Noise And Vibration Analysis For Source Identification Of Three-Cylinder Diesel Engine Using Fea And Experimental Techniques

Vivek Varma, Deepak Hujare, Rajesh Askhedkar

Abstract: The noise and vibration analysis plays a major role for determining the root cause for various faults in a diesel engine. The components of I.C engine produce noise and vibrations due to variation in loads and unbalanced forces which reduce the life of machine. So, it is essential to identify the major sources of vibration and its location and also to reduce the noise to avoid failures in the system. In the present study, experimental modal analysis is carried out for various 3-cylinder engine components to determine resonant frequencies in the system, which have been performed by using roving excitation technique. Then the results of experimental modal analysis are verified by using FEA. The experimental vibration analysis has been carried out at the speed near to the resonance conditions to find out the amplitude of vibrations. A structural borne noise mapping is carried out to know the exact locations of the noise sources. So, based on the results of noise and vibration analysis, structural modifications have been made in the corresponding components of an engine and the improved results have shown that amplitudes of vibration have been reduced by 35 to 40% and the structural borne noise level also reduced by 3 to 4 dB (A).

Index Terms: Borne Noise, FEA, FFT, Internal Combustion engine, Modal analysis, Roving excitation technique.

1. INTRODUCTION

INTERNALcombustion engine is the core of the automobile. A caution should be taken to the machine components of an engine. The components of I.C engine which generate noise and vibration during cycle are crankshaft, piston, gears, cylinder head and connecting rod. The cylinder head is the central components of the engine, to which all the exterior components are connected to it. However, the meshing error of gears is one of them which generates noise and transfers through gear tooth impact to all the gears in the engine. The high radiated noise and vibrations by diesel engine take over its advantages which reduces its power generating efficiencies in large scale. So, it is necessary to find the reasons of high radiated noise and vibration in diesel engine and subsequently their solutions. Some of the research work related engine noise and vibration is reported below.Investigated the act of the 4-cylinder diesel engine was modelled and tested by bench tests. To study the vibro-acoustic characteristics of a cylinder block, the FEA method was used to determine the natural frequency and its corresponding modes, and the results are proved by experimental modal analysis. The flexible multi-body dynamic model was fabricated by using different components, appliesdifferent boundary conditions and applies different forces on the block to obtain results close to the definite condition. The optimization process is used to move some natural frequencies, and this new block of cylinders is modelled, and bench test results display that thenoise level of the block is reduced [1].Studiedan example,a 6-cylinder in-line diesel engine was considered; gives some ideas about using test methods and MBD models to determine accurate sources of noise and vibration. To acknowledge the noise near field sound intensity and a near field acoustic holography techniques were used to identify hydraulic tensioner at the noise radiating component. By added an additional ribs near to the oil pump area with deducing of its thickness near to the radiating noise from the engine and realized up to 2dB(A)of sound deduction [2].In this paper they investigated the theoretical procedure to identify the noise and vibration of the engine block combined with the rotating crankshaft and drives the injection fuel pump was described. For these techniques, they proposed four methods, such as FEM, numerical integration technique, and some averaging methods. They described two gear backlash variation situations first at the working stage and second at zero. Their results showed that the variation in gear backlash affects the engine. Modification is done by combining the shaft with gear train to the engine block by oil film and stiffener reduces the noise level [3].Studied an engine modification of different components, including isolated oil sump, bearing beam and ladder frame to enhance acoustic noise levels of engine. A 6-Cylinder engine with rated power of 160 kw, having deep skirt type cylinder block. The results of engine modifications were analyzed by sound intensity, operating modes, sound pressure and mobility methods. The twoparallel plates were installed in between the isolated oil sump results showed that the noise levels were decreased at different frequencies for different components, overall reduction in sound power with isolated sump and ladder frame- Around 4.5 kHz [4]. Investigated sources, the transfer path; vibration modes and also noise radiation properties that can handle noise radiation from engine front and oil sump, timing transmission case and belt pulley. B & K 2- channel system used for data acquisition. 2031 type transducer and 4165 type microphonewereused to analyze different parameters.Regulartiming transmissioncasing made of cast aluminiumwith a stiffnessvalue of 12 give maximum noise radiation at frequency 630Hz - 4KHz.So, it is replaced by 5mm plexi-glass plate with reduction insound power of around 2-5 dB in a frequency range 500-630Hz [5]. Implemented a vibration isolator to the engine that can decrease the vibration transferred from the engine to the structure [6].

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2. METHODOLOGY

The methodology adopted here for reduction of noise and vibration of engine gear cover. Furthermost, the modal analysis of various engine components (i.e. crankcase, oil sump, gear cover) has been performed by using roving excitation method to determine its mode shapes and its resonant frequencies. Based on the resonant frequencies, the experimental vibration measurement has been carried out to verify the amplitude of vibration using the accelerometer. Similarly, the sound intensity technique was used to identify the noise source locations on the engine. As, per the results structural modifications has been made. For this current work we consider 3-cylinder inline water-cooled diesel engine. To which the measurement has been made at maximum rpm of 3000 which has power rating of 22 to 24 Hp. Methodology of experimental modal analysis and reduction of noise and vibration of diesel engine gear cover is shown below Fig.1 and Fig.2 respectively.

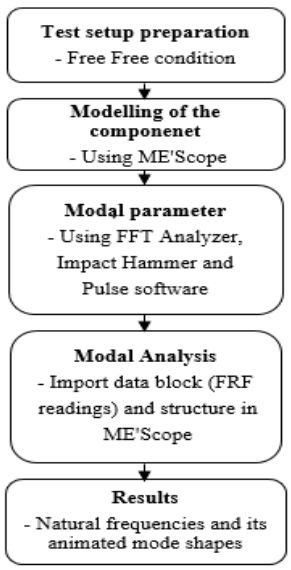


***Fig.1**. Modal analysis of engine components*

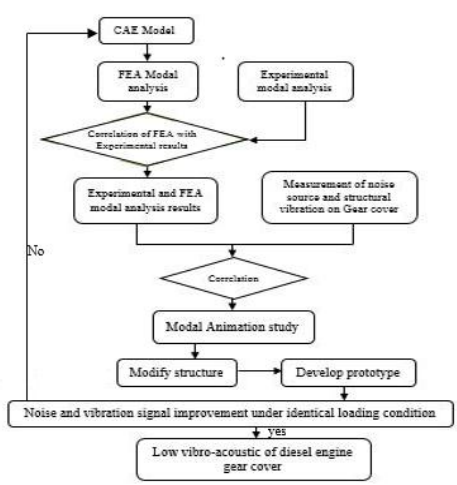


***Fig.2.** Methodology for noise and vibration reduction of diesel engine gear cover*

2.1 Theoretical Background

The equation of motion for an undamped system expressed in matrix notations is written as:

$[M]\{\ddot{u}\} + [K]\{u\} = \{0\}$(1)

Where,

[M] = Structural mass matrix

[K] = Structural stiffness matrix

$\{\ddot{u}\}$ = Nodal acceleration vector

{u} = Nodal displacement vector

For a linear system, free vibrations will be harmonic of the form:

$\{u\} = \{\phi\}_i \cos \omega_i t$

Where:

$\{\phi\}i$ = eigenvector representing the mode shape of the ith natural frequency

ω_i = ith natural circular frequency (radians per unit time)

t = time

Thus equation (1) becomes:

$-\omega_i^2[M] + [K]) \{\phi\}_i = \{0\}$ (2)

This equality is satisfied if either $\{\phi\}$ i = {0} or if the determinant of ([K] – w2 [M]) is zero. The first option is the trivial one and, therefore, is not of interest. Thus, the second one gives the solution:

?[K] 2 ω^2[M]?= 0 (3)

The above equation(3) represents the Eigen-value problem which may be solved for n eigen values and eigen vectors, which are the modes and natural frequencies of the structure.

2.2 Noisesource identification technique

This is the most significant step to measure the noise source identification in any components and structure. In this paper a sound intensity technique was used by placing one closely spaced microphone and single channel FFT Analyzer to measure the noise locations easier. Intensity of sound gives the frequency domain from the imaginary parts of cross section between the microphone and structure. The sound pressure level 'L_p' to be calculated as,[9]

$$L_p = 10 log_{10}(\frac{p^2}{p_{ref}^2})$$

Where p^2 is mean squared pressure and $P_{ref}$is reference pressure, it is equal to 2×10^{-5} Pascal for air borne sound.

2.3 Experimental setup of modal analysis

Tostudy the vibration characteristics of a component experimental modal analysis is performed. Every component has its own frequency at which it vibrates after providing initial displacement, this frequency is called as natural frequency of that component and the vibration pattern is called as mode shape. In this project three-cylinder diesel engine components are used for experimental modal analysis to find their natural frequency and mode shapes. A crankcase, oil sump, gear cover, are considered for experimental modal analysis. Fig.3 shows an actual view of Gear cover held in space at free-free condition for modal analysis. Mode shapes of Gear cover with first twisting and bending mode at their natural frequency 390 & 715 Hz as can be seen in Fig.4 and Fig.5 respectively. For experimental modal analysis all measuring equipment's are of B & K in NVH laboratory.

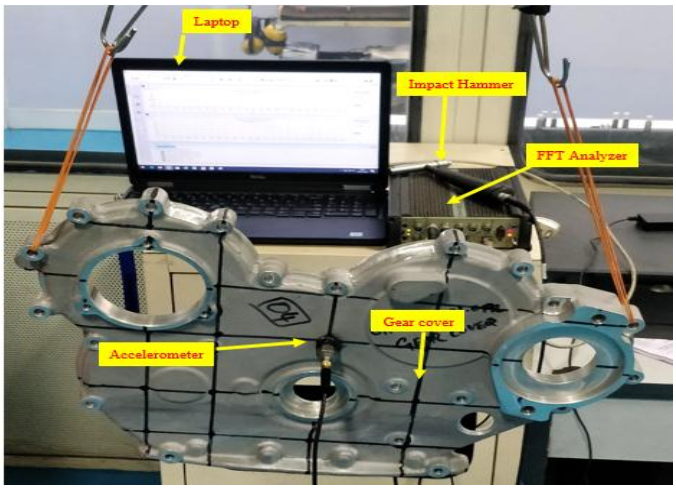


Fig.3. *Experimental setup of Modal analysis*

2.4 Engine components mode shapes

Dynamic analysis of engine parts provides the basic information for the component development process. The modal analysis is the opening assessment of the dynamic components of the engine. In this analysis, natural frequencies and mode shapes of the engine components are found as shown in below Fig. [4]& Fig. [5].

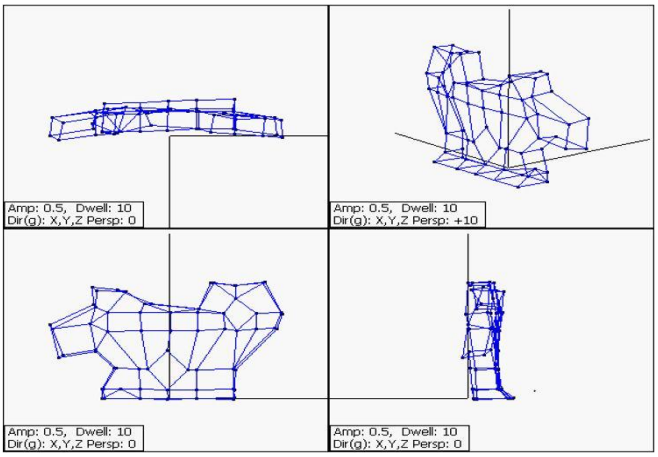


Fig.4. *Mode shape of Gear cover at 390 Hz*

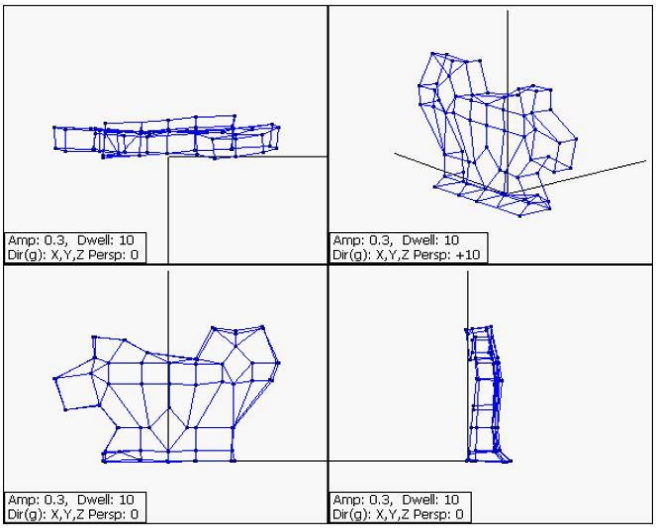


Fig.5. *Mode shape of Gear cover at 715 Hz*

2.5 Experimental modal analysis results

Experimental modal analysis results were identified on above-mentioned components which are having maximum plane surfaces. According to many research papers the primary source of structural noise radiation is through components having plane surfaces. Mostly, the vibration occurs from rotatry shaft, gears, reciprocating motion of pistons, etc. To perform experimental modal analysis single input and single output (SISO) technique is used. B & K impact hammer were used to perform hammering on components and a uni-axial accelerometer used to receive the impact responses of hammer. For data acquisition a 5 channel B & K Fast Fourier Transform (FFT) system is used, which is connected to computer as a data receiving and display unit. The results of FRF spectra can be seen in Fig.6 for Gear cover. Obtained natural frequencies are shown below Table I and the mode shapes of gear cover description is shown in below Table II.

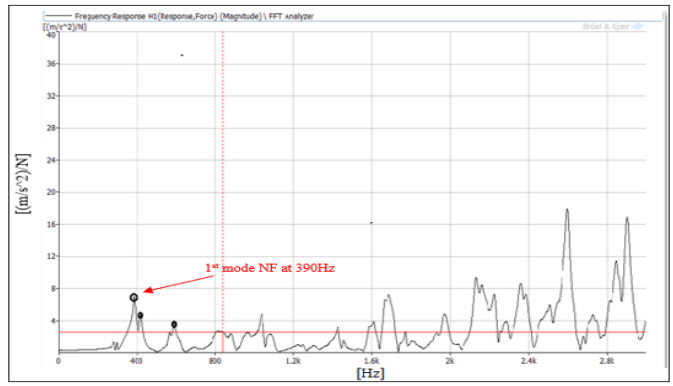


Fig.6 *Frequency response spectra of Gear cover*

Table I *Natural frequencies of Engine components*

NATURAL FREQUENCIES (HZ) EXPERIMENTAL DATA			
Sr. No.	Crankcase	Oil sump	Gear cover
1.	610	382	390
2.	1140	385	715
3.	1220	746	1010
4.	1260	947	1294
5.	1380	971	1794

Table II *Natural frequencies of Gear cover by FEA*

Mode No.	Natural Frequency (Hz)	Descriptions of mode shape
1.	390	Gear cover twisting
2.	715	Twisting and bending of Gear cover
3.	1010	Twisting of Gear cover
4.	1294	1st Global mode of Gear cover
5.	1794	Twisting in Oil filling cap side (LHS)
6.	1930	Twisting in RHS
7.	2380	2nd Global mode of Gear cover
8.	2465	3rd Global mode of Gear cover
9.	2990	Twisting and bending of Gear cover
10	3090	Twisting of Gear cover

2.6 Simulation in ANSYS

The finite element method (FEM) is the most popular simulation method to predict the physical behaviour of systems and structures.Although the FEM was originally

conceived to discover a solution to structural mechanics issues. Now it can be applied to a wide range of engineering fields where the physical description results in a mathematical formulation having typical differential equations that can be numerically solved. The Modal analysis of the engine components i.e. oil sump, gear cover, crankcase is performed in ANSYS 19.0. The various mode shapes obtained are then compared with those experimental tests in the NVH lab. CADmodelling of engine components is done by using CREO Software.

2.6.1 Meshing Efficiency

The final mesh settings were achieved after improving the mesh through a series of mesh size trials. Since modal analysis in FEA is performed in free-free condition, with no constraints and no forces engaged in this method, therefore meshing size will not impact the final results. Fig.7 shows the meshing of a gear cover.

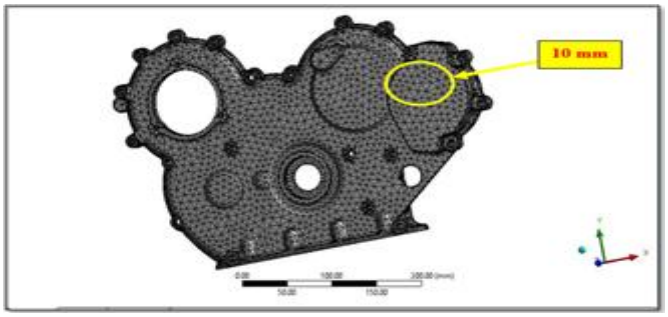


Fig.7 *Meshing of gear cover*

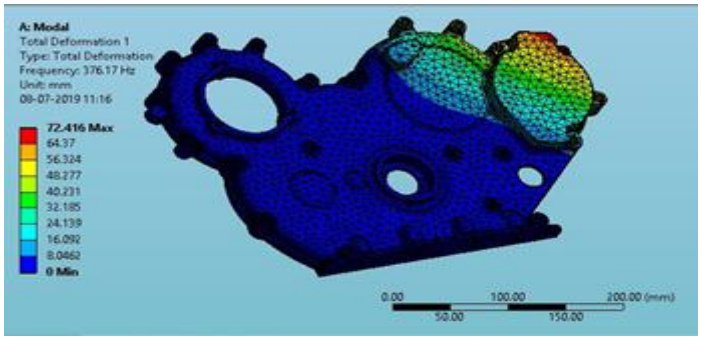
In the above figure shows the meshing of gear cover with an element size of 10mm which gives fine solution. This allows the structure to get reduction in percentage error comparing to experimental modal analysis results.

2.7 ANSYS Simulation results

FEA analysis is a time-consuming technique, but the results obtained in FEA can correlate with experimental method. Both natural frequencies and mode shapes will have to match with both results with an acceptable error. Gear cover (thickness = 4.5mm) is made of Aluminium alloy having density 2800 kg/m^3, Young's modulus 68800 MPa and Poisson's ratio 0.3. Oil sump (thickness = 2.3mm) is made of structural steel having density 7850 kg/m^3, Young's modulus 200000 MPa and Poisson's ratio 0.26. The FEA results of gear cover and oil sump were mentioned in below Table IIIand Fig.8.

Table III Natural frequencies in FEA

NATURAL FREQUENCIES (Hz)		
Sr. No.	Gear cover	Oil sump
1.	376.17	371.91
2.	675.58	372.78
3.	984.54	734.43
4.	1326.20	931.79
5.	1803.50	954.80
6.	1834.20	994.25



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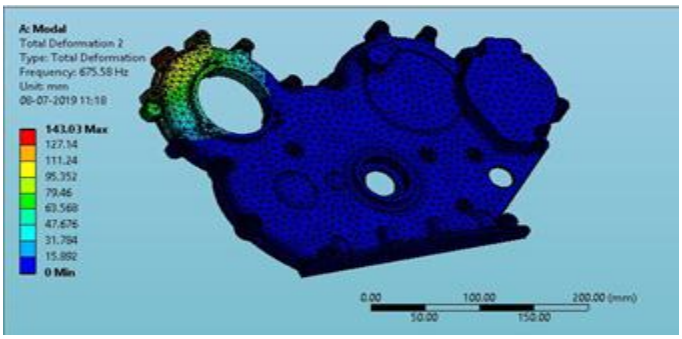

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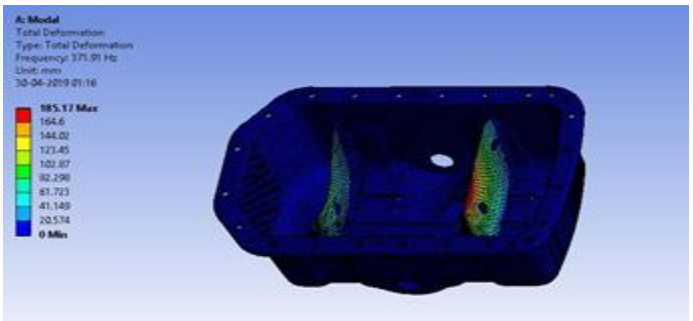

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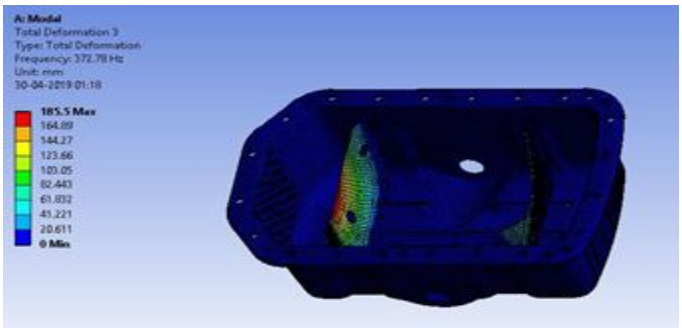


(d)

Fig.8 *Mode shapes of gear cover and oil sump natural frequencies GC (a) 376.1, (b) 675.5, and OS (c) 371.91, (d) 372 Hz*

The natural frequencies obtained from FEA modal analysis of gear cover is shown in above Table III. In Fig.8(a) and 8(b) of gear cover describes that at mode 1 the top right side of gear cover is twisting with a maximum deformation of 72.41 mm and at mode 2 shows left side of gear cover bending and twisting with a total deformation of 143.03 mm.

3. EXPERIMENTAL ANALYSIS OF ENGINE AT RUN UP CONDITION

The experimental study of engine structural vibration provides a major contribution to understand and to control many vibration Phenomenon which takes place in this experiment. The main aim of experimental vibration analysis is to determine the nature and its induced vibration response level from structure. The experimental vibration testing is carried out which is explained below;

3.1 Experimental vibration testing

The vibration level of mounting locations i.e. Crankcase surface, FIP side, Gear cover side, Rocker cover were measured at maximum rotating speed. The amplitude of structural vibration and its operating conditions were measured to check the harshness of the vibrations. So, that it will be very helpful to find out the source of induced vibration at engine rotational frequencies. The overall vibration level to be measured at perilous locations. The below Fig.9 shows the experimental vibration location at gear cover side.

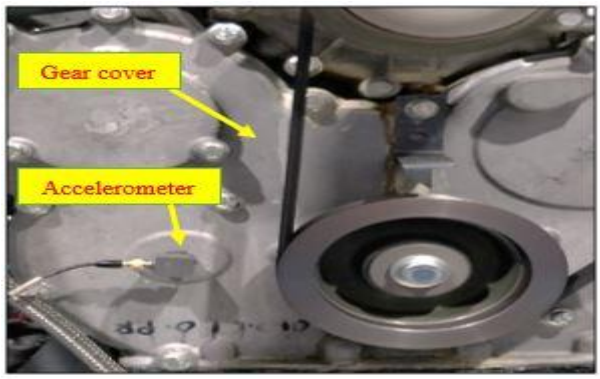


Fig.9 *Experimental vibration location at Gear cover side*

As shown in the above figure that the accelerometer is placed near to the gear end side to check the amplitude of vibration at two different speed one is at 1800rpm and other is at 3000rpm.The obtained results of experimental was further compared with gear mesh frequency of engine. To which it displays that the structure is resonating near to the gear mesh frequency. The gear mesh frequency is calculated and the actual gear cover maximum vibration results is shown below Table IV.

$$\text{Gear mesh frequency} = \frac{\text{RPM} * \text{No. of teeth}}{60}$$

Table IV *Actual maximum vibration levels at gear cover*

Direction	Frequency (Hz)	At 1800 rpm Gear mesh frequency (Hz)	Amplitude (m/s^2)
X	1060	1080	15.48
Y	1084	1080	5.66
Z	1076	1080	5.16
		At 3000 rpm	
X	1806	1800	18.25
Y	1810	1800	9.59
Z	1794	1800	14.27

4. EXPERIMENTAL METHOD TO DETERMINE NOISE INTENSITY

A sound intensity measurement technique is used to measure the noise of engine at run-up condition. It takes lot of time to complete the experiment but it gives good and reliable noise sources at any surface. The Instruments used to identify the noise source are microphone, 5- channel FFT analyzer and a computer system for data collection. Engine is running at 3000 rpm, to which a microphone is holding at a distance of 0.5m from engine which capture the noise and gives the noise frequencies of the running engine. The experimental noise source measurement setup and the contour plot results as shows in Fig.10 and Fig.11. The overall noise level of 93 dB(A) was observed for whole grid structure. The major contribution of noise level frequencies is 500 Hz, 800 Hz, 1250 Hz, 1600 Hz and 2000 Hz. The experimental noise results are shown in below Table V.

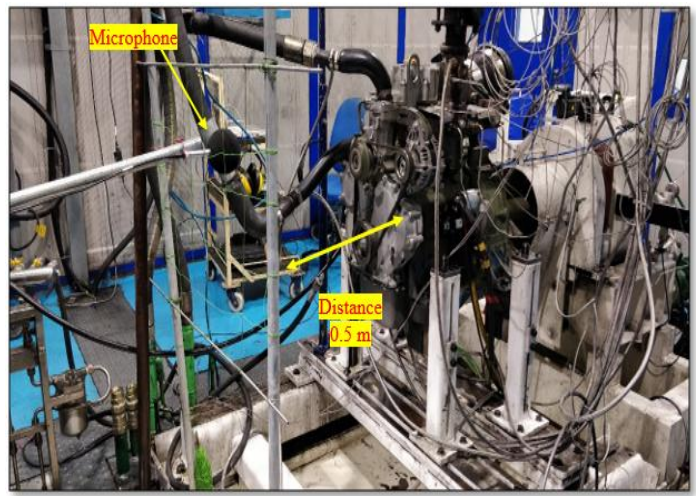
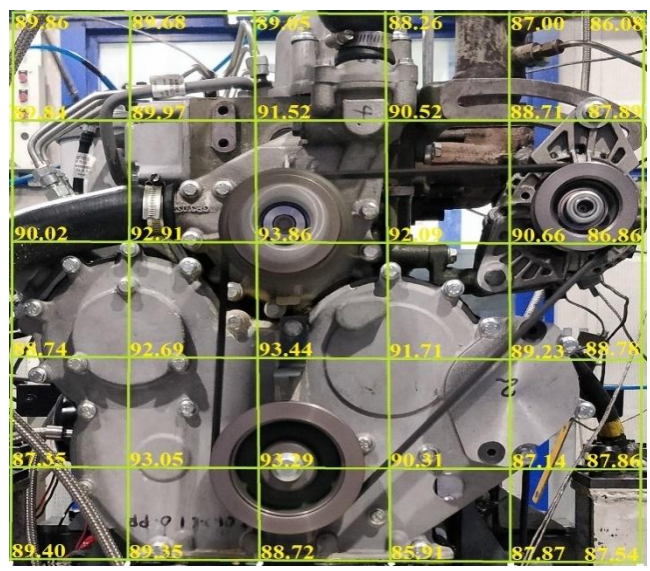
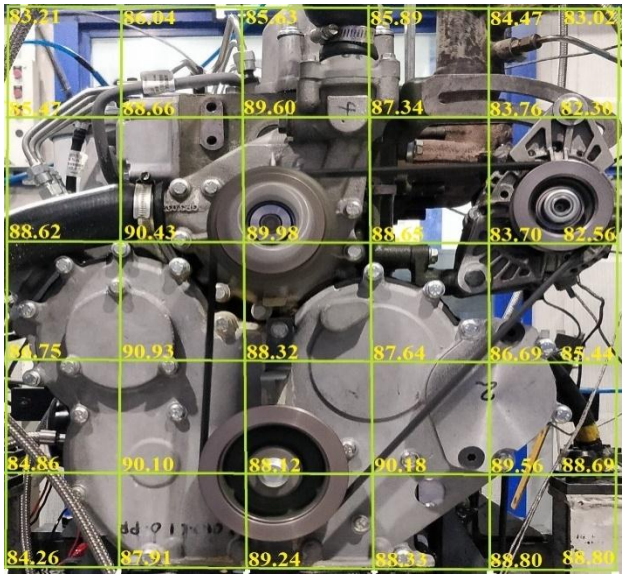
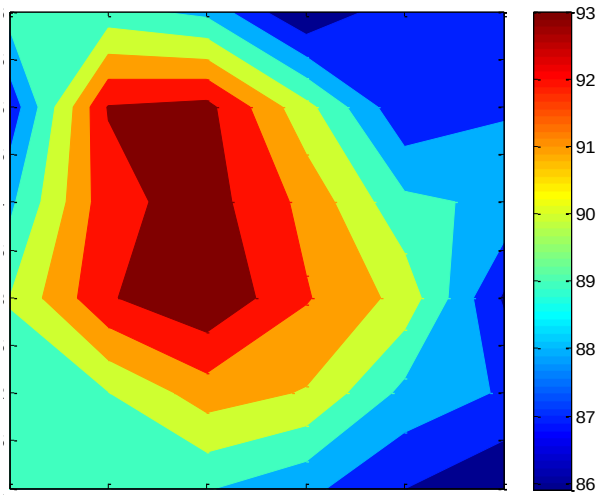


Fig.10 *Experimental setup for noise source identification*







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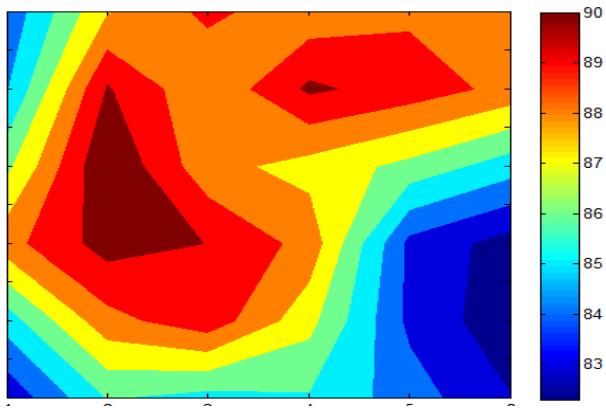


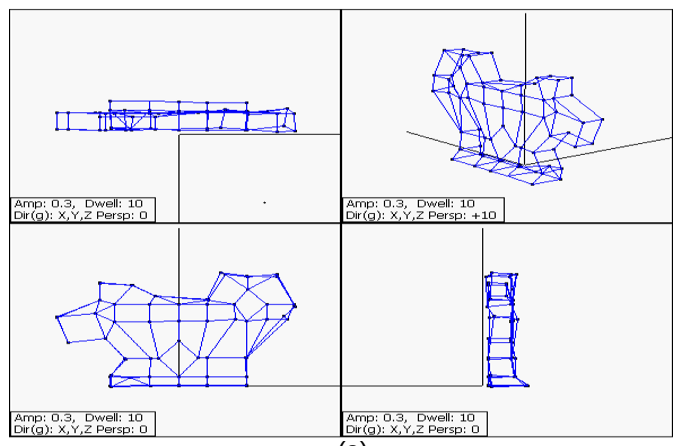
(b)

Fig.11 *(a) Overall noise level (b) Noise level at 1600 Hz of gear cover with grid structure*

Table V *Experimental noise source results*

Sl. No	Frequency (Hz)	Noise level [dB(A)]
1.	Overall	93.86
2.	2000	88.81
3.	1600	90.93
4.	1250	85.29
5.	800	85.61
6.	500	79.09

The above table describe the noise levels along with their frequencies at which the 1600 Hz illustrates major noise contribution of 90 dB(A) which is nearer to the gear mesh frequency. Due to mating of gear at high speed leads to create structural borne noise. The obtained results are marked in contour plots as can be seen in above Fig.11 are taken by using MATLAB 14a version software. It gives overall broad view towards the noise levels radiated by different region on a gear casing. As we know engine is running at 3000 rpm and its gear mesh frequency is 1800 Hz as mentioned above in experimental vibration testing. This running frequency is close to the natural frequency of gear cover i.e. 1794 Hz. The mode shapes calculated usingexperimental and FEA modal analysis can be seen in Fig.12.



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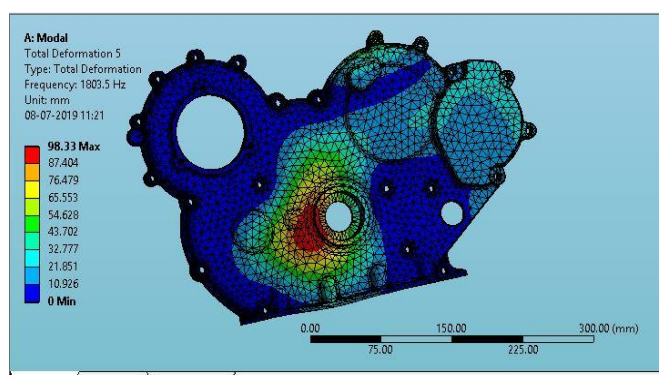


(b)

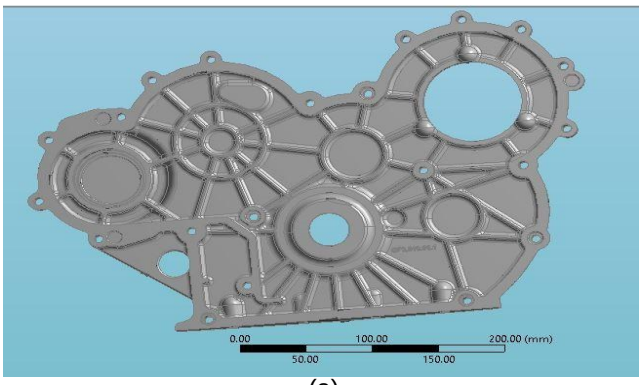
Fig.12 *(a) Experimental mode shape at 1794 Hz, (b) FEA mode shape at 1803 Hz.*

5. MODIFICATION ON GEAR COVER

5.1 Structural modification

The experimental results of noise and vibration analysis at gear cover side displays that the structure is vibrating more

nearer to the gear mesh frequency. Due to mating of gears at maximum rpm it produces noise too. Although, the mode shapes of gear cover shows bulged at the frequency of 1794 Hz experimentally and 1803 Hz by FEA are quite close to GMF. Based on the results it indicates that the structure should be modified to decrease overall noise and vibration at the above locations. The structural modifications has been done by adding an additional stiffeners instead of increasing whole masses. The Gear cover modifications in FEA and experimentally shows the reduction of amplitude of vibration and structural borne noise from the gear cover which is shown in below Fig.13 and Table [6], [7].



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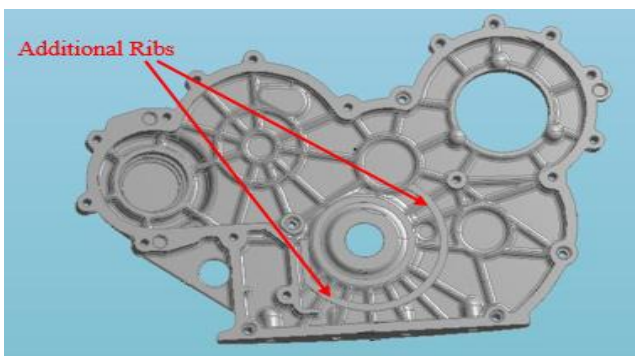


(b)

(c)

Fig.13 *(a) Actual gear cover (b) modified gear cover in FEA (c) manufacturing modified gear cover*

Table VI *Modified vibration level of gear cover*

Direction	Frequency (Hz)	Gear mesh frequency (Hz)	Amplitude (m/s^2)
	At 1800 rpm		
X	1060	1080	9.44
Y	1084	1080	3.33
Z	1076	1080	3.09
	At 3000 rpm		
X	1806	1800	10.95
Y	1810	1800	5.75
Z	1794	1800	8.56

Table VII Modified noise level results

Sl. No	Frequency (Hz)	Noise level [dB(A)]
1.	Overall	89.05
2.	2000	84.80
3.	1600	88.44
4.	1250	84.96
5.	800	82.61
6.	500	76.79

The modified FEA mode shapes results of gear cover at resonance shifting locations as shown in below figure 14.

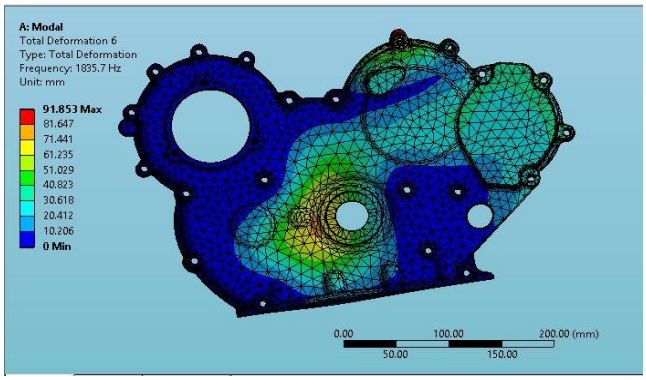
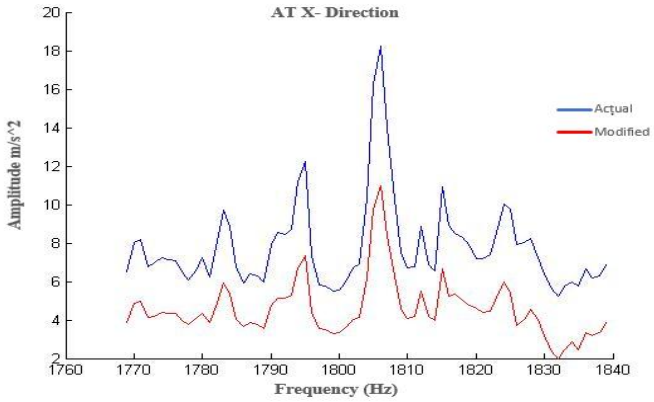


Fig.14 *Modified FEA mode shapes at 1836 Hz*

5.2 Sensitivity analysis for vibration on gear cover

The purpose of sensitivity analysis is to check the above structural modifications and a desired change in the structural properties. The experimental vibration analysis is carried out at 3000rpm to check the percentage of reduction in vibration. The comparison of Actual and modified gear cover results is shown below Table VIII. Along with their combined graph presented as shown below figure 15.



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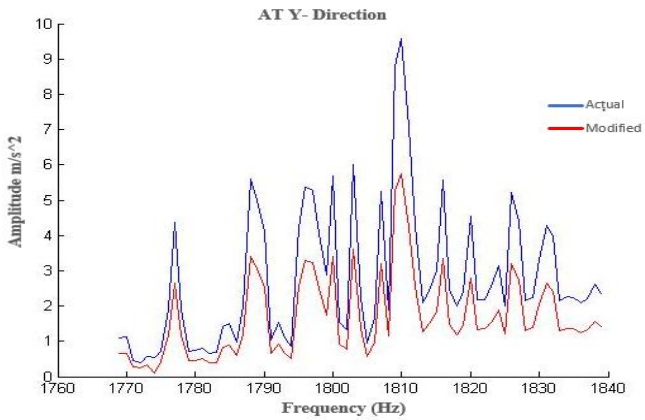
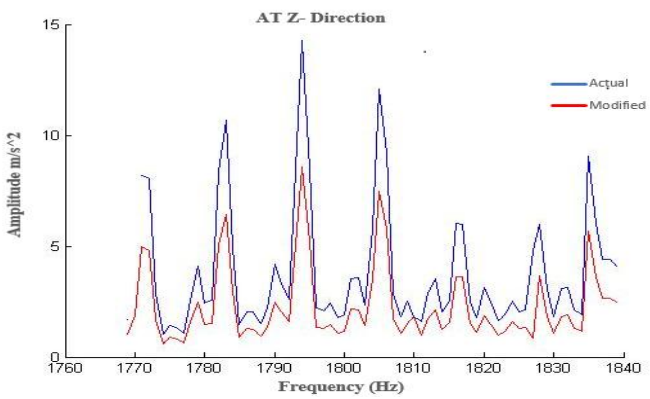


(b)



(c)

Fig.15 *Experimental Comparison of amplitude of vibration of actual and modified gear cover at 3000rpm (a) x-direction (b) y-direction (c) z-direction*

Table VIII

Experimental vibration comparison table between actual and modified gear cover

Direction	Actual data Freq (Hz)	GMF (Hz)	Amp. (m/s^2)	Modified data Freq (Hz)	GMF (Hz)	Amp. (m/s^2)	% Redn
At 1800 rpm							
X	1060	1080	15.48	1060	1080	9.44	39.85 %
Y	1084	1080	5.66	1084	1080	3.33	40.01 %
Z	1076	1080	5.16	1076	1080	3.09	40.19 %
At 3000 rpm							
X	1806	1800	18.25	1806	1800	10.95	40.39 %
Y	1810	1800	9.59	1810	1800	5.75	40.09 %
Z	1794	1800	14.27	1794	1800	8.56	39.39 %

GMF* = Gear mesh frequency
Amp* = Amplitude
Redn* = Reduction

The experimental vibration analysis was checked as it shows the overall reduction of induced vibration are 35-40%. Table VIII shows the results of actual and modified gear cover with percentage of reduction.

5.3 Noise Sensitivity analysis of gear cover

The main motive of sensitivity analysis is to check the modified data with actual one with a reduction of errors. The results of FEA and experimental shows the frequency of 1803 Hz and 1794 Hz are close to the gear mesh frequency of 1800 Hz with a percentage error of -0.51% which is quite acceptable as per companies' benchmark. The noise level comparison of Actual and modified gear cover results is shown below Table IX and the Fig.16 shows the 3-4 dB(A) noise reduction of overall gear cover along with the modified contour plots.

Table IX *Experimental noise level comparison table between actual and modified gear cover*

Sl. No.	At 3000 rpm: Actual results Freq (Hz)	Actual results Noise level [dB(A)]	Modified results Freq (Hz	Modified results Noise level [dB(A)]	Redn Level [dB(A)]
1.	Overall	93.86	Overall	89.05	4.8
2.	2000	88.81	2000	84.80	4.0
3.	1600	90.93	1600	88.44	2.4
4.	1250	85.29	1250	84.96	1.3
5.	800	85.61	800	82.61	3.0
6.	500	79.09	500	76.79	2.3

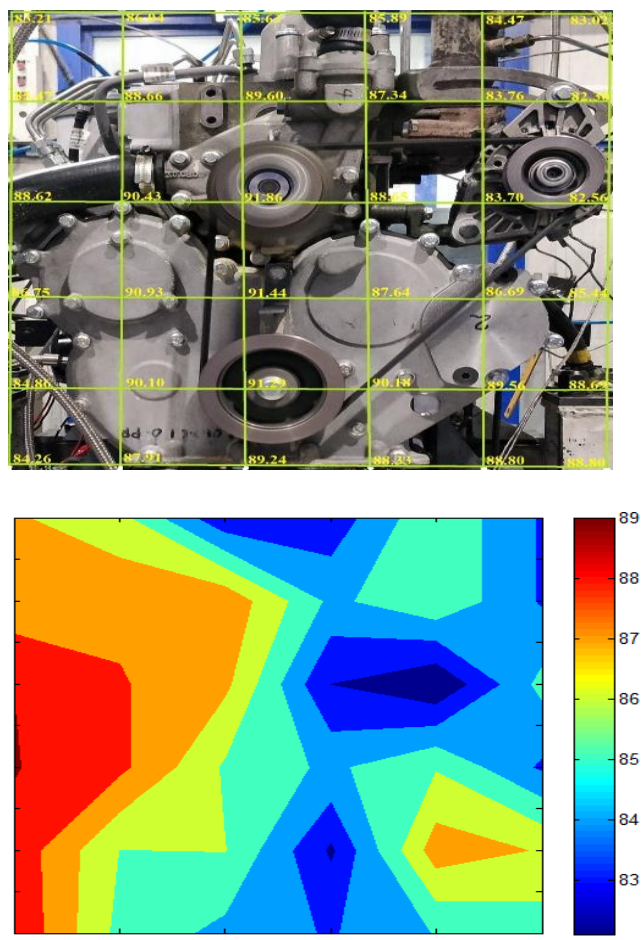


Fig.16 *Modified noise level contour plot*

6. RESULTS AND DISCUSSIONS

A successful attempt has been made to achieve the solution for the reduction of overall structural noise and induced

vibration on gear cover by using the right combination of FE modal analysis and experimental techniques.

A. The Experimental modal analysis of actual and modified results comparison of gear cover is shown in below Table X with a percentage error below ±10% which is quite acceptable as per companies' benchmark. Modal analysis is performed to find out the resonance frequency from that we found the frequency of 1794 Hz is close to the gear mesh frequency. Which is resonating noise and vibration on gear cover.

Table X

Experimental modal analysis results of actual and modified gear cover

Sl. No.	Actual Exp.	Actual FEA	Actual % Error	Modified Exp.	Modified FEA	Modified % Error
1.	390	376.17	3.51	375	381.46	-1.60
2.	715	675.58	5.42	655	684.32	-4.42
3.	1010	984.54	2.54	984	998.32	-1.42
4.	1294	1326.20	-2.41	1280	1343.9	-4.92
5.	1794	1803.5	-0.51	1784	1836.60	-2.91
6.	1930	1834.2	4.97	1900	1865.80	1.84

B. The experimental vibration results indicate that the induced vibration is coming from structure of gear cover. Due to coinciding the gear mesh frequency, leads to creates both noise and vibration. To reduce this induced vibration a structural modification has been done on gear cover which shows the reduction by 35 to 40%. As the sensitivity analysis is prescribed as above section v and the % reduction of amplitude graph is shown below Fig.17.

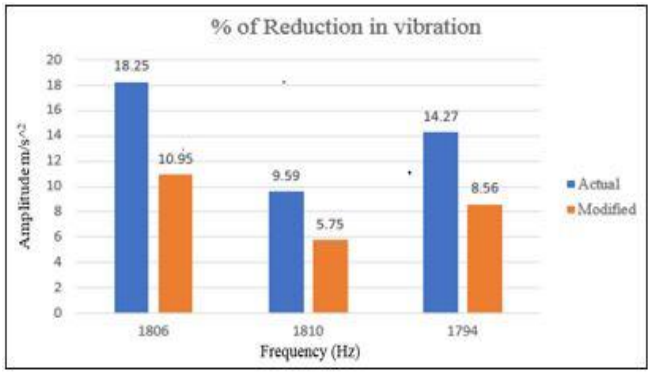


Fig.17 *comparison of vibration graph of actual and modified gear cover*

As it shows the unmodified gear cover amplitudes in three different direction (x,y,z) at 3000 rpm are 18.25, 9.59 and 14.27 m/s^2 at frequencies 1806, 1810,and 1794 Hz. While in case of modified its shows an amplitude of 10.95, 5.75, and 8.56 m/s^2 at maximum rpm. The modified vibration spectra in z direction is shown below Fig.18.

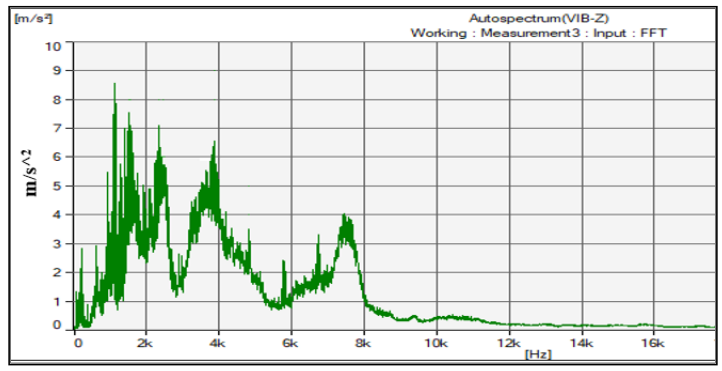


Fig.18 *Modified gear cover spectra in z direction*

C. Similarly, the structural borne noise is coming from the same frequency level of 1794 Hz experimentally. To reduce the structural borne noise, a modification is done by adding an additional stiffener which shows the reduction of structural borne noise by 3 to 4 dB(A). The above Fig.16 shows the effectiveness of structural modifications contour plot.

7. CONCLUSION

From the results and discussion, we come to the conclusion that the Experimental techniques is very useful to detect the vibration and noise analysis problem and help in reduction of overall noise and vibration level. The experimental and FEA modal analysis were performed which gives structure mode shapes and to find out the resonance frequency. A sensitivity analysis was measured, which show less than ±10 % Error. The experimental vibration analysis shows the structural modifications which is carried out from modal analysis, are quite useful for reduction of overall vibration level of gear cover by 35 to 40 %. Similarly, the sound intensity method is performed to analysis the noise level and the modification shows the reduction of noise up to 3 to 4 dB(A).

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