Performance and Analysis for the Vibration Reduction of Steering Wheel Assembly of Agricultural Tractor

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Abstract— One of the most important aspects of agricultural tractor operator comfort is the absence of steering wheel vibration. Tractor vibration control concerns are being investigated as part of this project's management remit. When trying to pin down the root of a problem, the planning and study of the controlling system is crucial. The steering vibration of a Power trac 439 DS was studied. To reduce tremors, the tuned mass damper idea is used. Testing is performed on a variety of damping materials to see which is most effective in reducing vibration, and the outcomes are dissected in MATLAB Simulink utilizing a two-level of-opportunity model that is exposed to an essential feeling.

Isolating the tractor's steering box from the wheel improves the vehicle's ride quality and makes the driver more comfortable by decreasing hand-arm vibration syndrome (HAVS). A damper may be used to lessen the amount of vibration felt by the driver at the agricultural tractor's steering wheel.

Index Terms—Steering Wheel, Vibration, Frequency, Damper

I. INTRODUCTION

Today, the safety and well-being of the driver are the primary concerns when designing a vehicle. Previously, the convenience of the tractor driver was not prioritised. However, times have changed, and modern tractor operators want the same conveniences as their city-dwelling counterparts. Tractors may now be used in many different settings. The vibrations are then transferred to the driver's hands through the steering box and the wheel. In most cases, operators were exposed to two distinct forms of vibration:

Vibrations felt throughout the body may be adjusted by moving the vehicle's seat, floor, or pedals.

• Vibrations felt through the hands when turning a steering wheel or other controls.

[3] Vibrations that are too strong may be harmful to human health. Hand-arm vibration disorder is a catch-all term for a group of symptoms and diseases. The purpose of this research is to show how a damper and MATLAB simulation may lessen tractor vibration.

II. LITERATURE REVIEW

An exhaustive literature review was performed to comprehend the prior efforts in the same area. Piezo crystal material is used to dampen both the vibrations felt in the user's hands and the body as a whole [1] by V.K. Tiwari and K.P. Vidhu. There are two isolators made of piezoelectric material utilised for this. Farm tractor comfort may be improved with the use of hydro-pneumatic and semi-dynamic taxi suspension, as shown by Kyuhyun sim and Ji won Yoon [2]. Kandavel Farm tractors often have a shaky steering wheel due to the vibrations caused by the engine and road; a systematic strategy is shown by Govari Shankar and Shrikant Samant [3]. They used six sigma design principles to lessen vibrations. A method for measuring the damping of vibration was developed by Anant Sakthivel and Rakesh B.Verma [4]. Tractors of many types were the subject of an extensive (40-50Kw). The simulation was carried out in ADAMS, and two different types of dampers—radial and axial—were utilised.

III. EXPERIMENTAL PROCEDURE

There are three known contributors to hand-felt vibrations. That's why we separated the vertical, longitudinal, and transverse axes (X, Y, and Z) into their own sets of estimates. Since the strength of vibration is highest in the vertical direction, it's the only one that gets measured when analysing a steering wheel's vibration. On a farm, they stopped and began the tractor. All calculations were performed in neutral to provide the most accurate results. At first, the measurements were taken while the engine speed was gradually raised from 750 rpm (idling) to 3000 rpm (maximum). [4]

This study aimed to describe the vibration presentation level experienced by the driver's hands as a result of vibrations transferred via the driver's arms from the tractor's steering wheel. It was the farm tractor that did the exploring. Hand-arm vibration levels were evaluated in two different driving scenarios:

- At Neutral Condition
- At Running Condition

ISO 5349-2001 specifications were used for the estimating system. All three axes of estimation occurred concurrently for these levels.

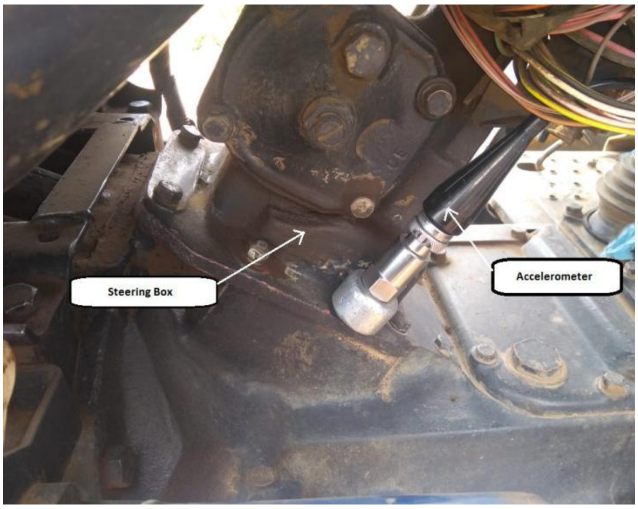

Fig.1.Accelerometer mounted on steering box

Fig.2.a steering-wheel-mounted accelerometer

IV. FREQUENCY WEIGHTING AND CALCULATION

A fundamental measurement for depicting the power of the vibration felt by the driver is the root-mean-square (rms) recurrence weighted speed increase communicated in meters each second squared. Utilizing the r.m.s. speed not entirely set in stone from the 33% octave band examination, we can get the same recurrence weighted r.m.s. speed increase up ah, w. Accomplishing this includes:

$$a_{\mathrm{h,\,w}} = \left[\sum_{j=1}^{n}(W_{\mathrm{hi}}a_{\mathrm{hi}})^2\right]^{1/2} \quad (1)$$

ISO 5349-1:2001 [5] specifies a weighting factor, for the ith 33% octave band and an associated r.m.s. acceleration, ahi, for evaluating vibration exposure across all three axes. The root-mean-square of the three components is equal to the total vibrational energy, which we will refer to as ahv.

$$a_{hv} = \sqrt{a_{hwx}^2 + a_{hwy}^2 + a_{hwz}^2} \quad (2)$$

wherea_{hwx}, a_{hwy} and a_{hwz} Speed increase in the X, Y, and Z tomahawks as a component of recurrence. total vibrational pressure and term of openness. Day to day "openness time" alludes to the aggregate timeframe that the hands are presented to vibration over the span of the average working day. For the sake of this discussion, the average daily vibration level will be defined as the sum of the energy-weighted vibrations felt over the duration of 8 hours.

$$A(8) = a_{\mathrm{hv}}\sqrt{\frac{T}{T_0}} \quad (3)$$

T is the aggregate everyday openness in seconds over the ahv, and T0 is the reference season of 8 hours (28800 seconds). In order to determine one's daily vibration exposure while working in an environment subject to several activities producing various vibration magnitudes, one must satisfy the following criterion: A (8).

$$A(8) = \sqrt{\frac{1}{T_0}\sum_{i=1}^{n} a_{\mathrm{hvi}}{}^2 . T_{\mathrm{i}}} \quad (4)$$

Where is the complete number of vibration openings, T_{iis} the time in a moment or two, and $_{ahv}$ is the all out vibration an incentive for the ith activity. [4]

Where ahv is the complete vibrational energy consumed during the ith activity, n is the absolute number of vibrational openings, and Tiis the time in seconds for the ith activity. [4]

V. MATHEMATICAL MODEL

Hub and spiral dampers are mounted on the motor connector plate and the transmission case, separately, to downplay vibrations. A sum of four equal dampers were utilized, every one of which was addressed by a spring with solidness k1 and a damper with damping coefficient c1. An elastic damping cushion, addressed by a couple of springs with firmness k2 and a damper with damping coefficient c2, is set between the guiding box and the segment base. That's why in Figure 3 we see a 2-degrees-of-freedom vibration model for this system. Since the input to this system is the engine's vibrational motion, we may classify it as an excitation issue with a support structure. Thomson's transmissibility formula for a 1-DOF support motion system was used as a starting point for this derivation. This may be thought of as a support motion issue, given the input to the system is the vibration motion of the engine. This derivation is based on Thomson's transmissibility statement for a 1DOF support motion system. In order to calculate how much energy is transferred from the engine to the wheel, we may use the following expression:

For simplicity, we will refer to the displacement of the steering box as x1, and the harmonic motion of the base support as y. The distance the steering column and wheels are moved is denoted by x2. The masses of the steering column-wheel assembly (m2) and the steering box (m1) are denoted by the symbols m1 and m2. Below is a schematic depicting the free body dynamics of two particles with masses m1 and m2.

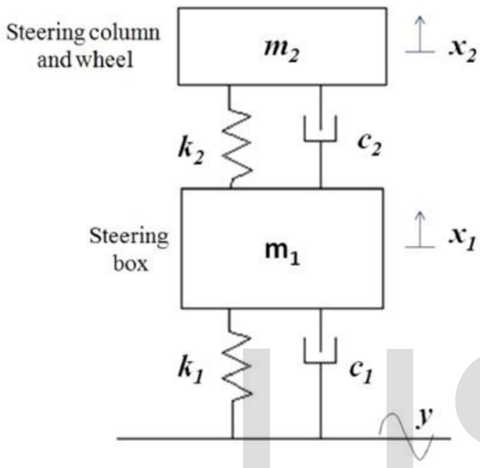

Fig.3.Motion Support with Two DOF

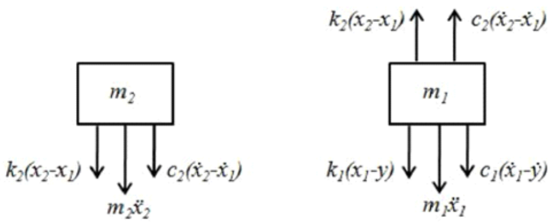

Fig. 4. Masses m1 and m2 in free-body diagrams

m2's motion equation is denoted by Eq (5),

$$m_2\ddot{x}_2 + c_1(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) = 0 \qquad (5)$$

Similarly, Eq. (6) and Eq. (7) may be used to get the equation of motion for m1 (7),

$$m_1\ddot{x}_1 + k_1(x_1 - y) + c_1(\dot{x}_1 - y) = k_2(x_2 - x_1) + c_1(\dot{x}_2 - \dot{x}_1) \qquad (6)$$

$$m_1\ddot{x}_1 - k_2(x_2 - x_1) - c_2(\ddot{x}_2 - \dot{x}_1) + k_1x_1 + c_1\dot{x}_1 = k_1y + c_1\dot{y} \qquad (7)$$

Combining equations (5), (6) and (7) into matrix form,

$$\begin{pmatrix} m_1 & 0 \\ 0 & m_2 \end{pmatrix}\begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{Bmatrix} + \begin{pmatrix} c_1 + c_2 & -c_1 \\ -c_2 & c_2 \end{pmatrix}\begin{Bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{Bmatrix} + \begin{pmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{pmatrix}\begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix} = \begin{Bmatrix} k_1y + c_1\dot{y} \\ 0 \end{Bmatrix} \qquad (8)$$

Assuming,

$y = Y e^{i\omega t}$, $x_1 = X_1e^{i(\omega t-(\varphi))}$ and $x_2 = X_2e^{i(\omega t-(\varphi))}$

$$-\omega^2\begin{pmatrix} m_1 & 0 \\ 0 & m_2 \end{pmatrix}\begin{Bmatrix} X_1 \\ X_2 \end{Bmatrix} + i\omega\begin{pmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{pmatrix}\begin{Bmatrix} X_1 \\ X_2 \end{Bmatrix} + \begin{pmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{pmatrix}\begin{Bmatrix} X_1 \\ X_2 \end{Bmatrix} = \begin{Bmatrix} k_1Y + i\omega c_1Y \\ 0 \end{Bmatrix} \qquad (9)$$

The Y-intercept is used to divide the two sides.

$$\begin{pmatrix} k_1 + k_2 - m_1\omega^2 + i\omega(c_1 + c_2) & -k_2 - i\omega c_2 \\ -k_2 - i\omega c_2 & k_2 - m_2\omega^2 + i\omega c_2 \end{pmatrix}\begin{Bmatrix} X_1/Y \\ X_2/Y \end{Bmatrix} = \begin{Bmatrix} k_1 + i\omega c_1 \\ 0 \end{Bmatrix} \qquad (10)$$

In order to find X2/Y,

$$\frac{X_2}{Y} = \frac{\begin{vmatrix} k_1 + k_2 - m_1\omega^2 + i\omega c_1 + c_2) & k_2 + i\omega c_2 \\ -k_2 - i\omega c_2 & 0 \end{vmatrix}}{\begin{vmatrix} k_1 + k_2 - m_1\omega^2 + i\omega(c_1 + c_2) & -k_2 - i\omega c_2 \\ -k_2 - i\omega c_2 & k_2 - m_2\omega^2 + i\omega c_2 \end{vmatrix}} \qquad (11)$$

$$\left|\frac{X_2}{Y}\right| = \sqrt{\frac{\left(k_1k_2 - c_1c_2\omega^2\right)^2 + \left(\left(k_1c_2 + k_2c_1\right)\omega\right)^2}{\left(k_1k_2 - \left(m_1k_2 + m_2k_1 + m_2k_2 + c_1c_2\right)\omega^2 + m_1m_2\omega^4\right)^2 + \left(\left(k_2c_1 + k_1c_2\right)\omega - \left(m_1c_2 + m_2c_1 + m_2c_2\right)\omega^3\right)^2}} \qquad (12)$$

These recipes might be utilized to decide the worth of the contagiousness from the motor to the controlling wheel. Therefore, by plugging in a range of values for the stiffness and damping coefficients, we can calculate the amplitude at the wheel. [4]

VI. ANALYSIS AND TRACTOR TESTING

The data was analysed in two phases:

- Stage 1: Utilize an FFT analyzer to determine the precise amount of vibration being generated by the steering wheel and the gearbox.
- Stage 2: Produce a known level of vibration using an electrodynamic shaker machine and compare it to the measured value when a damper is present. The first step included installing the accelerometer on the steering box and wheel and analysing the data using an FFT analyzer.

  Afterwards, an electrodynamic shaker machine was used for the second phase. The peak and RMS values of the estimated velocities and frequencies from the selected tractor were analysed in this step, and the results were used to control the electrodynamic shaker. Once the dampers were applied, the final values were measured using an FFT analyzer.

  In the table below, we compare the results of measurements taken with and without a damper.

Position and Direction	Type	Tractor ideal condition (Without damper)	Tractor running condition (without damper)	Tractor ideal condition (with damper)	Tractor running condition (with damper)
Steering wheel (X axis)	A (m/s^2)	6.27	5.19	3.59	3.31
	V (mm/s)	6.05	29.52	4.20	17.84
	D (µm)	17.01	59.62	12.56	29.15
Steering box (X axis)	A (m/s^2)	15.89	11.67	3.57	6.36
	V (mm/s)	4.20	24.73	3.15	13.59
	D (µm)	12.56	61.41	7.15	35.03
Steering box (Y axis)	A (m/s^2)	8.63	14.42	1.71	5.88
	V (mm/s)	1.01	7.44	0.88	4.14
	D (µm)	2.83	27.45	2.00	11.76
Steering box (Z axis)	A (m/s^2)	3.43	12.45	1.02	4.99
	V (mm/s)	0.59	14.23	0.37	8.40
	D (µm)	1.84	29.89	1.01	13.15

Using the root-mean-squared velocity value seems arbitrary. Given that it is the average of all square roots, its square root is, the r.m.s. speed is consistently more noteworthy than nothing. Obviously, this number must be positive. This is why the average number was selected for velocity.

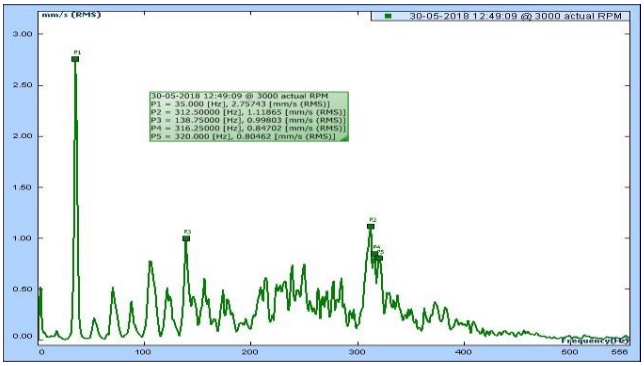

Fig.5. While the car is in neutral, you may read behind the wheel (without damper)

At the point when the vehicle is in unbiased, the vertical component of the RMS velocity is shown in Figure 5. The highest root-mean-squared speeds measured were 6.05 m/s.

As can be seen in Figure 6, the vehicle's neutral state results in a vertical velocity RMS value of.

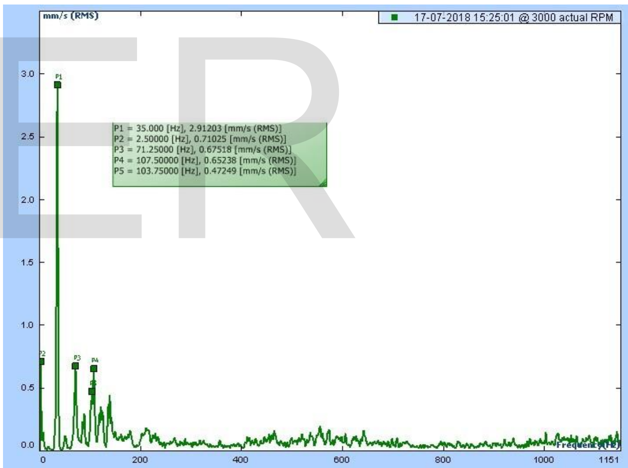

Fig. 6. While the vehicle is in neutral, you may read behind the wheel. (with damper)

The greatest root-mean-squared speeds measured were 4.20 m/s. While compared to the vibration levels experienced when steering with no damper, the levels experienced with the damper significantly reduce the vibrations felt in the hands.

VII. SIMULATIONS USING MATLAB

As specified in the MATLAB input file, the damper's mass, stiffness, and design processes make use of damping coefficients.

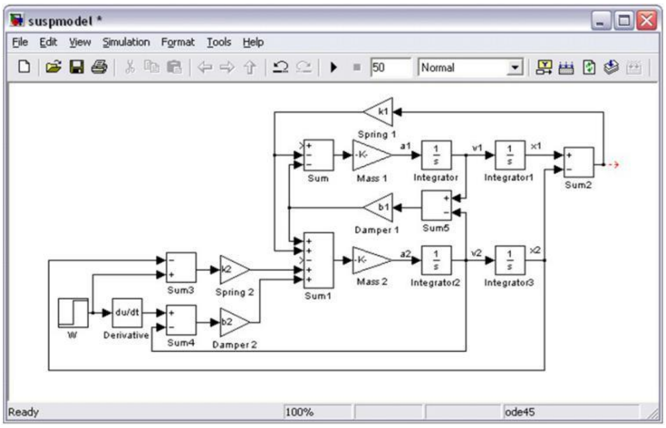

Fig.7. Use of MATLAB for Simulations

In the table below, you can see the settings that the MATLAB model for damper concepts used to get those results.

Input	Value
The m1 mass of the control module (kg)	4 kg
How much m2 are in the steering column and gearbox? (kg)	11 kg
Cushioning pad stiffness, k1 (kg)	1700 N/m
k2 is the damping material's stiffness (kg)	550 N/m
Coefficient of damping for padding, c1	1Nm/s
Damping coefficient of cushioning pads, c_2	0.79Nm/s

After measuring the vibration, the findings were compared to those obtained.

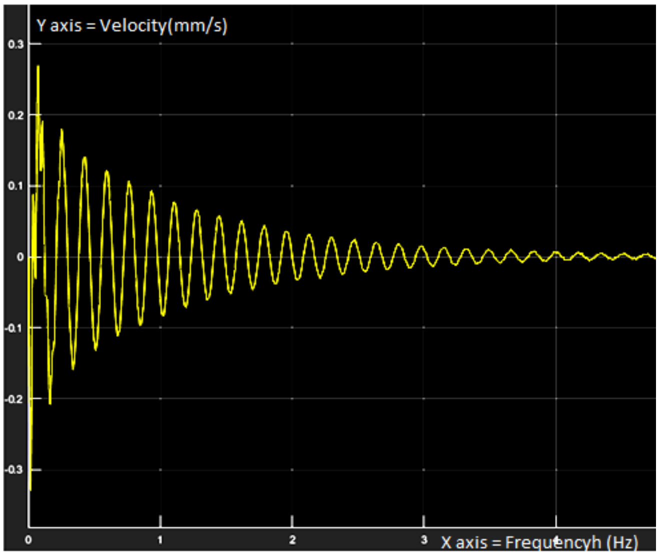

Fig.8.MATLAB Simulation Output

VIII. VALIDATION OF RESULTS

The FFT spectrum analysis graph shows a maximum displacement of 0.2815 mm, but the MATLAB value for displacement is 0.27 mm.

IX. EXPERIMENTAL RESULTS

Acceleration and frequency spectrum for the selected operating circumstances were collected; both the steering wheel and the steering box were noticeably vibrating. Neutrally, the steering wheel has an acceleration of around 6.27 m/s2, but after isolation, that value drops to roughly 3.59 m/s2. The acceleration value of the steering box was also observed to reduce from 11.67 m/s2 under normal operating conditions to 6.36 m/s2 when isolation was applied, for a total reduction in acceleration of roughly 5.31 m/s2.

X. CONCLUSION

The peak acceleration is reduced by 54.49 percent and the daily vibration exposure is cut in half (Steering box and steering wheel, respectively) using the damper. Steering wheel vibration was predicted with an accuracy of 85-90% using the created 2-DOF mathematical model and MATLAB simulation.

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