Bio-Dynamic Response of Tractor Occupant's Exposed to Whole Body Vibration

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Abstract **-** *During the daily commute, humans are being continuously exposed to vibrations in one form or the other as could be observed in trucks, buses and other heavy vehicles. Human vibrations can be pleasant or unpleasant depending on the situation. Operators of on road and off-road vehicles have been associated with discomfort and a number of physical and nervous disorders (low back pain as well as long term health degradation) due to regular and long-term vibration exposure. It is also shown the degeneration of the lumbar spine with long-term exposure to whole-body vibration. If the exposure of human body to the vibrations exceeds a certain limit (stated by ISO), that may be hazardous to the health of the human body. The measurements of whole body vibration are commonly expressed in terms of the frequency weighted acceleration (measured on the vibrating surface which is in contact with the body) and the exposure time. In the present study, the human response to the whole body vibration generated by tractors is studied by mathematically modelling and analysis of an occupant tractor system in Wolfram SystemModeler software, the results of which are compared with that of the experimental results to check for the level of agreement between the two. By comparing those results with that of the standards set by the ISO it can be shown whether the vibration exposure is within the safe limit or not.*

Keywords **-** *Whole Body Vibration, Mathematical modelling, Tractor vibration, Vibration exposure.*

I. INTRODUCTION

Vibration is basically a "to and fro" motion of a body about its mean position. It is highly preferable or deeply unpleasant depending on the situation [1]. The effect of the vibration on the human body can be called as human vibration. This human vibration can be classified as either Whole Body Vibration or Hand Arm Vibration depending on how the vibration is transmitted to the human body. When the vibration is transmitted to the body through supporting surface like back and buttocks, it's called as Whole Body Vibration [2].

It was shown by Ghuman Kuljit [3] that this exposure to the vibration for a prolonged period of time causes a number of musculoskeletal disorders like Ischemic Lumbago which is condition causing a pain in the lower back region of the spine and degeneration of the lumber spine. He stated that the physical problems became more severe when exposed to the Whole Body Vibration (WHB) encountered in the heavy vehicles, agricultural vehicles, moving bed machinery and trailers. The risk associated with the vibration exposure to the human body increases with the duration of the vibration exposure [3]. One such heavy vehicle discussed by Bongers et al., is a forklift used for lifting heavy objects. Bongers et al., showed that the operators who are involved in operating the crane for more than five years suffered from physical issues such as fatigue, backbone disorders, degeneration of the vertebral disc and nervous issues like reduction in tactile sensitivity and tingling sensation and decrease in the ability to perform precise motor movements thereby confirming the proportionality of the duration of exposure to the degree of health effects. He also stated that there is strong association between the age of the operators and the vibration risks [4]. The health risk of the whole body vibration also depends upon the posture of the operators during the vibration exposure. This was stated by Bovenzi and Betta [5] in his investigation on postural stresses which an operator is subjected to in the driving conditions. Cann et al., [6] conducted a study on transport truck operators to determine which truck characteristics influence the level of whole body vibration exposure. Various variables were considered like road condition, truck type, driver experience, and truck mileage and seat type. It was found that road condition was a significant predictor of the frequency weighted acceleration in all the three direction showing that the health risks due to the WBV is seriously associated with the road conditions in which the vehicle is operated [7]. Ji-Geng Yan [7] studied the effects of the vibration induced by motor vehicle and found that prolonged vibration exposure results in chronic brain edema and shrunken neurons which can be strongly associated with the neural function impairment. Usually the measurement of the vibration and its quantification is done as per the standard ISO 2631 – 1 for vehicles [7].

ISO 2631 - 1 states the procedure for measuring vibration exposure and also provides the endurance limit in the form of duration of the vibration exposure for the human being which is shown in the figure 1 [8].

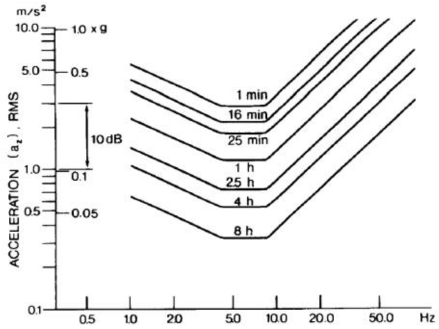

Fig I: The range of human endurance against vertical acceleration for Z-axis by different duration ISO 2631 [17]

Owing to serious health issues that are associated with the exposure to the whole body vibration, various steps were taken to investigate the human being's exposure to whole body vibration and also to study its effects on human beings with the aim of reducing the health hazards that are associated with the whole body vibrations. Exposure to the Whole body vibration can be reduced by various methods.

One such method is providing a proper cushioning in the operator's seat. Musa Marul et al., [9] tried using three different cushion material wool, sponge and cotton on the driver seat of a agriculture tractor and studied the extent to which the vibration is transmitted to the human body is each case. The vibration responses were measured using a tri-axial seat pad accelerometer placed at the operator/seat interface and compared against the standard ISO 2631. It was concluded that the wool cushion provided with the better isolation when compared to the other two [9]. Xiaoxu Ji et al. [10] proposed four different seat/cushion combinations and developed a neural network algorithm to identify the vibration attenuation properties of each seating conditions by implementing them in some forestry and mining vehicles. It was found that air inflated cushion has better vibration attenuation properties of the seat thereby reducing the health risks associated with the whole body vibration exposure [10].

Other way of reducing the health risks associated with the whole body vibration is to alter the suspension parameters of the vehicles by either modelling it mathematically or numerically such that the vibration transmitted to the body is attenuated or reduced. Zehsaz et al., [11] conducted a study whose main objective is to reduce the vibrations that is transmitted to the tractor's cabin. Initially the vertical acceleration of the driver's cabin is measured experimentally and it was used as the dynamic input to the finite element model of the tractor's cabin. Finally by using an iterative technique the suspension parameters are optimised and it was found that there was a significant improvement in terms of comfort and reduced fatigue [11]. Gunston et al. [12] tried to model the suspension seats whose dynamic behaviour was measured experimentally in the laboratory. The two different models were lumped parameter model which shows the response of the individual components of the seat and the Bouc–Wen model which has a nonlinear degree of freedom. Both of these models were stimulated by providing the dynamic inputs and the results were compared against the measured values. It was concluded that the lumped parameter model for the development of the suspension seat design [12]. These models can then be used to alter the suspension parameters to reduce the health risks associated with the whole body vibration. Cho-Chung Liang et al. [13] investigated on various lumped parameter models of the seated humans without back rest which are exposed to the vertical vibrations. The study [13] involved validating and synthesising the data of various models with different degree of freedom from published literature. Based on the correlation between the analytical and experimental results it was found that a four degree of freedom model was a best fit to the test results thereby making it a recommended model for studying the biodynamic response of the seated humans subjected to the whole body vibration [13]. Gao et al. [14] developed a lumped parameter model specifically a half car model of a car to study the response of the car to the uncertain random rad excitation input. The influence of the vehicle's parameters mass and stiffness, on the dynamic behaviour of the car was studied in detail using this model [14]. The lumped parameter model can be developed for the combination of the vehicle and human being together and it's called as coupled lumped parameter model [1].

The studies and investigations have been carried out by considering the human and vehicle as two separate lumped parameter model and very less focus is made on the coupled lumped parameter model and the studies are carried out by considering the lumped parameter model as non-linear system. Hence this paper focuses on modelling the tractor and it's occupant as a linear coupled lumped parameter model with 8 degrees of freedom on the whole (4 degrees of freedom for human occupant and 4 degrees of freedom for tractor) to which a dynamic excitation inputs are given. The results of the solved model are then compared against the experimental results to check for the conformity between those two. The results are then compared against the standard ISO 2631 to check whether the vibration exposure is within the safe limit or not.

II. MATERIAL AND METHODS

A. Mathematical Modelling

The human body subjected to Whole Body Vibration (WBV) can be modelled mathematically with the different parts considered as different lumped mass systems with a spring and a damper that represents the parameters of the system. The ISO 2631-1 suggests the use of acceleration in metre per second squared (ms^{-2}) as the parameter for the measurement of the vibration level that the human body is being subjected to due to the vibrations experienced by the Tractor/heavy vehicle drivers. The mathematical model is used to calculate the vibration level that the human body is exposed to for a given force/ the road profile. The system modelled is a composite of the human body and the tractor as the vibration is dependent on both the systems. The modelling is done with the following considerations:

i. Human body is modelled as a 4 Degrees of Freedom (DOF) system which is more aligned with the experimental results measured on human body as concluded by Cho-Chung Liang and Chi-Feng Chiang [13] by considering the different DOF systems along with the seat of the vehicle.

ii. The tractor is modelled as a Half-car with 4 Degrees of freedom with the Chassis, Front and Rear tyres as the lumped masses along with the seat as stated by Gao et al. [14]. The stiffness values of the tyres and the stiffness and the damping constant values of the suspension for the body and the seat obtained from [13], [14] and [15].

The considerations made above results in the mathematical model being developed with 8 Degrees of Freedom (Head, Upper Torso, Lower Torso, Viscera, Seat, Chassis, Front and Rear tyres). This results in an 8x8 matrix for the Mass, Stiffness and Damping constant in the form of a Second order differential equation.

The track for the test is assumed to be a sinusoidal track with the phase difference between the Front and the Rear tyres taken as 160 degrees and the amplitude of the track is taken as 0.05m as determined by Patil and Palanichamy [15]. The mathematical model is solved using Wolfram Mathematica and Wolfram System Modeller software. The results obtained are to be compared with the experimental values to ascertain the conformity of the model used.

B. Experimental Test and Instrumentation

TABLE I
Technical Specification of Mahindra MKM-NST 575 DI Tractor

Maximum Power	33.556 kW @ 1900 rpm
Engine	Model: MDI2500 Type: Four Stroke Direct Injection Water Cooled Cylinders: 4
Transmission	Sliding Mesh Eight speed Gear Box with eight forward speeds and two reverse speeds
Wheel and Tyre Size	Front: 6.0 x 16” Rear: 13.6 x 28”
Dimensions and Mass	Wheel Base: 1910mm Mass (with fuel, oil and water): 1870kg

The steps involved in analysing the mathematical model for the conformity includes an experiment to determine the values of the vibration experienced by the driver in a tractor.

Fig II: Tractor Used for Experimentation

Fig III: Seat Pad Accelerometer

The experiment involves the proper selection of the instrument for measuring vibration according to ISO 2631, and measurement of the vibration and analysis.

i. Tractor: The experiment is performed with a Mahindra MKM-NST 575 DI tractor, the specification of which is listed in the table 1. The road conditions for the test consist of both rough and smooth conditions. The rough condition signifies the use of a field with and without plough.

ii. Instrumentation: The instrument involves a seat pad accelerometer comprising of a Tri-axial accelerometer mounted in a rubber pad which is placed on the seat on which the driver sits. The sensitivity of the accelerometer is 100mV/g (±5%) with a range of ±60g. The values are taken from the Human Vibration Meter (HVM – VM30-HAWB). The vibration is measured in terms of acceleration using the seat pad accelerometer and is obtained from the vibration meter.

iii. Measurement and Data Extraction: The data of the measured vibration are stored in the vibration which can be extracted. This gives the mean value of the readings observed. The experiment is performed as per ISO 2631 and the readings are compared with the mathematical model to analyse the accuracy of the mathematical model developed.

Fig IV: Human Vibration Meter

III. THEORY

A mathematical model is to be developed in order to calculate the amplitude of the vibrations the human body is subjected to, based on the different conditions of the road profile, vehicle speed and parameters. The model was developed by considering a half car as the parameters differ between the front and the rear set of wheels.

A. *Assumptions for Mathematical Model*

The assumptions in the mathematical model which facilitate the ease of solving and influence the results are listed below [15]:

- The profile of the road or the field of the experimentation was considered to be sinusoidal, with amplitude of 0.05 m or 5 cm for facilitating the mathematical solution for the model.
- The tractor movement was considered to be only in the longitudinal plane that passes through the centre of gravity. The wheels were considered separately along with the chassis mass with different stiffness values for the front and the rear wheel.
- The couples and the forces due to the rotating wheels were assumed to be a minimum and are neglected.
- The displacements of the masses were considered to be sufficiently low for the tractor's suspension and tyre movements to be within their linear range always as this is used to make the equations of motion simpler to solve owing to the complexity of the solution for a non-linear system of equations.

The composite model of tractor-occupant was subjected to the sinusoidal vibrations with a phase difference of 160° between the front and the rear wheels caused due to the ground reaction forces that the tractor is subjected to in its motion on the field of observation. While deriving the governing equations of motion, only vertical motion of the tractor was included neglecting the effects of pitches and rolls. The stiffness and damping characteristics of the torsos, viscera and the head were represented by linear springs and linear dashpots. The equation of motion for each mass consisted of both the inertial term and the forces exerted on the mass by the springs and dashpots occasioned by the relative motion of the connected masses. The governing second order, coupled, linear, ordinary, differential equations of the various masses of the composite model are listed below.

B. *Mathematical Model*

i) *Head*

$$m_h\,(\ddot{x}_1)+ C_h(\dot{x}_1 - \dot{x}_2)+ k_h(x_1-x_2)=0 \ldots\ldots\ldots(1)$$

ii) Upper Torso

$m_u\ (\ddot{x}_2)+\ C_h(\dot{x}_2\ -\ \dot{x}\ )+\ C_{uv}(\dot{x}_2\ -\ \dot{x}_3\ )+\ C_u(\dot{x}_2\ -\ \dot{x}_4\ )+k_h(x_2\text{-}x_1)+\ k_{uv}(x_2\text{-}x_3)+\ k_u(x_2\text{-}x_4)=0$……(2)

iii) Viscera

$m_v\ (\ddot{x}_3)+\ C_{uv}(\dot{x}_3\ -\ \dot{x}_2\ )+\ C_{vl}(\dot{x}_3\ -\ \dot{x}_4\ )+\ k_{uv}(x_3\text{-}x_2)+\ k_u(x_3\text{-}x_4)=0$…………………………………(3)

iv) Lower Torso

$m_l\ (\ddot{x}_4)+\ C_u(\dot{x}_4\ -\ \dot{x}_2\ )+\ C_{vl}(\dot{x}_4\ -\ \dot{x}_3\ )+\ C_l(\dot{x}_4\ -\ \dot{x}_5\ )+k_u(x_4\text{-}x_2)+\ k_{vl}(x_4\text{-}x_3)+\ k_l(x_4\text{-}x_5)=0$…….(4)

v) Seat

$m_s\ (\ddot{x}_5)+\ C_l(\dot{x}_5 - \dot{x}_4\ )+\ C_s(\dot{x}_5 - \dot{x}_6\ )+\ k_l(x_5\text{-}x_4)+\ k_s(x_5\text{-}x_6)\ =0$…………………………………………(5)

vi) Chassis

$m_c\ (\ddot{x}_6)+\ C_s(\dot{x}_6 - \dot{x}_5\ )+\ C_{cr}(\dot{x}_6 - \dot{x}_8\ )+\ C_{cf}(\dot{x}_6 - \dot{x}_7\ )+k_s(x_6\text{-}x_5)+\ k_{cr}(x_6\text{-}x_8)+\ k_{cf}(x_6\text{-}x_7)=0$……(6)

vii) Front Tyre

$m_f\ (\ddot{x}_7)+\ C_{cf}(\dot{x}_7 - \dot{x}_6\ )+\ +k_{cf}(x_7\text{-}x_6)+\ k_f(x_7)=k_f(A)\sin(\omega t)$………………………………….(7)

viii) Rear Tyre

$m_r\ (\ddot{x}_8)+\ C_{cr}(\dot{x}_8 - \dot{x}_6\ )+\ k_{cr}(x_8\text{-}x_6)+\ k_r(x_8)=k_r(A)\sin(\omega t\text{-}\alpha)$…………………………….(8)

The values of the different symbolic constants stated above are extracted from the literatures in order to obtain the results of the mathematical model that is developed to check the accuracy of the model. The results are obtained by solving the system of differential equations that form an 8x8 matrix for the mass, stiffness and the damping constant. The values of the acceleration, velocity and displacement and their corresponding graphs with respect to time can be obtained from the Wolfram System Modeller software. The results can further be compared to the experimental values to obtain the conformity of both.

The matrix form of the mathematical model is given in the figure 5. The matrix equation is solved in order to obtain the mathematical solution of the model developed and the results are obtained on solving it using Wolfram SystemModeler software.

TABLE II
Values of Mass, Stiffness and Damping Coefficient

Part	Mass kg	Stiffness Nm^{-1}	Damping Const Nsm^{-1}
Head	4.17	134400	250
Upper Torso	15	192000	909.1
Viscera	5.5	10000	200
Lower Torso	36	49340	2475
Seat	4.537	2943	184.8
Chassis –Front	1794.4	66824.4	1190
Chassis – Rear		18615	1000
Front Tyre	140.4	101115	-
Rear Tyre	87.15	101115	-

$$\begin{pmatrix} m_h & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & m_u & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & m_v & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & m_l & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & m_s & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & m_c & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & m_f & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & m_r \end{pmatrix} \frac{d^2}{dt^2} \begin{pmatrix} x1[t] \\ x2[t] \\ x3[t] \\ x4[t] \\ x5[t] \\ x6[t] \\ x7[t] \\ x8[t] \end{pmatrix} + \begin{pmatrix} c_h & -c_h & 0 & 0 & 0 & 0 & 0 & 0 \\ -c_h & c_h + c_u + c_{uv} & -c_{uv} & -c_u & 0 & 0 & 0 & 0 \\ 0 & -c_{uv} & c_{uv} + c_{vl} & -c_{vl} & 0 & 0 & 0 & 0 \\ 0 & -c_u & -c_{vl} & c_l + c_u + c_{vl} & -c_l & 0 & 0 & 0 \\ 0 & 0 & 0 & -c_l & c_l + c_s & -c_s & 0 & 0 \\ 0 & 0 & 0 & 0 & -c_s & c_{cf} + c_{cr} + c_s & -c_{cf} & -c_{cr} \\ 0 & 0 & 0 & 0 & 0 & -c_{cf} & c_{cf} & 0 \\ 0 & 0 & 0 & 0 & 0 & -c_{cr} & 0 & c_{cr} \end{pmatrix} \frac{d}{dt} \begin{pmatrix} x1[t] \\ x2[t] \\ x3[t] \\ x4[t] \\ x5[t] \\ x6[t] \\ x7[t] \\ x8[t] \end{pmatrix} +$$

$$\begin{pmatrix} k_h & -k_h & 0 & 0 & 0 & 0 & 0 & 0 \\ -k_h & k_h + k_u + k_{uv} & -k_{uv} & -k_u & 0 & 0 & 0 & 0 \\ 0 & -k_{uv} & k_{uv} + k_{vl} & -k_{vl} & 0 & 0 & 0 & 0 \\ 0 & -k_u & -k_{vl} & k_l + k_u + k_{vl} & -k_l & 0 & 0 & 0 \\ 0 & 0 & 0 & -k_l & k_l + k_s & -k_s & 0 & 0 \\ 0 & 0 & 0 & 0 & -k_s & k_{cf} + kc_{cr} + k_s & -k_{cf} & -k_{cr} \\ 0 & 0 & 0 & 0 & 0 & -k_{cf} & k_{cf} + k_f & 0 \\ 0 & 0 & 0 & 0 & 0 & -k_{cr} & 0 & k_{cr} + k_r \end{pmatrix} \begin{pmatrix} x1[t] \\ x2[t] \\ x3[t] \\ x4[t] \\ x5[t] \\ x6[t] \\ x7[t] \\ x8[t] \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ A\ k_f\ \mathrm{Sin}[t * w] \\ A\ k_r\ \mathrm{Sin}[t * w - \alpha] \end{pmatrix}$$

Fig V: Matrix Form of Mathematical Model

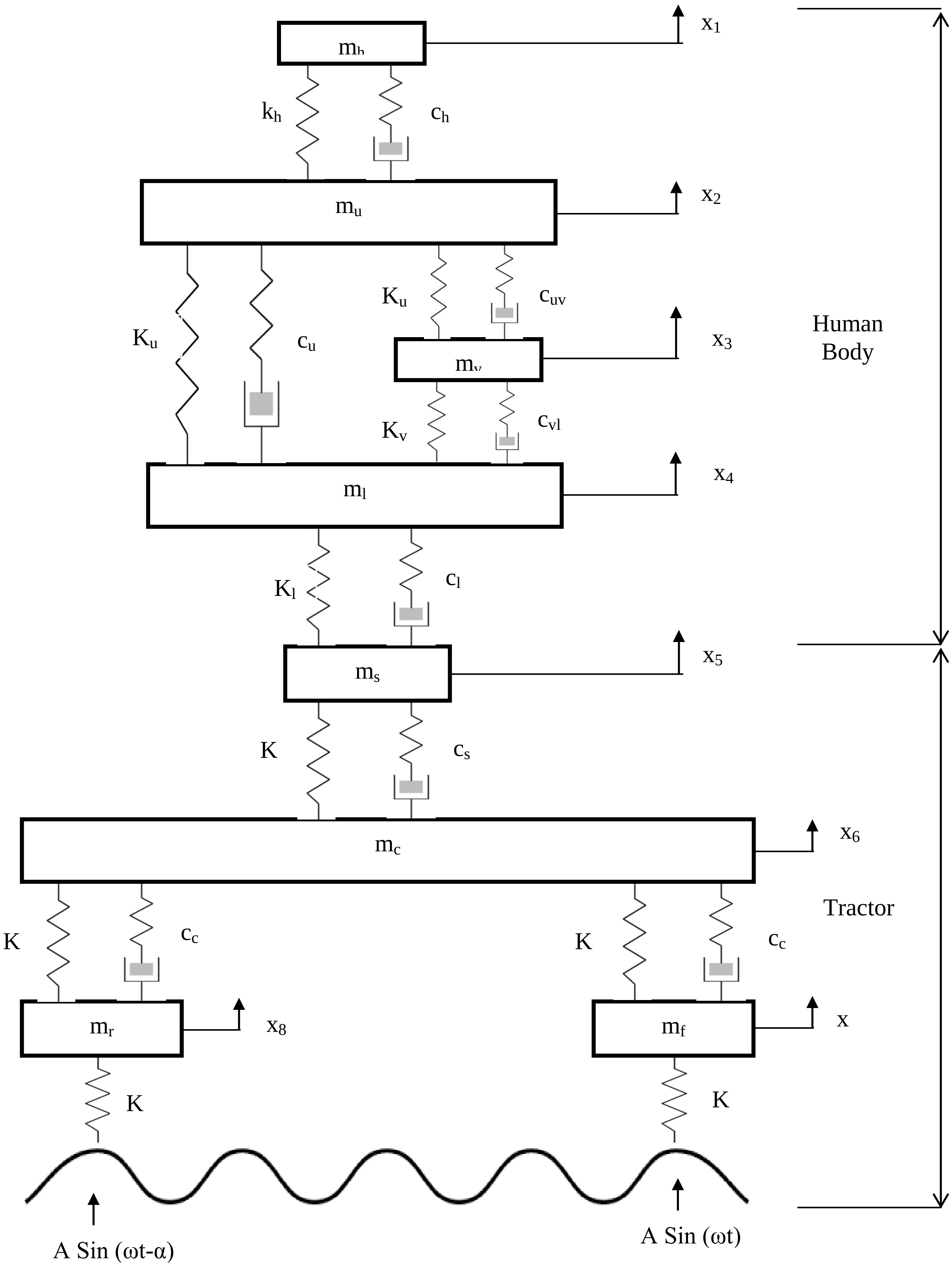

Fig VI: Mathematical Model of Tractor Occupant

C. Experimental Analysis

The experimental analysis is performed according to ISO 2631 [17] by using a seat pad accelerometer by placing it on the driver's seat of the tractor. This ensures the measurement of the maximum amplitude of vibration that is transferred to the human body. The figure 7 shows the experimental set-up used.

Fig VII: Experimental Setup

The prime focus is that human vibration is accurately measured so that an assessment can be made of: (a) the discomfort produced by the vibration, (b) the possible danger involved in being exposed to the vibration and (c) the conformance of the mathematical model that is developed. The exposure to vibration is mainly due to the fact that if people are over-protected it could impose limitations on their freedom of movement, resulting in reduced efficiency, but over-exposure to vibrations can cause accidents in the short term and/ or physical damage after a long-term exposure. This stresses the importance on the analysis of the vibrations that the human is exposed. The accuracy of human vibration measured depends on the analysis and recording equipment used. The transducer which is used for vibration measurements is the piezoelectric accelerometer, exhibits better all-round characteristics and stability than any other type of vibration transducer, and its response is linear through the frequency range of interest.

The whole body vibration is mainly caused by the vibration transmitted from the vehicle to the body through the seat. For this purpose a seat transducer - a tri-axial seat pad accelerometer placed into a rubber pad is used which can be positioned at the excitation point without disturbing the original position of the person or reducing his/her comfort. To measure vibrations transmitted to a vehicle driver, the driver may either sit on the transducer or strap it onto his back. The measurment was made for a duration 120 seconds to ensure reasonable statistical precision and the mean values of the readings were taken. ISO 2631 - Evaluation of human exposure to whole body vibration is used to identify the range in which the vibration should lie which is provided in the table below.

TABLE III
Whole Body Vibration Limit

Exposure Action Value	0.5 ms^{-2}
Exposure Limit Value	1.15 ms^{-2}

For whole-body vibration interval RMS is measured for each coordinate direction though the prime focus is on the z-direction, in which the vibration is transmitted to the human body. The vibration measurements are noted from the VM30-HAWB which provides the mean value of the vibration during the time period of measurement. The test parameters are tabulated below:

TABLE IV
Test Parameters

Instrument Used	VM30-HAWB
Accelerometer	KB103SV-100 for whole-body measurement (10mV/ms^{-2})
Frequency	10 Hz – 1250 Hz
Duration	120 seconds
Machine	Mahindra MKM-NST 575 DI tractor

The track used for measurement was selected to simulate the working conditions of the tractor. The two track conditions used were (a) Rough track (Agricultural Field) with/ without plough and (b) Smooth track (Plain road) without plough.

Fig VIII: Tractor Operating on an Agricultural Field (Rough Track) with Plough

The figures 8 and 9 show the operation of the tractor with and without plough on a rough track - agricultural field. The figure 10 shows the operation of the tractor under smooth conditions. The tests were performed on the same driver to ensure the uniformity of the readings measured. The results obtained were compared with the results of the mathematical model and the inferences are discussed in the subsequent section.

Fig IX: Tractor Operating on an Agricultural Field (Rough Track) without Plough

Fig X: Tractor Operating on a Smooth Track without Plough

IV. RESULTS AND DISCUSSIONS

A. Validation of Coupled Lumped Mass System

The results obtained by stimulating the mathematical model in Wolfram System Modeller was validated by comparing the acceleration values against the results obtained from that of the experiment conducted. The results obtained from the dynamic analysis of mathematical model are shown in the figures 11 to 19. The extent to which the results correlate against one another was represented in the form of percentage of deviation which helped in analysing the conformity of the mathematical model developed. The deviations were noticed due to the assumptions that were made in the modelling of the system mathematically and approximations made to solve the mathematical model.

B. Mathematical Results

Graphs were obtained by stimulating the mathematical model in system modeller with the tractor running on rough roads and smooth roads at different speeds. To get proper results for specific set of road conditions appropriate dynamic inputs were given. It was assumed that the variables in each of the stimulation were the frequency of the road profile and the engine vibrations. It was also found that the frequency of the road profile varies between 0.5 to 11 Hz depending on the road conditions [15].

The value of the acceleration is obtained from the graphs and is compared with the experimental values. The experimental results are given in the subsequent section for the different conditions.

TABLE V

Acceleration Values on Rough Track with Plough

Rough Track, With Plough, Duration:120 s	
At 1000 rpm, 1st gear	
Interval RMS (A(T)) in m/s²	X = 0.1 m/s²
	Y = 0.5 m/s²
	Z = 0.3 m/s²
At 1100 rpm, 1st gear	
Interval RMS (A(T)) in m/s²	X = 0.2 m/s²
	Y = 0.3 m/s²
	Z = 0.3 m/s²
At 1000 rpm, 2nd gear	
Interval RMS (A(T)) in m/s²	X = 0.3 m/s²
	Y = 0.4 m/s²
	Z = 0.6 m/s²

TABLE VI

Acceleration Values on Rough Track without Plough

Rough Track, Without Plough, Duration :120 s	
At 1000 rpm, 1st gear	
Interval RMS (A(T)) in m/s²	X = 0.1 m/s²
	Y = 0.3 m/s²
	Z = 0.2 m/s²
At 1100 rpm, 1st gear	
Interval RMS (A(T)) in m/s²	X = 0.1 m/s²
	Y = 0.5 m/s²
	Z = 0.2 m/s²
At 1000 rpm, 2nd gear	
Interval RMS (A(T)) in m/s²	X = 0.2 m/s²
	Y = 0.5 m/s²
	Z = 0.4 m/s²

C. Experimental Results

The tables 5, 6 and 7 represent the peak RMS acceleration value obtained during the experiment using a seat pad accelerometer and human vibration meter. The RMS acceleration value were measured under three different road conditions for around 120 seconds – Tractor running on rough track with and without plough and tractor running on smooth track without plough. Under each of the track condition, the tractor is operated at three different speeds. The X, Y, Z represents the RMS acceleration along three different directions. However the acceleration along the Z direction is considered for comparison against the mathematical results as the model was solved by assuming the movement to be only in the longitudinal plane. It can be seen from the table 5, 6 and 7 that the RMS acceleration in Z direction is much higher when the tractor is operated with plough rather than without plough. This means that health risks associated with the vibration generated by tractor is much higher when it is operated with plough. Another important thing that can be observed here is that when the engine RPM is increased under the same gear, the RMS value of the acceleration in the Z direction also increases. Thus the vibration transmitted to the human body also increases. At higher gear however the speed of the vehicle increases, hence the frequency of encountering the bumps in the track increases. This in turn increases the transmission of vibration to the human body. The results obtained by solving the mathematical model are taken from the graphs in figures 11 to 19. The peak value under each case is taken for comparison against the experimental results to check for conformity.

From the table 8, it can be seen that the deviation between experimental and mathematical results are less thereby ensuring the conformity between both the results. The variation is observed to be between 1.67% and 10.33%. The average variation of the results was found to be 5.64%, thereby the model can be used for changing the parameters of the spring and damper system and analyse the vibration exposure to the human body.

V. CONCLUSIONS

In this study, the mathematical model was developed for analysing the vibrations experienced by a tractor occupant using a Half-car model. The experiments were conducted according to the standards mentioned in ISO 2631. The simulations of the experimental conditions were performed using the System Modeller software and the values obtained for the acceleration was compared with the experimental results. It was found that the vibration measured experimentally were similar to the simulation results obtained from the mathematical model. The deviations from the experimental values were found to vary from 1.67% to 10.33% with the average deviation value of 5.64%. This deviation accounted from the assumptions made in simplification of the model in order to improve the solvability and to reduce the complexity of the model. The acceleration values obtained both experimentally and mathematically are found to be within the safe exposure limit according to the standards.

TABLE VII
Acceleration Values on Smooth Track without Plough

Smooth Track, Without Plough, Duration :120s	
At 1000 rpm, 1st gear	
Interval RMS (A(T)) in m/s²	X = 0.2 m/s²
	Y = 0.5 m/s²
	Z = 0.3 m/s²
At 1200 rpm, 2nd gear	
Interval RMS (A(T)) in m/s²	X = 0.2 m/s²
	Y = 0.3 m/s²
	Z = 0.7 m/s²
At 1000 rpm, 3rd gear	
Interval RMS (A(T)) in m/s²	X = 0.3 m/s²
	Y = 0.6 m/s²
	Z = 0.7 m/s²

TABLE VIII
Comparison between Experimental and Mathematical values

Description	Experimental result (m/s^2)	Analytical result (m/s^2)	Deviation (%)
Rough track, with plough, 1st gear, 1000rpm	0.3	0.274452	8.5
Rough track, with plough, 1st gear, 1100rpm	0.3	0.323303	7.76
Rough track, with plough, 2nd gear, 1000rpm	0.6	0.581213	3.1
Rough track, without plough, 1st gear, 1000rpm	0.2	0.179322	10.33
Rough track, without plough, 1st gear, 1100rpm	0.2	0.217270	8.63
Rough track, without plough,2ndgear, 1000rpm	0.4	0.385979	3.50
Smooth track, without plough, 1st gear 1000rpm	0.3	0.284784	5.07
Smooth track, without plough, 2nd gear, 1100 rpm	0.7	0.684845	2.16
Smooth track, without plough, 3rd gear, 1000 rpm	0.7	0.711755	1.67

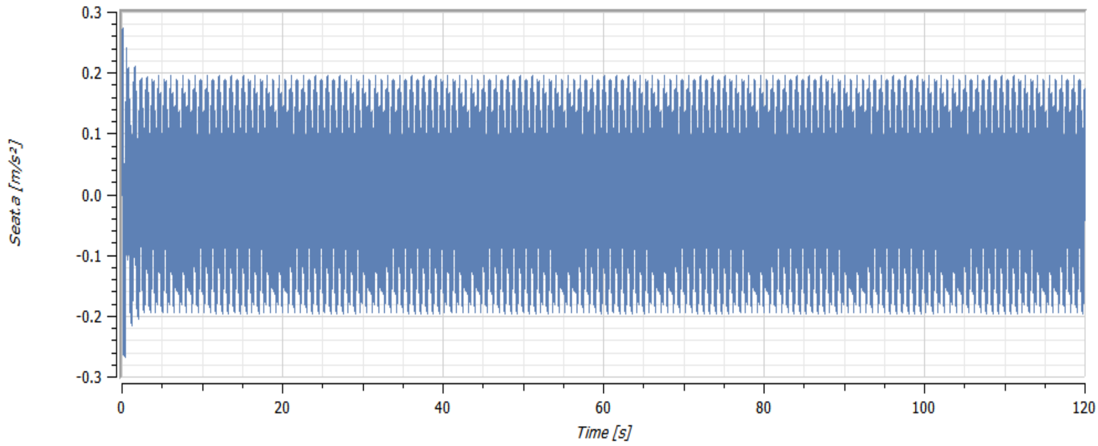

Fig XI: Seat Acceleration Vs Time Measured under Rough Track with Plough, 1st gear, 1000 RPM

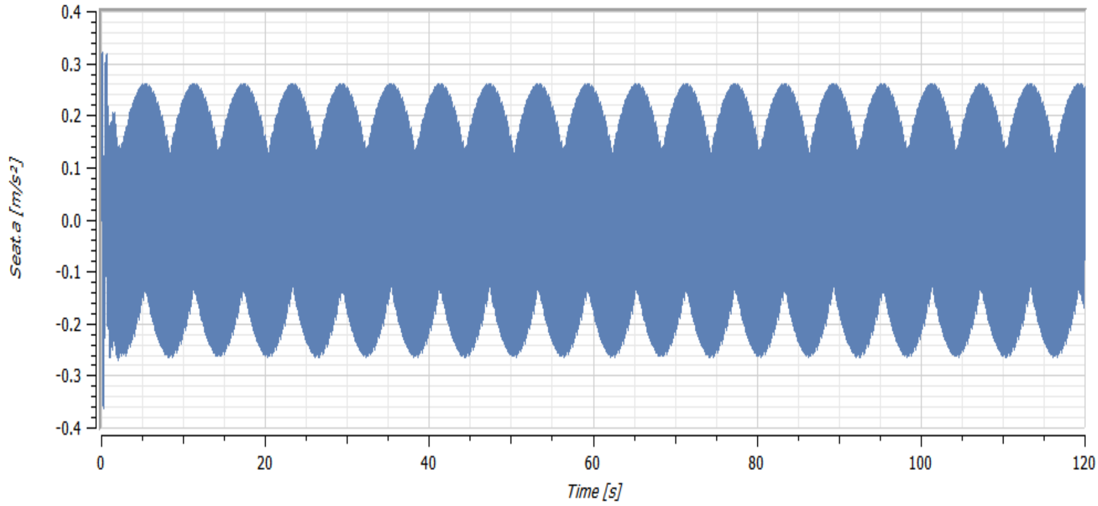

Fig XII: Seat Acceleration Vs Time Measured under Rough Track with Plough, 1st gear, 1100 RPM

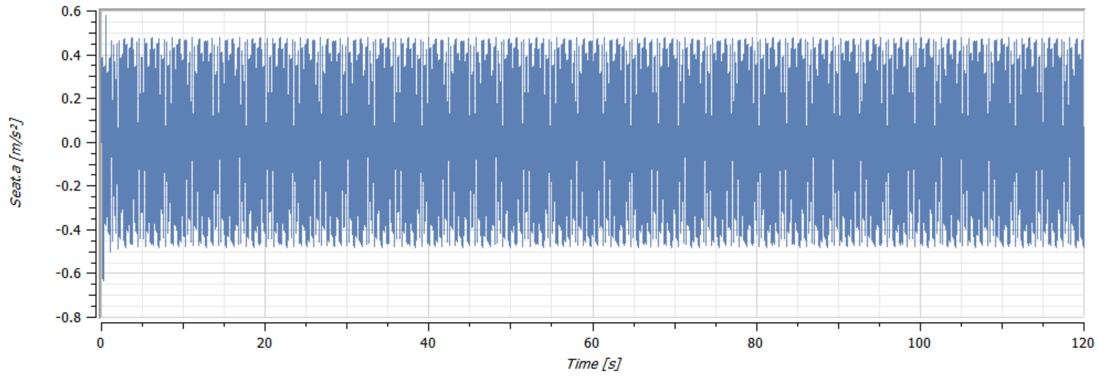

Fig XIII: Seat Acceleration Vs Time Measured under Rough Track with Plough, 2nd gear, 1000 RPM

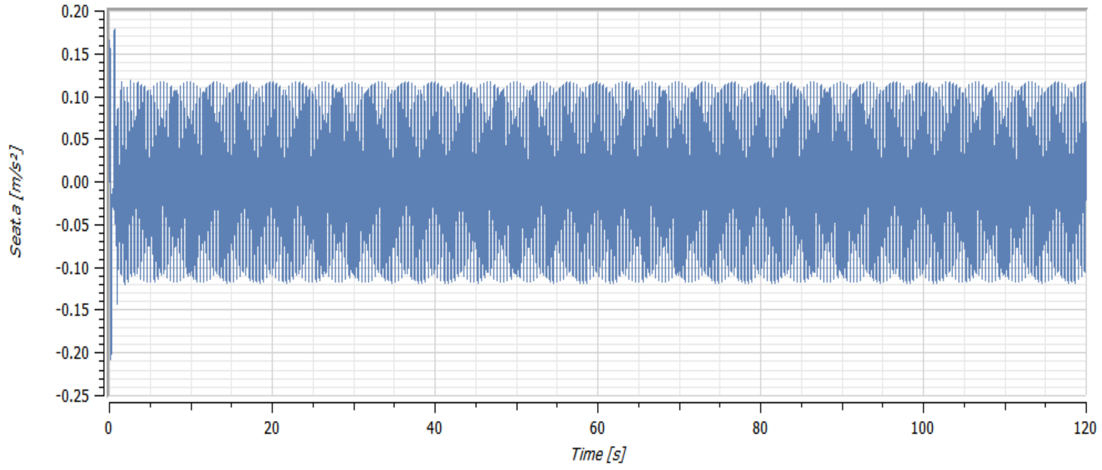

Fig XIV: Seat Acceleration Vs Time Measured under Rough Track without Plough, 1st gear, 1000 RPM

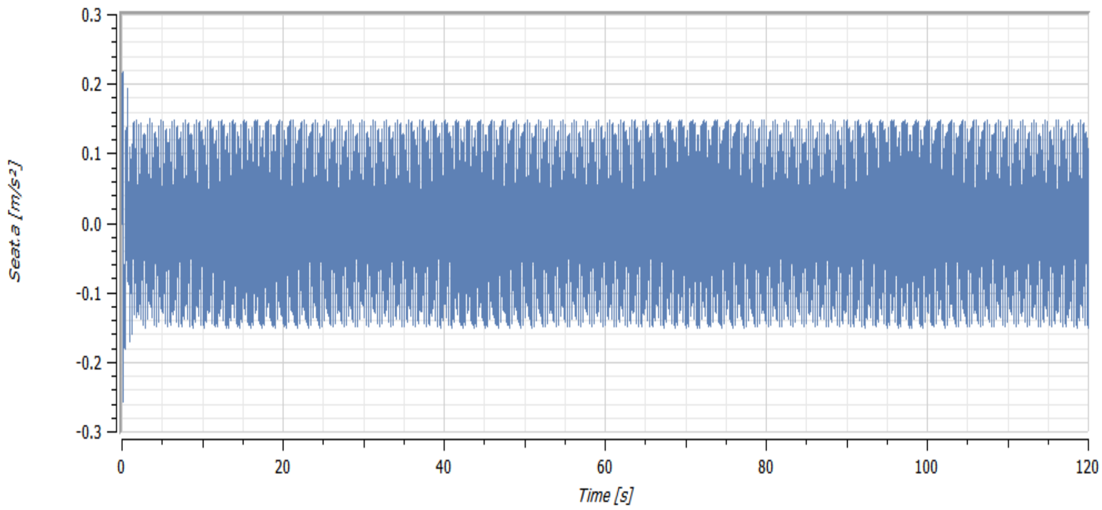

Fig XV: Seat Acceleration Vs Time Measured under Rough Track without Plough, 1st gear, 1100 RPM

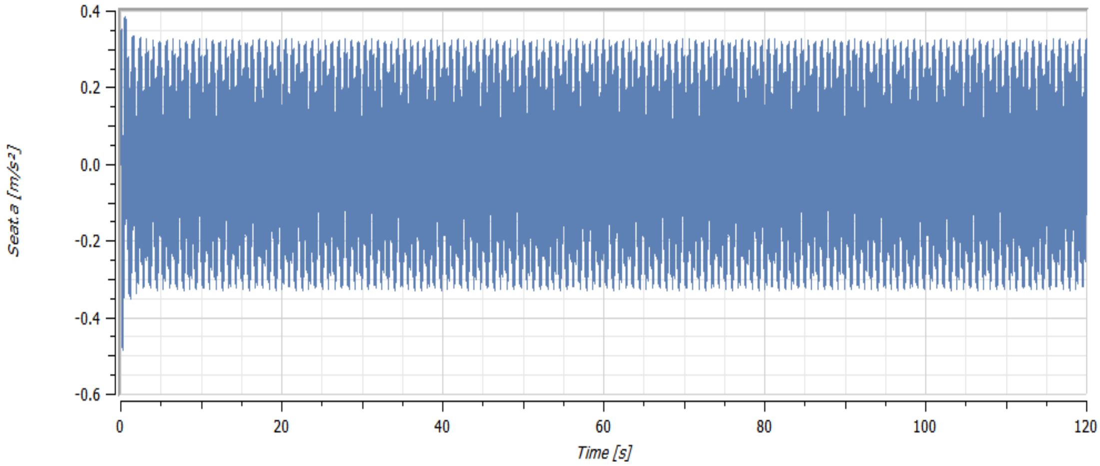

Fig XVI: Seat Acceleration Vs Time Measured under Rough Track without Plough, 2nd gear, 1000 RPM

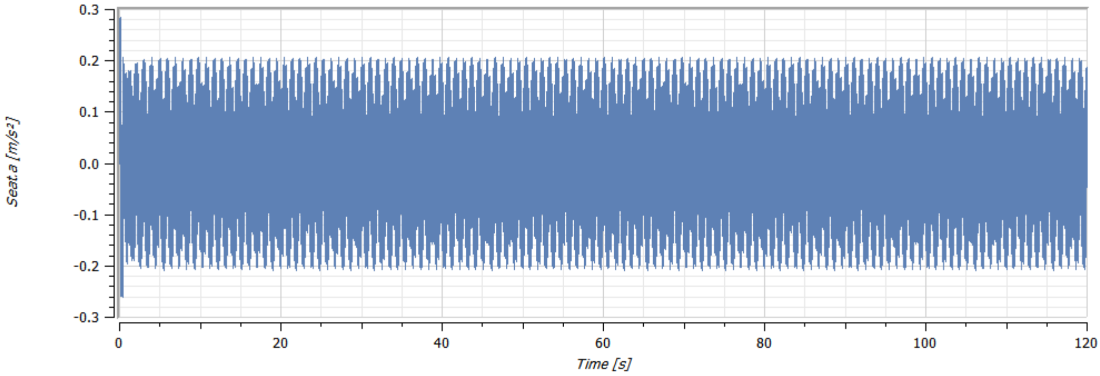

Fig XVII: Seat Acceleration Vs Time Measured under Smooth Track without Plough, 1st gear, 1000 RPM

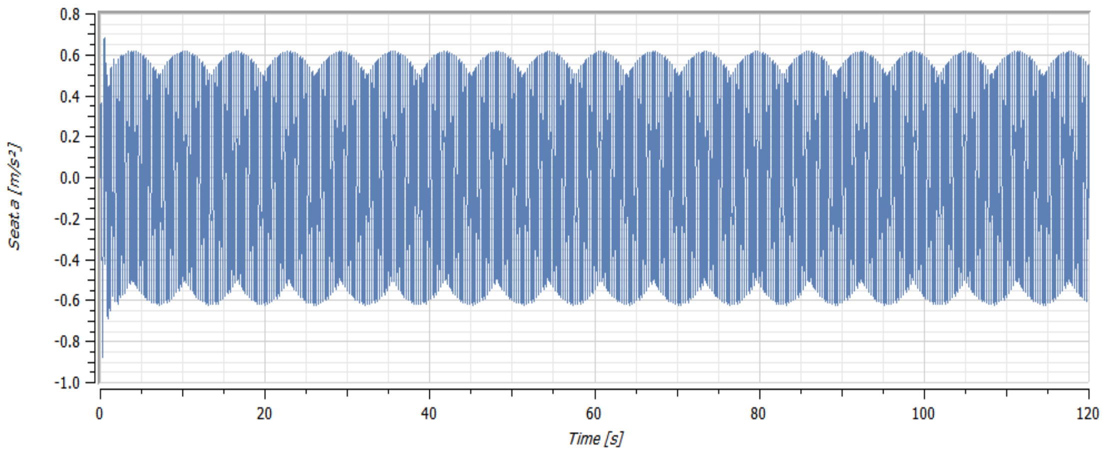

Fig XVIII: Seat Acceleration Vs Time Measured under Smooth Track without Plough, 2nd gear, 1100 RPM

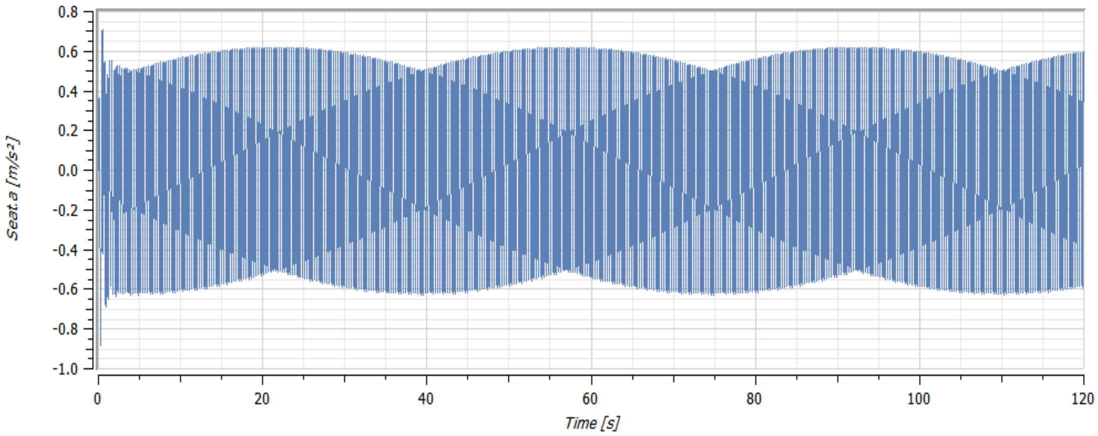

Fig XIX Seat Acceleration Vs Time Measured under Smooth Track without Plough, 3rd gear, 1000 RPM

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