry machines travel over rugged forest terrain containing rocks, logs, stumps and abrupt transitions, the operators experience harsh ride vibration. This whole-body vibration is a problem in forestry machines and can cause both injury and discomfort to the operators. In order to evaluate forestry machine design with improved ride characteristics, it is useful to determine ride limiting operating speeds. Currently, although useful criteria like VDI 2057, ISO 2631, and BS 6841 have been used to assess vibration exposure, little information is available on how to use them in determining the ride limiting speed of off-road vehicles like forestry machines. However, a criterion of a 6-watt level of absorbed power would be more useful to determine the ride limiting speed because the absorbed power criterion is a crude representation of the operators' vehicle control response to ride vibration.