Dynamic performance and ride comfort evaluation of the seat suspension system in a small agricultural tractor to attenuate low-frequency vibration transmission

H. P. W. Jayasuriya[1*], Kiattisak Sangpradit[2]

(1. *Dept. of Soils, Water and Agricultural Engineering, Sultan Qaboos University, PO Box 34, PC 123, Al-Khod, Oman*;
2. *RajaMangalaUniversity of Technology, Tanyaburi, Thailand*)

Abstract: To obtain safety and ride-comfort conditions for tractor operators, the tractor seat suspension system was analyzed and modified to attenuate the vibration transmission.   Experiments were conducted on a standard test track to investigate the effect of seat vibration on operators using a small 4-wheel tractor, and to develop a suitable seat suspension system following ISO guidelines.   Dynamic and ergonomic effects of vibration on operator using conventional seat and the new design were compared ensuing standard test procedures.   The supportive observations on ergonomic aspects were also made, viz. the operator heart rate, postural comfort survey, and Cornell ergonomic seating evaluation.   The vibration of existing seat in vertical direction had the highest amplitude followed by the lateral and longitudinal components sequentially while the new design could significantly reduce the vertical vibration and performed well within the vibration frequencies between 2 to 8 Hz with variable damping constants 1,177.7, 716.5, 695.5, 334.7 and 69.4 ns m^{-1}, and spring constant of 14,343 N m^{-1}.   The operator could be able to work satisfying 8 h fatigue-decreased proficiency limit, whereas the existing seat's proficiency limit was found slightly below 4 h.   Postural comfort and Cornell surveys showed significant improvements in the modified seat transferring it into comfortable zone.

Keywords: ride-comfort, ergonomics, suspension, postural-comfort, attenuation

1   Introduction

With the increase in the use of advanced machineries in agriculture, the mental and physical stress on the operator as well as the occupational hazards and diseases are found to be increasing, leading to impair the performance of operators (Huangand Suggs, 1961). Numerous options may be considered to improve the ride of an agricultural tractor, viz., suitable tires, primary suspension at the front and rear axles, cab suspension, and seat suspension.   Large diameter and soft tires

reduce the forces transmitted in the bounce and pitch modes but this option is infeasible due to limitation on the tire sizes (Deboli and Potescchi, 1986; Rakheja and Sankar, 1984).   Tractor seat design can be used as a means to modify loads on the body structures to reduce operator's discomfort.   The tractor seat system must also withstand the rigorous forces during different field operations and still maintain their dimensional stability, firmness and essential characteristic that enable the seat to retain its contoured shape (Mehta, 2000; Monarca et al., 2009).

Vibrations experienced by the operator especially at the tractor seat lie within the vulnerable range and the analysis of this ride vibration is a complex issue particularly for off-road conditions, as the vibration is in

category of multi-degree of freedom system and caused by many components, in the system (Braghin et al., 2008; Matthews, 1966; Yu, et al., 2008). The resonance frequencies of some human body parts and organs lie between 0 and 80 Hz range (Huang and Suggs, 1961), the most critical range being 0 to 8 Hz (abdominal mass, shoulder girdle 4-8 Hz; heart 4-6 Hz; entrails 3-7 Hz; spine 3-4 Hz; head 20-30 Hz; eye ball 30-80 Hz etc.). Operating speed, surface conditions, wheel air pressure, tire stiffness, mounting mechanism of the seat-cab or body parts could cause infinitely variable results. Suspension seats fitted to most tractors reduce the vertical component of vibration, but the levels are still undesirably high (Kumar et al., 2001; Marsili, et al., 2002) and there is little potential for further improvement using this technique.

The evaluation of ride comfort level is based on various standard criteria developed worldwide. Aswas described by Els (2005), Deboli et al., (2012), and in ISO 5008 (1979;2002), ISO 2631-1 (1985), ISO 2631-1 the second edition (1997), are mostly used in Europe; BS 6841 (1987), BSISO 5008 (2002) are mostly used in United Kingdom; Verein Deutcher Ingenieure, VDI 2057 (2002) is used in Germany and Austria; Average Absorbed Power (AAP) Pradko and Lee (1966) used in USA. Most of these standards cover the effect of low frequency range, generally below 80 Hz. In most parts ofAsia with small land holdings, demand for power tillers was saturated; whereas, the demand for small riding tractors, which are generally modified from power tillers, is increasing.

Due to the low field capacity and more custom hiring available, the operators used to work long hours under adverse conditions. Most tractor seats, especially the models sold in the developing countries, are mounted only on helical springs without any dampers to attenuate the vibration transmission (Braghin et al., 2008; Braghin et al., 2011; Marsili, et al., 2002; Yu et al., 2008). Moreover, these small riding tractors are still lack of studies on their vibration effect; in general they are not tested on the standard test tracks. Therefore, it is necessary to assess these typical conditions for safety and comfort of small tractor operators following international standards. This study investigates the feasibility, and magnitude of seat vibration attenuation by incorporating the damper-effect in the system and, the transmission characteristics leading to develop a suitable low-cost seat suspension system following ISO guidelines, and ergonomic considerations.

2   Methodology

Experiments on the measurement of vibration level was conducted using the standard test track built as per the ISO 5008 (1979) standardas is shown in Figure 1. The main objective of this study was to measure and analyze the factors affecting the physical workload and postural discomfort of the operator from ergonomic point of view, the results then used to design the new seat suspension system followed by performance evaluation tests on ergonomic factors and vibration transmission characteristics.

Figure 1   Standard 35 m test track constructed for experiment in this study following ISO 5008 (1979) guidelines for low frequency vibrations

Kubota Tractor model L2050 (3 cylinder, 4-WD, 25 hp, diesel engine, mass 950 kg) and wheels having recommended air-pressure (30 psi at front tires and 16 psi at rear tires) was used (Matthews, 1964). Vibration measurements were obtained using the vibration analyzer through the installed accelerometers (in Figure2c for vertical direction) at two positions, i.e., seat and seat base of the tractor, to measure vibration in three different directions, i.e., longitudinal (pitching), lateral (rolling) and vertical (bouncing).Instruments included, RION Vibration Meter VA-12 With FFT analysis function, three Kyowa PV55 piezo-electric accelerometers, frequency

range of 1 to 80 Hz and sensitive to vibration level of 0.05 m s^{-2} and capable of measuring vibrations is 5 m s^{-2} RMS with a crest factor as great as threewithout distortion and with an accuracy of ±0.05 m s^{-2}. Data could be downloaded to the desktop computer installed in the data acquisition system for analysis.Three operators were used having body masses 68 (Operator-1), 89 (Operator-2) and 57 (Operator-3) kg covering the typical range 55-98 kg, to drive the tractor on standard test track in order to optimize parameters such as damping constants and spring constant of suspension system.

2.1 Design of new suspension system

According to the standard code ISO 2631 (ISO, 1985), predominating frequencies can be recognized which correspond to excessive vibration and these frequencies are used for designing the seat suspension system.

Design of seat suspension system was consisted of coil springs and pneumatic damper to achieve the fundamental frequencies closest to the predominant frequencies observed to reduce the vibration in frequency range that exceed the standard limit of 8 h or 4 h operating threshold limit. The design steps were: (1) to study the seat suspension mechanisms proposed by Lines (2000) by comparing the advantage and disadvantage of each mechanism.   (2) Select the most appropriate mechanism that fits into the existing tractor.   (3) Compute the dimension of spring and damper to achieve the target range of spring constant, dumping coefficient. The suspension system was constructed according to the design shown in the Figure 2 and consisted of two coil springs, adjustable pneumatic damper to vary the damping characteristics in different experimental conditions.

a. Adjustable pneumatic damper and springs   b. New seat after assembly   c. Accelerometer on seat base

Figure 2   Construction of new seat suspension system

2.2 Performance evaluation on vibration characteristics

Performance tests were conducted for old and new seats by obtaining data in terms of frequency and amplitude on rough standard test track by portable *Fast Fourier Transformation* (FFT VA-11) analyzer for measuring the vibration that meets the standard requirements.   The analysis for the exiting seat was done with two operator samples (Operator-1, 68 kg and Operator-2, 89 kg) for the frequency component smoothened by 1/3 octave band weighted average method (Figure 3), and comparing the transmission ratio of all frequency and amplitudes with ISO 2631 frequency-amplitude standard curve selecting the point where

frequency-amplitude characteristicsexceed the standard curve.   This design point was then used to calculate the spring constant (k) and damping coefficient (c).

The analysisabove was repeated for the newly designed seat to make sure the seat suspension system possesses the required characteristics, frequencies correspond to 4 or 8 hours' threshold limits as appropriate. Test results were analyzed with the standard requirements and the resulting frequency-amplitude characteristics were compared with ISO 2631 standard characteristics. Further adjustments were made appropriately for spring and damping constants until the test results meet the satisfactory range.
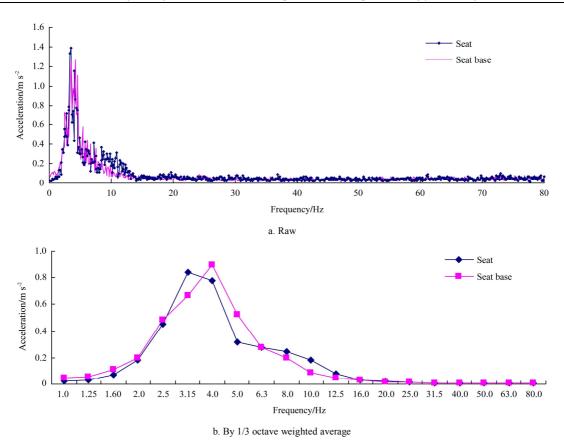
a. Raw

b. By 1/3 octave weighted average

Figure 3   Frequency-accelerations characteristics of the existing seat and seat-base along vertical direction (bouncing mode)

with Operator-1 of 68 kg

2.3   Performance evaluation on ergonomic aspects

In present research, the driving speed used for rough standard test track was 5 km h^{-1} as per the ISO 5008 (1979) and BSISO (2002) standards.   The mass of each operator, which is likely to affect the vibration level (ISO 5008, 1979; BSISO, 2002) was kept between the range of 55 to 98 kg considering typical Asian conditions. Operators had body masses 68 kg (Operator-1), 89 kg (Operator-2) and 57 kg (Operator-3) respectively. Before the tests were conducted, operator's basic physicals such as heart rate (Astrand and Rodahl, 1986; Nielsen and Meyer, 1987), weight, height etc.   Other comfort information were obtained through two types of questionnaires; Cornel Ergonomic Seating Evaluation and Postural Comfort Survey (Corlett and Bishop, 1976) to assess the improvement in newly designed seat.

3   Results and discussion

3.1   Results obtained for the existing seat

The purpose of measuring the vibration of existing seat was to determine the predominant frequencies that correspond to excessive vibration.   These frequencies were then used for designing the seat suspension system. The vibration was measured along three directions i.e. longitudinal, lateral and vertical, which corresponds to vibration modes of pitching, rolling and bouncing respectively.   Two operators were tested in this experiment, with the weight of 68 kg (Operator-1) and 89 kg (Operator-2), respectively.

3.2   Vibration on existing seat for operator-1

At low frequency, the existing seat was able to attenuate the vibration transmission, particularly in longitudinal direction and just above 4 h limit in lateral direction.   The vibration in vertical direction at 4 Hz gave the highest amplitude; where seat had lower amplitude than the seat base, which concealed that the existing seat could reduce vibration tosome extent.   In longitudinal direction, vibration of the existing seat was lower than 8 h working period of ISO acceptable limit (Figure 4a) while in lateral direction (Figure 4b), vibration of the existing seat satisfied 4 h working limit. The vibration in vertical direction was higher than 4 h working limit (Figure 4c).   Thus the operators can only work for not more than fourworking hours with the

existing seat as per the ISO2631 (ISO2631-1, 1997 and ISO2631-2, 2003) comfort limit.   Seat had excess vibration along its vertical and lateral directions requiring damping.   At 4 Hz, the vibration in longitudinal and lateral directions had transmissibility more than one, while vibration in vertical direction had transmissibility less than one.   Therefore, the vibration in longitudinal and lateral directions increased the amplitude, whereas the vibration in vertical direction reduced the amplitude.

3.3   Vibration on existing seat for operator-2

Similar trends on vibrations were observed for Operator-2, though they differed in magnitude.   The vibration results of operators -1 and -2 were the same

along longitudinal direction of the existing seat and lower than 8-h ISO acceptable line (Figure 4a).   While in lateral direction (Figure 4b), vibration of the existing seat was about 4 h limit.   Vibration in vertical direction was higher than 4-h limit (Figure 4c).   It is obvious from observations above that using the existing seat, the operators should work less than fourworking hours in order to meet the ISO2631 (ISO2631-1, 1997 and ISO2631-2, 2003) standards.   These circumstances prompted corrections necessary for attenuating the transmission along vertical direction probably by the use of dampers in the suspension system.

a. Longitudinal direction (pitching

b. Transverse direction (rolling)
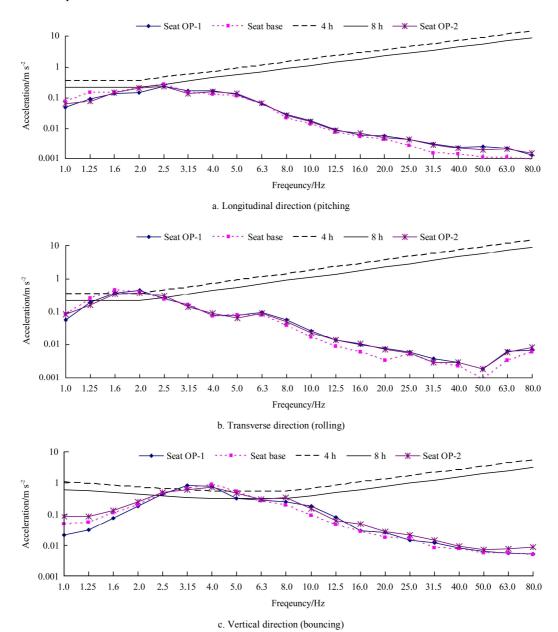

c. Vertical direction (bouncing)

Figure 4   Comparison of seat and seat-base vibration characteristics with ISO2631 acceptable limits of 4 h and 8 h for the Operator-1 (68 kg) and Operator-2 (89 kg)

3.4   Optimumspring constant and damping constant

The seat tested for both Operators -1 and -2 had the highest peak at 4 Hz in vertical direction, therefore, the new seat suspension design point was fixed at 4 Hz and spring constant was calculated by Equation (1):

$$k \le 4\pi^2 f^2 m \qquad (1)$$

where, k = spring constant (N m^{-1}); f = frequency (Hz).

At f = 4 Hz, m = 68 kg, the spring constant was 42,952 N m^{-1} and at f = 4 Hz, m = 89 kg the corresponding spring constant was 56,217 N m^{-1}. The estimated spring constant was then used to select the helical springs from the market.

Matthews (1966) stated that the best transmissibility ratio on heavier drivers should be 0.6. Thus, transmissibility ratio of (F_r) = 0.6 was used to estimate the damping constant using Equation (2):

$$F_r = \left[\frac{k^2 + \omega^2 c^2}{(k - m\omega^2)^2 + \omega^2 c^2} \right]^{\frac{1}{2}} \qquad (2)$$

where, F_r = transmissibility ratio; k = spring constant (N m^{-1}); ω = angular velocity (rad s); c = damping constant (ns m^{-1}); m = mass (kg).

Damping constant was calculated based on operator weights at f = 4 Hz, first for lower side, 68 kg, the damping constant was 340 ns m^{-1} and other for upper side, 89 kg, the damping constant was 1,177 ns m^{-1}. Damper was selected from the available adjustable pneumatic dampers in the market, according to the calculated range.

The transmissibility ratio of this design found to be less than 1 for both operators, which mean that the designed system could effectively reduce the vibration. The best damping constants that could reduce the vibration up to 30% for operator-1 and up to 52% for operator-2 was 69.4 ns m^{-1}.

3.5   Comparison of seats for fatigue-decreased proficiency limits as per ISO 2631

The newly designed seat was tested with different damping constant, i.e., at 69.4, 334.7, 695.5, 716.5, and 1,177.7 ns m^{-1}, whereas the spring constant was fixed at 14,343 N m^{-1}. The amplitude of the modified seat at every damping constant decreased drastically. For operator-1, Figure 5a shows the reduced vibration characteristics of the newly designed seat suspension

system comparing with the ISO2631 (ISO2631-1, 1997 and ISO2631-2, 2003) acceptable levels of eightand four working-hour threshold limits for damping constants at 1,177.7 ns m^{-1}. Acceleration showed peak at 4 Hz for both the seats, however, modified seat showed the improvement with working limits. As the damping constant at 1,177.7 ns m^{-1} could give the highest acceleration, for every damping constant the newly designed seat suspension system should support the operator to go beyond 8 h limit.
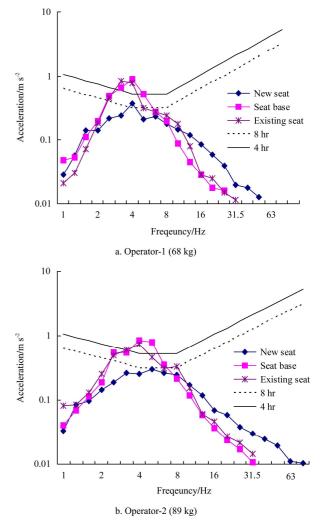

a. Operator-1 (68 kg)

b. Operator-2 (89 kg)

Figure 5   Comparison of two seat suspension systems with damping constants at 1,177.7 ns m^{-1}
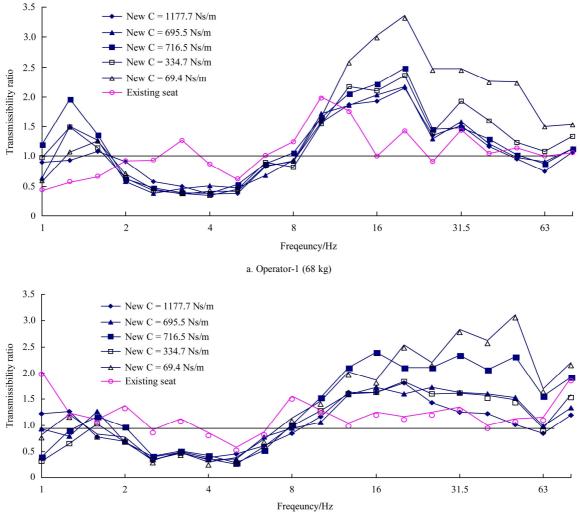
The results for operator-2 showed similar trends (Figure 5b). For the designed seat, the damping constant at 69.4 ns m^{-1} was the best point in reducing the vibration. Figure 5b shows that the reduced vibration of the newly designed seat suspension system comparing with the ISO2631 (ISO2631-1, 1997 and ISO2631-2, 2003) acceptable levels at eight working hours and four

working hours for damping constants at 1,177.7 ns m^{-1}. Compared with operator-1, at every damping constant the newly designed seat supports the operator-2 to be able to work for 8 h working range. Same as operator-1, at frequency 4 Hz showed the highest amplitude.

3.6   Comparison of seats for transmissibility characteristics

For the operator-1, the modified seat gave the lowest transmissibility ratios between 2 to 8 Hz (Figure 6a) and the damping constant of 334.7 ns m^{-1} was the best point in attenuating vibration.   At highest amplitude, the existing seat had transmissibility ratio of 0.86, while the newly designed seat with damping constants of 69.4, 334.7, 695.5, 716.5, and 1,177.7 ns m^{-1} had lowest transmissibility ratios of 0.41, 0.34, 0.51, 0.51, and 0.36, respectively.   Thus designed seat at any of those five damping constants could considerably reduce the vibration.

Operator-2 data showed similar trend as the operator-1; at highest amplitude, the existing seat showed lowest transmissibility ratio of 0.86.   While the modified seat with damping constants of 69.4, 334.7, 695.5, 716.5, and 1,177.7 ns m^{-1} showed lowest transmissibility ratios of 0.30, 0.35, 0.35, 0.42, and 0.39, respectively (Figure 6b).   This specifies that the modified seat at any of those five damping constants could considerably reduce the vibration for both operators.   For the existing seat, transmissibility was more than 1 between 2.5 and 3.5 Hz for both the operators.   The existing seat could only attenuate the vibration from the seat base by 12%, while the modified seat had the best transmissibility value of 65% at damping constant of 334.7 ns m^{-1} and spring constant 14,343 N m^{-1}.   The newly designed seat was able to attenuate the vibration 53% more than that of existing seat.
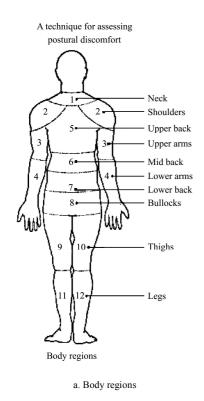
a. Operator-1 (68 kg)

b. Operator-2 (89 kg)

Figure 6   Comparison of transmissibility of two seat suspension systems; new suspension system with 5 damping constants

3.7 Comparison of seat vibration based on heart rate

Three operators were tested for heart rate measurement at three conditions, i.e., during rest, driving tractor on smooth surface and on standard test track. Measurements taken during the tests are given in Table 1. Heart rates were not significantly different when evaluated by F-test during the operator's on rest and when driving the tractor on smooth surface. However, the heart rates were significantly higher for existing seat than the modified one when operators were driving the tractor on the standard test track.

Table 1   Heart rate of three operators measured at three conditions i.e. during rest, driving tractor on smooth surface and on standard test track

Existing seat				
Test condition	Air temperature /⁰C	Heart rate (beat per min)		
		Operator-1	Operator-2	Operator-3
Rest	33.2-34	90	96	68
On smooth surface	33.3-34	91	98	70
On standard test track	33.5-34	106	111	93
Modified seat				
Test condition	Air temperature /⁰C	Heart rate (beat per min)		
		Operator-1	Operator-2	Operator-3
Rest	33.5-34	83	92	67
On smooth surface	33.8-34	84	95	71
On standard test track	33.8-34.5	95	97	73

3.8 Postural comfort survey and cornell ergonomic seating evaluation

Postural comfort survey (Corlett and Bishop, 1976) was conducted on three operator samples as per the guidelines, viz. taking response on fatigue at various locations of the body as indicated in Figure 7a. Figure 7b shows the combined scores from three operators and the average in 1-10 scale indicating the comfort level, 1 being very comfortable to 10 being very uncomfortable (Figure 7c). Overall comfort score of the existing seat 5.05 out of 10 was under 'uncomfortable zone' where as the new seat, 3.13 out of 10 was almost near 'very comfortable zone' as shown in the Figures 7b and 7c. From the comfort score, the new seat reported to be comfortable for most of the operators' abdomen, shoulder, hip, low back and upper back. At the mid-back, modified seat showed reduced comfort due to the armrest pressing the waist in rolling mode which could transfer pain to the mid-back. The seat cushion and backrest have two important functions, which have been researched to only a limited extent (Donati et al., 1984); the effect of the shape in providing postural support, and the effect as part of the overall suspension system.

A technique for assessing postural discomfort

Neck — 1
Shoulders — 2
Upper back — 5
Upper arms — 3
Mid back — 6
Lower arms — 4
Lower back — 7
Bullocks — 8
Thighs — 9 10
Legs — 11 12

Body regions

a. Body regions

Area of body	Existing seat	New seat
Neck	15	8
Chest	14	9
Abdomen	22	10
Upper back	21	12
Mid back	16	17
Low back	23	13
Shoulder	22	11
Upper arm	14	10
Elbow	14	8
Forearm	12	10
Wrist	15	8
Hand	11	8
Finger	10	8
Hip	22	9
Thigh	13	9
Knee	11	9
Lower leg	11	8
Ankle	11	10
Foot	11	8
Total (570)	288	185
Scale 1 - 10	5.05	3.15

b. Combined postural comfort survey score by three operators; 57, 68 and 89 kg

Very comfortable	1
	2
	3
Some comfortable	4
	5
Uncomfortable	6
	7
Very uncomfortable	8
	9
Intolerable	10

c. 1-10 scale indicating the comfort levels

Figure 7   Results of the postural comfort survey

The modified seat was affirmed more comfortable than the existing seat according to Cornell Seating Evaluation criterion.   In the scale 0-10, corresponding to unacceptable to excellent, score of the existing seat was very low in chair adjustment (Table 2) and ease-of-use because the existing seat was unable to be adjusted.   The overall operator scores for the existing seat were 1.27, 1.53 and 1.33 while the modified seat scored 7.27, 8.07 and 7.87, respectively by the operators 1, 2 and 3.   The ride comfort of the operator is important in the long run as the terrain induced vibrations have the potential to cause discomfort to the operator, and even cause permanent physical damage (Gerke and Hoag, 1981).

Table 2   Comparison of seats as per the Cornell ergonomic seating evaluation criteria

	Existing seat (out of 30)			Modified seat (out of 30)		
	Operator 1	Operator 2	Operator 3	Operator 1	Operator 2	Operator 3
Chair adjustment score	0	0	0	20	23	21
Comfort score	5	5	6	23	23	22
Ease-of-use score	0	0	0	21	23	26
Body support score	6	7	6	20	25	21
Overall chair experience score	8	11	8	25	27	28
Total out of 150	19	23	20	109	121	118
For scale 0 - 10	1.27	1.53	1.33	7.27	8.07	7.87

Note: Scale: Unacceptable (0) --- Average (5) --- Excellent (10).

4   Conclusions

The vibration of existing seat in vertical direction showed highest amplitude followed by the vibration in lateral and longitudinal directions sequentially.   The operator's fatigue-decreased proficiency limit for the existing seat was near 4 h level of ISO 2631(ISO2631-1, 1997 and ISO2631-2, 2003) standard.   The modified seat showed excellent results within the frequency range 2 to 8 Hz, which is found to be the critical low-frequency vibration range of typical tractor vibration modes.   The newly designed seat with damping constant settings (c) 1,177.7, 716.5, 695.5, 334.7, 69.4 ns m^{-1} and with spring constant (k) 14,343 N m^{-1} facilitated the operator to pass the 8 h fatigue-decreased proficiency limit of ISO 2631 standard.

The comparison between the modified seat and the existing seat revealed that the new seat could attenuate the vibration transmission significantly than the existing seat along the vertical direction.   Heart rates of three operators were not much different while driving the tractor on smooth surface.   However, the heart rate increased significantly when they were driving the tractor on the standard test track.   From the postural comfort survey, the new seat comforted most on the operators' abdomen, shoulder, hip, low back and upper back.   At the mid-back, new seat decreased the comfort because the armrest pressing the waist in rolling mode transferring pain to the mid-back.   The Cornell ergonomic seating evaluation results showed that the modified seat is more comfortable than the existing one.

No survey records or health statistics could be found on the after-effects of long-term operation of such tractors in the region.   As the cost of this modification is found to be negligible (cost of the developed seat US$100) compared to the cost of the 4-wheel tractor, the regional tractor manufacturers should consider the implementation of this safety standard.   The experimental results have limitations with the number of operators used in the set of experiments.   The experiments were conducted with three operators having body masses distributed within the ISO2631 (ISO2631-1, 1997 and ISO2631-2, 2003) recommended range.   The aim was to research about the vertical (bouncing) mode of vibration within the recommended range.   Replications at each specific level could not be accommodated.

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