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STEERING WHEEL IDLE VIBRATION IMPROVEMENT ON TRACTOR

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ABSTRACT

High amplitude level of steering wheel vibration is observed during idle running tractor validation tests. It is found that the tractor's platform firewall area's lateral mode is close to 1.5th order excitation frequency of three cylinder engine and this resonance has contributed to high y-direction vibrations at idle condition. In order to comply with the requirements of ISO 5349 Measurement and evaluation of human exposure to hand-transmitted vibration and to satisfy the comfort level by investigation of human subjective response, decreasing the magnitude of vibration is taken as a target. To achieve this target, three iterations are evaluated by comparison of modal frequency response plots. Increasing the lateral stiffness of the firewall area is main purpose. The validation of chosen iteration by finite element model is made through vibration testing and subjective comparisons.
Keywords: Engine Idle Vibration, Steering Wheel Vibration, ISO 5349.

TRAKTÖRDE MOTOR RÖLANTİ DEVRİNDE MEYDANA GELEN DİREKSİYON TİTREŞİMLERİNİN İYİLEŞTİRİLMESİ

ÖZ

Traktörün doğrulama testleri sırasında motor rölanti devrindeyken direksiyonda yüksek genlikli titreşimlerin oluştuğu gözlemlenmiştir. Platform ile motoru ayıran bölgenin yanal modunun üç silindirli motorun 1.5'inci motor yanma frekansına yakın olduğu gözlemlenmiştir ve bu rezonansın motor rölanti devrinde çalışmaktayken y yönünde oluşan titreşimleri arttırdığı gözlemlenmiştir. ISO 5349 kişilerin maruz kaldığı, elden vücuda iletilen titreşimlerin ölçülmesi ve değerlendirmesi standardına uymak, insanın subjektif tepkisiyle oluşan konfor düzeyini karşılamak amacıyla titreşim genliklerini düşürmek hedef alınmıştır. Bu hedefe ulaşmak amacıyla, sonlu elemanlar modeli üzerinden hazırlanan üç farklı iterasyon frekans tepki fonksiyonları üzerinden karşılaştırılmıştır. İterasyonlarda yanal yönde rijitliği arttırmak ana amaçtır. Sonlu elemanlar analizi yöntemiyle seçilen iterasyonun doğrulaması fiziki testlerle ve subjektif değerlendirmelerle yapılmıştır.
Anahtar Sözcükler: Titreşim, Motor, Rölanti, Ağırlıklandırma, Direksiyon, ISO 5349.

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1. INTRODUCTION

Vibration is a fundamental topic in automotive industry and perception of vibration is important to manufacturers since it is a factor for optimizing the comfort and quality metrics. Manufacturers of vehicles are interested in reducing the vibration levels of seat and steering wheel to increase the competitiveness of brand name. On the other hand, customers seek to enhance their workforce with reasonable prices and optimized products.

Besides health effects of vibration exposure in daily life, discomfort caused by steering wheel and seat have become more important. Therefore, the perception of vibration is needed to be investigated in the range of human perception thresholds and comfort. Subjectivity of steering vibration can be evaluated by experts in idle running tractors, but to be able to make more objective evaluations, vibrations signals can be recorded with tri-axis accelerometers by data acquisition systems and some signal filters are used to reflect human perception of vibration magnitude and frequency.

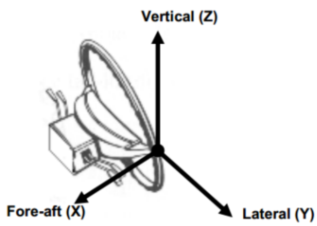


Figure 1. Three axes of vibration measured on a steering wheel

The axes used in measurement can be seen in Figure 1, the X- axis is taken along the fore-aft direction of the tractor, the Y-axis is taken along lateral direction, and the Z-axis is taken along the vertical direction.

During driving, steering wheel power spectral densities can reach frequencies of up to 350 Hz with vibrational energy mostly present in the range between 10 and 60 Hz. [1] They are typically characterized by low frequency excitation in the range from 8 to 20 Hz due to 1st order tire non-uniformity forces and tire-wheel unbalance, and due to 2nd order engine and mechanical unbalance in the frequency range from 20 to 200 Hz during engine idle. [2]

In order to quantitatively assess the perceived intensity of steering wheel vibration, the recorded acceleration values are traditionally weighted according to the frequency, so as to represent the differences in human sensitivity with respect to frequency. [3] Several studies have conducted by researchers to develop equal sensation curves and these have contributed to the definition of W_h frequency weighting, which is currently used in ISO 5349. In this study, both objective and subjective parameters are used in the assessment of steering vibration. W_h frequency weighting is used in objective evaluations, although there are some suggestions that W_h weighting underestimates the perceived intensity of hand-arm vibrations, this weighting is a well-accepted standard for calculations. Researchers have been working on new weighting method called W_s and they suggest that the only internationally standardized frequency weighting W_h is not accurate to predict human perception of steering wheel hand-arm vibration. [4] On the other hand, in automotive industry W_h frequency-weighting is widely used and W_s is not yet standardized internationally.

In our study, steering wheel is directly mounted with bolted connection to the four wheel drive tractor's firewall area and we dealt with vibration responses created by engine excitations at idle running condition. Finite element model of tractor's platform is also used in modal analysis and in obtaining modal frequency response. In this finite element model, platform and steering wheel model is reduced to shell elements and concentrated masses including inertial properties and these masses are connected to platform with rigid elements. The reduced model built in Hypermesh has a manageable size with 86250 nodes and 86120 elements.

2. MATERIALS AND METHODS

2.1. Method and Test Condition

Vibration control and data acquisition are performed by means of LMS Test Lab. software and a 40-channel LMS Scadas Recorder. The acceleration obtained at the steering wheel is measured using a PCB Piezotronics ICP accelerometer located on the top left side of the wheel. The measurement is carried out at idle running condition of the tractor.

Recorded raw data is analyzed for averaged vibration spectral weighting according to ISO 5349 where vibration transmitted to driver's hand is described with Root Mean Square (RMS) single axis acceleration value of frequency-weighted vibration in meters per second squared (m/s^2) and denoted by a_{hw}.

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

W_{hi} is the weighting factor for the i th one-third-octave band shown in Figure 2, a_{hi} is the r.m.s. acceleration measured in the i^{th} one-third-octave band, in meters per second squared (m/s^2). The final evaluation of vibration exposure is found by combining all three axes with the root-sum-squares of three components and denoted by a_{hv}.

$$a_{hv} = \sqrt{a_{hwx}^2 + a_{hwy}^2 + a_{hwz}^2} \tag{2}$$

The daily vibration exposure can be calculated by using vibration total value and duration of the exposure time. Eight hours energy equivalent frequency-weighted vibration total value can be calculated using equation (3).

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

T is referred as total daily duration of the exposure to the vibration total value in second and T_0 is the reference time duration of eight hours in seconds.

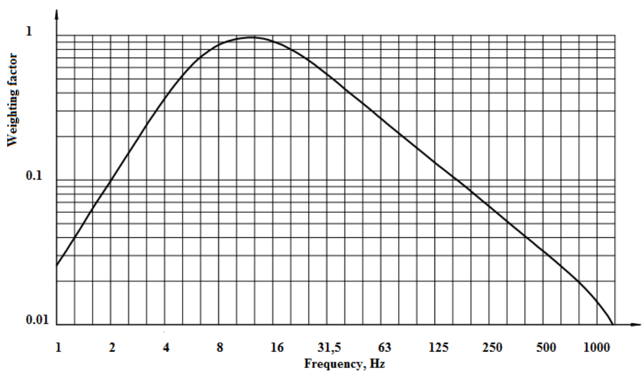


Figure 2. Frequency-weighting factor curve for hand-transmitted vibration [10]

2.2. Analyzing the Vibration Signals

In the analysis phase for steering wheel vibration, both spectrally weighted and non-weighted steering wheel accelerations are calculated. Bellmann M. used specially selected narrow frequency bands around prominent engine orders in his studies.[5] Frequency bands for each engine order can be chosen according to idle Revolutions Per Minute (RPM) and cylinder number.

A measured averaged spectrum in three dimensions of the steering wheel is shown in Figure 3. The cut-off frequency is chosen 100 Hz because above that limit the absorbed energy in signal decreases. Addition to this, sensitivity of human body decreases with increasing frequency. [13]

The peaks at spectrum in Figure 3 can be interpreted as engine orders. Amplitude modulation in three-cylinder diesel engine at idle is the result of half order engine harmonics. Power stroke in three-cylinder engine occurs in every one and a half crankshaft revolutions. The reciprocating motion of the pistons and of the connecting rods, combined with the rotational motion of the crankshaft, generate inertial forces that act on the engine block. At low engine speeds the combustion gas forces are greater than the mechanical inertial forces, but at high speeds the opposite is true. [6]

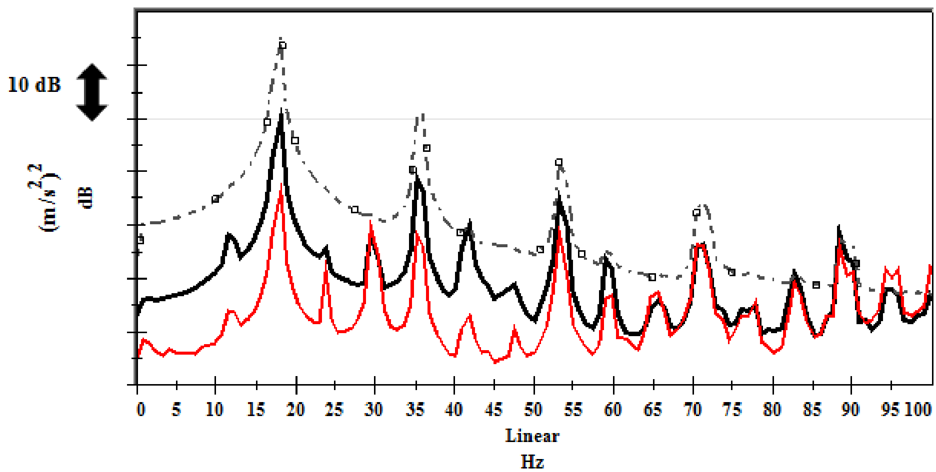


Figure 3. Averaged Auto-power Spectrum for steering vibrations in all three directions, continuous black line shows the x- direction vibration, black dashed line shows the y-direction vibration and continuous red line shows the z-direction vibration

Idle running r.p.m. is 650 for this measured three cylinder engine tractor and 1.5th engine firing frequency is calculated as 16.25 Hz and the peak amplitude in spectrum overlap with this frequency. Most sensitive direction and frequency can be chosen as y-direction and 16.25 Hz from the magnitudes of each peak at following engine firing order frequencies.

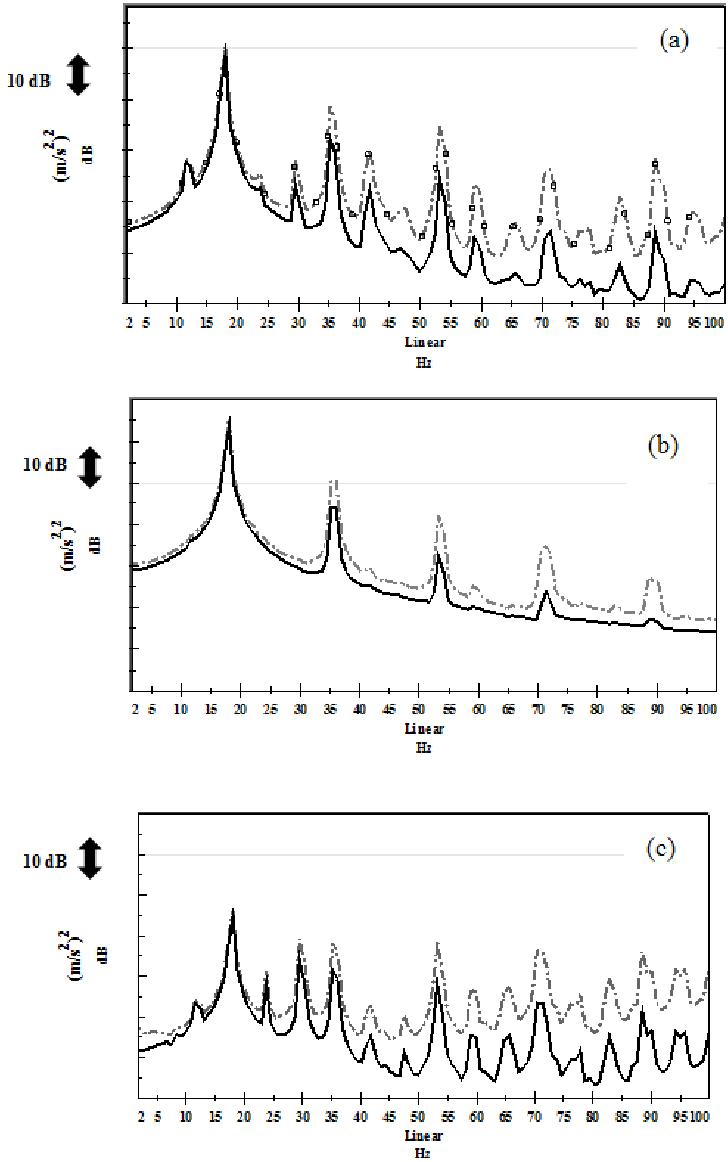


Figure 4. Averaged power spectrum for steering vibrations, grey dashed line shows vibration at idle, continuous black line shows W_h weighted vibration at idle, (a) plot refers to x direction, (b) plot refers to y direction, (c) plot refers to z direction

As seen in Fig.4 in comparison of W_h weighted and non-weighted spectrums, the amplitudes are tend to decrease more at higher frequencies, this is because of the low pass characteristic of weighting factor defined in ISO 5349 for hand-arm transmitted vibrations. Higher y- direction amplitude is observed in comparison with other directions. The root-sum-square of the three directions can be calculated with the formula stated as vibration total value in ISO 5349.

Vibration total value is calculated by LMS Signal Calculator Tool and the final value for weighted directional values in m/s^2 can be seen in Table 1 for different runs. During an ordinary working day the driver spends approximately 6.5 h in effective work. The effective work consists of two dominant operations during which the tractor is either idling or at full load. The duration of idling is approximately 1.5 h. [7]

Table 1. Measured accelerometer values of steering wheel vibration at idle in m/s^2

Acceleration Direction	First Run	Second Run	Third Run	Mean Value
x	1.40	1.42	1.41	1.41
y	6.50	6.80	6.65	6.65
z	0.74	0.70	0.72	0.72
VTV at Idle				6.83
A(8) at Idle				2.95

This calculated level of vibration exposure is lower than vibration total value, this is an expected outcome due to low exposure time at idle. Although the estimated year is higher than eight years for episodic finger blanching (vibration-induced white finger) disease according to Table 2, this does not mean not to take any action for excessive vibration.

Table 2. Values of daily vibration exposure A(8) which may be expected to produce episodes of finger blanching in %10 of persons exposed for a given number of years D_y [5]

D_y ,years	1	2	4	8
A(8)	26	14	7	3.7

European Union Physical Agent (Vibration) Directive (EU PA(V)D- EEC:2002) defines two criteria for Hand-arm vibrations. The first limit which must not be exceeded is Exposure Limit Value (ELV), and the second limit which refers to need to take action for reducing the exposed vibration level is Exposure Action Value (EAV).

Table 3. Vibration Exposure Values Specified by EU PA(V)D [8]

		8-hour energy-equivalent RMS acceleration- A(8) (m/s^2)
Hand-Arm Vibration	Exposure Action Value (EAV)	2.5
	Exposure Limit Value (ELV)	5

Our calculated value for 8-hour equivalent RMS acceleration is above the limit of EAV with 2.95 m/s^2. Daily exposure level exceeds the EAV and this means that the level of risk to health will increase. The root cause of vibration on steering wheel is investigated and the amplitude of vibration reduced to minimum level both for not to exceed EAV and designing more ergonomic tractors. A research done for Health and Safety Executive in UK suggests that the majority of agricultural vehicles will exceed the EAV during most normal (full) days and exceeding the limit creates a need for health monitoring to workers who are subjected to high levels of vibration. [9]

2.3. Root Cause Finding Analysis

It is observed during experimental test that tractor's platform firewall area, where the steering assembly is mounted, has resonated at idle. Steering wheel modal analysis is performed and it has been found that the first mode is above 20 Hz. Steering wheel modal analysis results are indicated that excessive vibration is a consequence of resonating firewall. As a second step, tractor's platform modal analysis is performed and natural frequencies between 10Hz to 20Hz are extracted. The frequency range is chosen so that, to see if there is any resonance frequency coupling with the 1.5^{th} order idle engine firing frequency. The methodology used for performing analysis can be seen in Figure 5. The FE Model is built in Hypermesh using shell elements and concentrated masses which are connected with rigid elements. Bolted connections are also made using rigid elements.

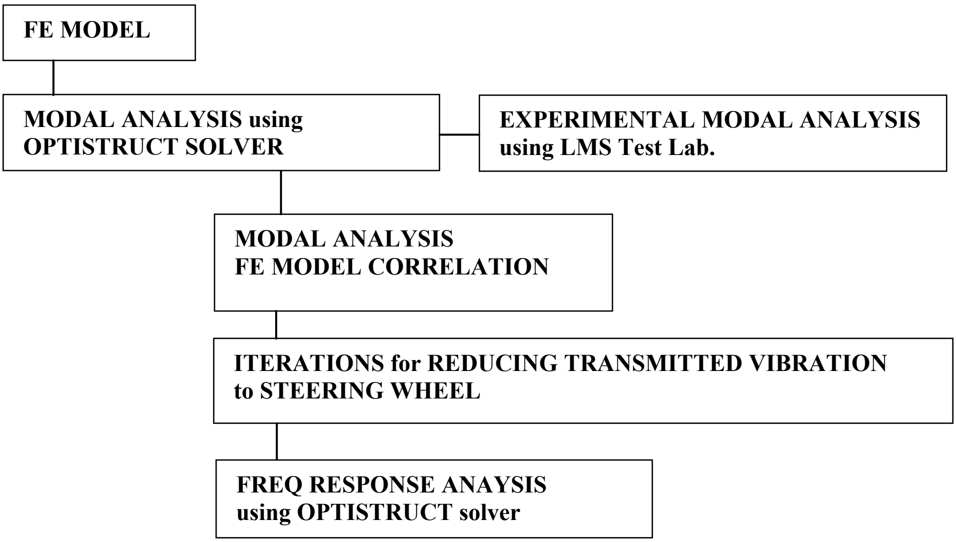


Figure 5. Methodology used in FE Analysis

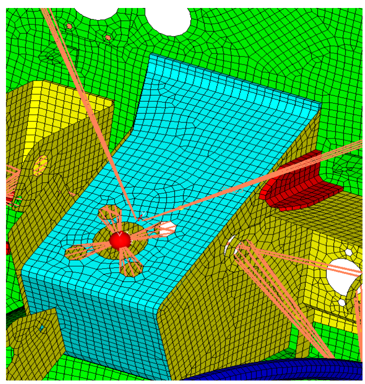


Figure 6. FE Mesh Model for Platform Structure **Figure 7.** PAS Pump Connection

The configuration of whole model is shown in Figure 6. The model consists of mostly sheet metal parts welded together. The seat is also mounted with rigid elements to the platform. The details in each sheet part like chamfers, radiuses and small features are neglected and are suppressed in geometry. The suppression of small details has not only reduced the numerical computation time but also refined the mesh quality. Steering wheel assembly is modeled with concentrated mass and inertial properties are assigned to them, the red node in Figure 7 has resembled the steering wheel mass and inertia properties. The red node in Figure 7 is used for modal frequency response plots. The four spiders around the red hole resembled the power assisted steering (PAS) pump bolted connection.

The finite element model is correlated with experimental modal analysis. First three natural resonance frequencies and mode shapes are matched with experimental results. For experimental modal analysis LMS Test Lab. Modal Analysis tool is used. The structure is excited separately from x, y and z directions with a modal hammer and most sensitive directions are found as y direction based on FRF plots. For experimental modal analysis correlation auto-MAC matrix is used. The Modal Assurance Criterion is a common tool to check the separation of modes and similarity of mode shapes. Computing the MAC values between a certain set of modes with itself (called an auto-MAC) gives a quality measure of the modal parameter estimation. [11] Calculated auto-MAC matrix for experimental modal analysis can be seen in Fig. 8. Auto-MAC values are assigned in percentages; %100 refers to perfect correlation and diagonal terms in matrix have to be at least %90 for acceptance. By correlating the modes with themselves it can be concluded whether there are sufficient measurement points used in experiment or not.

As seen in Figure 8 diagonal terms show %100 percentages of correlation indicating that number and location of the measurement points are sufficient to separate modes from each other.

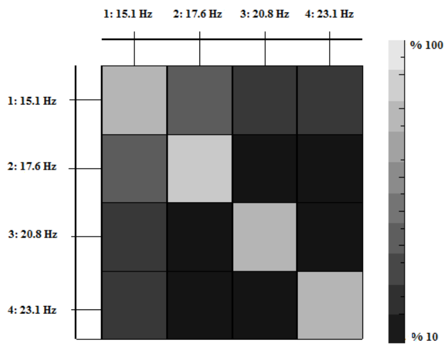


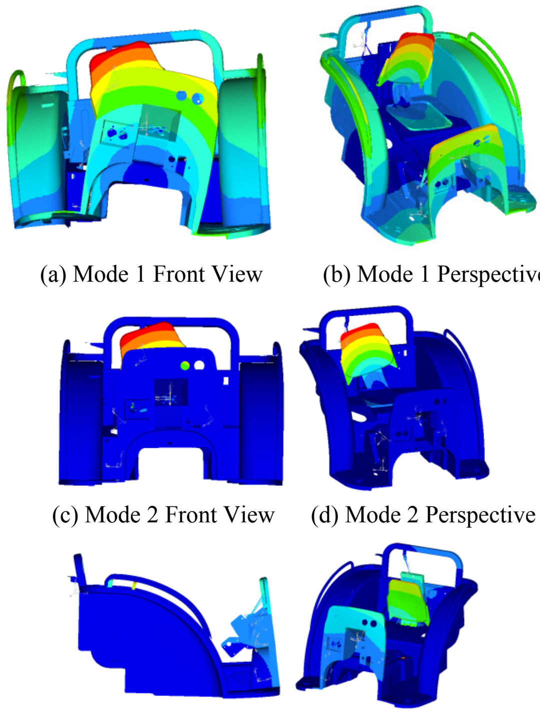
Figure 8. Auto-MAC Matrix for Experimental Modal Analysis

MAC and auto-MAC plots are widely used for correlations, but lack of giving information about natural frequency correspondence. The most obvious comparison can be done by simple tabulation of the measured values versus calculated values. The tabulated comparison can be seen in Table 4.

Table 4. Natural Frequencies and Mode Shapes

FE Modal Analysis Results			**Experimental Modal Analysis Results**		
Mode No	**Frequency (Hz)**	**Type of Mode**	**Frequency (Hz)**	**Type of Mode**	**Modal Damping**
1	14.8	Platform Torsion	15.1	Platform Torsional	% 4.7
2	17.5	Seat Lateral	17.6	Seat Lateral	% 5.85
3	20.5	Firewall Fore -Aft	20.8	Firewall Fore -Aft	% 4.3

The tractor's platform has three natural frequencies between the ranges 10-20 Hz. It is found that first torsional mode of platform couples with 1.5th order idle engine firing frequency. This torsional mode of platform has caused lateral vibration on firewall area and this resonance contributed to high y-direction on steering wheel at idle condition for 3-cylinder engine tractor. As shown in Figure 9, when the mode shapes of each natural frequency are examined lateral movement of firewall area has showed up at mode 1.



(a) Mode 1 Front View (b) Mode 1 Perspective View

(c) Mode 2 Front View (d) Mode 2 Perspective View

(e) Mode 3 Side View (f) Mode 3 Perspective View

Figure 9. Mode Shapes for Platform

3. RESULTS AND DISCUSSIONS

Root-cause analysis has indicated the low lateral stiffness of firewall area has affected the steering wheel vibration. Vibration test results are also supported the analysis results with high amplitude vibration levels on firewall. The firewall area is too weak at welded joints to platform. When a symmetric plane dividing the firewall area is considered from front view, residual shape simply resembles an L-beam with a thick upper side and thin lower side. This L-shaped cross-section in Figure 10 is stiffened from the corner side by welding 5mm triangular gusset plates. Gusset plates are widely used by structural designers to strengthen the joint. The gusset plate size and thickness are determined by iterations and FRF plots.

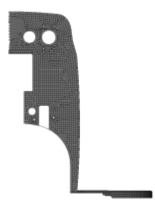


Figure 10. Cross-section of Firewall Area

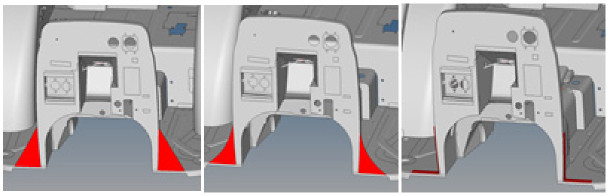


Figure 11. Design Iteration Summary (a) Iteration 1with a triangular 5 mm thickness gusset, (b) Iteration 2 with a rounded triangular 5 mm thickness gusset, (c) Iteration 3 with an angular 5 mm thickness beam

In order to compare each iteration, platform modal and frequency response analysis are performed with free-free boundary condition. Unit force in three main directions (i.e. X,Y,Z) is applied at the platform-chassis mount attachments. Frequency Response Function plots are calculated by exciting the four platform mounts with a unit force in frequency domain. Response node is selected as steering column mounting base, see Fig.7. Only the front right mount excitation and response in Y direction have taken into account for evaluations because the highest sensitivity is observed on this path.

Frequency response results are compared with base design (without any support bracket, see Figure 12). It is aimed to decrease the resonance peak amplitude at 15 Hz, which is predicted to be related to high steering wheel vibrations at idle condition. Based on FRF plots as shown in Figure 12, Y direction vibration amplitude has decreased at the steering column attachment node in Iteration 1. Iteration 2 is also acceptable regarding acceleration amplitudes which are slightly higher than Iteration 1. Iteration 3 is also improved the steering column attachment node Y direction acceleration response, but not provided similar decrease in amplitude level as Iteration 1 and Iteration 2.

Iteration 1 realization is tested on the tractor, as seen in Figure 13 peak amplitude at 15 Hz dropped significantly. Simulation results are validated with test results.

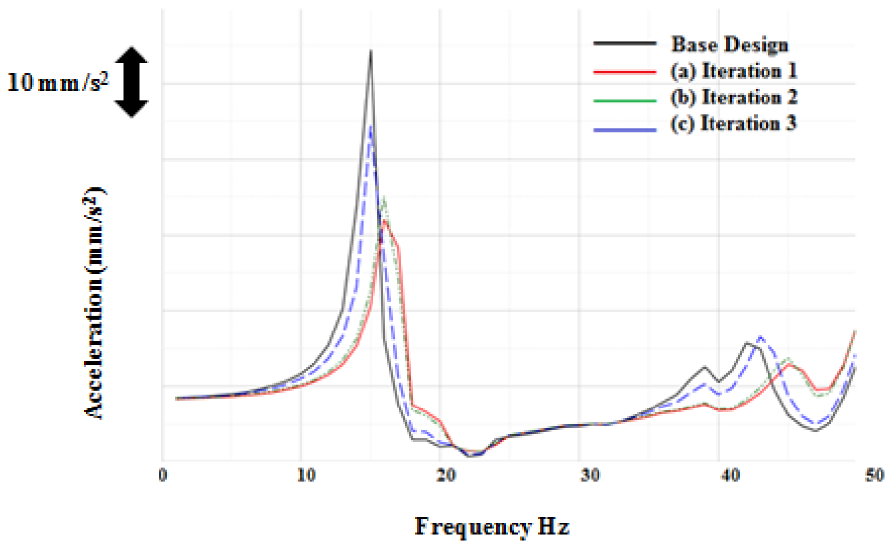


Figure 12. Frequency Response Function Plot in Y-direction btw. 0-50 Hz (Excitation From Front Right Attachment Y direction – Response on Steering Column Attachment Y direction)

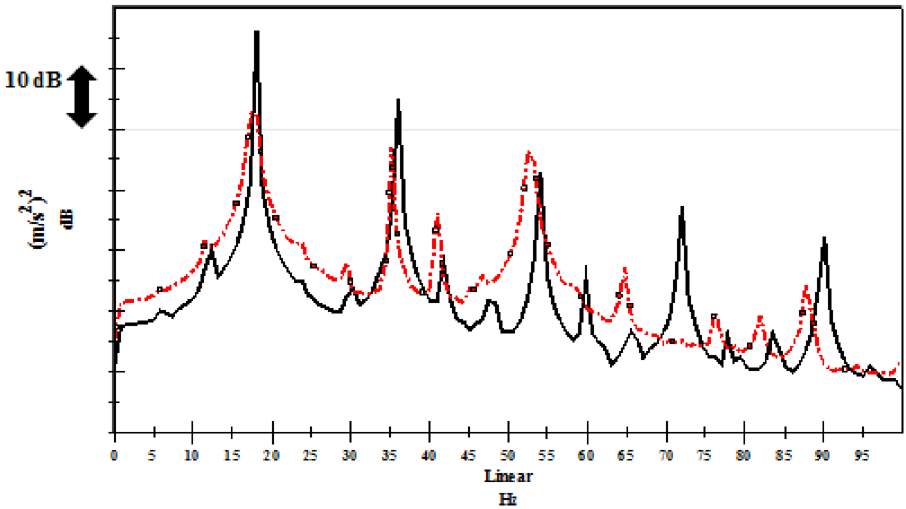


Figure 13. Averaged power spectrum for steering vibrations, black continuous line shows y-direction vibration at idle for base design, red dashed line shows y-direction at idle for Iteration 1

Based on measurements results overall vibration level on y direction has dropped from 6.65 m/s^2 to 1.96 m/s^2. The W_h weighted vibration dose levels are also shown in Table 6. Vibration exposure total value and 8-hours RMS accelerations are also calculated according to ISO 5349. Vibration exposure total value has dropped from 6.83 m/s^2 to 2.44 m/s^2. The calculated 8-hour energy equivalent RMS calculation has also dropped from 2.95 m/s^2 to 1.05 m/s^2. When we compare the daily vibration exposure using the predicted 10% prevalence of vibration-induced white finger graphic in Figure 14, it is obvious that before the improvement the acceleration level transmitted from the steering wheel to driver's hand will probably cause finger blanching in 10% of expose persons after 10 years. Calculated A(8) value 1.05 m/s^2 for the improved platform with gussets has prolonged the predicted time for white-finger disease to more than 20 years.

Table 5. W_h weighted RMS values for each directions measured on tractor with added gussets (Iteration 1)

Acceleration Direction	First Run	Second Run	Third Run	Mean Value
x	1.34	1.28	1.31	1.30
y	1.91	1.95	2.04	1.96
z	0.63	0.68	0.72	0.67
VTV at Idle				2.44
A(8) at Idle				1.05

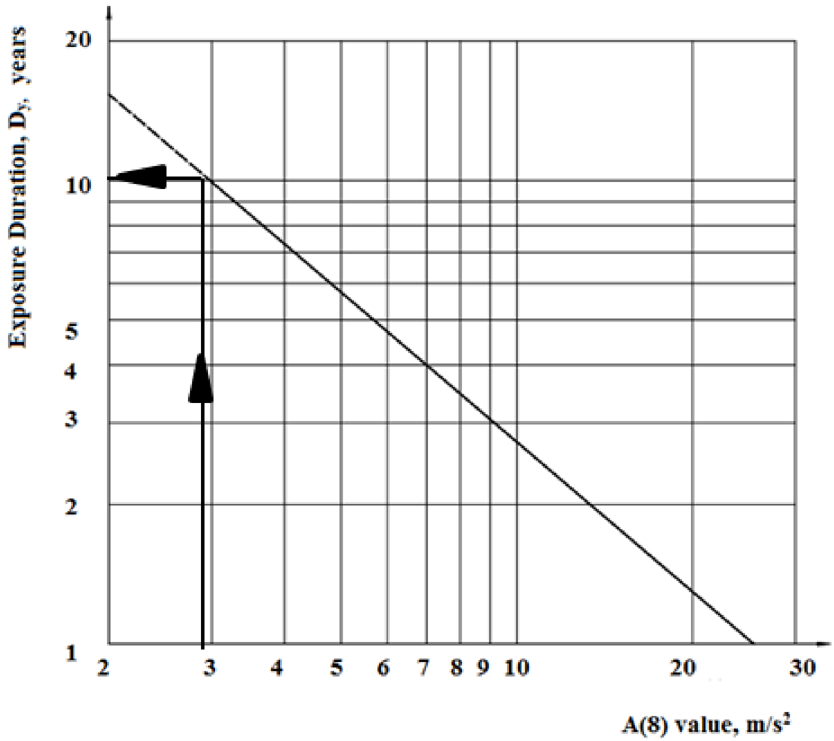


Figure14. Vibration total values and total daily exposure (for preventing white finger disease) for idle according to ISO 5349-1, 20

Sensible and visible vibration on steering wheel is tested by subjective tester in order to rate the vibration quality between the base and improved idle running tractors. Six highly experienced subjective test participants consisting of five men and a female are available. All testers have given similar subjective ratings therefore, assessments are summarized together.

Seat position is adjusted in order to simulate the real driving posture. Gripping position and type have kept same by testers. The assessment results for perceived vibration are given according to Vehicle Evaluation Rating System (VER scale) shown in Table 6. This scale is used to assess performance of a subjective nature which cannot be easily measured or quantified.

Table 6. Vehicle Evaluation Rating System

Rating Index	1 2 3 4 5 6 7 8 9 10							
NVH	Not Acceptable	Objectionable	Moderate	Light	Very Light	Trace	Very low	None

The base tractor's steering wheel vibration mean value is rated 4-objectionable. On the other hand, the improved tractor's steering wheel vibration mean value is rated 7-very light by test participants. The increased lateral stiffness by gussets is both subjectively and objectively reduced the vibration of steering wheel at idle.

4. CONCLUSION

Based on modal analysis results, in order to decrease the vibration levels on the steering wheel at idle operating condition for 3-cylinder engine tractor, iteration 1 design proposal is implemented to the platform structure. Steering wheel vibration level is reduced to acceptable limits not only for occupational health and safety considerations, but also for comfort expectations at idle. The improvement made on platform is verified by simulation and experiments. It is also found that the lateral vibration mode of seat couples with main engine firing frequencies at idle. The future work has planned for human exposure to whole body vibration for seat.

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