Experimental and Simulated Analysis of Tractor Seat during Tillage

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Abstract

The tractor operators are subjected to negative vibrations because most of the tractors are without any type of suspensions. Because drivers have to work for long hours, these vibrations are hazardous to their health. The present study is focused to analyse the seat of tractor experimentally by collecting the real time data from tractor seat while working in fields with implements during tillage process. Tri-axial accelerometer was used to collect the actual data from seat base and was stored in data collection unit. Based on the real-time data a dynamic model of tractor along with seat was developed in MATLAB-Simulink for further analysis of vibrations. Measurement of vibrations is the basis for developing a good tractor seat with reduced vibrations. Although, there still remained many problems to be solved but this study will provide a good base to work in this direction.

Keywords -Tractor, Implements, Vibration, Modelling and Simulation.

1. INTRODUCTION

Vibrations in the range of 12 Hz affect all organs of operator, whereas vibrations above 12 Hz have a local effect on organs. Low frequency (less than 6 Hz) movement, such as movements of tires over an irregular road, can cause body resonance [1]. Biodynamic research has shown that prolonged exposure to vibrations increases the risk on health [2].

Many characteristics of mechanical vibrations strongly influence the operators. These characteristics include the direction of vibrations, point of contact with the body, frequency, and duration [3, 4, and 5]. If external force frequency matches with natural frequency, resonance occurs [6] due to which undesirable oscillations of large amplitude are produced. When a system is disturbed and can vibrate on its own, the natural frequency is the frequency of the vibrations without damping or external force [7, 8]. If damping is present, it is referred to as the system's damped natural frequency. Most seats exhibit low-frequency resonance, resulting in greater magnitude of vertical vibrations on the seat [10, 11].

MATERIALS AND METHODS

The tractor seat was used to collect real-time data. The experiment was conducted for two different implements. First time data was taken from tractor seat when the tractor was moving with plough and second time disc harrow (Figure 1) was used as implement for tillage process.

Figure 1. Tractor with Chisel Plough and Disc Harrow

During both cases vibrations were measured with tri-axial piezoelectric accelerometer (Figure 2). Vibrations measuring accelerometer was fitted on the seat base of test tractor to measure vibrations on seat of tractor. The frequency sensitivity range of an accelerometer was 2-8000 Hz. Every time, acceleration levels were recorded simultaneously in three perpendicular directions, X longitudinal (pitching), Y lateral (rolling), and Z vertical (bouncing/heave) on the base of the tractor seat.

Figure 2. Location of accelerometer on the base of tractor seat.

A vibration measurement setup in the X, Y, and Z directions was created (Figure 3). At a frequency of 10 Hz, the vibration measuring accelerometer could quantify the vibration of the X, Y, and Z axes

Figure 3. Tri-axial Sensor and Data Acquisition unit setup used for the measurement of vibrations

The tractor was made two runs at two constant speed during both cases and the real time data was recorded separately for each case along horizontal, transverse and vertical directions.

2. DATA ANALYSIS

Accelerations measured are shown in Figure 4 in horizontal (x), transverse (y) and vertical (z) direction. The speed of tractor was kept constant at 2.30 Km/hr. As the Figure shows when tractor was made to work with tillage implements (chisel plough) the measured RMS values of acceleration was highest in transverse (y) direction and lowest in horizontal (x) direction and the vertical (z) direction values lies in between that of lateral (y) direction and longitudinal (x) direction.

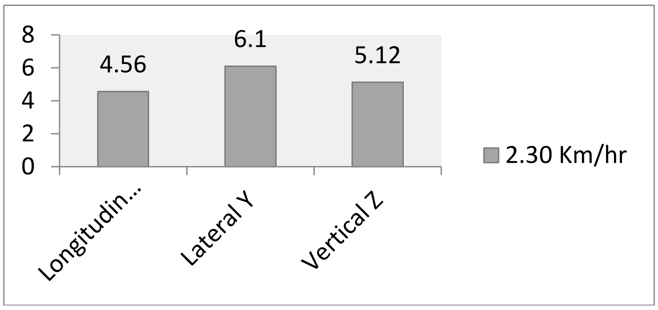


Figure 4. RMS acceleration in all three directions at tractor’s seat with implements

3. IMPLEMENTATION- SIMULATED MODEL OF TRACTOR

A full tractor vibration model along with implements was simulated. The tractor-implement structure was represented by the characteristics like MOI and mass of tractor with implements. Figure 5 depicts the dynamic model of the tractor and implement. Due to surface roughness, the tractor's front and rear tyres experience displacement excitations qfz(t) and qrz(t), respectively. A displacement with respect to the centre of mass is introduced by the change in amplitudes of qfz(t) and qrz(t). Roll, yaw, and pitch movements of the tractor were introduced as a result of this displacement with respect to the mass centre. Rough road force between front axle and rear axle with a time lag of q_{fz}(t)and q_{rz}(t)of the tractor-implement system at an instant i can be expressed as

$$q_{fzi}(t) = q_{rzi}(t + \tau) \qquad (1)$$

Where q_{fzi}(t) and q_{rzi}(t) are displacement excitations, Ʈ is time lag and can be calculated as

$$\tau = \frac{l_{bf} + l_{br}}{v} \qquad (2)$$

Where lbf is the distance between the chassis and the front axle, and lbr is the distance between the chassis and the rear axle. v represents the velocity of the tractor-implement system.

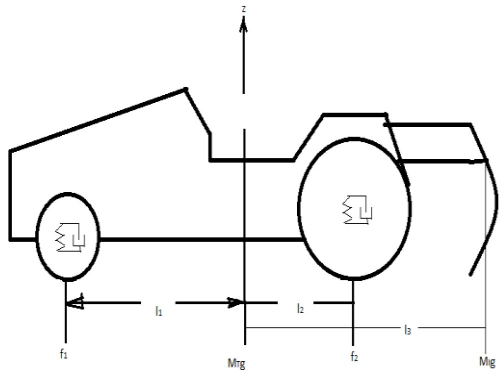


Figure 5. Schematic diagram of Tractor Implement System

The motion equations can be written as follows:

$$(M_T + M_I)\ddot{z} = f_1 + f_2 - (M_T + M_I)g \tag{3}$$

Wheel displacement can be calculated as

$$z_1 = z + \left(l_1 + l_3 \frac{M_I}{M_T + M_I}\right) \tag{4}$$

$$z_2 = z + \left(l_2 - l_3 \frac{M_I}{M_T + M_I}\right) \tag{5}$$

Equations (3) and (4) can be used to calculate the effect of implement on wheel displacement. So M_T will be used to express the effective mass of the tractor and implement. Equations (1) to (4) represents roll, yaw, and pitch effect on wheel displacements. Tractors and other off-road vehicles do not have primary suspension. This complete model is made up of three parts: four wheels, a body, and a vibration model. Figure 6 depicts the physical model of the entire tractor, including the driver's seat.

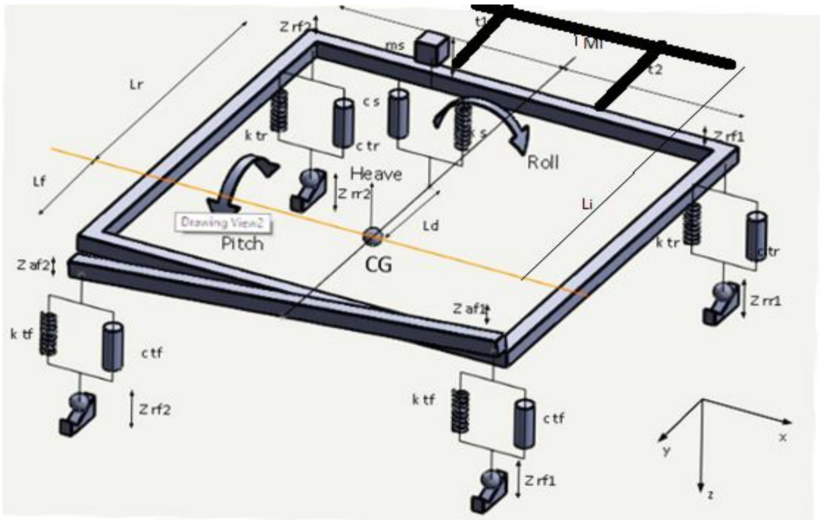


Figure 6. Complete tractor implement vibration model

4.1 Mathematical Full Tractor with Suspended Seat Model

In mathematical modelling, the motion of equations were derived for each of the five degrees of freedom. It is well known fact that the force of spring is proportional to the variation of the length of the spring and that the strength of the shock absorber that is damper force is proportional to variation of the length of damper over time. The vibratory forces transmitted by each wheel at the point of contact between the axle and the wheel can be formulated as follows:

$$k_t(z_a - z_R) + c_t(\dot{z}_a - \dot{z}_R) \quad (6)$$

The mathematical equations for the forces transmitted to chassis are given as:

$$F_{f1} = k_{tf}\left(z_{af1} - z_{Rf1}\right) + c_{tf}\left(\dot{z}_{af1} - \dot{z}_{Rf1}\right) \quad (6a)$$

$$F_{f2} = k_{tf}\left(z_{af2} - z_{Rf2}\right) + c_{tf}\left(\dot{z}_{af2} - \dot{z}_{Rf2}\right) \quad (6b)$$

$$F_{r1} = k_{tr}(z_{ar1} - z_{Rr1}) + c_{tr}(\dot{z}_{ar1} - \dot{z}_{Rr1}) \quad (6c)$$

$$F_{r2} = k_{tr}(z_{ar2} - z_{Rr2}) + c_{tr}(\dot{z}_{ar2} - \dot{z}_{Rr2}) \quad (6d)$$

The mathematical equations of the body of tractor are the following:

(i) for the linear acceleration of the centre of gravity of the tractor in Z direction

$$\ddot{z}_{cg} = -\frac{1}{m_t}\left(F_f + F_{r1} + F_{r2}\right) \quad (7)$$

(ii) for angular accelerations of the tractor body about X-axis which crosses the center of gravity of the tractor

$$\ddot{\theta} = \frac{1}{I_{xxt}}\left(F_f l_f - (F_{r1} + F_{r2})l_r\right) \quad (8)$$

(iii) for angular accelerations of the tractor body about Y axis

$$\ddot{\emptyset} = \frac{1}{I_{yyt}}(F_{r2}t_2 - F_{r1}t_1) \quad (9)$$

The calculation of the tractor body's bouncing, rolling, and pitching motion results in an acceleration of the tractor's driver's seat base. If the distance between the tractor's centre of gravity and the seat base is known, the elements of the seat base can be calculated. The equation of motion of seat base of tractor is given as (Ahmadi, 2014):

$$\ddot{Z}_{seat\ base} = \ddot{Z}_{cg} \quad (9a)$$

$$\ddot{Y}_{seat\ base} = \mathrm{R}\ddot{\phi} \quad (9b)$$

$$\ddot{X}_{seat\ base} = \mathrm{R}\ddot{\theta} \quad (9c)$$

$\ddot{Z}_{seat\ base}, \ddot{Y}_{seat\ base}, \ddot{X}_{seat\ base}$ – acceleration of tractor seat base in z direction, y direction and z direction (m/s^2) respectively.

The accelerations of the points of force application to the tractor body (comprising the acceleration of the pivot point of front axle and acceleration of the joints of the tractor rear wheels of the tractor body) may be computed as following (Sarami 2009):

$$\ddot{z}_{afc} = \ddot{z}_{cg} - l_f\ddot{\theta} \quad (10)$$

$$\ddot{z}_{ar_1} = \ddot{z}_{cg} - l_r\ddot{\theta} + t_1\ddot{\phi} \quad (11)$$

$$\ddot{z}_{ar2} = \ddot{z}_{cg} - l_r\ddot{\theta} - t_1\ddot{\phi} \quad (12)$$

The accelerations in linear direction of the joints between front axle and front wheels are indicated as follows:

$$\ddot{\alpha} = \frac{1}{I_{xxa}}\left(F_{f1}t_1 + F_{f2}t_2\right) \quad (13)$$

Finally, the mathematical equations for the accelerations in linear of the axles of front wheel joints can be computed as follows:

$$\ddot{z}_{af_1} = \ddot{z}_{afc} + t_1\ddot{\alpha} \quad (14)$$

$$\ddot{z}_{af2} = \ddot{z}_{afc} + t_2\ddot{\alpha} \quad (15)$$

4. FULL TRACTOR MODELLING USING SIMULINK

Table 1 Full tractor vibration model parameters.

Notations	Description	Values	Units
k_{tf}	stiffness of tractor front tire	90,000	N/m
c_{tf}	damping of tractor front tire	1,000	N-s/m
k_{tr}	stiffness of tractor rear tire	90,000	N/m
c_{tr}	damping of tractor rear tire	2,000	N-s/m
m_t	Tractor weight	2,800	Kg

t_1	Tractor center of gravity from left portion of chassis	0.87	M
t_2	Tractor centre of gravity from right portion of chassis	0.87	M
l_r	Distance of center c.g. of tractor from rear portion of chassis	0.91	M
l_f	Distance of c.g of tractor from front portion of chassis	1.35	M
I_{xxt}	Tractor MOI about the x axis	600	kg-m^2
I_{yyt}	MOI of tractor about y axis	2,000	kg-m^2
I_{xxa}	Tractor front axle MOI about pivot point	5	kg-m^2
m_s	Mass of seat	35	Kg
l_d	Seat's longitudinal distance from the centre of gravity	0.69	M
k_s	stiffness of seat	8000	N/m
c_s	Damping of seat	130	N-s/m

Figure 7 illustrates a MATLAB-Simulink computer model of the entire tractor.

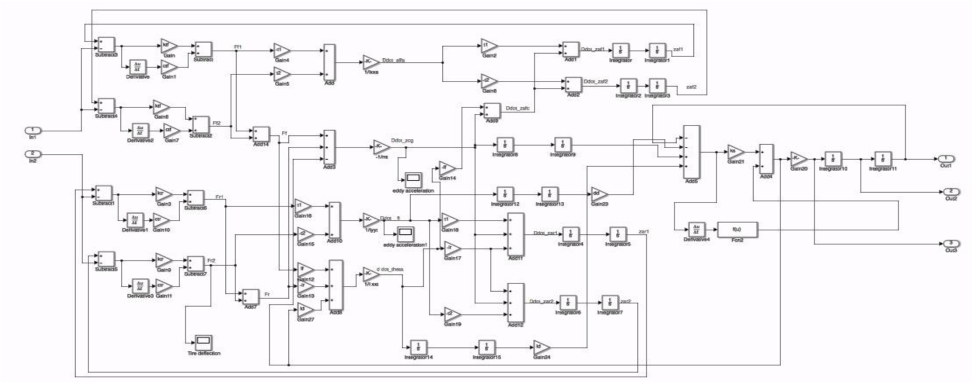
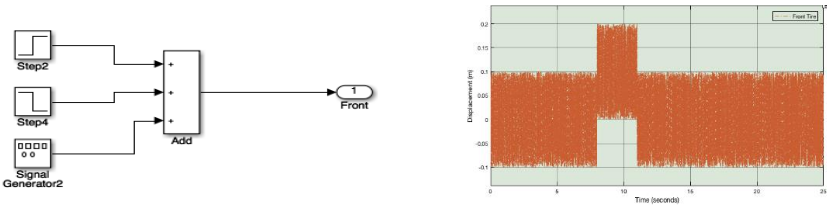


Figure 7. Full Tractor-Implement Simulink Model

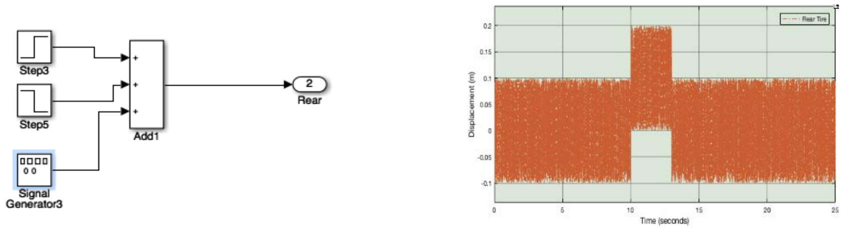
The source block or input is the signal generator representing the sine input with 0.01m amplitude and 35 Hz frequency and step function with the height of 1 unit up at 5 sec and 1 unit down at 7 sec for front wheel were selected. The steps up and down for the rear wheel took 8 and 10 seconds, respectively. The road inputs were applied to the front and rear wheels of the modelled tractor at times t0 and t1, respectively. The time intervals t0 and t1 are considered in relation to the distance between the front and back axles. (Wheel Base), WB, and the speed (v) of the tractor as:

$$t_1 - t_2 = \frac{Wheel\ Base\ (WB)}{Velocity\ of\ tractor\ (v)} \qquad (16)$$

Figure 8 (a) and (b) show the developed Simulink model of disturbance input for the front and rear wheels, respectively.



(a)



(b)

Figure 8. Road Input (a) Front wheels (b) Rear wheels

5.1 Simulated Dynamic Tractor-Implement Model Validation

In the quantitative comparison, as the input displacement functions, a step function was chosen. To simulate a moving tractor at 3.5 km/h and a wheel base of 1970 mm (as for the test tractor) over a step-like hurdle, the rear wheels were delayed for two seconds as follows:

$$t = \frac{1970(mm)}{3.5(\frac{km}{hr})} = \frac{1.97}{0.972(m/s)} = 2.026s \qquad (17)$$

Based on the time domain results, a comparison of the simulation acceleration and the real test acceleration was performed using Root Mean Square (RMS) values.

5.2 Comparison of Simulated and Experimental Results

The modified inputs resulted in RMS errors of 1%, 2.2%, and 5.2% in the X, Y, and Z directions, respectively. As a result, the modified model demonstrated the model's dependability in simulating the seat base acceleration caused by a tractor moving over a step-like random irregularity. The results of the modified model obtained in this investigation were compared with the calculation of the RMS value of acceleration and the estimation of the simulation error used by other researchers to validate the developed vibration model. The modified model produced excellent results, with simulation errors of 1%, 2.2%, and 5.2%, which were found to be lower than 10% of other studies' developed models. The results indicate that the computer model developed could accurately model the induced vibrations caused by the field irregularity transmitted to the tractor seat. As a result, this model can be used to simulate the test required for the design of the tractor seat suspension controller.

5. RESULTS AND DISCUSSION

The tractor parameter values were entered into a MATLAB Simulink model of the tractor with implements, and the RMS values of the simulated model were compared to the experimental RMS values. To reduce the percentage error in measured and simulated values, some parameters were redefined. The modified inputs resulted in RMS errors of 5.2%, 1%, and 2.2% in the X, Y, and Z directions, respectively. As a result, the modified model demonstrated that the model is reliable in simulating seat acceleration caused by a tractor moving over random irregularity. The results of the modified model obtained in this investigation were compared with the calculation of the RMS value of acceleration and the simulation error used by other researchers to validate the developed vibration model. The modified model produced excellent results, with simulation errors of 5.2%, 1%, and 2.2%, which were found to be less than 10% of other studies' developed models.

6. CONCLUSION

Because the differences between simulated and measured values are less than 10%, this model can accurately predict tractor seat vibrations during tillage. Because the model is comparable to the actual tractor-implement system, a variety of other studies can be conducted by changing the input parameters such as speed and type of tillage implement in the simulated model without actually performing on the tractor. Further research into designing a controller to reduce tractor seat vibrations can be conducted using this model.

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