Print ISSN: 0976-2876 Online ISSN: 2250-0138

Available online at: http://www.ijsr.in

INDIAN JOURNAL OF SCIENTIFIC RESEARCH

DOI:10.32606/IJSR.V13.I1.00005

Received: 28-04-2022 Accepted: 16-07-2022 Publication: 31-08-2022

Indian J.Sci.Res. 13 (1): 43-49, 2022 Original Research Article

MODAL ANALYSIS OF GM 16-645 E3 ENGINE BLOCK USING FINITE ELEMENT METHOD

ALI SOLEIMANI[a1], SAEID SHAHRAKI[b], MOHAMMAD ALI REZVANI[c] AND ASGHAR NASR[d]

[a]Shiraz Urban Railway Organization, Shiraz, 7193689711, Iran

[b]Department of Mechanical Engineering, University of Zabol, Zabol, Iran

[cd]Iran University of Science and Technology, Tehran, Iran

ABSTRACT

In present article, natural frequencies of GM 16-645 E3 engine block has been considered using modal analysis. At first, engine block has been simulated and after that combustion and present mechanical load frequencies have been calculated using available computational relationships. Also, vibration level of block was acquired by finite element method. Thereafter, block was analyzed by modal analysis, by using three different methods of AMS, subspace and lonczos. Acceleration test was done to validate the results. combustion frequencies and main harmonics have been calculated using Fast fourier transform. Frequencies resulting from tests had an excellent accordance with results of finite element method. Finally weak parts of the block enduring external cyclic forces are discussed from the strength point of view (related to mode shapes and frequency amplitude).

KEYWORDS: Modal Analysis, Engine Vibration, Block, Diesel Engine

By development of diesel engines technology and necessity of noise reduction in environmental standards, block is the base and main part of the engines (Guo and Gao, 2012), Effect of secondary stimulating forces on body in different situations causes vibration of block and other parts installed on it and this oscillation causes noise radiation (Tan Daming, 1993). Generally by analyzing harmonic response of engine block by modal analysis, it is possible to identify weak parts susceptible to fatigue fracture caused by external periodic forces (Vulli *et al.*, 2009). Noise radiation has a direct relation with oscillations. Radiated noises are the reason of most of parts' vibrations. So block vibrations have a remarkable influence on creating noises. Due to fast advance of computer technology, it is possible to use finite element method to reduce noises and vibrations (Lalor, 2007). To deal with complicated structures with variable loads, finite element is more effective than any other methods to compute modal analysis of the model. Rahmatalla has investigated the integrity of old bridges by using modal analysis (Rahmatalla, 2014). Carlucci *et al.* (2006) perfomed an experiment on a four cylinder diesel engine (D.I, FIAT 1929 cm3 TDID 154 D 1.000) with GARRETT TD 2502 turbo charger. This system was equipped with a control unit and improvement of injection parameters like injection strategy (number of injection systems) ,injection timing and injection pressure. By measuring block acceleration, they were able to reduce engine noises to some extent. Jie mao (2012) reduced the noises by adding some beams to block of a four cylinder engine. Primarily he identified the disturbing oscillations and then analyzed natural modes by finite element method. At first, he tested and investigated finite element method and then compared the disturbing frequency modes obtained from tests with modes obtained from finite element method. Through this procedure, he approved the validity of finite element method. Then he stimulated the engine block by boundary conditions like internal pressure of cylinder, impact force of valves, tangent force of valves and latitudinal forces. He optimized the system in axial direction (TS) and anti-axial direction (ATS) and reduced the sound power of radiated noises to 1 dB. By the way, there is a lot of investigation by modal analysis on different fields. This present article deals with dynamic harmonic response of engine block by using modal analysis.

LOCOMOTIVE ENGINE TEST

Locomotive engine test has been done to validate finite element method. This test is performed in pull force office of railway on GT26 locomotive that was ready to undergo load test.

EXPERIMENT AND MEASURED PARAMETERS

Acceleration test on locomotive engine block was performed in selected points using accelerometer as shown in figure 1 and measured data were acquired using a Data logger TMR 7200 device and sent to a computer.

[1]Corresponding author

Figure 1: Measured points with accelerometer

As it's seen from figure 1, first accelerometer was located in back part of locomotive and the second accelerometer was located near the generator at the middle of the engine. Measurement was done in a 30 seconds duration that during this range of time, engine proceeds from it's minimum round to maximum (513 to 900 RPM).

Figure 2: Location of point 1 for acceleration measurement

RESULTS AND DISCUSSION

To analyze the acceleration results, results of experiment must be taken to frequency field. One way is to use Fourier's fast transform (FFT,DFT). Also to obtain velocity and displacement amounts, we should integrate the data with respect to time. To meet this purposes, a specific code has been written in Labview software to transform data to acceleration, velocity and displacement in time and frequency fields. A medium of this software is shown in figure 3.

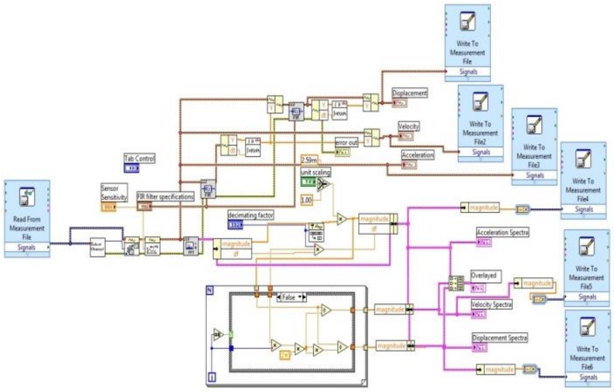
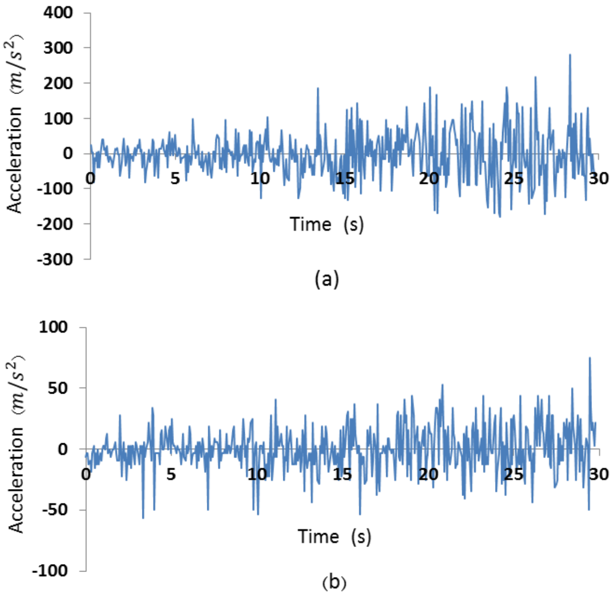


Figure 3: A medium of Labview software for transforming accelerometer datas

Generally, vibration are measured as amount of root of velocity squares, that is proportional to energy content of vibration. Sometimes, amplitude and acceleration of vibration are also measured.

Vibration measurement is used to recognize proper functioning of most of industrial equipments and if according to standards, vibration were trespassing some defined ranges, the device must be reviewed and fixed.

In this article, vibration level obtained from tests has been compared to ISO 10816-6 standard. According to this standard, locomotive engine lays between 6 and 7 level. So if it works properly, having oscillations between 1.1 to 112 sensitization range is admissible. So according to this standard and subjects expressed above, vibration of engine up to 70.1 m/s2 is admissible. Block vibration for points 1 and 2 in three directions of X,Y,Z has been measured and shown in figures 4 and 5.



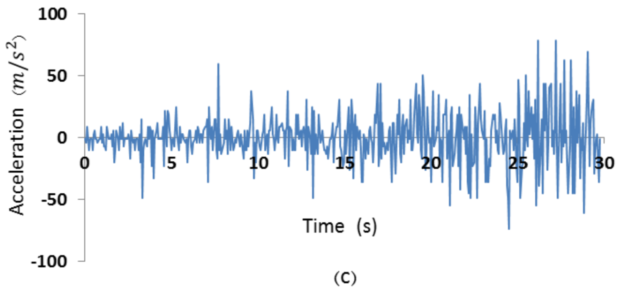


Figure 4: Acceleration time diagram for point 1 a) in X direction b) in Y direction c) in Z direction.

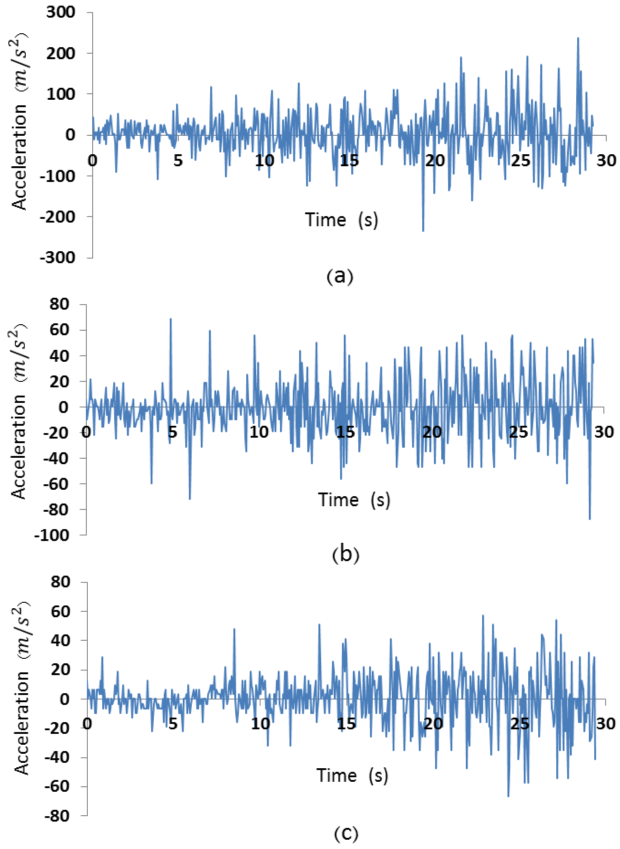


Figure 5: Acceleration time diagram for point 2, a) in X direction, b) in Y direction, c) in Z direction.

Table 1: Amount of root of squares obtained from tests for selected points

RMS (m/s^2)	X Direction	Y Direction	Z Direction
Point 1	64	18	20
Point 2	56	21	17

According to ISO 10816-6 standard, it can be said that engine is working properly and Results of simulation are compared in following sections.

Stimulation frequency and mechanical loads in engine block

Important parameters of GT26 diesel engine are listed in table 2.

Table 2: Main parameters of diesel engine

Parameters	Values
Number of cylinders	16
Cylinder diameter	9.0625 in
Idling condition	900rpm
Maximum torque condition	900rpm/3290hp
Compression ratio	14.5:1

In this section, engine stimulating frequencies and present loads on block (caused by combustion) are considered and Minor loads have been ignored.

Stimulation Frequencies of Engine Vibration

Each combustion in each cylinder, exerts a torque pulse around crankshaft axis to engine block. In every 2 rounds of crankshaft rotation, each cylinder combusts once. So firing frequency of engine is equal to half of the multiplication result of engine speed (RPM) in amount number of cylinders (Baranescu, 1999).

$$Firing\ Frequency = \frac{speed\,(rpm)}{60} \times \frac{cylinder\ number}{2} \quad (1)$$

According to the fact that diesel engine has a minimum speed of 513 rpm and maximum speed of 900 rpm, it can be expressed:

$$Firing\ Frequency_{\min} = \frac{513(rpm)}{60} \times \frac{16}{2} = 68.4Hz \quad (2)$$

$$Firing\ Frequency_{\max} = \frac{900(rpm)}{60} \times \frac{16}{2} = 120Hz \quad (3)$$

That working frequency of engine is between 68.4 to 120 Hz. So sound analysis must be performed in this range. Stimulation of each combustion is not a simple sinusoidal function. There are also forces on engine with the same harmonic frequency of combustion frequency. Additionally combustion of different cylinders is not identical and because of small differences in fuel injection time, injection velocity and cylinder volume, in high death point, pressure and productive power of cylinders are different, but in this project, these combustion forces are assumed to be identical.

Mean Effective Pressure

One of the important operative parameters is the mean effective pressure (MEP) and it's the result of dividing work done during each cycle by piston scanned volume (Pulkrabek, 2004).

$$mep(Kpa) = \frac{P(KW) \times n_R \times 10^3}{V_d(dm^3)N(rev/s)} \quad (4)$$

$$mep = 141psi = 141 \times 6.895Kpa = 0.972Mpa \quad (5)$$

Maximum Present Pressure in Cylinder

This pressure is obtained from available standards for diesel engines.

$$P_{\max} = 1350\,psi = 1350 \times 6.895Kpa = 9.308MPa \quad (6)$$

Forces Resulting from Combustion

$$F = P_{\max} \times A_{piston} = 9.308 \times 10^6 \times 0.0416 = 387.212KN \quad (7)$$

Calculation of the Force of Main Bearing

Force of main bearing is obtained from the formula 8 (Norton, 199).

$$F = P_{\max} \times A_{piston} - F_1 \quad (8)$$

$$F_1 \cong m_A\left(r\omega^2 \cos \omega t\right) + m_B\left[r\omega^2\left(\cos \omega t + \frac{r}{l}\cos 2\omega t\right)\right] \quad (9)$$

$$F_1 = 40\left(0.127 \times 64.23^2\right) + 60\left[\left(.127 \times 64.23^2\right)\left(1 + \frac{0.127}{0.5588}\right)\right] = 59538.36N \quad (10)$$

After getting F_1 force, force of main bearing is :

$$F = 387212 - 59538.36 = 327673.64N \quad (11)$$

Force on Each Cylinder Head Retaining Bolts

$$F_{bolt} = \frac{P_{\max} \times A_{piston}}{4} = \frac{9.308 \times 10^6 \times 0.0416}{4} = 96.803KN \quad (12)$$

Finite Element Model of Block

The engine block has been made according to the available sketches in Rail Traction company using Solidworks software. Then by exerting mechanical loads, natural frequency of block and vibration level has been calculated by Abaqus software. Also by sealing the holes that are connected to the block by screws of bearing cap, boundary conditions are applied in three directions. Finite element model of block contains 208019 tetrahedron elements and block is made of steel 37 which it's mechanical properties are including Density 7800 kg/m^3, Young modulus 210 GPa and Poison's ratio 0.28.

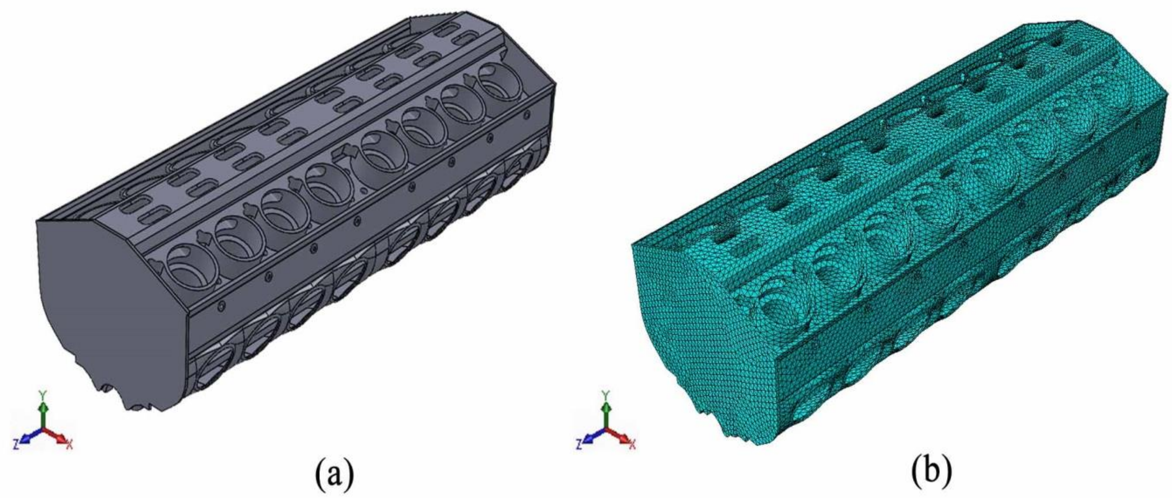


Figure 6: a) 3 D model of block b) finite element model of block.

COMPARISON OF SOFTWARE RESULTS WITH EXPERIMENTAL RESULTS

To validate finite element method, simulation results are compared with test results. Figure 13 shows vibration level obtained from experiments and simulation. Mean amount of root of squares obtained from experiments and simulation for point 1 are 94.43 and 84.15 dB respectively and are 94.43 and 90.68 dB for point 2. Error percentage for points 1 and 2 is 6 and 4 percent respectively. So validity and precious of finite element method can be used for other simulations. In figures 14 and 15, test and simulation results of vibration for two selected points in x,y,z directions are compared. Amount of mean root of squares and their error percentage are listed in tables 3.

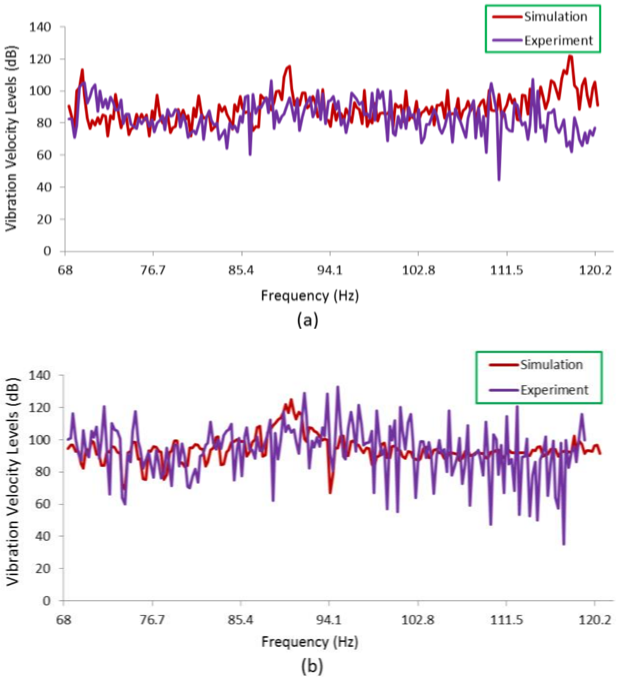


Figure 7: Comparison of vibration levels by experiment and simulation a) for point 1 b) for point 2

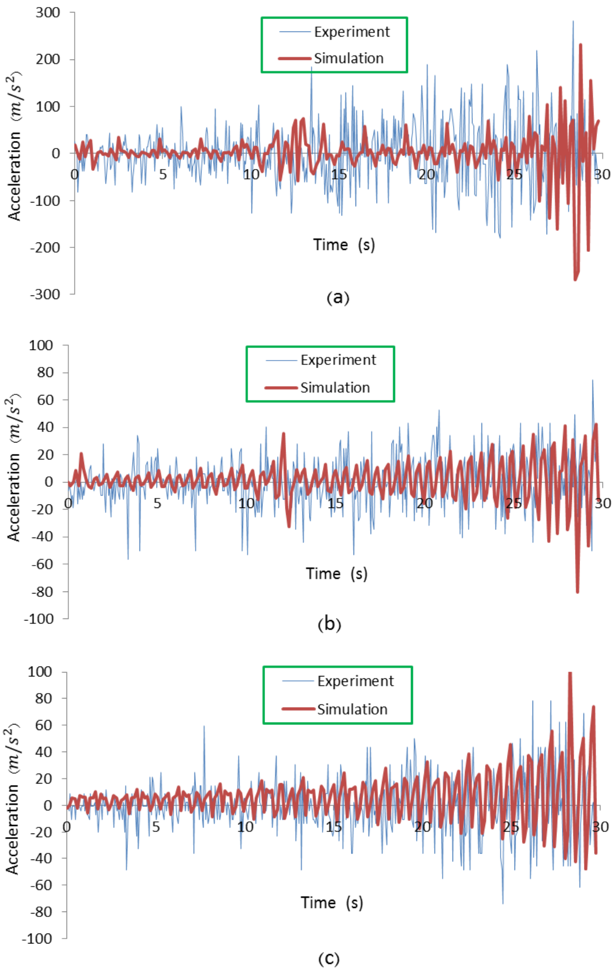


Figure 8: Comparison of acceleration of oscillations by experiment and simulation for point 1 a) in x direction b) in y direction c) in z direction

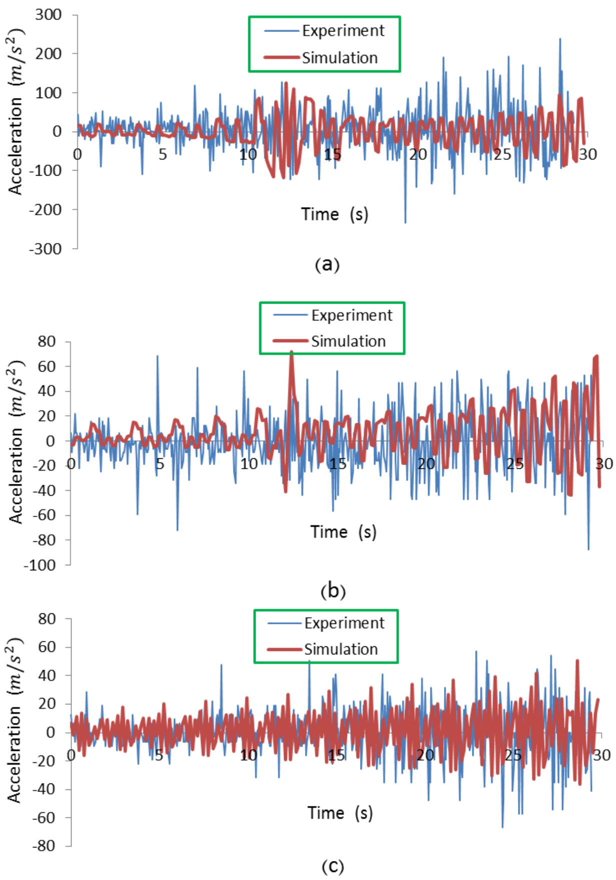


Figure 9: Comparison of acceleration of oscillations by experiment and simulation for point 2 a) in x direction b) in y direction c) in z direction

The maximum error is in x direction and it's related to ignoring effects of generator and engine coupled forces in simulation. Amount of error in the other directions is little and negligible.

Actually the reaction of vibration torques is around the torsion axis (roll) and more than any other factors, it causes engine vibration in the roll axis direction. Basically, torque pulses around the roll axis is the main source of vibration in big engines. So the most important natural frequency of engine set and vibration insulators, is the natural frequency of roll and for isolation, it's necessary to design this frequency less than combustion frequency.

Table 3: Mean root of square of acceleration for points 1 and 2 with mm/s^2 dimension

Point Number	Direction	Simulation	Experimental	Error Percent
1	X	49.082	64.13	23.46
	Y	14.83	18	17.62
	Z	19.96	20.03	0.36
2	X	42.92	56.35	23.84
	Y	20.34	21.89	7.038
	Z	16.72	17.048	1.915

FINITE ELEMENT MODAL ANALYSIS OF ENGINE BLOCK

Kang *et al.* (1998) have compared different numerical solution methods by computer, using modal analysis. They performed their analysis on a crankshaft and compared Lonczos, subspace and AMS (Automatic Multi-level Substructuring) methods and concluded that AMS method is the best one to obtain natural frequencies and mode shapes. Similarly, first ten natural frequencies of block were calculated numerically by abaqus software using Lonczos, subspace and AMS methods. These frequencies were calculated without excitation and results are listed in table 4.

Four types of first frequency modes is presented in figure 10. As seen in table 4, answers obtained from Lonczos and subspace methods are equal and difference of these methods is only in their solving time. Because the block of these type of engines is not uniform and solid, first modes shape occurs in low frequencies. Also their first frequencies is related to their weak parts, as shown in figure 5 and maximum variations is related to red area that is almost 32 micrometer.

Table 4: First ten natural frequencies of engine block by Lonczos, subspace and AMS methods

Mode order/method	Lanczos	Subspace	AMS
1	125.72	125.72	125.87
2	167.31	167.31	167.75
3	173.75	173.75	174.3
4	179.84	179.84	180.41
5	192.43	192.43	193.01
6	265.96	265.96	267.58
7	269.58	269.58	271.02
8	279.73	279.73	281
9	295.21	295.21	297.05
10	312.22	312.22	314.17

First mode has been occurred at frequency of 125.87 Hz. This mode shows torsional deformation around z axis. Second mode has been occurred at frequency of 167.75 Hz, showing bending caused deformation around x axis. Also this mode shows that upper edges of block are weak and these parts are more susceptible to vibration compared to other parts. Third and fourth mode occur at frequencies of 174.30 Hz and 180.41 Hz respectively and show bending caused deformation around x and y axes.

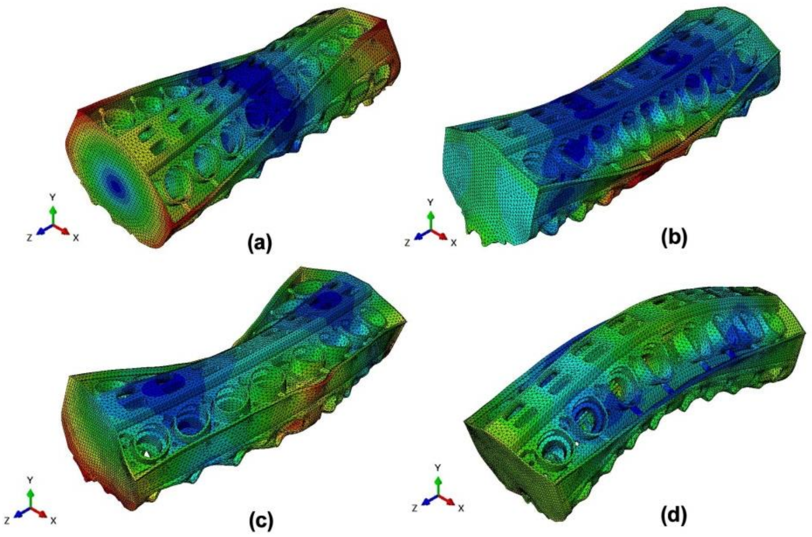


Figure 10: Four mode shapes of block. Firs Mode shapess: (a) first mode shape; (b) second mode shape; (c) third mode shape; (d) fourth mode shape

According to figures 10-a and 10-c and occurred distortions at upper parts of block, we can conclude that upper parts of block are weak and susceptible to more vibration. Also middle parts of block having blue color, have more strength than the other parts.

To validate obtained frequencies, fast Fourier's transform has been done on test results and presented in figure 11. As seen in this figure, main peaks are related to combustion frequency and harmonics of block. frequency of high value peaks is related to engine harmonics (or natural frequency of engine), that it's first harmonics are occurred almost at frequency of 123 Hz and is in the range of simulation answer and it has only a difference of 1.6 percent with the computational solution. This confirms validity of finite element method.

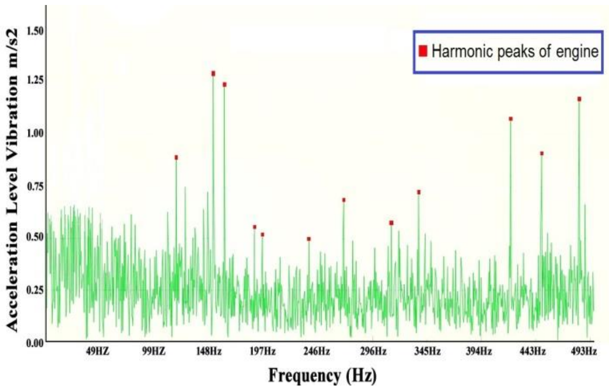


Figure 11: FFT of acceleration of point 2, main peaks are related to combustion frequencies and block harmonics

CONCLUSION

In this project GM 16-645 E3 engine block has been investigated. At first engine block was simulated and using available computational relationships, natural frequencies and mechanical loads present in block were calculated. After that by applying these forces to block and using finite element method, natural frequency and vibration level of block were obtained. To validate applied analysis, acquired results of analysis were compared to results of acceleration test performed in Rail Traction company. According to the obtained results, maximum error was in x-direction and its amount was about 23 percent and its reason was ignoring the effects of engine coupled forces. Additionally according to results, it's understood that the maximum vibration torque is around the torsion axis X and It more than any other factors, causes the vibration of engine in this direction. Additionally from modal analysis results, it's understood that the upper parts of engine are weak and susceptible to more vibration.

REFERENCES

Baranescu R. and Challen B.J. 1999. Diesel engine reference book.Reed Educational and Professional Publishing Ltd., 260301.

Carlucci A.P., Chiara F.F. and Laforgia D., 2006. Analysis of the relation between injection parameter variation and block vibration of an internal combustion diesel engine. Journal of Sound and Vibration, **295**(1): 141-164.

Guo P.P. and Gao W.Z. 2012. FEM Analysis on Block Vibration of Diesel Engine. Applied Mechanics and Materials, **155:** 1086-1089.

International Organization for Standardization, Geneva, 1995. Mechanical vibration-evaluation of machine vibration by measurements on nonrotating parts: part 6: reciprocating machines with power ratings above 100 kW.

Kang Y., Sheen G.J., Tseng M.H., Tu S.H. and Chiang H.W., 1998. Modal analyses and experiments for engine crankshafts. Journal of sound and vibration, **214**(3): 413-430.

Lalor N., 2007. Finite Element Optimization Technique of Diesel Engine Structures, SAE 820437.

Mao J., Hao Z.Y., Jing G.X., Zheng X. and Liu C., 2012. Sound quality improvement for a four-cylinder diesel engine by the block structure optimization. Applied Acoustics.

Norton R.L., 1999. Design of machinery: an introduction to the synthesis and analysis of mechanisms and machines (pp. 104-5). WCB McGraw-Hill.

Pulkrabek W.W., 2004. Engineering fundamentals of the internal combustion engine, **2:** New Jersey: Pearson Prentice Hall.

Rahmatalla S., Hudson K., Liu Y. and Eun H.C., 2014. Finite element modal analysis and vibration-waveforms in health inspection of old bridges. Finite elements in analysis and design, **78:** 40-46.

Tan Daming, 1993. Engine Vibration Control, Southwest Jiao Tong University Press.

Vulli S., Dunne J.F., Potenza R., Richardson D. and King P., 2009. Time-frequency analysis of single-point engine-block vibration measurements for multiple excitation-event identification. Journal of Sound and Vibration, **321**(3): 1129-1143.