Translational vibration evaluation of tractor seats for ride comfort

Yatih Nupur[1*], Tewari V. K.[2], Thangamalar R.[3], Sweeti Kumari[4], Ashok Kumar[4]

(1. *Department of Agricultural and Food Engineering, IIT Kharagpur, India;*
2. *Department of Agricultural and Food Engineering, IIT Kharagpur, India;*
3. *TAFE R&D, Chennai, India;*
4. *Department of Agricultural and Food Engineering, IIT Kharagpur, India*)

Abstract: An investigation was carried out on tractor seat design for better operator comfort during operation.   The project objectives were mainly aimed to increase the ride comfort by improving the seat suspension which is used in Indian tractors. Tests were conducted for haulage operation and also for field operation.   During haulage operation tests were carried out on existing seats used in Indian tractors with three new selected suspension systems of the two most selling models of MF 241 and MF 1035.   The model MF 241 equipped with a existing seat is mostly used in India, while the model MF 1035 consists of exported benchmark seat (for comparison) which is a seat whose suspension appeared to majority of operators is used for more comfortable and rest three selected seat suspensions were replaced to same existing seat used in India.   Tests were conducted with three different operators representing 5th, 50th and 95th percentile (SM$_1$, SM$_2$ and SM$_3$) of the population of India.   Best suspension system was selected, based on weighted RMS acceleration a_{wx}, a_{wy}, a_{wz} for each axis and the acceleration vector sum a_v performance.   The values of weighted RMS acceleration a_{wx}, a_{wy}, a_{wz}, and acceleration vector sum a_v of different suspension systems were compared with   exported benchmark seat as well as existing seat mostly used in India.   The changes in the vibration peak along with speed were compared graphically.   It was observed that the SP$_2$ suspension gives better result among three selected suspensions.   The results were comparable with exported benchmark seat in higher speed range which was available in developed countries.

Keywords: seat suspension, weighted R.M.S acceleration, spring constant

1   Introduction

Tractor operators are subjected to two general types of vibrations.   These are the relatively low frequency vibrations which are induced by the tires and terrain, and the higher frequency vibration which originates in the various machine components such as engine, gear train and accessories Dewangan (2007).   Since the higher frequencies are relatively easy to attenuate, the work in vibration control is directed towards the problems of terrain induced vibrations.   This frequency has its major

frequency components is in the low frequency range between approximately 1 to 10 Hz.

The operator seat is a very common component on self propelled agricultural machines.   Yet, in an engineering sense, the operator seat is an uncommon component in that it serves as the interface between the mechanical system, the machine, and the delicate and sensitive human biological system.   An operator seated on a mobile off-road machine, such as an agricultural tractor, is a bio-mechanical component in an otherwise mechanical system which is vibrating in response to terrain roughness.

Human factors considerations are important in designing and controlling the vibration exposure to provide the comfort level expected by the operator and to

protect against deleterious health and safety effects. An engineer responsible for vehicle ride characteristics and seat comfort must be take into consideration the dynamic response characteristics of the vehicle, the response of the seat which interfaces the operator to the vehicle chassis, and the biomechanical and physiological responses of operator. Like other components affecting the comfort, safety, convenience, and productivity, seats influence customer buying decisions. U.S. farmers spend an estimated total of more than two billion h per evaluating operator seats and other human factors features on their equipment (Mehta and Tewari, 2000). These evaluations affect buying decisions in the market place.

It was found that tractor drivers suffer from a number of disorders of the spine and supporting structures at low frequency and high amplitude vibrations that are encountered in their field and road activities (Dewangan, 2007). A survey conducted by orthopaedic surgeons in the United States showed that truck and tractor drivers suffer from a number of disorders of the spine and supporting structures. Infect high incidences of osteoarthritis. Traumatic fibrositis, herniated disks, coccygodynia, lumbosacral pain, abdominal pain, and intestinal disorders occur in drivers of tractors. Tewari and Mehta (2000) had reviewed about seating discomfort for tractor operators. A large number of studies have been conducted and researchers have suggested several criteria for evaluating discomfort and the suitability of a tractor seat in a given working condition.

Matthews (1964) investigated the effect of acceleration levels in the range of 0-0.35 g; 2.5 and 3.5 Hz on compensatory tracking ability, ability to maintain constant foot pressure and deterioration of visual acuity.

Farm holdings are shrinking continuously and farmers in India prefer to have lower hp range tractors. Tractors in 30-45 hp range are in most demand in India. During collecting the voice of operators, it was noticed that operators had dis-comfort due to vibration of the seat. The tractors designed and used in developed countries either have suspension systems or are provided with suspended cabs to reduce ride vibration levels. Therefore, the operators are more comfortable in the tractor seats of those tractors (Tewari and Mehta, 2000).

However, the tractors used in developing countries, such as India, are unsuspended and are without cabs due to cost considerations. The tractor seat appears to be the simplest, the most economic and the best location for isolating vibration and making the tractor ride comfortable. Consequently, suspension and cushioning of seats appear to be the most prominent area of developmental activity in the tractor seat design.

The primary functions of a tractor seat include the demand to locate the driver in a position from where he can operate the tractor safely with visions all around and experience reduced noise and vibration levels. The operator must also feel comfortable on the cushion material of the seat. Unfortunately, comfort is the more subjective in definition and there are no universal criteria or methods that can be used to measure comfort. The seat cushioning material plays a dominant role in supporting operator posture, isolating vibration and improving ride quality. Generally, the cushion material in a tractor seat should be neither too hard nor too soft. A hard cushion causes high pressure points directly underneath the ischial tuberosities (bony prominence at the bottom of the pelvis) and a soft cushion causes the driver to &sink in' which results in the femurs being pushed up and increases pressure in the hip joints (Dewangan, 2007). Consequently, dimensional instability with fatigue (*i.e.* thickness loss) should be minimized to avoid geometric changes in the seat.

Considering the above problems a research was proposed with the objectives to measure the translational vibration and to examine the present vertical dampening mechanism effectiveness and to compare the developed seat suspension system with the existing exported bench mark seat which has proven of operator's comfortable ride.

2   Types of tractor vibration

Vibrations primarily are of two types, i.e. sinusoidal and random. Sinusoidal vibration is regular in nature and predictable where as random vibration is irregular and unpredictable (Mehta 2002). The mechanical vibrations that affect the human body can be grouped into two directional vibrations i.e. rectilinear and rotational (or

angular).   The rectilinear vibration (vertical, longitudinal, Transverse) transmitted to the human body, according to appropriate directions in an orthogonal co-ordinate system, the origin of which coincides with the location of the heart (Figure 1).   Accordingly, three components of vibration a_x, a_y and a_z are considered.   Acceleration 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).
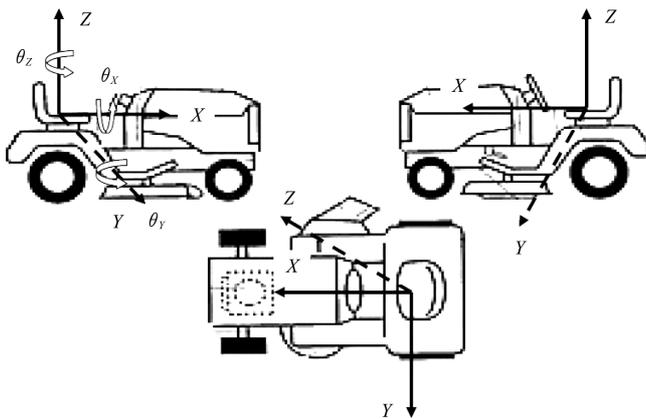
Rectilinear vibrations: X axis – longitudinal; Y axis – transverse; Z axis – vertical.
Rotaional vibrations: θ_X – roll; θ_Y – pitch; θ_Z – yaw

Figure 1   Directions of basic entric coordinate system for mechanical vibrations influencing thehuman body

Some studies of Janeway (1975) shows that the prominent vibration motion of a wheel tractor is vertical and that the seated operator is most sensitive to vertical vibration.   The effect of longitudinal and transverse mode depends on the type of task being performed by the tractor (Stayner et al., 1984).   The angular vibration effects the human body can be grouped into three directional vibration i.e. Yaw, pitch and roll (Figure 1), the rotational mode of vibration does not usually cause much discomfort (Griffin et al., 1982).   But in certain cases like, tractor moving on rough terrain, the pitching or rolling motions of the seat may be disturbing than rectilinear vibrations up and down.

Mehta and Tewary (2000) studied about role of cushioning material in a tractor seat in supporting operator posture, isolating vibration and improving ride quality.   And conclude that the vibration attenuation in a tractor seat is achieved by selecting proper suspension

and damping mechanism.   Servadio et al. (2007) studied the vibrations transmitted from the ground to the driver's seat by using method of ISO standards with a four-wheel-drive tractor, equipped with a front suspension axle and a suspended cab, operating at 11.1 and 13.9 m s^{-1}. Drakopoulos (2007) had studied about the current seat technologies in agricultural tractors and compared between earlier and current tractor seats.   Duke and Goss (2007) investigated on tractor driver seat performance with Non-linear Stiffness and on–off damper.

Stikeleather (1981) aimed to review the historical brief of seat, static and dynamic compatibility of operator/seat/machine, and evaluation of durability and reliability of suspension system.   Boileau and Rakheja (1990) had studied about vibration attenuation performance of different types of suspension seats for off-road forestry vehicles.   Sankar and Afonso (1993) studied about the design and testing of lateral seat suspension for off-road vehicles.   Prasad (1995) had investigated on Tractor seat design with regard to Indian operators.

3   Methodology

Tractors in 30-45 hp ranges are in most demand in India.   During collecting the voice of operators, it was noticed that tractor operators have some discomfort due to vibration of the seat.   The vibration is also influenced by speed of operation and ground terrain so the experiments were conducted under actual condition. Subjects for the study was selected on the basis of 5th, 50th, and 95th percentile (SM$_1$, SM$_2$ and SM$_3$ respectively) of the subject mass given in Table 1.   Out of galaxy of 30-45 HP range models, two models MF 1035 and MF 241 of TAFE tractors were selected for the study.   Test trailer having weight of 6 t, two bottom MB Plough and 36 blade rotavator having 1.5 m width were selected for the study under haulage operation and field operations respectively.

Table 1   BMI of Subjects

Subject	Height /m	BMI= Weight/height^2 /kg m^{-2}
SM$_1$	1.702	29.35
SM$_2$	1.820	22.42
SM$_3$	1.651	20.18

For haulage testing, test tractors were fitted with accelerometer and data logger was allowed to run over the marked patch.   Tests were conducted for six speeds. Different speeds were maintained by changing the throttle and gear.   Tests were repeated with three selected new seat suspension systems of SP_1, SP_2 and SP_3 respectively of the existing seat mostly in India and the specifications of the selected suspensions are given in Table 2.   After completing the haulage test, vibration characteristics of suspension systems were studied w.r.t exported benchmark seat which is a seat whose suspension appeared to majority of operators used for more comfortable and is fitted with exported tractors to the developed countries (The synthetic rubber foam cushion materials (thickness of 54 mm, density of 68.70 kg m^{-3})) (Mehta and Tewari, 2000).   According to ISO-2631,

characteristics were compared for 1/3rd octave band of frequency.   From the test results, the best seat suspension system was selected and used for the field testing of selected implements.

Table 2   Specification of Selected of spring pair

Sl. No.	No. of active coils	Mean dia./mm	Spring constant/kg mm^{-1}	Abbreviations
1	23	35	0.3	K1
2	24	39	0.55	K2
3	22	37	0.7	K3

3.1   Preparation of experimental setup

For measurement of vibration a uni-axial accelerometer was used in all the three axes (Z, X and Y direction respectively), a setup was made as per ISO 5007 for the mounting of accelerometer over the seat, HLC and base as shown in Figure 2.

Figure 2   Setup for measuring vibration in Z, Y and X direction

3.2   Evaluation of suspension system

Three types (SP_1, SP_2 and SP_3) of suspension systems with same cushion material were used for the vibration study.   Spring constant of the selected suspensions were different from the seat which was used in existing seat

used in India.   Model proposed by Mehta et al., (2000) was used for calculating transmissibility.

Figure 3 shows the three different types of suspension system SP_1, SP_2, and SP_3 selected for the study.   The value of spring constant K_1, K_2, and K_3 for three selected
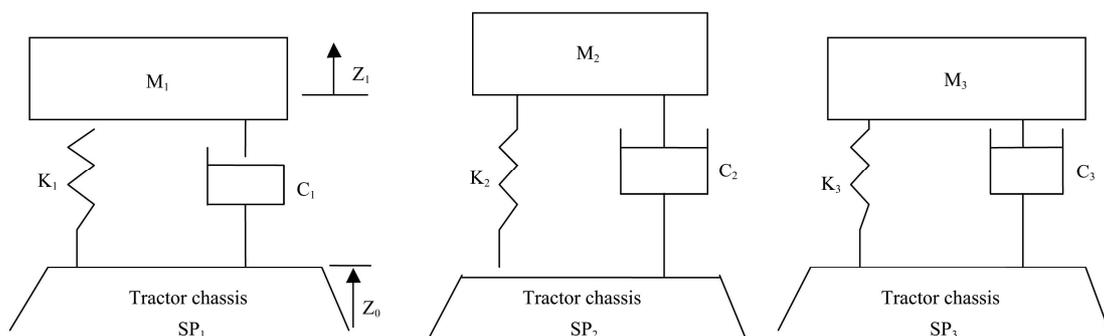

Figure 3   One DOF mechanical model of seat with man as rigid mass

suspensions were taken as 0.3, 0.55 and 0.7 kg mm^{-1} respectively, the value of damping coefficient (damping is an effect that reduces the amplitude of oscillations in an oscillatory system, particularly the harmonic oscillator) of dampers C_1, C_2 and C_3 was the same and it is of 1,465.935 Ns m^{-1}.   Apart from these suspension systems, vibration characteristics were also studied for existing seat used in India (K = 0.4 kg mm^{-1}, C = 1465.935 Ns m^{-1}) system as well as exported benchmark seat (thickness of 54 mm, density of 68.70 kg m^{-3}) for comparison of the test results.   For above simple model, the transmissibility calculated as follows (Equation (1)).

3.3   Data recording

For vibration measurement, a uni-axial piezoelectric accelerometer with FFT analyser, an eight-channel digital tape recorder (Josef Heim, Data Rec E8) and a real time frequency analyser (B&K, 2143) and UPS for long time charging of laptop were used.   A setup was made for the measuring vibration in Z, Y and X direction as discussed above.   For measurement of transmissibility the vibration was measured at the seat base Z_0 and that at the seat cushion, Z_1.   The values were measured for all sets of seats.   In accordance with ISO 2631-1, vibrations were analysed in 1/3 octave band from 1 to 80 Hz.   The root mean square (RMS) values, obtained for each 1/3 octave band, were used to calculate the global frequency-weighted acceleration a_{wx}, a_{wy} and a_{wz} for each axis, and then combined to obtain the value of the acceleration vector sum a_v, in m s^{-2}, where (Equation (2) and Equation (3)):

$$a_w = [\sum (w_i a_i)^2]^{1/2} \qquad (2)$$

where, a_w = frequency weighted r.m.s acceleration w.r.t the orthogonal axes x, y, z; W_i = weighting factor of the ith one-third octave band; a_i = acceleration level given by the accelerometer

$$a_v = (k_x^2 a_{wx}^2 + k_y^2 a_{wy}^2 + k_z^2 a_{wz}^2)^{1/2} \qquad (3)$$

where, k_x, k_y, k_z, are multiplying factors $k_x = 1.4$, $k_y = 1.4$, $k_z = 1$ (for seat)

4   Results and discussion

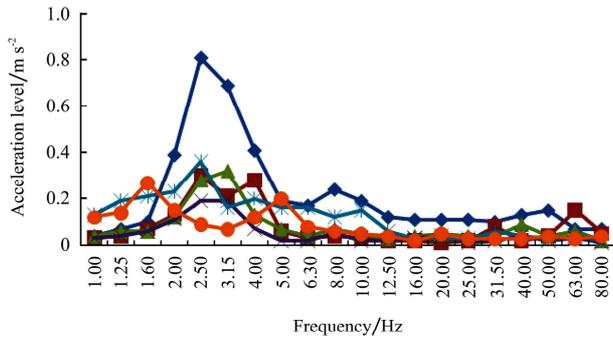### 4.1   Vibration characteristic of existing seat used in India for MF 241

Based on the test results, a comparison was made between the vibration characteristics of existing seat i.e. mostly used seat in India and exported bench mark seat at 11.67, 19.03 and 24 km h^{-1} respectively and is shown in Figure 4.   It was observed that there is no much difference of R.M.S vibration along X and Y axis between exported benchmark and existing seat in India and also there was no sharp peak along these two axes only the major difference was along Z axis values of peak vibration along Z axis increases with speed value of peak vibration along Z axis for existing seat, it was 0.81, 1.02 and 1.15 m s^{-2} for 11.67, 19.03 and 24 km h^{-1} of speed respectively.   Acceleration peak along Z axis occurs between 2-6.3 Hz for existing seat in India.   Peak for exported benchmark seat along Z axis was below 0.4 m s^{-2} all speeds.   Acceleration level along X and Y axis was under 0.3 m s^{-2} for exported benchmark as well as existing seat at all the speeds except for 19.03 km h^{-1} of speed peak vibration along X axis for existing seat used in India was 0.68 m s^{-2}.
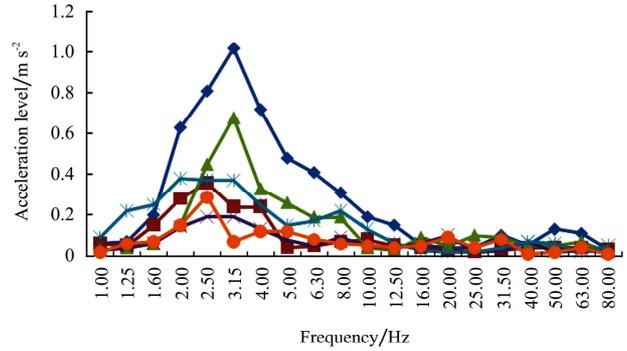
The comparison of vibration characteristics between existing seat mostly used in India with suspensions SP_1 and exported benchmark seat was studied.   It was observed that value of peak vibration along Z axis increases with speed for existing seat used in India it was 0.78, 0.82 and 1.07 m s^{-2} for 11.67, 19.03 and 24 km h^{-1} of speed respectively.   Acceleration peak along Z axis occurs between 2-6.3 Hz for existing seat in India.   Acceleration level along X and Y axis was under 0.3 m s^{-2} for benchmark as well as existing seat in India at all speeds except for 24 km h^{-1} of speed, peak vibration along X and Y axis at this speed for existing seat used in India was 0.63 and 0.68 m s^{-2} respectively.   Also a comparison made for of vibration characteristics between existing seat used in India with suspension system SP_2 and benchmark seat and is shown in Figure 5.   It was observed that value of peak vibration along Z axis was 0.6, 0.7 and 0.75 m s^{-2} for 11.67, 19.03 and 24 km h^{-1} of speed respectively.   At 24 km h^{-1} of speed two peaks occur along Z axis.   Acceleration peak along Z axis occurs between 2-6.3 Hz for existing seat.   Value of peak vibration along Y and X axis was below 0.4 m s^{-2} at all the speeds.   For the comparison of vibration characteristics between existing seat used in India with

suspension system SP_3 and benchmark seat and observed that, the same trend was followed for spring pair K_3 also but the value of peak was more than the domestic sear and peak was maximum at 24 km h^{-1} speed of tractor. Vibration was maximum in the range of 2.5-4 Hz for all speeds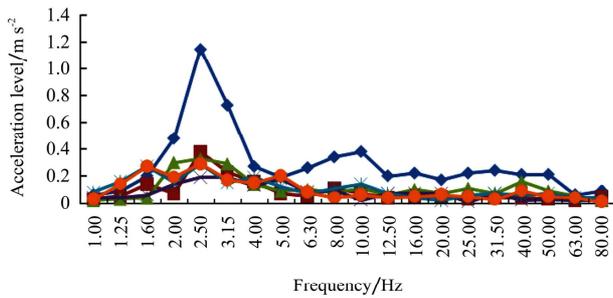 value of peak vibration along Z axis increases with speed for existing seat used in India it was 1.22, 1.30 and 1.53 m s^{-2} for 11.67, 19.03 and 24 km h^{-1} of speed respectively. Acceleration peak along Z axis occurs between 2-6.3 Hz for existing seat used in India. Value of peak vibration along Y and X axis was below 0.4 m s^{-2} at all speeds.

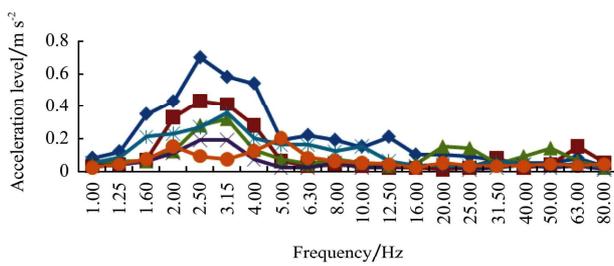
a. Vibration characteristic at 11.67 km h^{-1}

b. Vibration characteristic at 19.03 km h^{-1}

c. Vibration characteristic at 24 km h^{-1}

Figure 4   Comparison between existing seat used in India and exported benchmark seat

a. Vibration characteristic at 11.67 km h^{-1}

b. Vibration characteristic at 19.03 km h^{-1}

c. Vibration characteristic at 24 km h^{-1}

Figure 5   Comparison between SP_2 suspended existing seat and benchmark seat

4.1.1   Vibration characteristic of existing seat used in India for 1035 Model

A comparison was made between the vibration characteristics of existing seat i.e. mostly used seat in India and exported bench mark seat at 11.67, 19.03 and 24 km h^{-1} respectively and is shown in Figure 6.   It was observed that there is no much difference of RMS vibration along X and Y axis between benchmark and existing seat.   Value of peak vibration along Z axis for existing seat was 0.79, 0.9 and 1.08 m s^{-2} for 11.67, 19.03 and 24 km h^{-1} of speed respectively.   Acceleration peak along Z axis occurs between 2-6.3 Hz for existing seat.   Peak for benchmark seat along Z axis was below 0.4 m s^{-2} at all speeds.   Acceleration level along X and Y axis was under 0.4 m s^{-2} for benchmark as well as existing seat at all the speeds except for 24 km h^{-1} of speed peak vibration along Y axis for existing seat used in India was approx 0.5 m s^{-2}.   A comparison was made for vibration characteristics between existing seat used in India with suspension system SP$_1$ and benchmark seat, and observed that the value of peak vibration along Z axis increases with speed for existing seat used in India was 0.85, 0.92 and 1.68 m s^{-2} for 11.67, 19.03 and 24 km h^{-1} of speed respectively.   Acceleration peak along

Z axis occurs between 2-6.3 Hz for existing seat used in India.   Value of peak vibration along Y axis for existing seat used in India was approx 0.56, 0.45, and 0.46 m s^{-2} and value of peak vibration along X for existing seat used in India was 0.51, 0.44 and 0.61 at 11.67, 19.03 and 24 km h^{-1} of speed respectively, and also a comparison of vibration characteristics was made between existing seat used in India with suspensions SP$_2$ and benchmark seat was shown in Figure 7.   It was observed that value of peak vibration along Z axis increases with speed for existing seat used in India it was 0.58, 0.61 and 0.76 m s^{-2} for 11.67, 19.03 and 24 km h^{-1} of speed respectively.   Acceleration peak along with X, Y and Z axis occurs between 2-6.3 Hz for existing seat used in India.   Value of peak vibration along Y axis is 0.38, 0.13 and 0.34 m s^{-2} and that along X axis was 0.41, 0.3, and 0.44 m s-2 at 11.67, 19.03 and 24 km h^{-1} of speed respectively.   While for the comparison of vibration characteristics between existing seat used in India with suspensions SP$_3$ and benchmark seat, it was observed that same trend was followed for spring pair K$_3$ also but the value of peak was more. Peak value was maximum at 24 km h^{-1} speed of tractor. Vibration was maximum in the range of 2.5-4 Hz for all

a.Vibration characteristic at 11.67 km h^{-1}

b. Vibration characteristic at 19.03 km h^{-1}
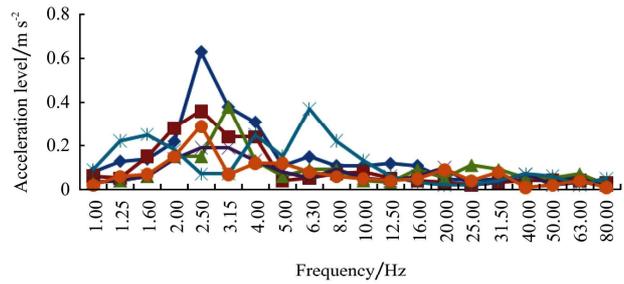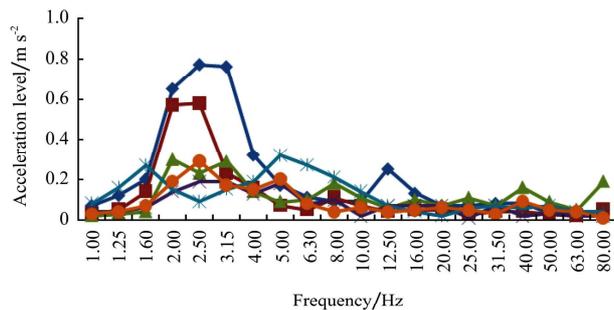

c. Vibration characteristic at 24 km h^{-1}

- DO-(Z axis)
- EX-(Z axis)
- DO-(X axis)
- EX-(X axis)
- DO-(Y axis)
- EX-(Yaxis)

Figure 6   Comparison between existing seat used in India and benchmark seat

a.Vibration characteristic at 11.67 km h^{-1}

b. Vibration characteristic at 19.03 km h^{-1}

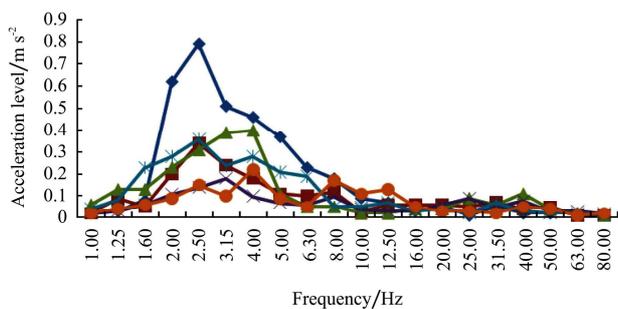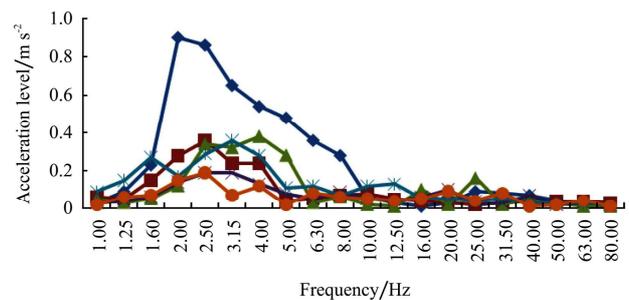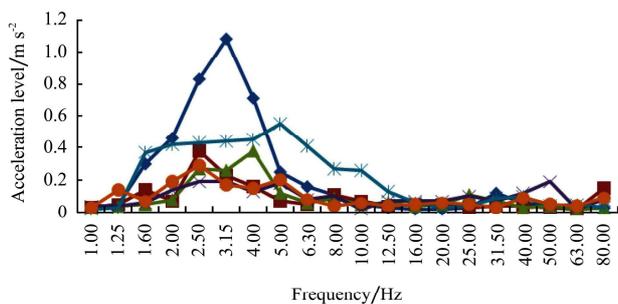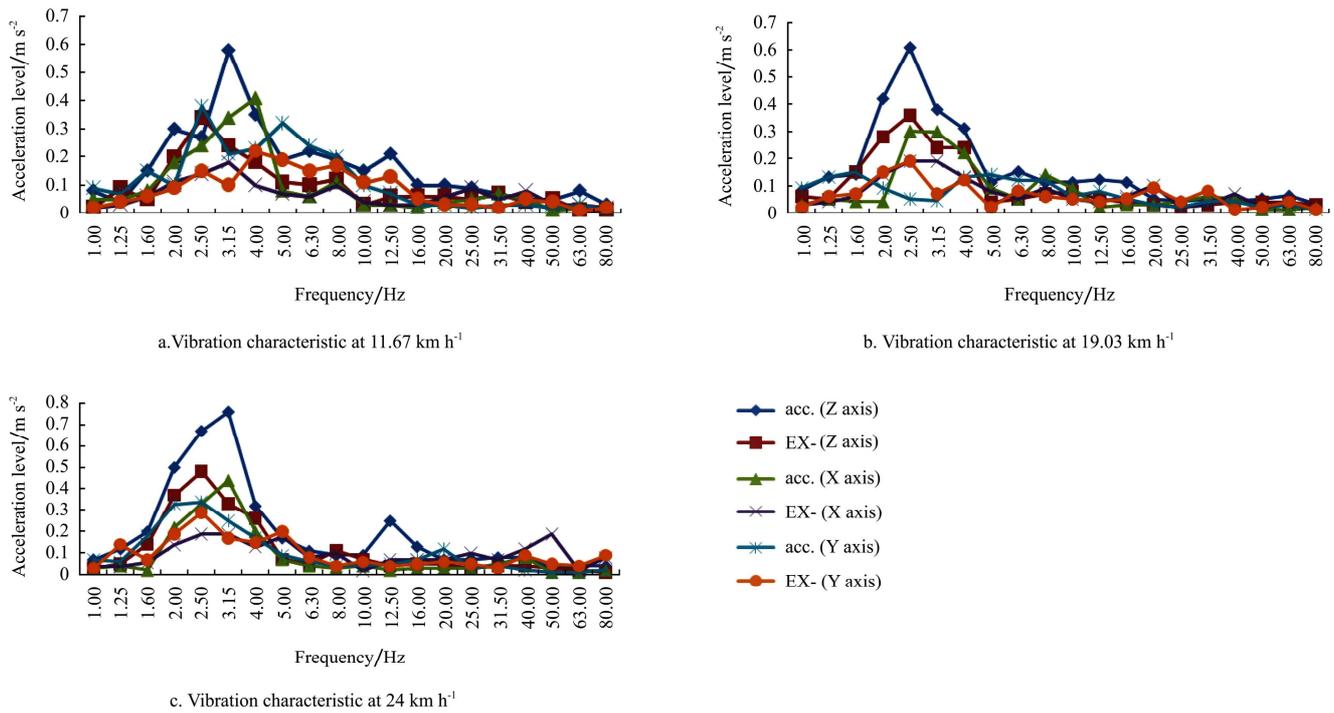c. Vibration characteristic at 24 km h^{-1}

- ◆ acc. (Z axis)
- ■ EX- (Z axis)
- ▲ acc. (X axis)
- ✕ EX- (X axis)
- ✱ acc. (Y axis)
- ● EX- (Y axis)

Figure 7   Comparison between SP$_2$ suspended existing seat and benchmark seat

speeds.   Value of peak vibration along Z axis increases with speed for existing seat used in India it was 1.19, 1.24, 1.49 m s^{-2} for 11.67, 19.03 and   24 km h^{-1} of speed respectively.   Acceleration Peak along Z axis occurs between 2-6.3 Hz for existing seat used in India. Value of peak vibration along Y and X axis was below 0.4 m s^{-2} at all speeds.

4.2   Transmissibility analysis

The   transmissibility   characteristics   of   selected suspension systems at 11.67 km h^{-1} of speed, were the value of 1.34 and 1.154 for MF 241 and MF 1035 respectively, shown in Figure 8 and Figure 9, the peak value occurs between the frequency range of 2-6.3 Hz for all conditions.   It has been observed that value of transmissibility is less than one for all frequencies other than resonance.   A seat is only required to provide good isolation of the vibration at the frequencies to which it will be exposed when in use.   The vibration isolation efficiency of a seat may be expressed simply by means of the seat effective amplitude transmissibility (SEAT) and its value may be considered to be the ratio of the frequencies – weighted ride experienced on the seat to the frequency – weighted   ride which would be experienced if the seat were rigid, expressed as percentage.
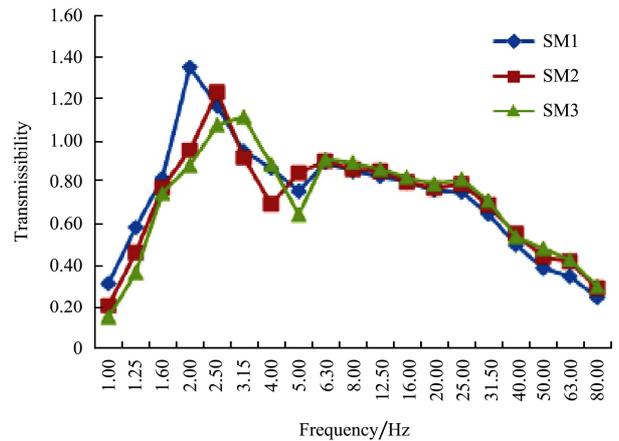

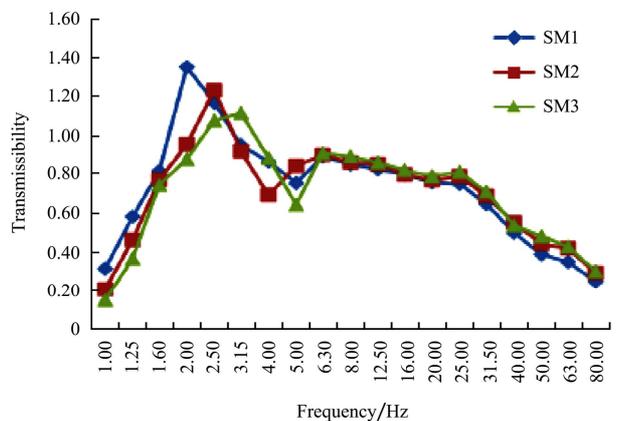Figure 8   Transmissibility of SP2 suspension at 11.67 km h^{-1} speed of MF241 model tractor

Figure 9   Transmissibility of SP2 suspension at 11.67 km h^{-1} speed of MF1035 model tractor

4.3   Analysis in 1/3 octave band from 1 to 80 Hz for the operator seat

The weighted RMS accelerations a_{wx}, a_{wy}, a_{wz} and the corresponding vector sum a_v were analysed for the different operator seat in different conditions. The values do not always increase as the forward speed increases. Table 3 and Table 4 show the values of frequency-weighted R.M.S. acceleration a_{wx}, a_{wy}, a_{wz} for each axis and the acceleration vector sum a_v for the MF 1035 model and MF241 respectively for all conditions of operator mass and speed.

Table 3   Values of the global frequency-weighted RMS acceleration a_{wx}, a_{wy}, a_{wz} and the acceleration vector sum a_v for the MF 241 model

				11.67					19.03					24.00		
		Domestic	SP3	SP1	SP2	Export	Domestic	SP3	SP1	SP2	Export	Domestic	SP3	SP1	SP2	Export
SM1	a_x	0.48	0.53	0.52	0.46	0.28	0.51	0.64	0.51	0.32	0.23	0.67	0.71	0.66	0.56	0.37
	a_y	0.43	0.43	0.43	0.48	0.4	0.36	0.52	0.36	0.48	0.3	0.42	0.57	0.42	0.42	0.41
	a_z	0.84	0.88	0.82	0.78	0.5	0.85	1.32	1.07	0.8	0.53	1.35	1.6	1.57	0.91	0.67
	a_v	1.23	1.3	1.25	1.22	0.85	1.22	1.75	1.38	1.14	0.75	1.74	2.05	1.92	1.34	1.02
	%a_z		5.13	-2.62	-6.79	-40.52		54.69	25.35	-5.99	-37.68		18.96	16.88	-32.12	-50.41
	%a_v		5.81	1.77	-0.75	-30.98		43.19	13.04	-6.89	-38.7		17.69	10.16	-23.03	-41.54
SM2	a_x	0.5	0.63	0.34	0.34	0.35	0.62	0.63	0.32	0.32	0.27	0.42	0.43	0.82	0.39	0.25
	a_y	0.34	0.47	0.38	0.46	0.25	0.63	0.37	0.42	0.43	0.26	0.44	0.35	0.6	0.46	0.28
	a_z	0.99	0.89	1.06	0.79	0.41	1.16	1.42	1.16	0.81	0.55	1.22	1.6	1.13	0.98	0.55
	a_v	1.3	1.42	1.27	1.12	0.73	1.7	1.75	1.38	1.1	0.76	1.49	1.78	1.81	1.29	0.76
	%a_z		-9.93	7.29	-20.26	-58.16		22.36	0	-30.61	-52.88		31.07	-7.77	-19.62	-55.11
	%a_v		9.38	-1.72	-13.59	-43.53		3.36	-18.73	-34.99	-55.24		19.32	21.35	-13.24	-49.11
SM3	a_x	0.52	0.52	0.3	0.49	0.43	0.37	0.51	0.5	0.47	0.42	0.47	0.66	0.3	0.37	0.67
	a_y	0.39	0.43	0.65	0.32	0.5	0.42	0.36	0.26	0.41	0.44	0.32	0.42	0.26	0.37	0.43
	a_z	0.84	0.87	0.83	0.72	0.35	0.99	0.88	0.91	0.8	0.38	1.21	1.05	1.22	0.91	0.65
	a_v	1.24	1.27	1.3	1.09	0.98	1.26	1.24	1.2	1.19	0.93	1.45	1.52	1.34	1.16	1.28
	%a_z		2.98	-1.67	-14.29	-58.81		-10.64	-8.31	-19.05	-61.8		-13.37	0.99	-25.33	-46.04
	%a_v		3.02	5.11	-11.76	-20.83		-1.28	-4.78	-5.7	-26.03		4.9	-7.38	-20.05	-11.46

Table 4   Values for the global frequency-weighted RMS acceleration a_{wx}, a_{wy}, a_{wz} and the acceleration vector sum a_v for the MF 1035 model

				11.67					19.03					24.00		
		Domestic	SP3	SP1	SP2	Export	Domestic	SP3	SP1	SP2	Export	Domestic	SP3	SP1	SP2	Export
SM1	a_x	0.38	0.49	0.58	0.49	0.33	0.50	0.63	0.58	0.47	0.33	0.51	0.72	0.52	0.43	0.39
	a_y	0.57	0.62	0.40	0.44	0.28	0.46	0.58	0.30	0.52	0.33	0.40	0.78	0.87	0.47	0.41
	a_z	0.78	0.80	1.00	0.70	0.55	0.88	1.19	1.15	0.66	0.58	0.97	1.25	1.57	0.80	0.68
	a_v	1.23	1.36	1.41	1.15	0.82	1.30	1.69	1.47	1.18	0.88	1.33	1.94	2.12	1.20	1.04
	%a_z		2.96	28.96	-6.90	-29.34		34.05	29.75	-25.00	-34.28		29.13	62.09	-17.25	-30.27
	%a_v		10.49	14.32	-6.83	-33.77		29.70	12.76	-9.11	-32.73		46.36	59.59	-9.52	-21.90
SM2	a_x	0.49	0.47	0.58	0.49	0.21	0.43	0.47	0.64	0.34	0.25	0.35	0.73	0.80	0.45	0.27
	a_y	0.50	0.45	0.57	0.43	0.23	0.50	0.42	0.52	0.26	0.24	0.77	0.66	0.65	0.48	0.36
	a_z	0.99	1.13	0.98	0.77	0.52	1.25	0.72	1.05	0.69	0.59	1.33	1.15	1.17	0.93	0.65
	a_v	1.40	1.45	1.51	1.20	0.68	1.55	1.14	1.55	0.91	0.76	1.78	1.79	1.86	1.30	0.90
	%a_z		14.55	-0.81	-22.02	-47.58		-42.41	-16.53	-44.97	-52.96		-13.73	-12.08	-30.46	-51.39
	%a_v		3.96	7.71	-14.45	-51.46		-26.69	-0.07	-41.58	-50.93		0.52	4.55	-26.84	-49.39
SM3	a_x	0.36	0.43	0.36	0.52	0.29	0.46	0.48	0.50	0.47	0.33	0.50	0.37	0.56	0.37	0.41
	a_y	0.36	0.37	0.36	0.39	0.36	0.57	0.49	0.57	0.32	0.26	0.57	0.36	0.57	0.42	0.39
	a_z	0.86	0.71	0.59	0.59	0.48	0.82	0.72	0.69	0.69	0.58	1.24	1.48	1.17	1.14	0.67
	a_v	1.12	1.06	0.93	1.08	0.80	1.31	1.20	1.26	1.05	0.83	1.63	1.64	1.62	1.38	1.04
	%a_z		-16.90	-31.00	-30.89	-44.06		-11.76	-15.32	-16.05	-28.43		19.02	-5.56	-8.30	-46.09
	%a_v		-4.80	-16.89	-2.88	-28.41		-7.98	-3.41	-19.64	-36.53		0.92	-0.67	-15.32	-36.05

Weighted RMS acceleration a_{wx}, a_{wy}, a_{wz} along three axes and acceleration vector sum a_v was shown in Table 3 for all type of seats. Percentage difference in a_v and a_{wz} with respect to existing seat used in India was also calculated. %a_{wz} was calculated because suspension system was used for damping the vibration along Z axis only so, due to change of suspension basically value of a_z will get affected. The analysis of five types of suspension system i.e. existing seat used in India without change in suspension system, than three pair (SP$_1$, SP$_2$, and SP$_3$) of changed suspension system and benchmark seat is shown in Table 3. It was observed that value of %a_v and %a_{wz} was always negative for all speeds and operator's i.e. vibration coming on existing seat with SP$_2$ suspension was lesser then the existing one. It can be observed from above tables that percentage improvement of a_v lies between 2.88 to 41.58 and a_z 6.90 to 44.97% for K$_2$ in different speeds.

Table 4 also depicted the percentage difference in a_v and a_z of different types of suspended seat with respect to existing seat used in India at three speeds of tractor. Findings were the same for MF 241 model also as MF 1035, K$_2$ was best among three pairs of suspension system. It was observed that the percentage improvement of a_{wz} lies between 5.99 to 32.12% and a_v 0.75 to 34.99% for K$_2$ in different conditions of speed and mass.

4.4　Field Testing

The field tests were conducted with best suspension system which is selected from the haulage results. Since speed of operation in field test was very low so vibration below 0.4 m s^{-2} was observed along entire axis.

4.4.1　Vibration characteristics of selected seat under field condition

The speed of operation for ploughing and tilling operation was maintained 5.95 and 3.35 km h^{-1} respectively. It was observed that peak along X axis is more than Z axis for the reason is that it may be longitudinal and lateral movement of the operator due to undulations of the ground. The vibration along Z axis was lesser in field operation; it might be due to less speed of operation. It was also observed that vibration beyond 8 Hz was less than 0.05 m (s^2)$^{-1}$ in all operations. The vibration characteristics of selected seat under field

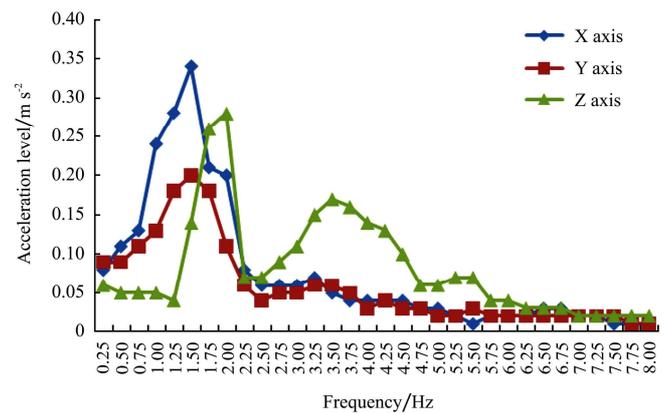condition are shown in Figure 10 and Figure 11.

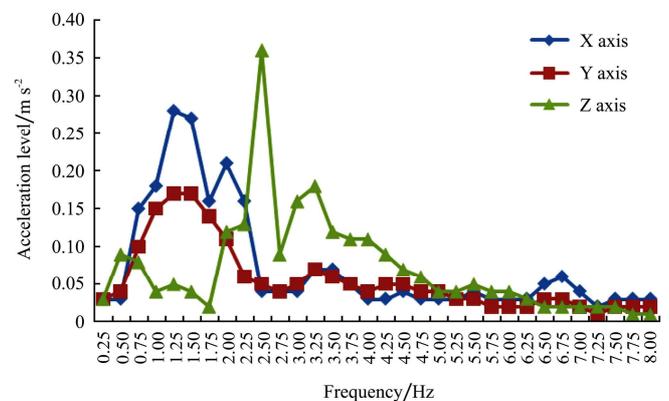Figure 10　Vibration characteristics of the seat in MB Plough operation

Figure 11　Vibration characteristics of the selected seat during tilling operation

5　Conclusions

Based on experimental results of this study following conclusions were drawn:

1) SP$_2$ suspension gives better result among three selected seat suspensions. The results are comparable with exported benchmark seat.

2) It was observed that there was no much difference of RMS vibration along X and Y axis between benchmark and existing seat and also there was no sharp peak along these two axes only the major difference was along Z axis.

3) Value of % a_v and % a_z was always negative for all speeds and operators i.e. the vibration coming on existing seat used in India with SP$_2$ suspension was lesser than the exported benchmark seat.

4) Percentage improvement of a_v lies between 2.88 to 45.15 and a_z 5.15 to 45.45 % for SP$_2$ in different

conditions of speed and mass.   It gives better performance at higher speed.

5) Peak transmissibility of SP_2 suspension at 11.67 km h^{-1} of speed was 1.34 and 1.154 for MF 241 and MF 1035 respectively.   Peak occurs between 2-6.3 Hz. for all conditions.

6) Field testing of existing seat used in India with SP_2 suspension also gives good results. Since speed of operation in field test was very low so vibration below 0.4 m s^{-2} was observed along entire axis.

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