

Study of Vibration from Steering Wheel of an Agricultural Tractor

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ABSTRACT

Steering wheel vibration is one of the major factors in determining the operator comfort in agricultural tractors. Main sources of steering wheel vibration are engine imbalance, resonance of steering system, lesser damping, road / field induced vibration, etc. The European Council directive 2002/44/EC recommendations are taken as guidelines in this work.

Steering vibration study was conducted on various tractor models (40 - 50 kW range) and one tractor was identified for improvement. Upon detailed analysis on that particular tractor, it was found that the resonance of steering system with engine excitation is the root cause for excessive vibration. Various methods to reduce vibration due to resonance were considered, such as shifting the natural frequency away from the second order engine frequency and increasing damping coefficient to reduce the vibration amplitude at resonance. Six different concepts were generated and analyzed using the design assessment matrix. Two concepts were selected, namely radial damper and axial damper concepts for further processing. These concepts were tested and a significant reduction in vibration levels was achieved. Axial damper concept offers higher vibration reduction when compared to radial damper concept.

These new design concepts were modeled as a 2 degree of freedom (2-DOF) system with support motion from engine vibration as input. Expressions were derived for vibration transmissibility from engine to steering wheel and coded in MATLAB. A simulation was done in ADAMS software. It was observed that steering vibration predicted by the mathematical model matched well with the measured values. Furthermore, this model could be employed as a tool in selection of optimal suspension parameters for the steering system.

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INTRODUCTION

Passenger comfort is a vital criterion in the present day vehicle design. Until recent times, tractors were considered as performance machines and the operator comfort was not given much importance. Now the scenario has changed and tractor owners also want to have an equal level of comfort, if not more than the passenger vehicles. Tractors generally operate in adverse environmental conditions and an ergonomically inferior tractor could make things worse for the driver. Operator comfort in tractor means many factors like space available for occupant, reach of controls, visibility, noise and vibration, temperature, etc. Out of these, vibration is one factor which not only makes the operator uncomfortable, but also leads to failure of various parts of the tractor. Tractor operators are exposed to two types of vibration: whole body vibration transmitted via seat, floor and foot pedal controls and hand transmitted vibration via steering wheel and hand control knobs.

There are two widely followed strategies for preventing the adverse effects of vibration. They are:

- Vibration control at excitation source itself
- Vibration isolation by taking appropriate action in the transmission path

While it is obvious that controlling the vibration at source is the best possible option, many a times, the vibration reduction objective also comes with other constraints of product design like packaging, design simplicity, product cost, product development lead time, etc. Hence appropriate strategy has to be chosen considering the constraints of product design.

Excessive exposure to hand transmitted vibration can induce disturbances in blood flow in the fingers and neurological functions of the hand and arm [1]. The term "Hand-arm vibration syndrome" (HAVS) is commonly used to refer such disorders like circulatory disorders, bone and joint disorders, neurological disorders, muscle disorders and central nervous system disorders [2]. Higher vibration of

tractor steering wheel may lead to this problem and hence this subject assumes great importance. ISO 5349-1:2001 standard [3] outlines the methodology to correlate the measured vibration from steering wheel to hand-arm vibration syndrome through two parameters namely,

- 8-hour energy equivalent frequency weighted vibration total value and
- Number of years in which 10% of the operators exposed to vibration may develop hand-arm vibration syndrome

This ISO standard has indicated guideline values on vibration. The European Council directive 2002/44/EC [4] on the minimum health and safety requirements regarding the exposure of workers who are likely to be exposed to risks from mechanical vibration during their work recommends daily exposure limit value (ELV) of 5 m/s^2 and daily exposure action value (EAV) of 2.5 m/s^2 for hand-arm vibration. Vibration values specified in this standard are taken as guidelines in this work. In recent developments, tractor manufacturers have been asked to measure and disclose the vibration levels at steering wheel, seat, floor, etc.

The objective of this work is to study the steering wheel vibration in detail and to reduce the same to protect the operator from adverse hand-arm vibration syndrome, develop a mathematical model for steering wheel vibration and to verify the mathematical model with experimental results, so that it will form a base for vibration prediction in future.

A detailed literature survey was conducted to understand the work carried out so far in related fields. Sugita and Asai [5] explained an experimental method for the reduction of steering wheel vibration. Mechanical impedance methods were applied to predict the resonant frequency of the steering system. Tewari and Prasad [6] presented a 3 DOF analytical model for tractor seat suspension system. Even though this is not directly relevant to the present work, the assumptions in the model development are inherited in the present work as both steering wheel and seat are excited primarily by the engine vibration. Dewangan and Tewari [7] explained that the discomfort and pain due to hand transmitted vibration leads to fatigue. As a cascading effect, operators take rest to overcome fatigue which reduces work output and duration of actual work. They have presented the results of vibration transmitted from the handle of a 6.7 kW hand tractor. Gowri Shankar, et al., [8] demonstrated a systematic approach towards reducing the steering wheel vibration of an agricultural tractor. However there was not much work done on the evaluation of vibration parameters as per commonly used ISO 5349:2001 standard and mathematical modeling.

METHODOLOGY

Steering wheel vibration on various tractors in the power range of 40 to 50 kW was compared and one 45 kW tractor was chosen for further study and improvement. Following methodology was adopted in this work.

- Comparison of steering vibration of various tractors in 40 to 50 kW power range.

- Selection of one tractor for further study and improvement
- Detailed experimentation on the tractor and analysis of results
- Identification of primary cause of vibration
- Concept generation, detailed design and proto type making of improved design
- Mathematical modeling of steering wheel vibration and simulation in ADAMS [9] software
- Comparison of present and improved design as per ISO 5349 standard.
- Comparison of computed and measured vibration.

EXPERIMENTAL SETUP

The test setup consists of a vibration transducer, data acquisition front end hardware, personal computer and software for data analysis. In a typical vibration test setup as shown in Figure 1, the accelerometer mounted on the vibrating body converts the acceleration into voltage which is fed to a signal conditioner. Then the signal is filtered and amplified as required and passed on to the personal computer. This signal is properly processed by appropriate software to obtain the output in the required format. The accelerometer used in this experiment is a light weight piezo-electric accelerometer (B&K Endevco Model 751-100 ISOTRAN), with integral electronics. Accelerometers were positioned in one or more of the locations such as engine rear end, steering wheel, steering column base and steering box base. Mounting of accelerometers was done in accordance with ISO 5349-2:2001 standard [10]. A four channel Bruel & Kjaer make digital data acquisition system (B&K Pulse 3560 C) was employed for recording the vibration measurements.

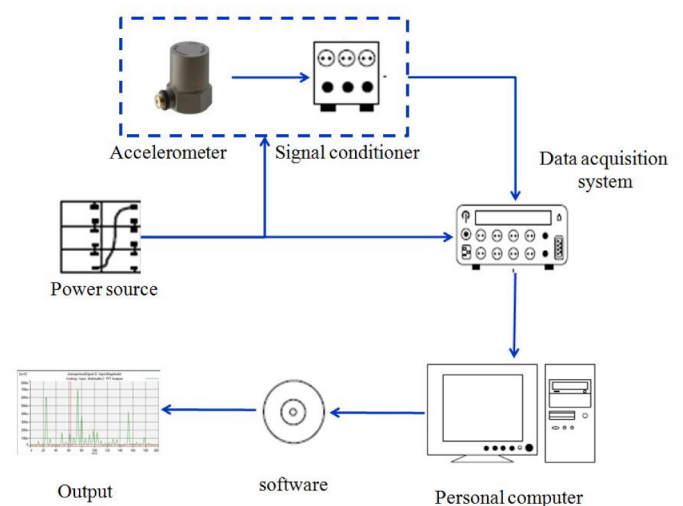


Figure 1. Schematic diagram of vibration test setup

EXPERIMENTATION PROCEDURE

It is known that the vibration entering the hand contains contributions from all three measurement directions. Therefore, measurements were made in all three directions as shown in Figure 2. In this paper, the X_h , Y_h and Z_h axes are termed as vertical, longitudinal and transverse axes respectively. The tractor was parked on a flat concrete surface and the engine was started. Measurements were taken with the gear in neutral position. Initially, the engine speed was increased from idling (750 rpm) to maximum speed (2650 rpm) slowly and steadily over a period of one minute and the measurements were made. The vibration was not significant at low speeds and it became significant only after 1800 rpm. Hence it was decided to take measurements at intervals of 200 rpm in the frequency range where vibration is higher (1800 - 2650 rpm) and to take fewer points below 1800 rpm. This was done considering the scarce availability of testing resources. At each engine speed, vibration was measured for a period of 10 seconds and stored in the computer for further analysis.

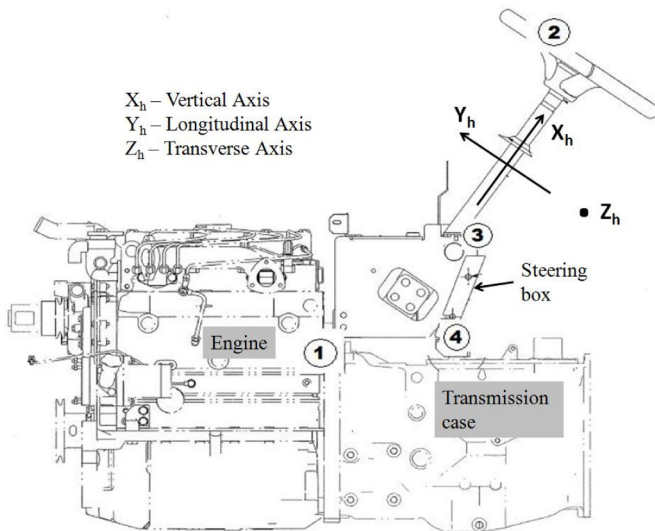


Figure 2. Location of accelerometers on the tractor

FREQUENCY WEIGHTING AND CALCULATIONS

As per ISO 5349-1:2001 recommendations, the most important quantity used to describe the magnitude of the vibration transmitted to the driver's hands is root-mean square (r.m.s.) frequency-weighted acceleration expressed in m/s^2 . The measurement of frequency-weighted acceleration requires the application of a frequency weighting curve (Figure 3) on the measured acceleration. The frequency weighting reflects the assumed importance of different frequencies in causing injury to hand.

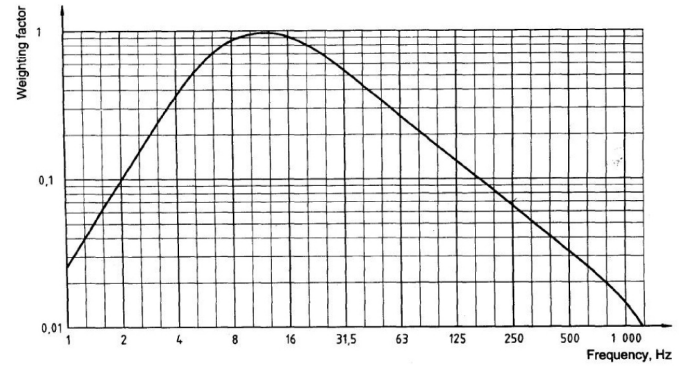


Figure 3. Frequency weighting curve for hand-transmitted vibration [3]

The r.m.s. acceleration values from one-third-octave band analysis can be used to obtain the corresponding frequency-weighted r.m.s. acceleration a_{hw} using the Eq. (1).

$$a_{hw} = \sqrt{\sum_i (W_{hi} a_{hi})^2} \quad (1)$$

where W_{hi} is the weighting factor for i th one-third-octave band and a_{hi} is the r.m.s. acceleration measured in the i th one-third-octave band. The evaluation of vibration exposure in accordance with ISO 5349-1:2001 is based on a quantity that combines all three axes. This is the vibration total value a_{hv} and it is defined as the root-mean-square of the three component values:

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

where a_{hwx} , a_{hwy} and a_{hwz} are the frequency weighted acceleration in x, y and z axes respectively.

The vibration exposure depends on the magnitude of the vibration total value and the duration of the exposure. Daily exposure duration is the total time for which the hands are exposed to vibration during the working day. The daily vibration exposure shall be expressed in terms of the 8-hour energy equivalent frequency-weighted vibration total value as

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

where T is the total daily duration of the exposure expressed in seconds to the a_{hv} and T_0 is the reference duration of 8 hours (28800 seconds).

If the work is such that the total daily vibration exposure consists of several operations with different vibration magnitudes, then the daily vibration exposure, $A(8)$ shall be obtained using Eq. (4).

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

where a_{hvi} is the vibration total value for the i th operation, n is the number of individual vibration exposures and T_i is the duration of the i th operation in seconds.

ISO 5349-1:2001 indicates that the time taken for a specific amount of total daily vibration to produce hand-arm vibration syndrome in 10% of persons exposed to vibration is given by Eq. (5).

$$D_y = 31.8(A(8))^{-1.06} \quad (5)$$

where $A(8)$ is the total daily vibration exposure and D_y is the number of years in which 10% of the persons exposed to vibration will develop hand-arm vibration syndrome.

BASE TRACTOR TESTING AND ANALYSIS

The chosen base tractor was tested for steering wheel vibration. The frequency spectra of the vibration measurements indicate that there exists a distinct peak in acceleration response at a frequency corresponding to second harmonic of engine speed. For example, in the steering wheel vertical vibration measurement at 1800 rpm, a peak was obtained at 60 Hz. The engine frequency is obtained by dividing the engine speed in rpm by 60. As the four stroke engine expels exhaust once in every two cycles, the number of cylinders is divided by 2 and the same is multiplied by engine frequency to obtain engine firing frequency. The second order excitation frequency (engine firing frequency of four cylinder engine) of the engine at 1800 rpm speed is $(1800/60) \times (4/2) = 60$ Hz. This phenomenon of peak occurring at engine second order frequency happened in all the measurements done.

Hence, as a preliminary measure, the second order acceleration values were analyzed in all three directions at each speed. However, other frequency components were also taken into account while calculating the $A(8)$ vibration total value and frequency weighting later. The steering wheel vibration is plotted as a function of the engine speed for all three axes and illustrated in Figures 4, 5, 6. All the vibration values are indexed to the maximum value and specified in this paper.

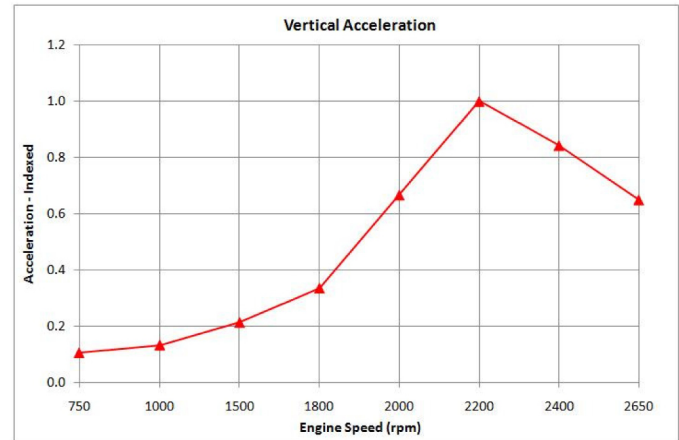


Figure 4. Base tractor steering wheel vertical acceleration

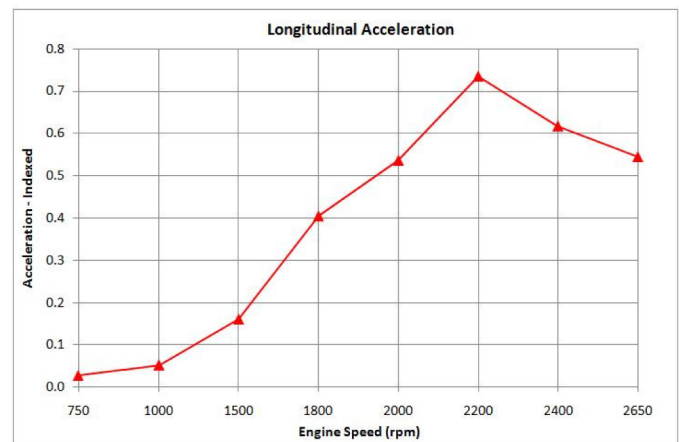


Figure 5. Base tractor steering wheel longitudinal acceleration

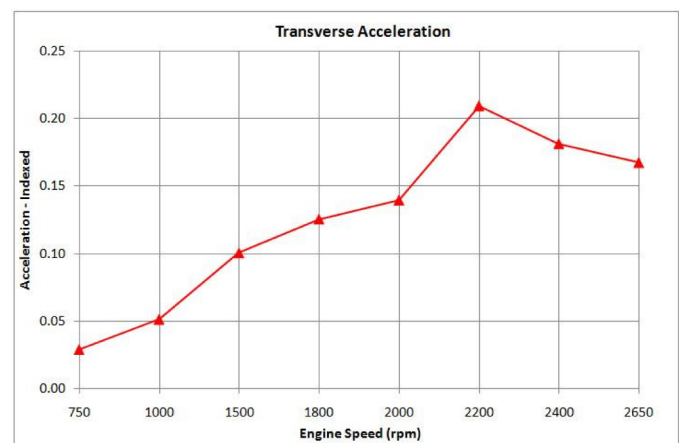


Figure 6. Base tractor steering wheel transverse acceleration

Vertical vibration is predominant in this tractor, followed by longitudinal and then transverse directional values. The reason for vertical vibration being the maximum is that the engine secondary imbalance acts in the vertical direction. A maximum response was obtained at 2200 rpm in the vertical direction. Maximum vibrations in the longitudinal and transverse directions were also observed at 2200 rpm. The vibration response reduces with further increase in speed.

Vibration measurement locations are numbered and shown in [Figure 2](#). The vibration transmissibility from engine (location 1) to steering box base (location 4) is almost unity in all the speeds. This is justified by the fact that those two points are rigidly connected by engine adapter plate and transmission case. Transmissibility from steering box base (location 4) to steering column base (location 3) is higher and varying as shown in [Figure 7](#). Similarly, transmissibility from steering column base (location 3) to steering wheel (location 2) is also higher as shown in [Figure 8](#). As there is no magnification from engine to steering box base, it is planned to omit this measurement point (location 4) both in analysis and further experiments to reduce testing duration.



Figure 7. Transmissibility from engine to steering column base-Base tractor

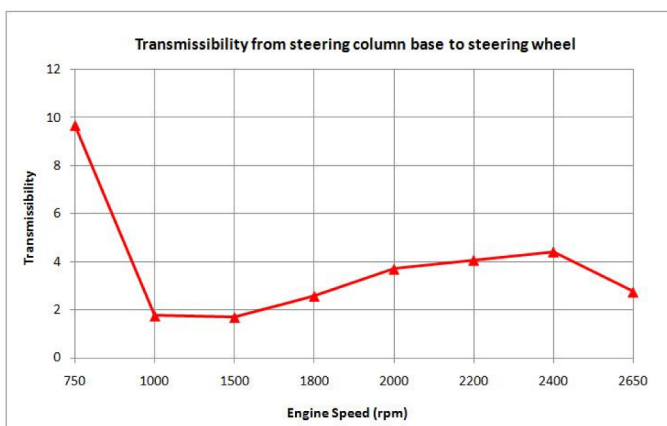


Figure 8. Transmissibility from steering column base to steering wheel-Base tractor

IDENTIFICATION OF VIBRATION SOURCE

Even though the vibration measurements show a hint of resonance around 2200 rpm, it was planned to find out the natural frequency and verify whether that is matching with any of the orders of engine firing frequency.

Impact Excitation of Steering System

The steering system was suspended freely using a weak spring. Then the system was excited by an impact hammer. The frequency spectrum of response ([Figure 9](#)) was captured and analyzed. The frequency corresponding to the maximum response refers to the natural frequency and this occurred at 75 Hz for the steering system. Engine second order excitation frequency at a speed of 2200 rpm is 73 Hz. Hence, the peak obtained at 2200 rpm is confirmed to be the effect of resonance of steering system with engine excitation.

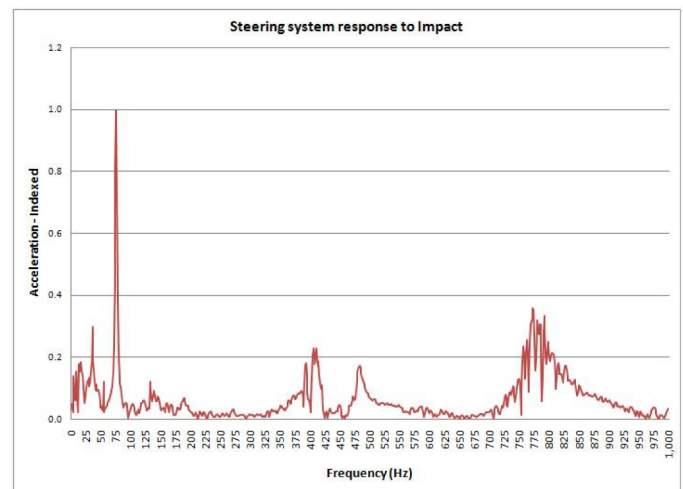


Figure 9. Steering system response to impact excitation

Ambient Excitation of Steering System

This is another method of confirming the resonance. The engine and steering wheel acceleration were measured at different engine speeds in steps of 100 rpm. The measured acceleration values have to be normalized by removing the effect of increasing speed leading to increase in acceleration. This can be accomplished by dividing the acceleration by ω_1^2 , where ω_1 is the angular velocity corresponding to the respective engine speed and then multiplying by ω_2^2 where ω_2 is the angular velocity corresponding to a reference engine speed. [Figure 10](#) shows the normalized acceleration of engine and steering wheel at a reference speed of 1800 rpm. If the normalized acceleration is invariant with increase in speed, then there is no resonance. A significant peak will indicate the occurrence of resonance at that particular speed.