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Enhancing Customer Comfort by Reducing Steering Wheel Vibration

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ABSTRACT

In today's world, the application of the tractor is essential for numerous operations like cultivation, ploughing, transportation and many more. During these applications, tractors may be subjected to different level of vibrations in various structure parts which effects the ride and comfort performance. In this paper, case study related to enhanced customer comfort by reduction of steering system vibration is presented. Systematic approach is used to understand the steering performance, replication in lab, incorporation of advance testing like ODS (Operational Deflection Shape) and virtual analysis to execute the different iterations. To solve FEA model faster, super element technique is used. With this approach, solution time is reduced by 95% as compared to full model approach. Target location stiffening is adopted based on the modal strain response to achieve the desired results. With modified design, there is significant vibration reduction. Same is confirmed with subjective evaluation also. Based on all these, recommended design modification is implemented, and positive customer feedback about increased comfort is received from multiple customers.

INTRODUCTION

Vibration reduction can be achieved by working either at source or path. For this, it is necessary to understand the machine dynamics. From the testing and analysis, it is possible to compute the detail dynamic properties of the machine. The vibration reduction decreases operator's stress and consequently improve the ride and comfort quality.

The engine is a major source of vibration. It leads to the vibrations of other attached components. The design and analysis play a major role for determining the root cause for the problem. Once the problem and its root cause are well defined, then the solution approach for addressing the problem becomes clear and in the right direction.

Dynamic response of a structure is determined by external loads and structure dynamic properties like modal frequencies, mode shapes, damping etc. Sugita and Asai [1] demonstrated that the vibration can be reduced by increasing the resonant frequency by different ways like increasing the stiffness, reducing the mass. Ankush Shinde et al [2] used isolator concept to reduce the vibration. They used experimental and analytical method in their work. G Pandiyanayagam et al [3] established the good correlation among the modal parameters. They also demonstrated the method to find out the damping ratio. Yuntao Chen, et al [4] showed good corelation between analysis and test of the electronic module. Storck, H et al [5] presented experimental modal analysis on an exhaust system of an off-road car. In V.K.Tewaria et al [6] work, appropriate isolators were installed in specific locations to reduce the effect of vibration level.

This paper focuses on the methodology to understand and predict the dynamic behavior of the system using various mechanical tests and simulations. The modal response of the model is acquired through experiments, and thus, determining its dynamic properties like mode shapes, modal frequencies. Next, the same model is analyzed by finite-element method, determining its dynamic properties. Thus, the results obtained through both the approaches are compared. Further, different designs are analyzed and tested towards vibration reduction. In the present work, the testing and analysis of tractor steering system is presented. The root cause of vibration, identification of path of vibration and effect of engine vibrations are discussed. Operational deflection shape (ODS) Testing is performed to understand the dynamic response in detail. Modal Test response of the steering system is corelated with the virtual modal analysis. Super element technique is used to reduce the solution time of FE Model. Different iterations are executed using virtual analysis. Modal frequency is shifted on higher side considering the engine operating RPM range and the excitation at different RPMs. Finally, significant vibration reduction is achieved with the recommended design.

MODAL ANALYSIS

Modal parameters like frequency, mode shapes etc are properties of the system, and are dependent on the mass, stiffness, damping and boundary conditions of the structure [7]. These properties can be computed analytically using Modal Analysis. Equation 1 is the governing equation of motion.

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = f(t) \qquad (1)$$

where M, C and K are mass, damping and stiffness matrices respectively. Here, inertial force is represented by $M\ddot{x}(t)$, $C\dot{x}(t)$ the damping force, $Kx(t)$ the restoring force and $f(t)$ an externally applied force. The displacement response of the system is represented by variable $x(t)$. In order to determine the inherent structural properties, there is no external force considered in modal analysis. Therefore, by substituting f(t) = 0, in Equation (1)

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = 0 \qquad (2)$$

Dynamic characteristics or modal properties can be obtained either through analytical or experimental techniques. These are extremely useful information while designing of almost any structure. There can be two methods to conduct modal analysis. One is the computational method and the other is the experimental method. Often, the results from experiment are used to verify computational results and to validate the computational model.

OPERATIONAL DEFLECTION SHAPE (ODS) ANALYSIS

An operational deflection shape (ODS) is an animation of the vibration pattern in a structure under various operating conditions. Both the amplitude and phase of vibration measurements are animated. It gives additional insight into vibration response that individual measurements alone do not. It helps to understand the directional behaviour of vibration response and then, to explore the possible design modifications effectively.

To perform an operational deflection shape analysis, three steps are required:

- Geometry: Create a geometry of the test object
- Measurement: Acquire data with consistent phasing at different points
- Analysis: Create animation utilizing the geometry and measurement data

Any type of measurement (order, frequency, time) can be animated. The key is that the phase relationship between all the different channels (corresponding to various points and directions) is preserved during the measurement.

FINITE ELEMENT METHOS AND SUPER ELEMENT TECHNIQUE

Finite element method (FEM) is the most effective and versatile tool for computational analysis. In the present work, commercially available software is used for modelling and analysis. Different steps are performed to execute the analysis like geometry modelling, mesh generation, defining materials, applying loads, obtaining solution and reviewing results. The appropriate boundary conditions are applied representing the actual scenario.
The Lanczos mode extraction method is used in this study because it is the recommended method for the medium to large models [8]. The enough number of modes are extracted to cover the frequency range of interest.

For executing different DOEs quickly, super element technique is used. It uses a reduced model method. In this method, full model result file is used and reduces solved results to reduced model, consisting of few elements. This reduced model contains all mass and stiffness matrix information for each mode. Thus, few nodes are required to be solved in reduced model, makes it extremely faster. In traditional approach, it requires to solve large model every time with small change in model at specific region corresponding to different DOEs. Super Element Technique is very strong and useful in this kind of scenario.

EXPERIMENTAL APPROACH

In Physical testing, the steering wheel is instrumented with triaxial accelerometers and engine rpm sensor. A set of accelerometers is installed to capture the detailed vibration behavior in ODS Analysis (Fig. 1). Data is recorded for engine sweep (from low idle to high idle engine RPM). A sampling rate of 16 kHz is considered. For test data analysis, frequency resolution of 1 Hz with Hanning window is applied.

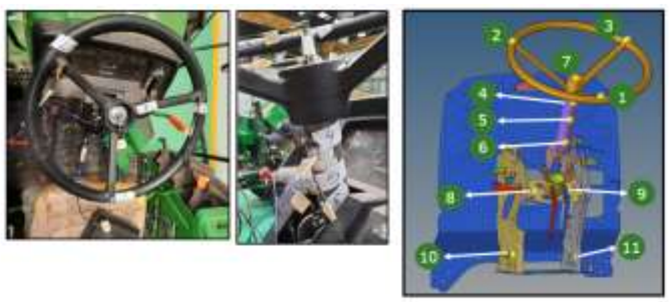


Figure 1: Instrumented Steering Assembly

In general, steering wheel vibrates more at its natural frequency, input force frequency and its amplitude. In the present study, higher vibration response is observed at ~45 Hz frequency with associated 1.5^{th} engine order excitation. (Fig. 2)

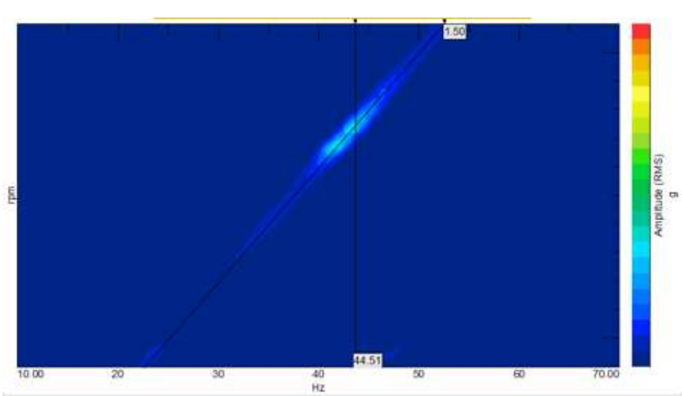


Figure 2: Baseline Vibration Response

Impact test is also executed to identify the modal frequency. The modal frequency is ~45 Hz (Fig. 3).

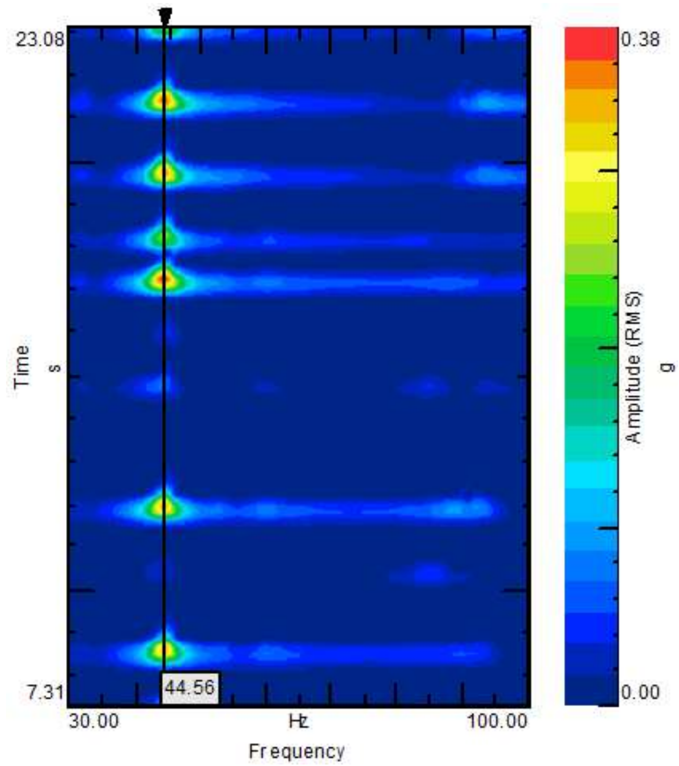


Figure 3: Steering Wheel Impact Test

ODS analysis is executed to understand the vibration pattern. Steering vibration response at ~45 Hz from ODS Analysis is shown in Fig 4.

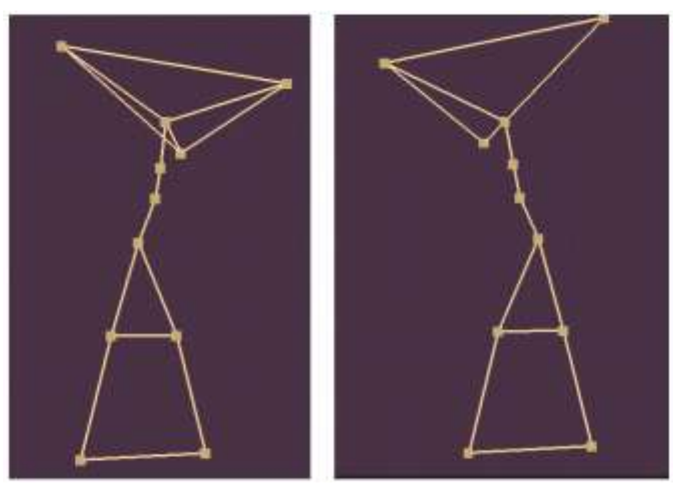


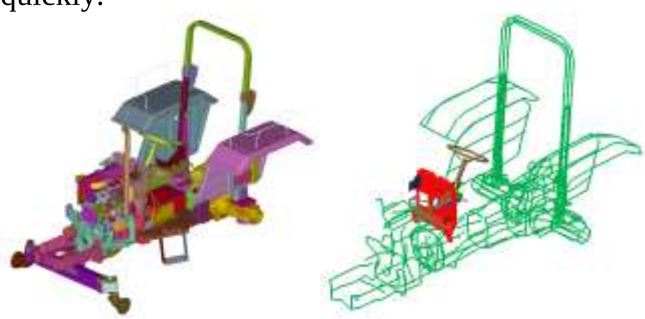
Figure 4: Operational deflection shape (ODS)

From impact testing, vibration response, and ODS analysis, it is clear that higher vibration is due to excitation of its natural frequency ~45 Hz because of 1.5th engine order excitation. These details are used for correlating the analysis model later.

ANALYTICAL APPROACH

In FEA (Finite Element Analysis) model, chassis, fenders, ROPS, platform, and other connected assembly are included (Full Model, as shown in Fig. 5). Mass and stiffness of the assembly are captured properly.

To solve FEA model faster, super element technique (previously discussed) is used. Other than steering assembly, remaining model consumes 95% of nodal count of full model which does not change in iterations but still needs to be solved. Considering this, all the remaining parts are included as super-element as shown in Fig. 5. With this approach, solution time is reduced by 95% as compared to full model approach for each iteration. This approach is very helpful to check different DOEs response quickly.



Full Model ***Superelement Model***

Figure 5: Full Model Vs Superelement Model

The FEA model is solved with free-free boundary condition. Modal analysis is performed to identify the frequencies and mode shapes. Modal frequency (~48 Hz) and associated mode shape are shown in Fig. 6.

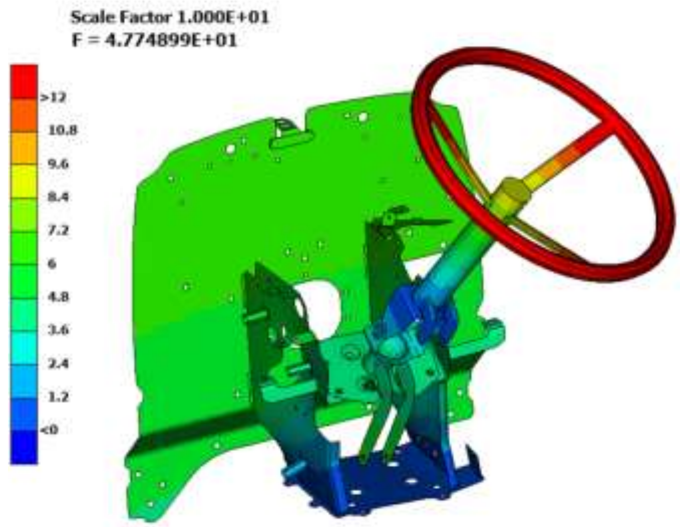


Figure 6: Baseline Modal Deformation Response

By comparing it with test results (Fig. 4), there is good correlation established between test and analysis.

The engine is a major source of vibration. For maximum engine RPM of 2300 and 1.5th Order engine excitation, frequency comes out to be 57.5 Hz. In baseline configuration, steering assembly mode with natural frequency of ~48Hz is in the engine excitation range. Mode shape from analysis and test shows that it is vibrating laterally and can be excited by rolling of engine. By all these, it can be inferred that lateral mode shape at ~48 Hz is responsible for steering higher vibration.

Considering maximum engine RPM of 2300, 1.5th order engine excitation and 10% variation, steering assembly frequency should be more than 64 Hz to avoid engine excitation frequency range. Corelated model is used for exploring different design modifications towards shifting of natural frequency on higher side, preferably out of engine excitation range.

DESIGN MODIFICATION

Modal Strain energy response plot (as shown in Fig. 7) is used to identify weaker section and targeted stiffening (encircled in the Fig 7).

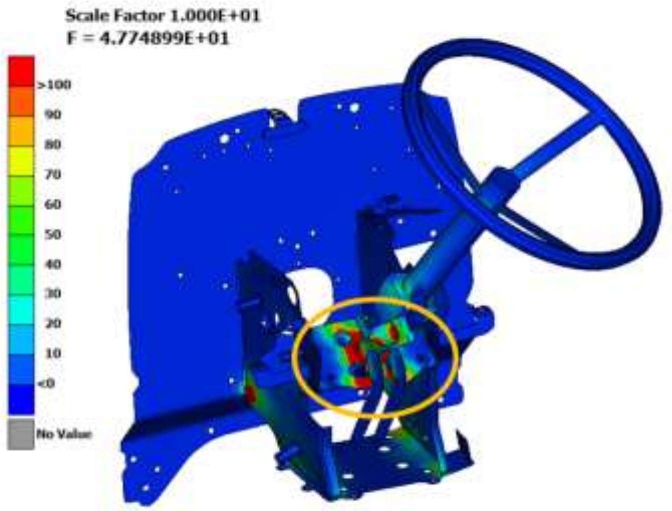


Figure 7: Baseline Modal Strain Energy Response

Various design options are analyzed to increase the frequency. In Option-1, plate thickness is increased by 4 mm for red plates as shown in Fig 8.

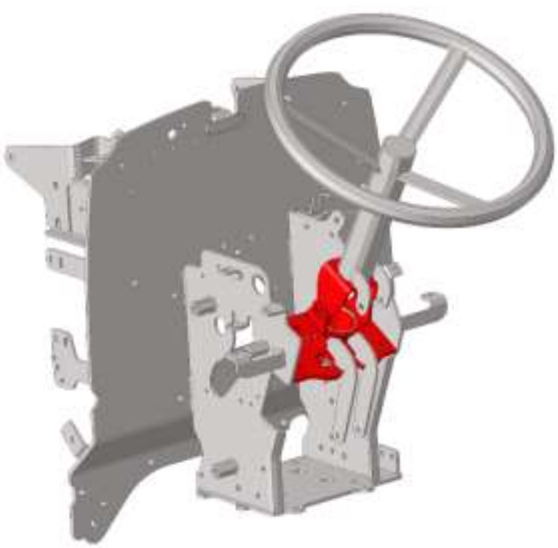


Figure 8: Option-1 Design Modification

With this, lateral modal frequency is shifted to 52 Hz (Fig. 9), is still in the engine operating range and less than the required limit. Also, there are multiple design assembly and manufacturing constraints for these design changes.

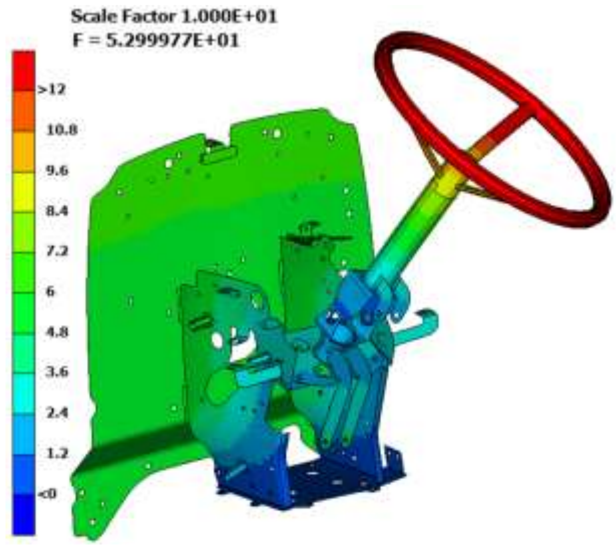


Figure 9: Option-1 Modal Deformation Response

In Option-2, gusset is added to cylindrical portion of steering in lateral direction which connects base plate and cylindrical portion as shown in Fig. 10.

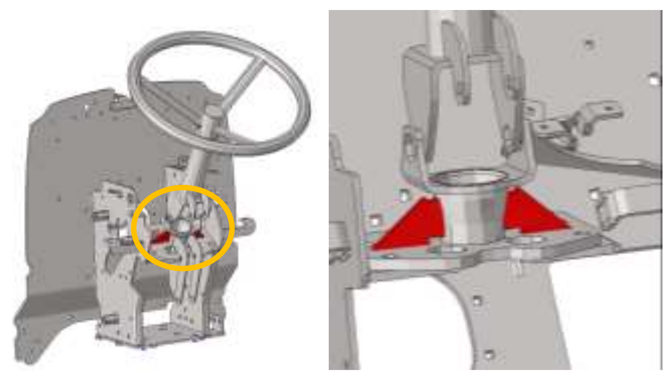


Figure 10: Option-2 Design Modification

With this, lateral model frequency is shifted to 58 Hz (Fig. 11), which again does not meet the acceptance limit.

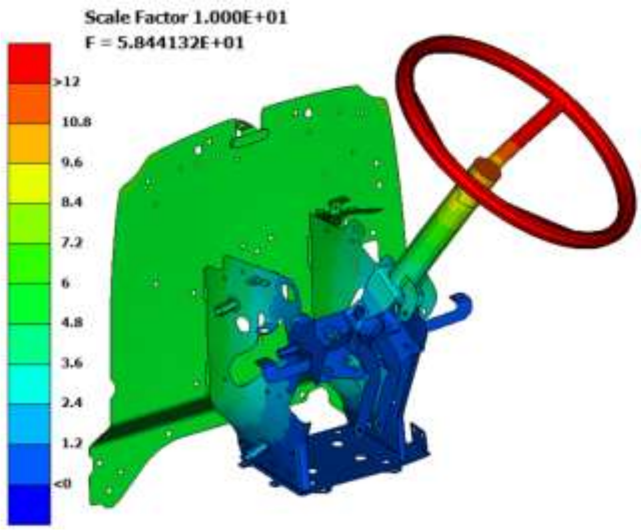


Figure 11: Option-2 Modal Deformation Response

With addition of small gusset, there is ~10 Hz change in the frequency. This indicates the sensitivity of the region for further targeted stiffening.

In Option-3, gusset is extended up to u-bend. Now, it connects, base plate, cylindrical portion and U-bend as shown in Fig. 12.

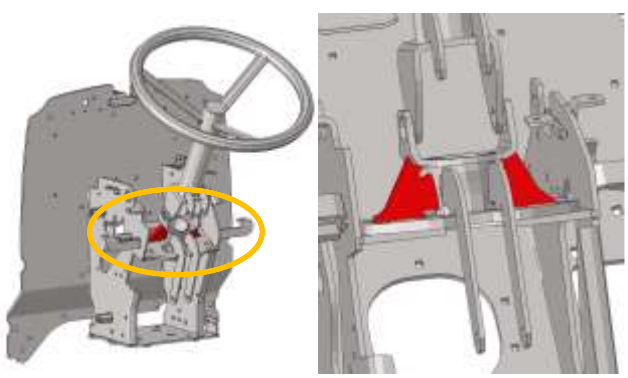


Figure 12: Option-3 Design Modification

With this, lateral modal frequency is shifted to 66 Hz (Fig. 13), meets acceptance limit. Changed behavior of mode shape is observed with reduced bending of steering at base mounting. This is also indicated by reduced strain values at cylinder location (Fig. 14). Considering all these, physical testing is recommended with this modification for verification.

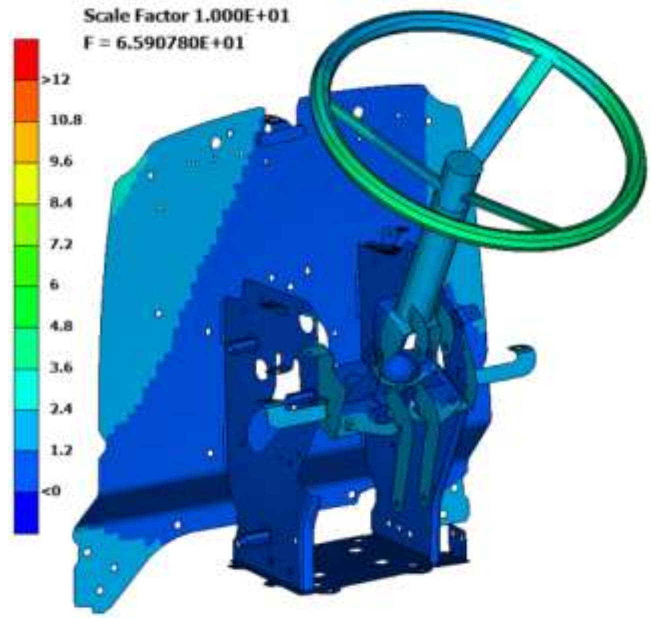


Figure 13: Option-3 Modal Deformation Response

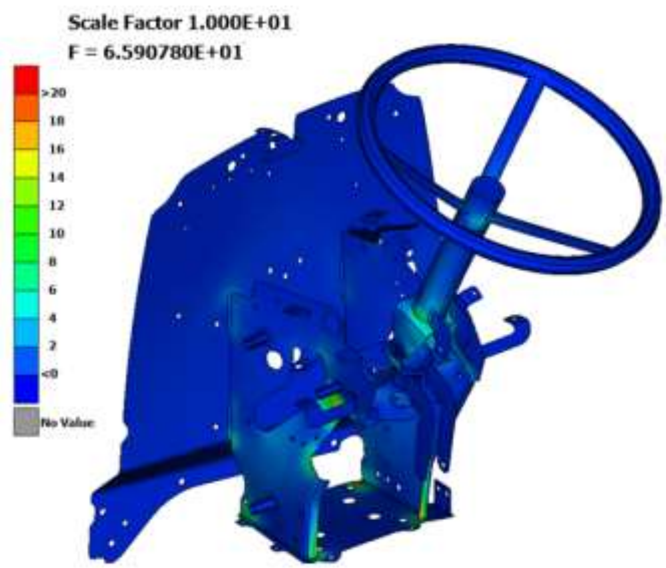


Figure 14: Option-3 Modal Strain Energy Response

VERIFICATION

Manufacturing feasibility of incorporating the recommended modification is checked and confirmed. Modified Design Model and Physical

Assembly are shown in Fig. 15. For physical testing, steering wheel is again instrumented with same set of triaxial accelerometers and engine rpm sensor. Data is recorded for engine sweep (from low idle to high idle engine RPM).

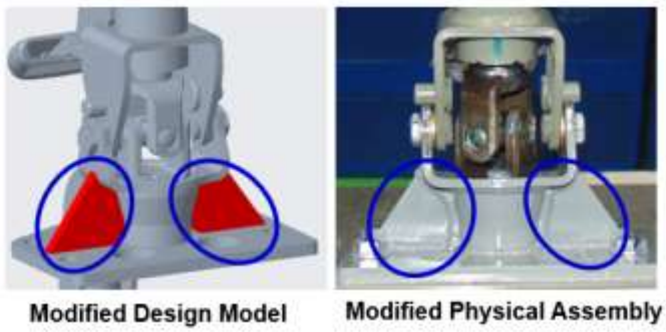


Figure 15: Recommended Design Configuration

The results of vibration level for baseline and modified assembly are shown in Fig.16. With modified design, there is significant vibration reduction. Same is confirmed with subjective evaluation also. Based on all these, recommended design modification is implemented. This resulted in enhanced customer comfort.

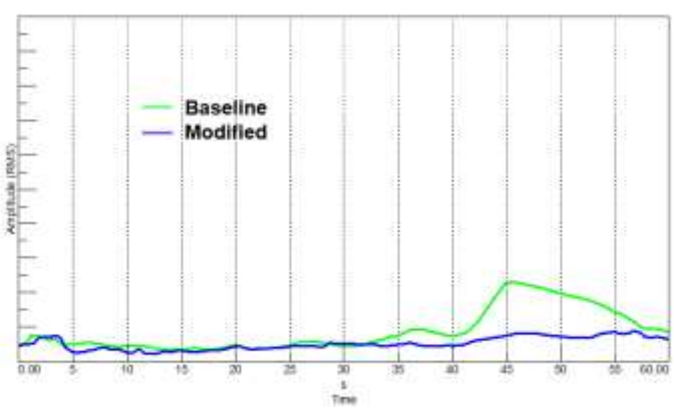


Figure 16: Steering Wheel Vibration Response : Baseline vs Modified Design

CONCLUSION

In this paper, it was explained how the analysis-led design assisted to reduce the vibrations, by executing the test and analysis, identifying the root cause and then, required modification. The entire engine system is usually connected with the different structures and therefore, lead to the vibrations of other components. In case of steering system, it was inferred that higher vibration was due to excitation of its natural frequency ~45 Hz because of 1.5^{th} engine order excitation. Modal frequencies and shapes were correlated from analysis and ODS test. Super element technique was used to reduce the solution time of FE Model and to execute the different DOEs quickly. From the strain plot, weakest section in the steering system was identified. Different designs were analyzed based on their dynamic response with addition of mass and stiffness in the identified region. With the proper stiffening at targeted location, Modal frequency was shifted in such a way that steering system frequency should not be excited because of dominant engine order excitation across the RPM Range. This resulted in enhanced customer comfort based on the feedback received from multiple customers.

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NOMENCLATURE

C	Damping matrices
f(t)	General force function
FE	Finite Element
FEA	Finite Element Analysis
FEM	Finite Element Method
FRF	Frequency Response Function
K	Stiffness matrices
M	Mass matrices
ODS	Operational Deflection Shape
RPM	Revolution per minute
S	Strain
T	Time Duration