

NOISE AND VIBRATION ANALYSIS ON V6 ENGINE

Submitted in partial fulfilment of the requirements for the award of degree

BACHELOR OF ENGINEERING

In

AUTOMOBILE ENGINEERING

by

K. SIVA THANIGAI (37860036)

N V. DEVENDRA SAI (37860007)



**DEPARTMENT OF AUTOMOBILE ENGINEERING
SCHOOL OF MECHANICAL ENGINEERING**

SATHYABAMA

**INSTITUTE OF SCIENCE AND TECHNOLOGY
(DEEMED TO BE UNIVERSITY)**

Accredited with Grade "A" by NAAC I 12B Status by UGC I

Approved by AICTE

JEPPIAAR NAGAR, RAJIV GANDHI SALAI, CHENNAI - 600 119

March - 2021



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Jeppiaar Nagar, Rajiv Gandhi Salai, Chennai – 600 119

www.sathyabama.ac.in

DEPARTMENT OF AUTOMOBILE ENGINEERING

BONAFIDE CERTIFICATE

This is to certify that this Project Report is the bonafide work of **K. SIVA THANIGAI (37860036), N V. DEVENDRA SAI (37860007)** who carried out the project entitled "**NOISE AND VIBRATION ANALYSIS ON V6 ENGINE**" under my supervision from **2020** to **2021**

Mr. R. RAJASEKAR, M.E.,

Internal Guide

Dr. V.K. BUPESH RAJA, M.E., Ph.D.,

Head of the Department

Submitted for Viva voce Examination held on _____

Internal Examiner

External Examiner

DECLARATION

We **K. SIVA THANIGAI (37860036)**, **N V. DEVENDRA SAI (37860007)** hereby declare that the project report entitled “**NOISE AND VIBRATION ANALYSIS ON V6 ENGINE**” done by under the guidance of **Mr. R. RAJASEKAR, M.E., Department of Automobile Engineering** at SATHYABAMA INSTITUTE OF SCIENCE AND TECHNOLOGY is submitted in partial fulfilment of the requirements for the award of Bachelor of Engineering in AUTOMOBILE ENGINEERING

DATE:

PLACE:

SIGNATURE OF THE CANDIDATE

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ABSTRACT

Internal combustion engines are undoubtedly one of the major powerhouses for an automobile. During the combustion process lot of energy is generated successively generates noise vibration and harshness.

Noise and vibration play a major role with the internal combustion engine. With design being made we will be needing to validate the design using finite element method. Finite element method allows an engineer to virtually test the components under various load conditions and take the result values into consideration to see the design how it is acting when a particular amount of load.

Excitation forces like piston slap force, compression force on the valve springs are calculated using Multi Body Dynamics (MBD) method. The calculated excitation forces are transferred into frequency domain with help of fast Fourier transforms. FEM analysis is used to calculate the vibro-acoustic behaviour of powertrain.

The analysis part majorly concentrates on analysis's like normal mode analysis (modal), frequency response analysis (harmonic response), modal acoustic analysis.

The major aim of this projects is to validate the model which is designed using solidworks in terms of acoustic behaviour and to reduce the noise and vibration (frequency) using above mentioned analysis.

KEYWORDS

Noise vibration harshness, Finite element method, 3d design, multibody dynamics, frequency response analysis, modal acoustics analysis and sandwich method.

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LIST OF ABBREVIATION

SYMBOL	DETAILS OF ABBREVIATION
FEM	Finite Element Analysis
MBD	Multi Body Dynamics
NVH	Noise Vibration Harshness
CAD	Computer Aided Design
CADD	Computer Aided Design and Drafting
PDE	Partial Differential Equation
ICE	Internal Combustion Engine
N&V	Noise and vibration
XFEM	Extended Finite Element Method
GFEM	Generalized Finite Element Method
DG	Discontinuous Galerkin
MFEM	Mixed Finite Element Method
FE	Finite Element
SAE	Society of automotive engineers
RPM	Rotations per minute
Db	Decibels A Piston slap force
CG	Center of gravity
MP	Mass of piston assembly
Mr	Mass of connecting rod analysis
b	Distance from connecting rod CG to pin
Io	Mass moment of inertia of connecting rod
T	Torque

SYMBOL	DETAILS OF ABBREVIATION
N	Newton
N	Nitrogen
C	Carbon
Si	Silicon (6)
Mn	Manganese
S	Sulphur
P	Phosphorous
Mo	Molybdenum
FRA	Frequency response analysis
FRF	Frequency response functions
SIMO	Single input multi output
MISO	Multi input single output
MIMO	Multi input multi output

CHAPTER 1 INTRODUCTION

1.1 NOISE VIBRATION HARSHNESS

NVH is one of the major areas that need to be concentrated in automobiles. Studies to lead during the cycle of work are related to noise vibration and stiffness generated by mechanical components. High noise levels can affect the rider's comfort and cause discomfort to individuals traveling in that particular automobile. Where high levels of vibration can cause damage to the component or make it less efficient. Modal acoustic studies relate to these parameters.

Noise, Vibration and Rigidity, also known as Noise and Vibration, abbreviated as NVH and N&V respectively, are used to measure the noise and vibration characteristics of vehicles, especially cars and trucks, and The name given to the field to be modified. Harshness is somewhat of a historical misnomer. Noise and vibration can be measured, but stiffness is a more subjective assessment.

There is a psychoanalytic measurement called stiffness but it does not correlate very well with many hardness issues. The internal NVH is the noise and vibration experienced by vehicle cabin occupants, while the external NVH is largely concerned with the noise generated by the vehicle, and includes drive-by noise. The noise arising from the fluctuations of fluid pressure and passage through the air is called airborne noise.

The noise emanating from the vibrating surface is called structure-generated noise. Noise is used to describe audible sound, with particular attention to the frequency range from 30 to 4000 Hz. Vibration is used to describe tactile vibration, with specific consideration of the frequency range from 30 to 200 Hz.

Each mechanical component when in use certainly produces some amount of N&V, but the design is designed to reduce the effect of N&V on that particular component to make it more durable and efficient. Reducing the N&V level can make the ride even smoother.

NVH problems in a compliant vehicle can be extremely difficult to check. The problem is that in order to repeat the NVH problem repeatedly, and especially if a resolution is being tested, the test, environment, and driving conditions must be the same each time. Often the problem does not appear even between identical vehicles. There is always a risk that inaccurate conclusions can be drawn regarding both the problem and the solution, and especially if a palliative determination gives only a minor improvement.

To identify the preferred strategy is to simulate the position of the vehicle with a model, then identify a solution, and apply the results to an actual vehicle for objective and subjective confirmation.

Therefore, when the model is assembled, engineering depends on the accuracy of the assumptions, especially with respect to the impact conditions and the vibration path in the driving compartment.

The engine NVH development process typically consists of two steps:

- Concept Design and Analysis phase
- Product design and test validation phase.

There are many types of sounds in automobiles. There are some notable noises,

- Any sound when rotating a corner
- Flapping.
- Manual transmission grinding during shifting.
- Hissing or sizzling under the hood.
- Knocking off the engine compartment.



Fig: 1.1: N&V resemblance of ICE

1.2 COMPUTER AIDED DESIGN

CAD computers / workstations are used to assist in design, modification, analysis, or optimization of design. CAD software is used to increase designer productivity, improve design quality, improve communication through documentation, and build a. For database creation. CAD output is often in the form of electronic files for print, machining, or other manufacturing operations. The term CADD (for computer aided design and drafting) is also used.

CAD software allows the user to create and analyze models in an easy way
Some of the widely used CAD software are

- Solidworks
- Catia
- Uni graphics
- Creo
- AutoCAD

1.3 FINITE ELEMENT METHOD

FEM has been a widely applied subject in the field of automobiles. The FEM finite element method (FEM) is a numerical technique that is used to perform the Fem of any physical phenomenon.

It is necessary to use mathematics to understand and quantify any physical phenomenon, such as structural or fluid behavior, thermal transport, wave propagation, and growth of biological cells. Most of these procedures are described using partial differential equations (PDEs). However, for a computer to solve these PDEs, numerical techniques have been developed over the last few decades and are one of the most prominent elements today [FEM].

1.3.1 Applications of fem

FEM started with significant promise in modeling many mechanical applications related to automobile and aerospace engineering. Applications of the finite element method are now only reaching their potential.

1.3.2 Different types of finite element method

As discussed earlier, traditional FEM technology has demonstrated deficiencies in modeling problems related to fluid mechanics and wave propagation. Several improvements have been made recently to improve the solution process and to extend the applicability of finite element analysis. Some of the important functions still in use include:

1.3.2.1 Extended Finite Element Method

The Bubnov – Galerkin method requires a continuity of displacement in elements. Although problems such as contact, fracture, and damage include discontinuities and jumps that cannot be directly controlled by the finite element method. To address this shortcoming, XFEM was born in 1990. XFEM works through the expansion of shape functions with heavy step functions. Additional degrees of freedom are assigned to nodes around the point of dissection so that the jump can be considered.

1.3.2.2 Generalized Finite Element Method

GFEM was introduced in the same way as XFEM in the 90s. It combines the features of traditional FEM and meshless methods. Shape functions are mainly defined by global coordinates and multiplied by division-unity to form local elemental shape functions. One of the advantages of GFEM is to prevent it from spreading again around the eccentricities.

1.3.2.3 Mixed Finite Element Method

In many problems, such as contact or incompleteness, obstacles are introduced using lag multipliers. These additional degrees of freedom arising from Lagrange multipliers are resolved independently. The system of equations is solved like a coupled system of equations.

In many problems, such as contact or incompleteness, obstacles are introduced using lag multipliers. These additional degrees of freedom arising from Lagrange multipliers are resolved independently. The system of equations is solved like a coupled system.

1.3.2.4 Hp-Finite Element Method

HP-FEM A combination of automatic mesh refinement (H-refinement) and an increase in the order of polynomials (P-refinement). This is not the same as doing separate h- and p-refinements. When automatic HP-refinement is used, and an element is divided into smaller elements (H-refinement), each element may also have different polynomial orders.

1.3.2.5 Discontinuous Galerkin Finite Element Method (DG-FEM)

DG-FEM has shown significant promise for using the idea of finite elements to solve hyperbolic equations, where traditional finite element methods have been weak. In addition, it has also shown improvements in bending and inelastic problems that are commonly seen in most material processes. Here, additional constraints are added to the weak form that includes a penalty parameter (to prevent interpretation) and conditions for other equilibria of stresses between elements.

1.4 MULTIBODY DYNAMICS

MBD is a study of the dynamic behaviour of interconnected rigid or flexible bodies, each of which can undergo large translational and rotational displacements. Multi-body dynamics tools are well established to simulate the structure and mechanism of geometric non linearity, large deflection, backlash, etc. These devices offer a variety of predefined components, linear as well as nonlinear connections.

The equations of motion for these systems can be solved in the time domain. Small time steps are required to properly cover the dynamic effect; The selection of the appropriate numerical solver and time increment is mandatory to obtain the correct result. Structural elements for multi body models are usually represented by significantly simpler elements than detailed Finite Element component models.

Nevertheless, some of these devices are nowadays capable of including subsystem Fay models; Both model assembly and computational effort for such approaches are greatly increased.

CHAPTER 2 LITERATURE SURVEY

2.1 “M. Menday”, Loughborough University, UK “Tribology and Dynamics of Engine and Powertrain” An introduction to noise and vibration issues in the automotive drivetrain and the role of tribology.- Chapter -21.

In this paper, we learned how to identify the drivetrain noise, vibration and harshness (NVH) issues, at impact zone (force acting point) noise and vibration mainly occur based on the condition of lubricant, high energy impact and engine irregularities due to inertia. It discusses the reason that why industries should have eagle eye on engine (or body) NVH issues which are:

- Vehicles are coming with higher engine torque rise rates so engine performance need to be focused.
- The vehicle power-to-weight ratio has increased over time
- Customer awareness and expectation has evolved.

2.2 “Haddad S D and Howard D A”, SAE paper 800517. Analysis of piston slap-induced noise and assessment of some methods of control in diesel engines

In this paper we learned more about the theoretical analysis of the piston slap dynamics which helps in predictions of some optimum piston design features for low piston slap-induced noise related to the estimated mechanical efficiency of the engine

The source for the noise in engine mainly is:

- piston slap - either mechanical impacts or impulsive oil film forces.
- timing gear impacts - mainly mechanical impact and electro hydrodynamic forces
- bearing impacts - impulsive oil film forces fuel injection system - mainly impacts
- valve system and accessories - lubricating oil pump, compressors, when fitted, the turbocharger unit itself, etc.

Piston slap excitation is the most important mechanical source of noise and vibration in the engine.

2.3 “Furuhama S” and “Hirukawa K” (1983), SAE 1984 Transactions ‘Piston slap motion and engine noise under low temperature idling operation of diesel engines’, - SAE paper Vol. 92, Section 1; 830066, pp.173-186 (14 pages)

This paper we came to know how the experiment (not software), the hammering vibration was generated manually with a small hammer near the point T4 under the engine stopped condition. These measured results gave a suggestion as to some significant contribution of the piston slap impulse on the diesel idling noise, conducted the experiment at different idle temperatures.

2.4 “Raghuveer Chandra” VNR Vignana Jyothi Institute of Engineering & Technology; Article “Design and analysis of V6 engine”

In this paper we came to know Noise and vibration play a major role with the internal combustion engine. With design being made we will be needing to validate the design using finite element method. Finite element method allows an engineer to virtually test the components under various load conditions and take the result values into consideration to see the design how it is acting when a particular amount of load. Excitation forces like piston slap force, compression force on the valve springs are calculated using Multi Body Dynamics (MBD) method. The calculated excitation forces are transferred into frequency domain with help of fast Fourier transforms. FEM/BEM coupled analysis is used to calculate the vibro-acoustic behaviour of powertrain. The analysis part majorly concentrates on analysis's like normal mode analysis, modal acoustic analysis, frequency response analysis and noise analysis. The major aim of this projects is to validate the model which is designed using solidworks in terms of acoustic behaviour.

2.5 “Shoichi Furuhamma” “Koji Hirukawa” SAE International Congress and Exposition “Piston Slap Motion and Engine Noise under Low Temperature Idling Operation of Diesel Engines”.

From this paper we came to know Diesel engines make a shrill noise called "idle knock" under low temperature idling operation. This causes a serious noise pollution problem in automobile diesel engines. It was clarified by this study that one important source of this noise was piston slap impulse. Piston slap motion was

measured under usual operating conditions and a condition with additional oil supplied into the piston clearance. The piston slap motion was calculated, taking into account the frictional resistances of the crank mechanism and squeeze action of oil film. It was concluded that only a negligible amount of oil existed in the piston clearance for the squeeze action.

CHAPTER 3 DESIGNING OF V6 ENGINE

3.1 TYPES OF ENGINE

The engine configuration describes the fundamental operating principles by which IC engines are listed. Piston engines are often classified by their cylinder layout, valves, and camshaft. Wankel engines are often classified by the number of rotors present. Gas turbine engines are often classified into turbojets, turbofan, turboprop and turboshafts.

Based on the cylinder layout ICE are classified into the following types

- Single cylinder engines
- In-Line engines
- V engines
- W engines
- X engines
- U engines
- H engines
- Radial engines
- Delta engines

For the following study we have chosen V6 engine because of its wide usage in high performance cars which makes it be precise in terms of performance and other parameters.

3.2 Engine layout

The number of pistons in a V engine can be vary of 2–24 depending on the power output and space constraints required. Since the number in the V6 engine indicates that 6 pistons (3 each side) are arranged.

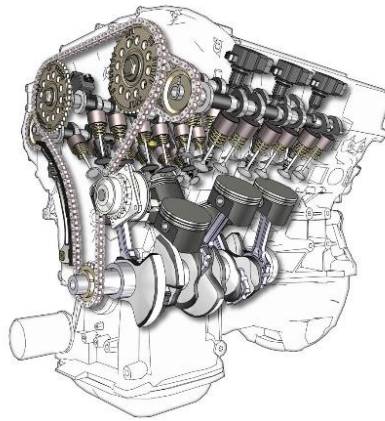


Fig: 3.1: V6 engine

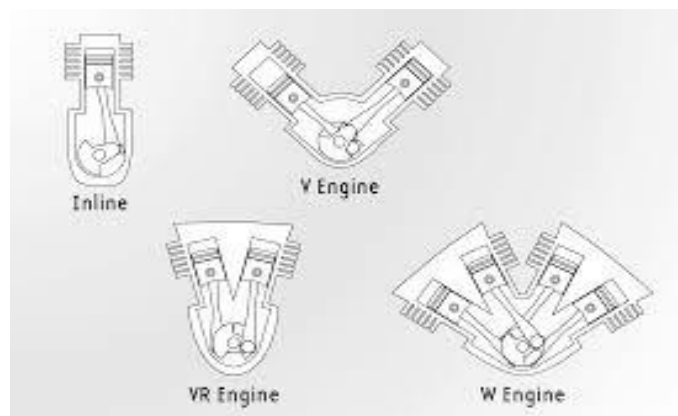


Fig: 3.2: Various engines based on cylinder design

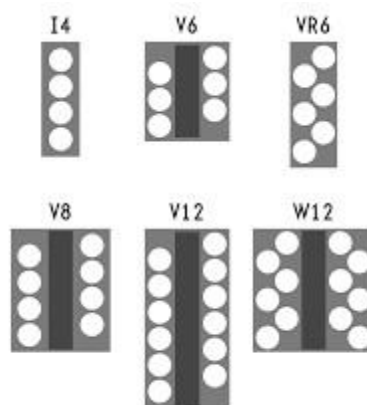


Fig: 3.3: Piston placements in Vee type ICE

3.3 COMPONENTS IN ENGINE

A typical ICE consists of the following parts

- Piston
- Connecting rod
- Gudgeon pin
- Crank shaft
- Camshaft
- Valve
- Rocker arm
- Cylinder block
- Valve cover
- Oil pan
- Cylinder head
- Timing gears

3.3.1 Piston

A piston is a component of reciprocating engines, reciprocating pumps, gas compressors, hydraulic and pneumatic cylinders, among other similar mechanisms. It is the moving component that is contained by a cylinder and is made gas-tight by piston rings, in an engine, its purpose is to transfer force from expanding gas in the cylinder to the crankshaft via a piston rod and/or connecting rod. In a pump, the function is reversed and force is transferred from the crankshaft to the piston for the purpose of compressing or ejecting the fluid in the cylinder. In some engines, the piston also acts as a valve by covering and uncovering ports in the cylinder. In a pump, the function is reversed and the force is transferred from the crankshaft to the piston for the purpose of compressing or ejecting the fluid in the cylinder. In some engines, the piston also acts as a valve by covering and exposing the ports in the cylinder.



Fig: 3.4: Piston and connecting rod assembly

3.3.2 Connecting rod

The connecting rod, also known as the con rod, is the part of a piston engine that connects the piston to the crankshaft. With the crank, the connecting rod converts the rotating speed of the piston to the rotation of the crankshaft. The connecting rod is compressed with the piston and required to transmit tensile forces and rotate at both ends.

The predecessor of the connecting rod is a mechanic linkage used by water mills to convert the speed of the rotating water by the water wheel to the rotating motion.



Fig: 3.5: Connecting rod

3.3.3 Gudgeon Pin

In internal combustion engines, the gudgeon pin connects the piston to the connecting rod, and provides a bearing for the connecting rod to move the piston. In very early engine designs, including those powered by steam, and many very

large stationary or marine engines, the Goodin pin is located in a sliding crosshead that connects to the piston via a rod.

A gudgeon is a pivot or journal.

3.3.4 Crank shaft

A crankshaft is a rotating shaft that (in combination with connecting rods) converts piston speed to rotational speed. The crankshaft is commonly used in internal combustion engines and consists of a series of cranks and crank pins to which the connecting rods are attached.

The crankshaft rotates within the engine block through the use of the main bearings, and within the connecting rods using the crankpin rod bearings. Crankshafts are usually made of metal, with most modern crankshafts constructed using forged steel.

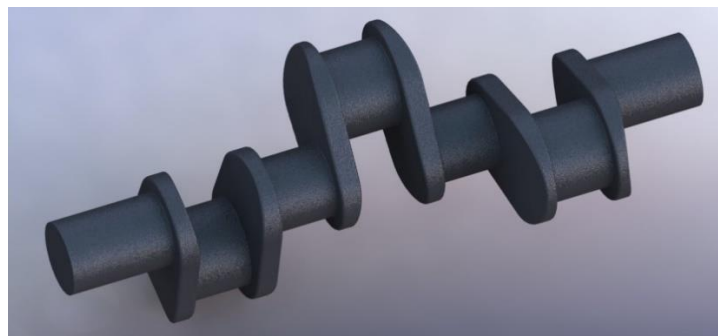


Fig: 3.6: Crank shaft

3.3.5 Cam shaft

A camshaft is a rotating object with pointed cams, which converts rotational motion to reciprocating motion. The camshaft is used in internal combustion engines (to operate the intake and exhaust valves), mechanically controlled ignition systems, and early electric motor controllers. Camshafts in automobiles are made of steel or cast iron, and are an important factor in determining the RPM range of the engine's power band



Fig: 3.7: Cam shaft

3.3.6 Cam

A cam is a rotating or sliding piece in a mechanical linkage specifically used to convert rotary motion to linear motion. It is often a part of a revolving wheel (such as an eccentric wheel) or shaft (a cylinder with an irregular shape) that collides with a lever at one or more points on its circular path.

3.3.7 Rocker arm

A rocker arm is an oscillating lever that conveys radial movement from the cam lobe into linear movement at the poppet valve to open it. One end is raised and lowered by a rotating lobe of the camshaft while the other end acts on the valve stem. When the camshaft lobe raises the outside of the arm, the inside presses down on the valve stem, opening the valve. When the outside of the arm is permitted to return due to the camshafts rotation, the inside rises, allowing the valve spring to close the valve.



Fig: 3.8 Rocker Arm assembly

3.3.8 Cylinder block

An engine block is the structure which contains the cylinders, and other parts, of an internal combustion engine. In an early automotive engine, the engine block consisted of just the cylinder block, to which a separate crankcase was attached. Modern engine blocks typically have the crankcase integrated with the cylinder block as a single component.

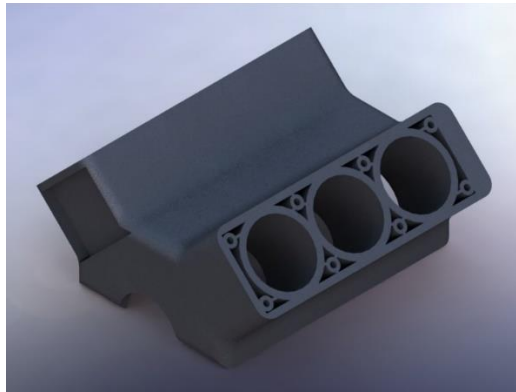


Fig: 3.9: Cylinder block

3.3.9 Cylinder head

In an internal combustion engine, the cylinder head sits above the cylinders on top of the cylinder block. It closes in the top of the cylinder, forming the combustion chamber. This joint is sealed by a head gasket. In most engines, the head also provides space for the passages that feed air and fuel to the cylinder, and that allow the exhaust to escape. The head can also be a place to mount the valves, spark plugs, and fuel injectors.



Fig: 3.10: Cylinder head



Fig: 3.11: Air turbo



Fig: 3.12: Valve cover

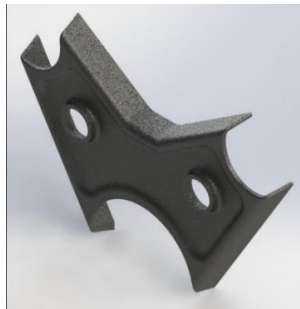


Fig: 3.13: Front cover

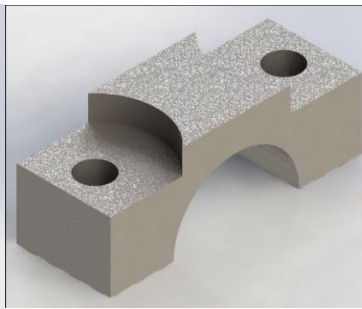


Fig: 3.14: Retainer

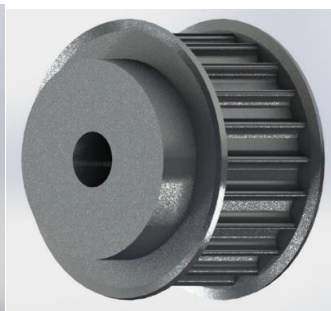


Fig: 3.15: Belt Wheels

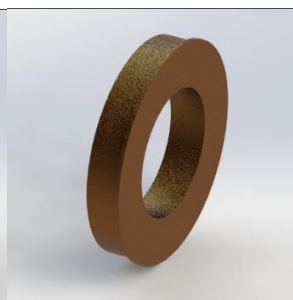
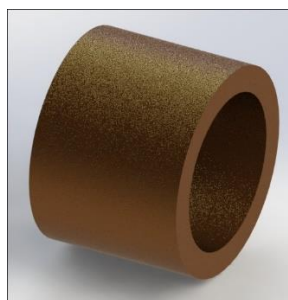


Fig: 3.16: Bushings



Fig: 3.17: Rocker Spring

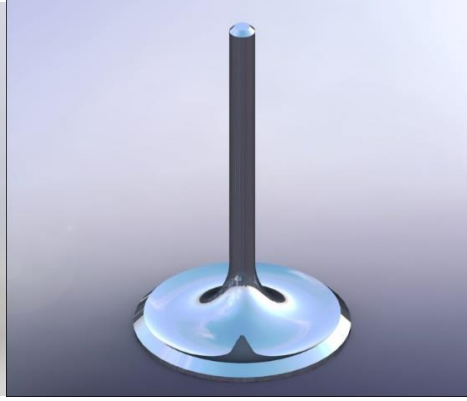


Fig: 3.18: Valve

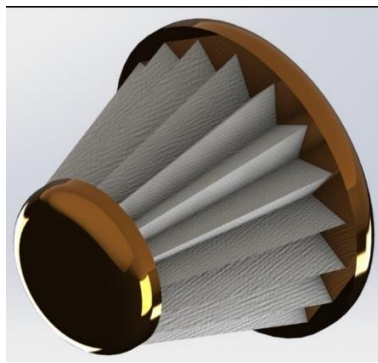


Fig: 3.19: Air filter



Fig: 3.20: Oil pan

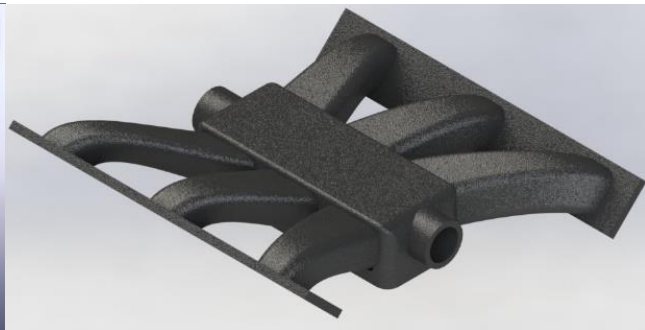


Fig: 3.21: Intake & Exhaust manifold

3.3.10 Timing belt

A timing belt, timing chain, or cam belt is a part of an internal combustion engine that synchronizes the rotation of the crankshaft and camshaft (s) so that the engine's valves open and close at the appropriate time during the intake and exhaust strokes of each cylinder. The timing belt or chain is also important in an interference engine so that the piston can be hit with the valve. A timing belt is

usually a toothed belt which is a drive belt with teeth on the inside surface. A time series is a roller chain.

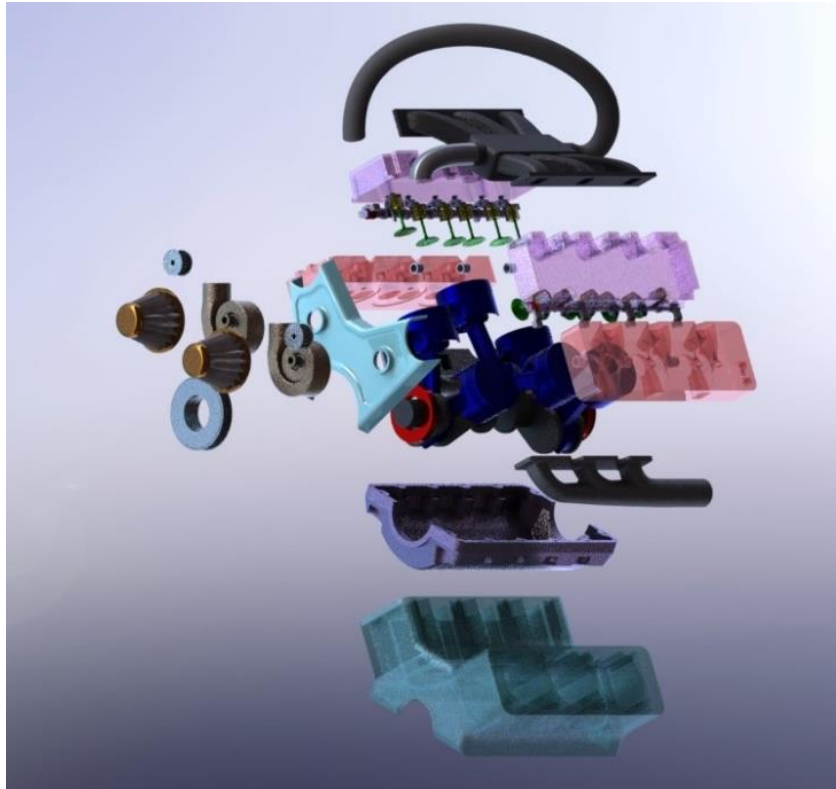


Fig: 3.22: Exploded view of engine assembly



Fig: 3.23: Isometric view of engine assembly

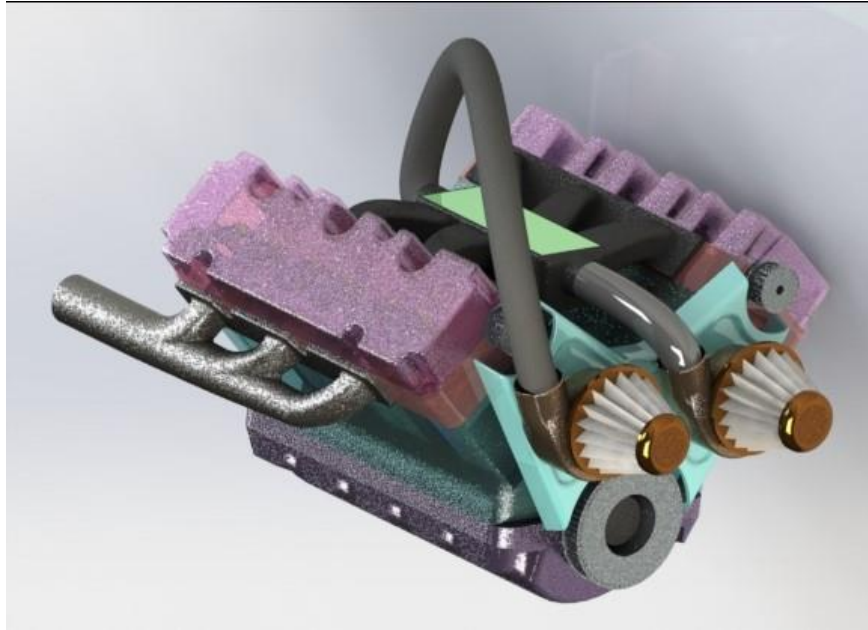


Fig: 3.24: Front view of engine assembly

Table: 3.1: Bill of materials

ITEM NO	PART NAME	QTY.
1	Engine block	1
2	Cam bushing	2
3	Crankshaft	1
4	Piston	6
5	Cylinder head	2
6	camshaft	2
7	Crankshaft bushing	4
8	Camshaft retainer	4
9	Spring	12
10	Engine valve	12
11	Rocker arm valve	12

12	Oil pan	1
13	Front cover	1
14	Belt wheel 2	1
15	Valves cover	2
16	Air filter	2
17	Air turbo	2
18	Exhaust manifold	1
19	Exhaust right	1
20	Belt wheel 1	2
21	Hoses	1
22	Hoses 2	1

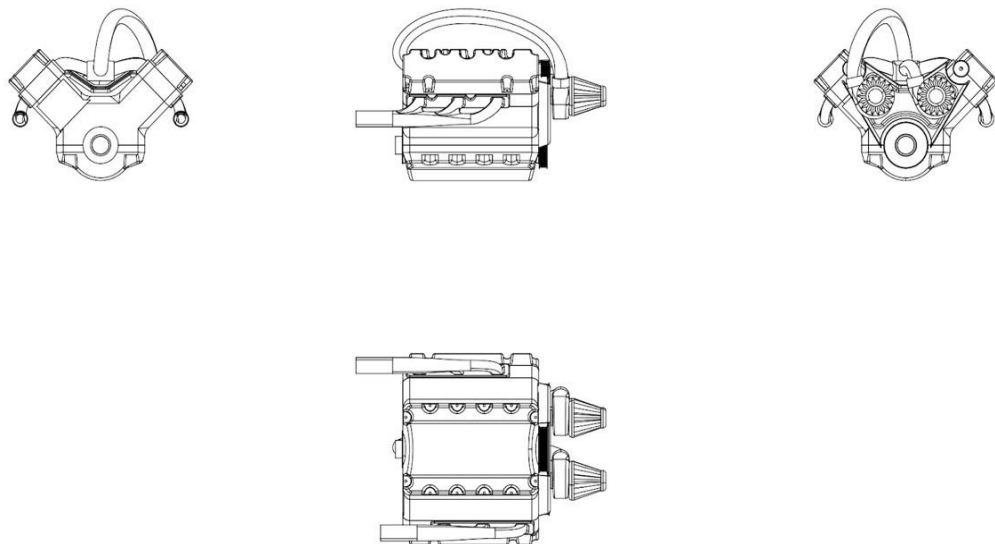


Fig: 3.25: Drawing sheet of Engine

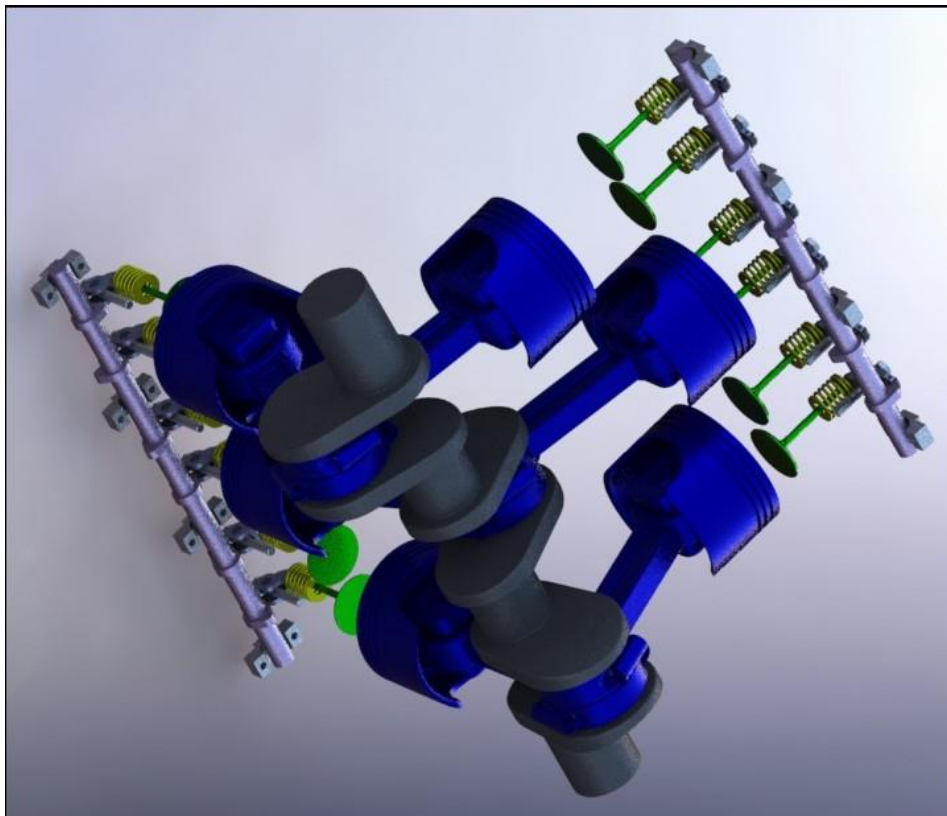
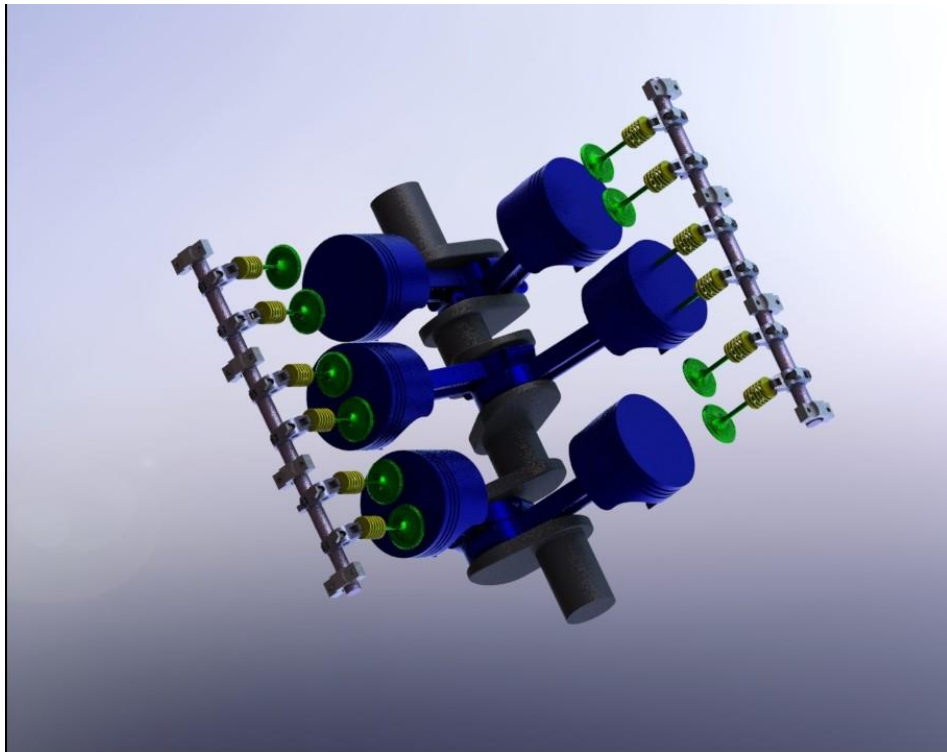


Fig: 3.26: Piston and valve train arrangement

CHAPTER 4 MULTIBODY DYNAMICS

4.1 NEED OF MBD ANALYSIS

When the ICE is up and running because if the mechanical components working to carry on the combustion cycles some forces are generated which in turn effects and N&V levels of the engine. These forces are can also be one of the reasons for failure of the components at some stages. To Know some important forces that are being acted during the combustion cycle MBD is carried out.

Altair Inspire motion is used to carry out the MBD of ICE.

4.2 PREPROCESSING

To carry out the analysis in inspire motion contacts, connections, couplings, actuators and rigid groups in order to define working of various components more precisely.

4.2.1 Input

To rotate the crank shaft in order to resemble the motion during a combustion cycle motor is given as an input with a max RPM of 6000. We have considered 6000 as our input value because it's in the mean rpm range so that we can get more accurate results.

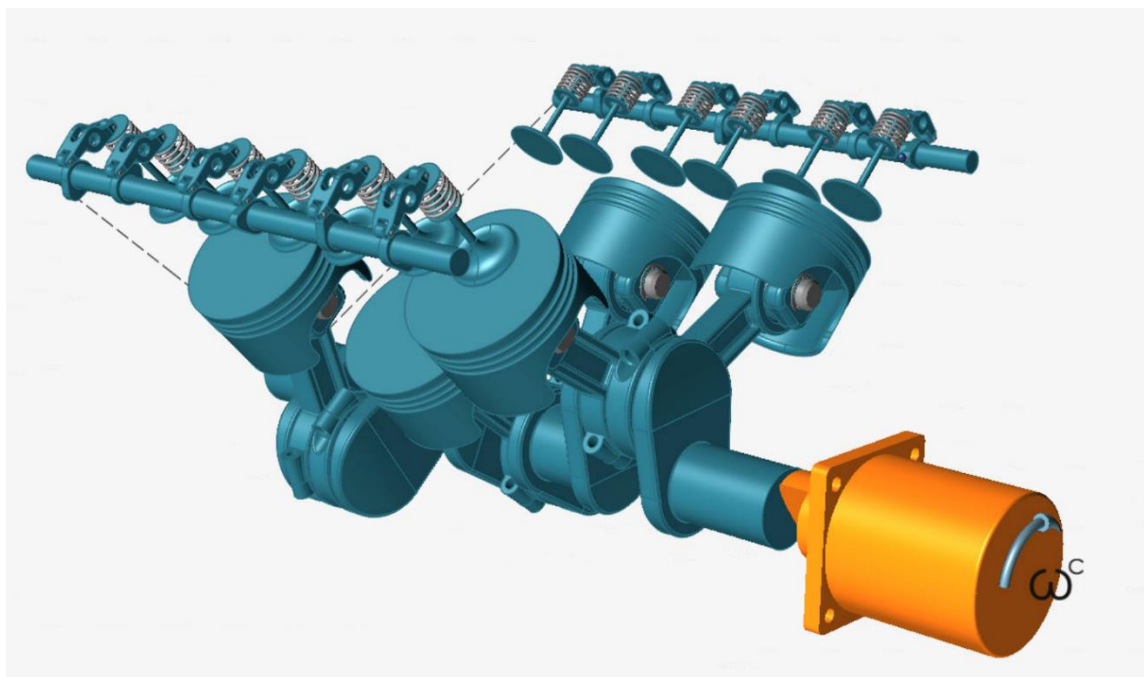


Fig: 4.1: Motor connected to crankshaft

Table: 4.1 Motor input (6000) rpm

Name	Parts	Type	Use Controller	Locked	Function	Value	Reverse Direction
Motor 3	crankshaft	Speed	☒	☐	Step	6000 rpm	☒

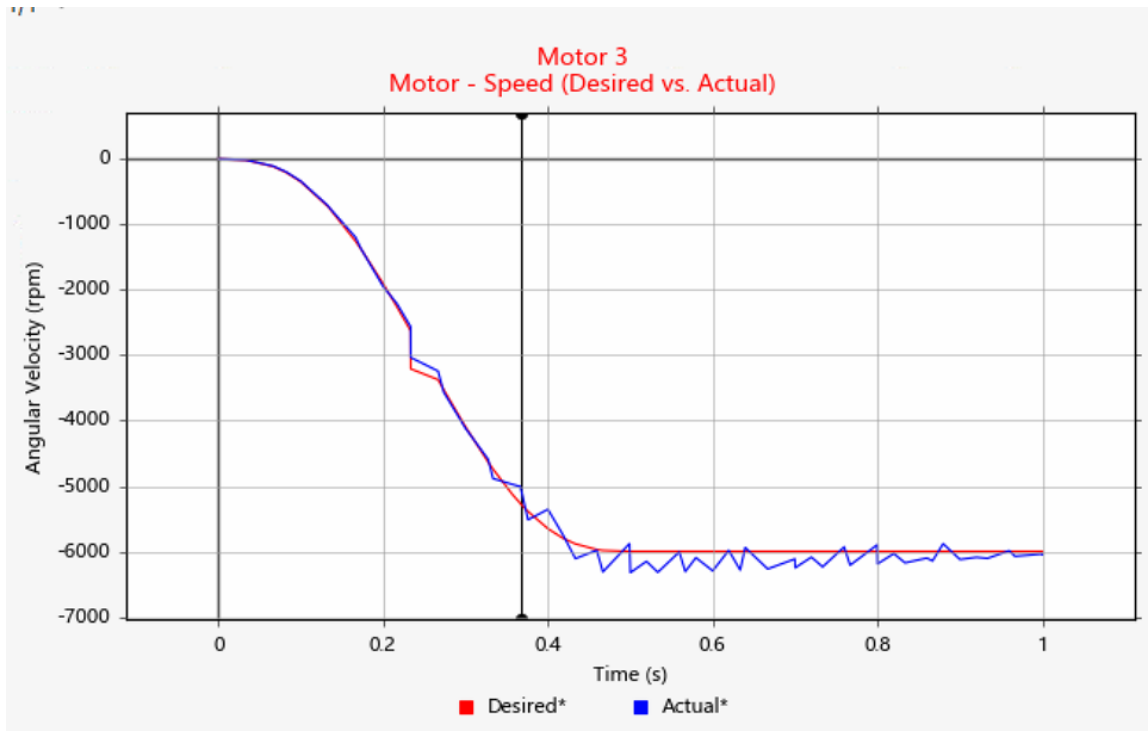


Fig: 4.2: Moto Desired v/s actual rpm

This curve shows the actual rpm which is reduced compared to desired rpm due to losses in engine.

4.2.2 Gravity

Gravity condition is defined in order to replicate more precise physical model

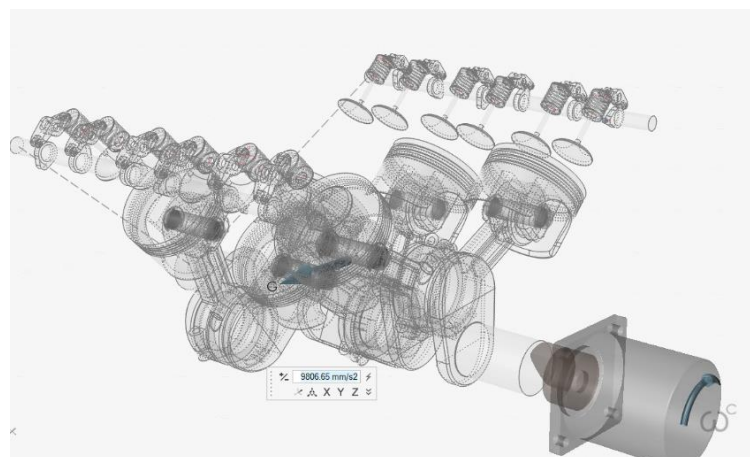


Fig: 4.3: Gravity

4.2.3 Actuators

A mechanical actuator functions to execute movement by converting one kind of motion such as rotary motion into another kind of motion such as translating motion. An example of such actuator is Crankshaft and connecting rod. This is implement into the analysis to make it more accurate and get a precise result.

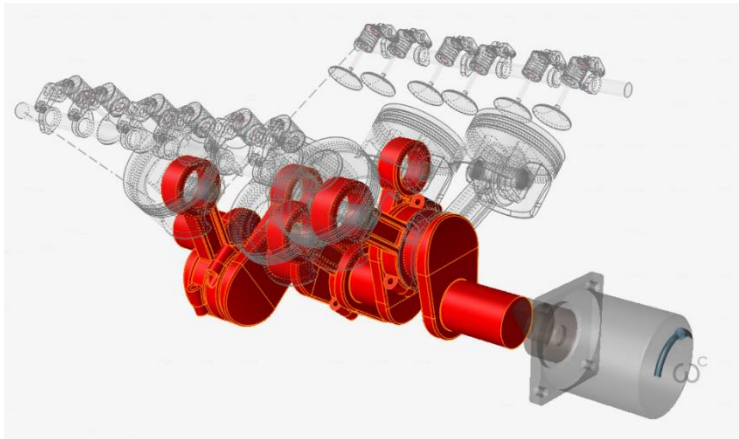


Fig: 4.4: Actuators

4.2.4 Coupling

A coupling condition generally couples/links two or more components in order to make then dependent to one another. In this analysis both camshaft and crankshaft are coupled in order to replicate the timing mechanism for proper movement of the components

Table: 4.2: Coupling input.

Couplers							
Name	Entities	Joint 1 Coupling	Scale 1	Joint 2 Coupling	Scale 2	Reverse Direction	
Coupler 3	Joint 264, Joint 265	Rotational	360 deg	Rotational	180 deg	☐	
Coupler 4	Joint 264, Joint 262	Rotational	360 deg	Rotational	180 deg	☐	

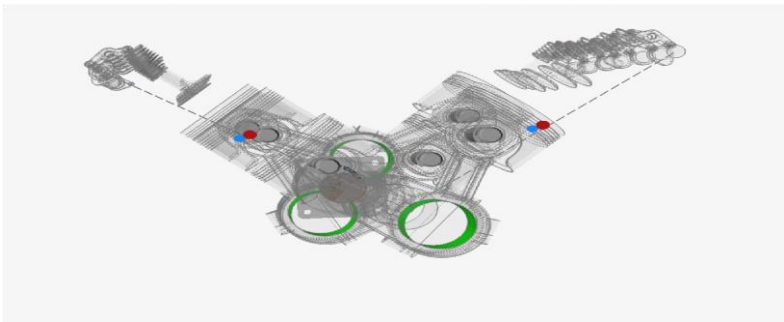


Fig: 4.5: Coupling

4.2.5 Rigid Groups

Rigid groups resemble the components that are rigid and cannot move. In this analysis rigid groups are provided for two reasons • To define the components which are actually rigid • To calculate the forces acting on the components

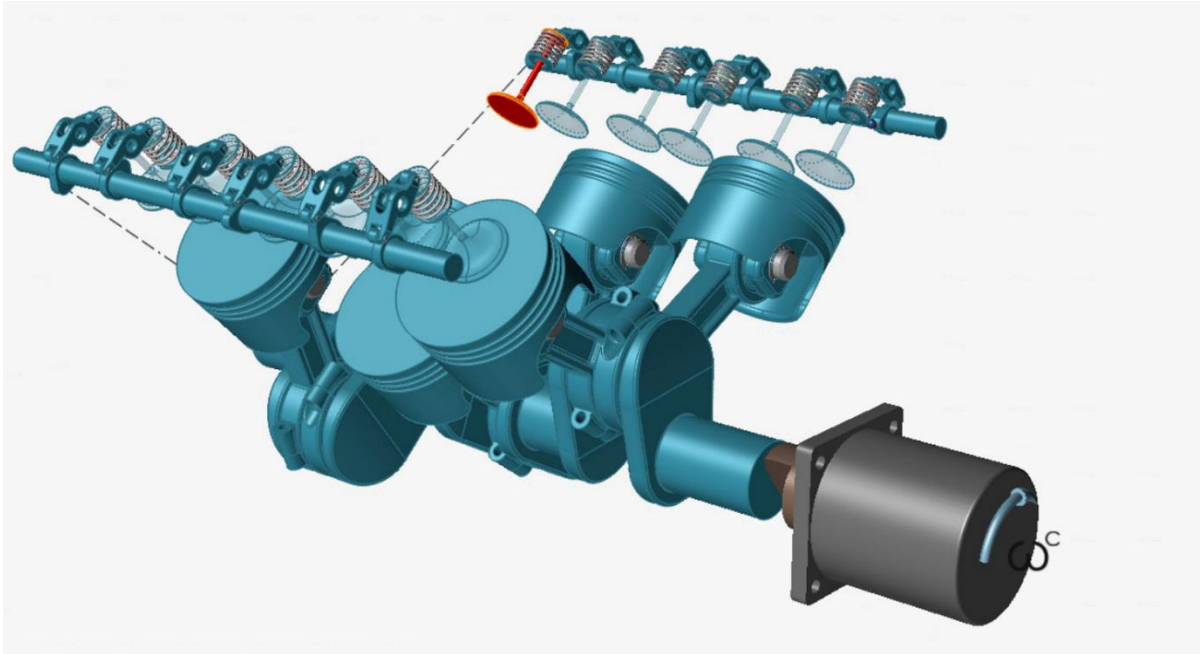


Fig: 4.6: Rigid connection for Valve

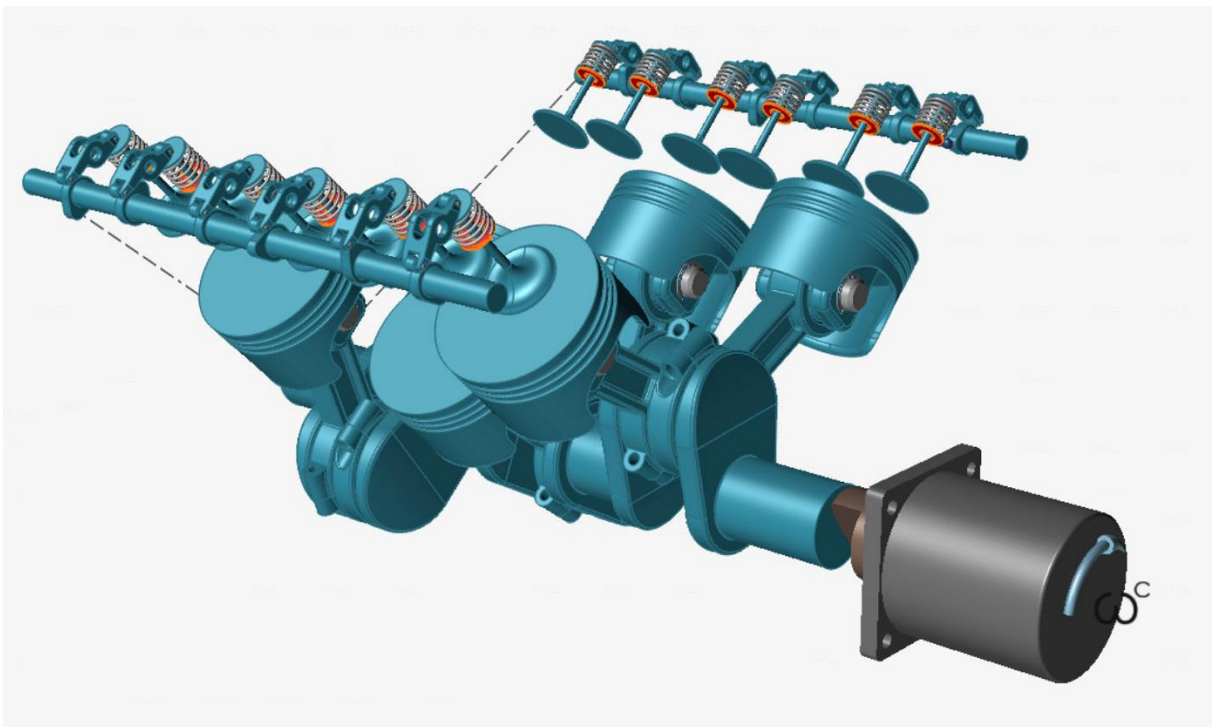


Fig: 4.7: Rigid connection for slicing planes

The rigid connections in the above are given in order to calculate the forces and various parameters the are related to the valve and springs.

4.2.6 Contacts

Contacts resemble the relation between two components which are in a way that they both touch each other. Contacts are giving as an input in this analysis to resemble more precise physical model.

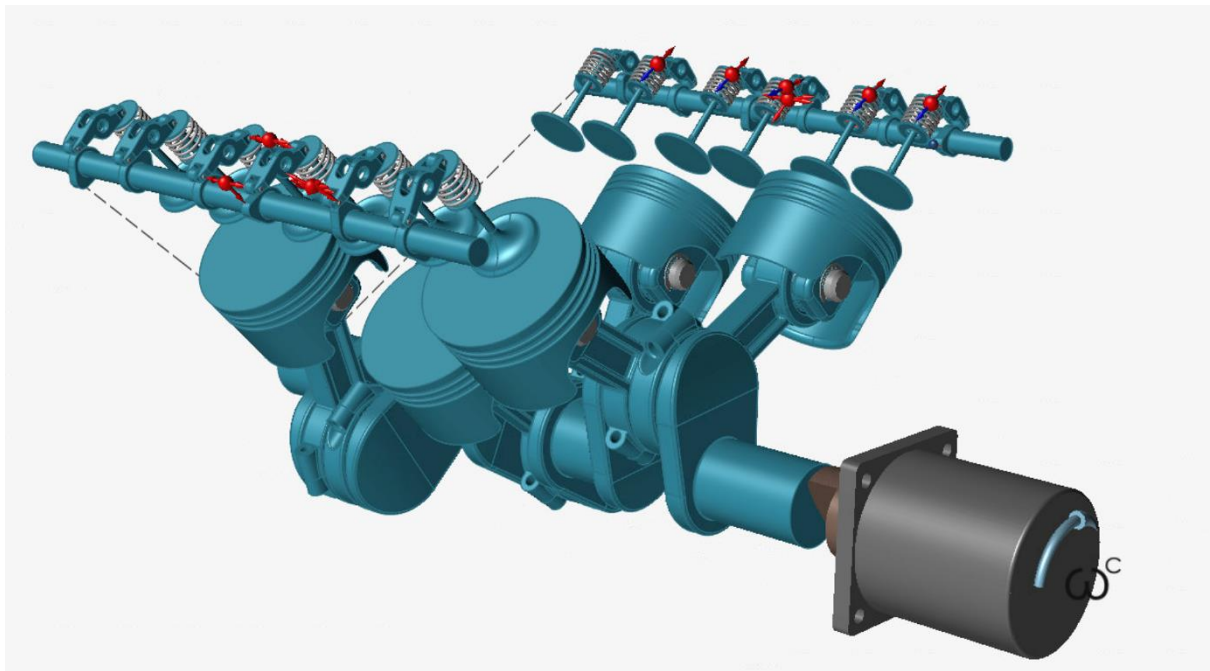


Fig: 4.8: Contacts

4.2.7 Joints

Joints are applied in order to define the components such that a more accurate resemblance to physical model is created.

Table: 4.3: Types of joints

Name	Connection Type	Parts	State	Detected Features	Search Distance	Material	Behavior
Joint 186	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 188	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 190	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 190	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 191	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 192	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 192	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 193	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 194	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 194	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 195	Pin	piston_head, piston_rod.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 195	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 196	Pin	piston_head, piston_rod.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 196	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 196	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 197	Pin	piston_head, piston_rod.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 197	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 198	Pin	piston_head, piston_rod.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 198	Pin	rocker_arm, rocker_arm.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default
Joint 199	Pin	piston_head, piston_rod.	Active	Aligned Holes	0 mm	Steel (AISI 304)	Default

4.3 RESULTS

Simulation is carried out for a time period of 2 sec in order to obtain the intended result.

4.3.1 Piston

After the simulation is completed displacement and joint force at piston is noted down.

For reference purpose results at 2 pistons out of 6 are considered.

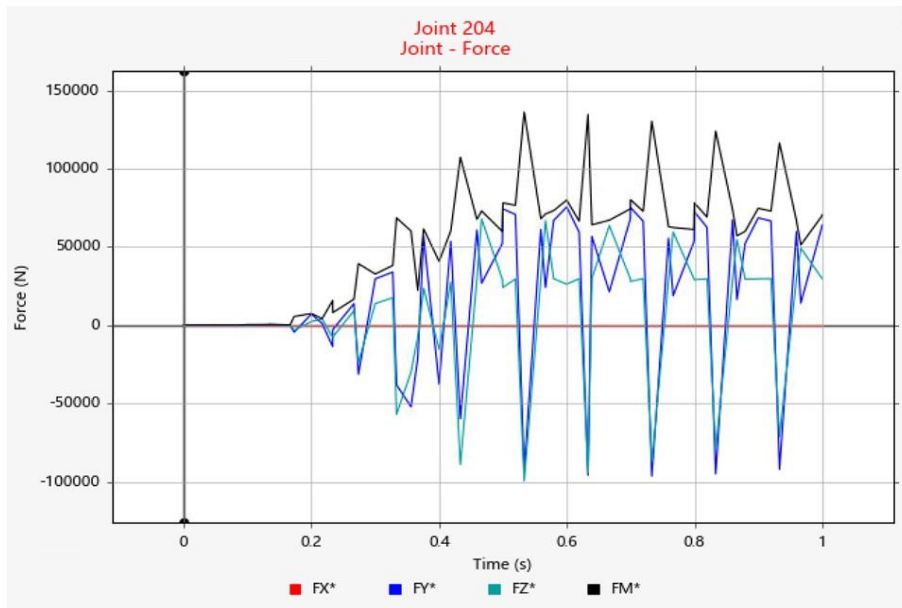


Fig: 4.9: Piston Force-1

The graph shows time vs. piston force curve, peak value and low FM= 140000N at 0.53s, FZ=60000N at 0.44s, FY= 60000N at 0.82s.

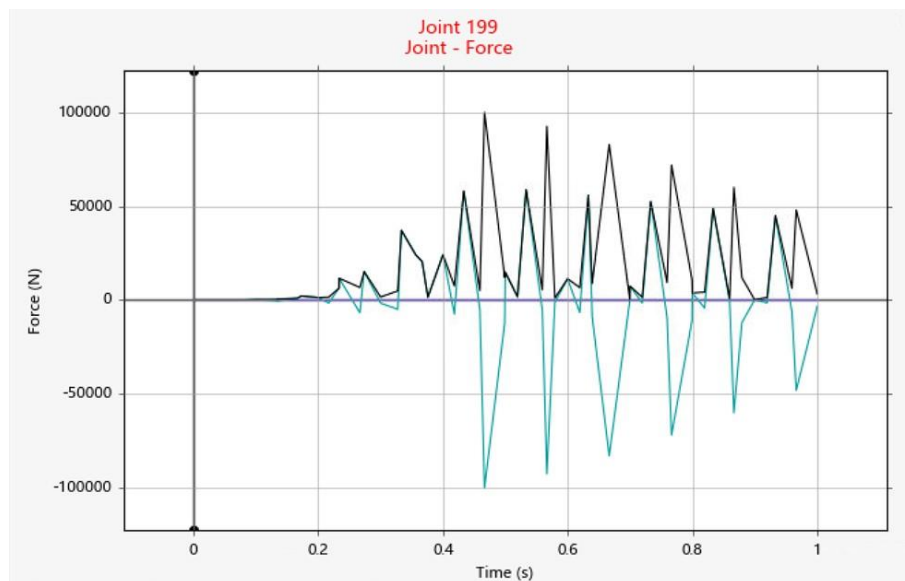


Fig: 4.10: Piston Force-2

The graph shows time vs. piston force curve, peak value and low FM= 100000N at 0.046s, FZ=52000N at 0.42s.

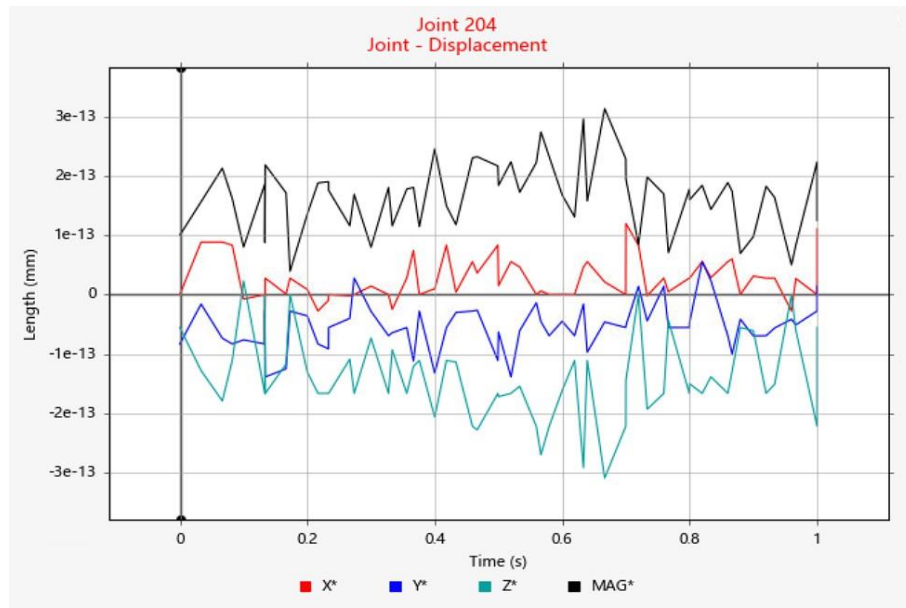


Fig: 4.11: Piston Displacement-1

The graph shows time vs. displacement curve, peak value $X = 1.2 \times 10^{-13}$ mm at 0.72s, $Y = 0.5 \times 10^{-13}$ mm at 0.82s, $Z = 0.1 \times 10^{-13}$ mm at 0.15s.

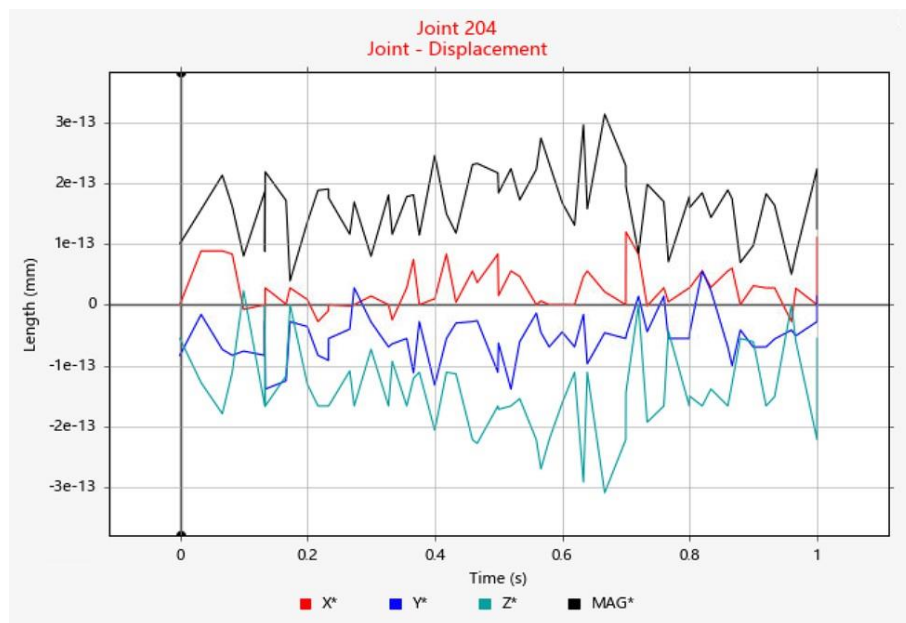


Fig: 4.12: Piston-2 Displacement-2

The graph shows time vs. displacement curve, peak value $X = 1.2 \times 10^{-13}$ mm at 0.72s, $Y = 0.5 \times 10^{-13}$ mm at 0.82s, $Z = 0.1 \times 10^{-13}$ mm at 0.15s.

4.3.2 Motor

As input rpm is given to the crankshaft there is a torque generated at the crankshaft with this we can validate the design of the crankshaft

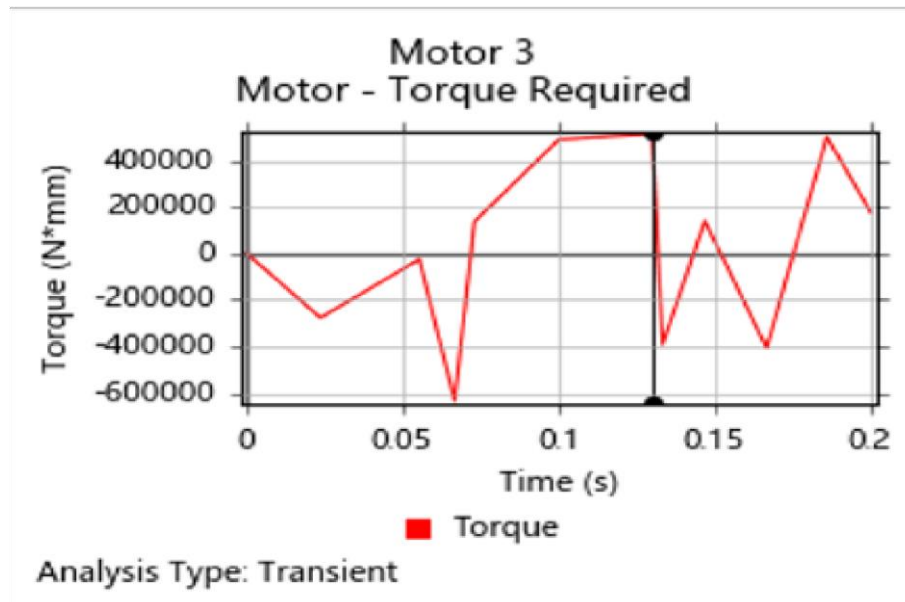


Fig: 4.13: Motor torque

The graph shows time vs. torque curve, peak value of torque= 40×10^4 mm

4.3.3 Roller

The forces at roller which is a part of the rocker arm are noted down. This part is taken into consideration because it is continuously in contact with the valve spring on one side and cam on other side and is subjected to loads. After simulation 3 rollers out of 6 are taken into consideration in order to represent the more accurate results.

From the results it can be said that the roller is subjected to a normal force in the range of 60-100N.

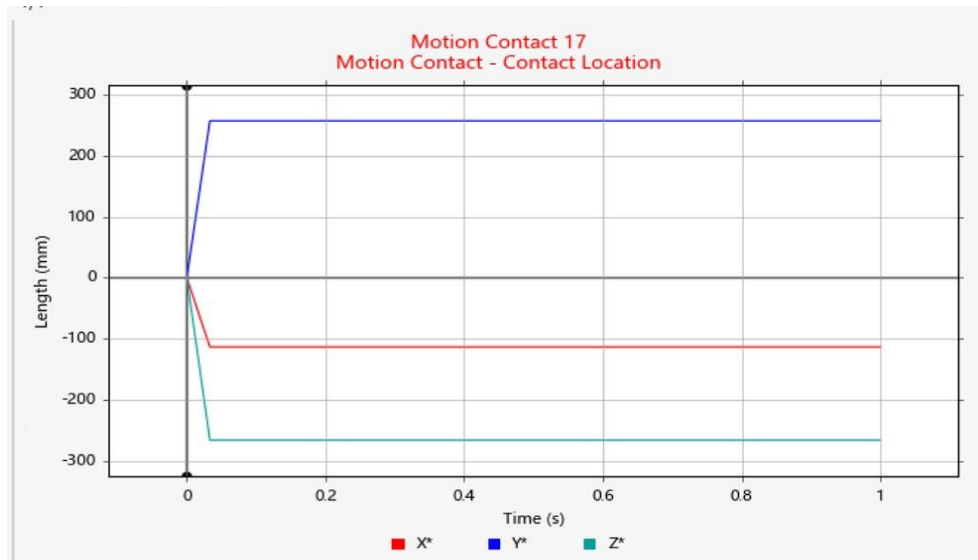


Fig: 4.14: Roller contact location-1

This graph shows length time vs. (displacement) time of roller at input loads ,X=110mm,Y=260mm and Z=270mm

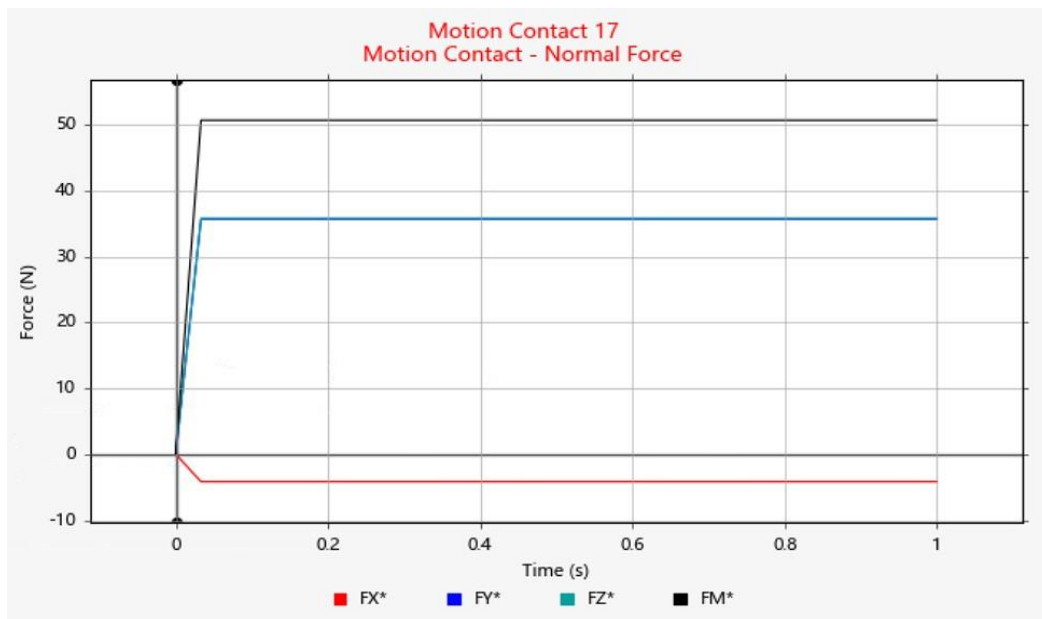


Fig: 4.15: Roller normal force-1

This graph shows time vs. length (displacement) of roller at input loads, FX=4mm, FY=36mm and FZ=51mm

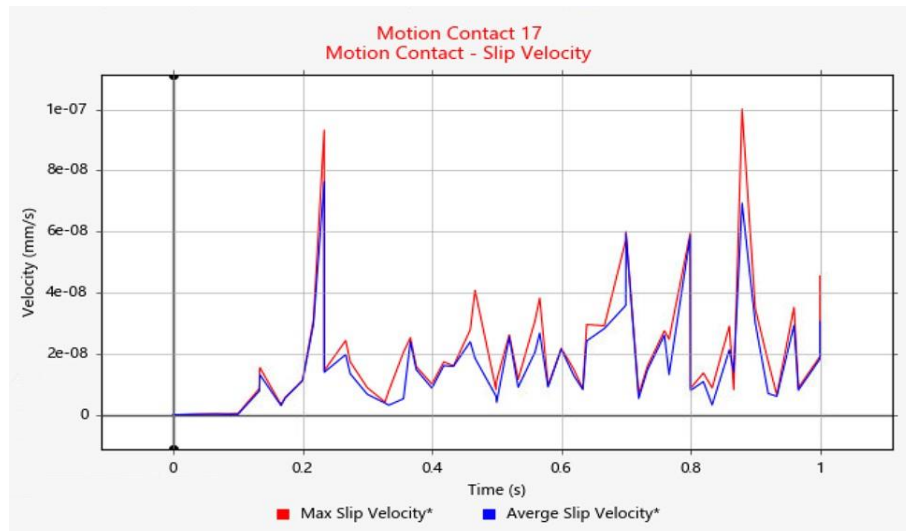


Fig: 4.16: Roller Slip velocity -1

This graph shows time vs. roller slip velocity (mm/s) of roller at input loads, Mass slip velocity= $1\text{e-}07$ mm/s at 0.84s, Average Slip Velocity = $7.8\text{e-}08$ mm/s.



Fig: 4.17: Roller Total torque-1

This graph shows time vs. Roller torque (N/mm) of roller at input loads, TX=24000 N/mm, TY=6000N/mm and TZ=8000N/mm, TM=24900N/mm.

4.3.4 Valves

With rigid connection given the forces and other parameters at the valves of the engine are noted. For accurate representation of the results 3 valves out of 12 are taken into consideration

With simulation being completed the results are noted and it can be observed that the displacement is in the range of 0-400mm

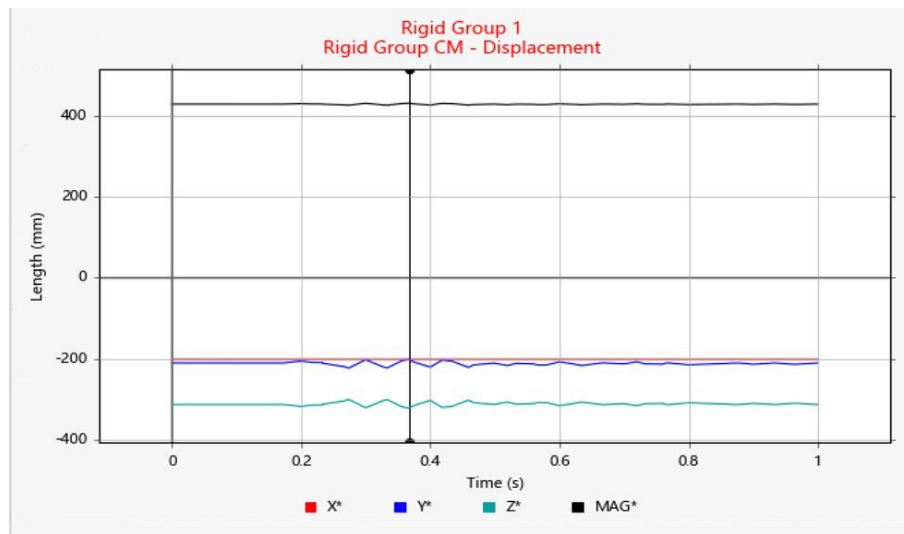


Fig: 4.18: Roller Displacement-1

This graph shows time vs. length (displacement) of roller at input loads, X=200mm, Y=200mm, Z=300mm and MAG =410mm

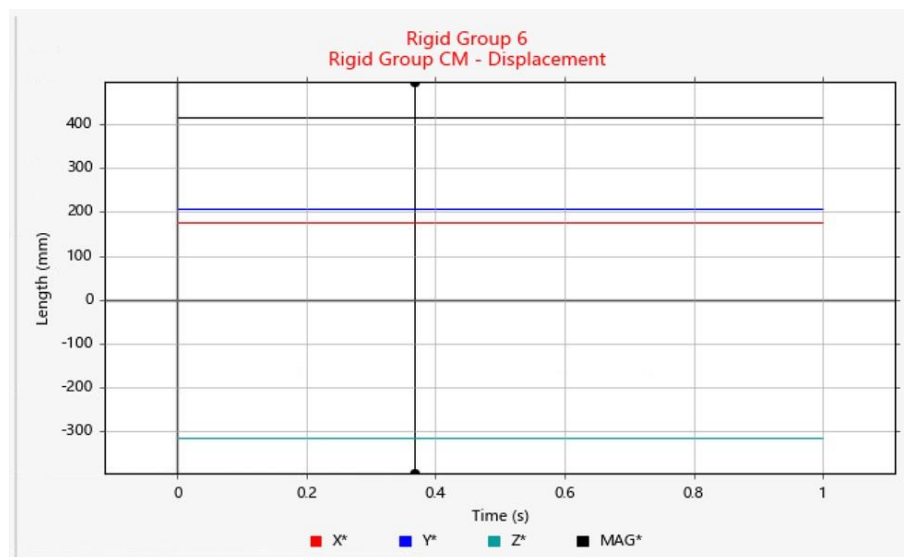


Fig: 4.19: Roller Displacement-2

This graph shows time vs. length (displacement) of roller at input loads, X=180mm, Y=200mm, Z=310mm and MAG =410mm

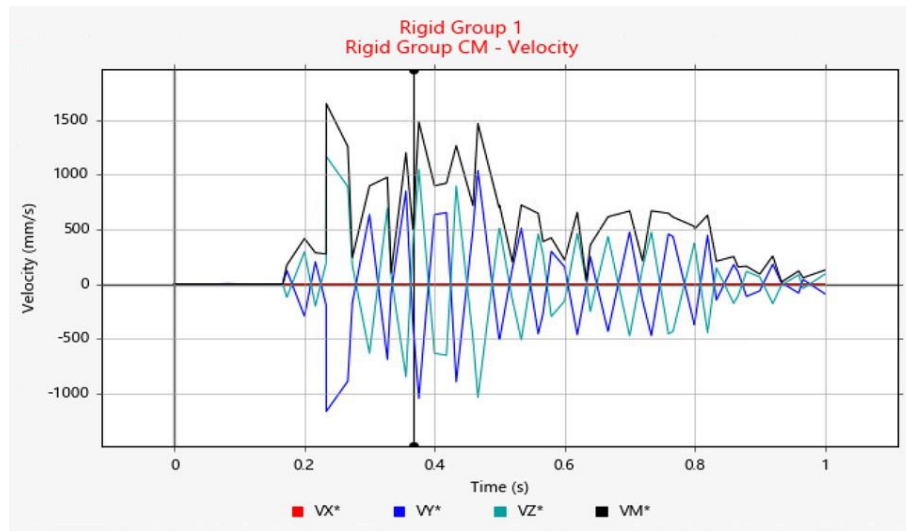


Fig: 4.20: Roller velocity-1

This graph shows time vs. velocity (mm/s) of roller at input loads, $VX=0\text{mm/s}$, $VY=950\text{mm/s}$, $VZ=1200\text{mm/s}$ and $VM=1700\text{mm/s}$

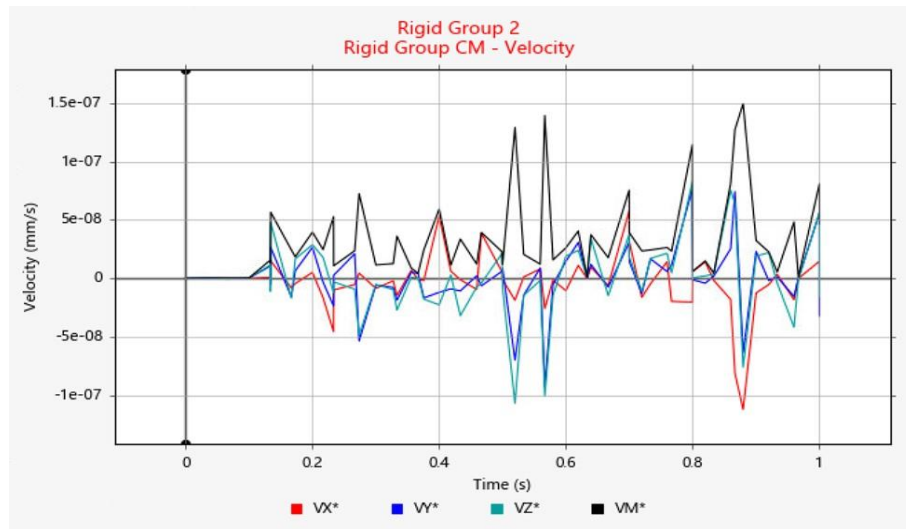


Fig: 4.21: Roller velocity-2

This graph shows time vs. velocity (mm/s) of roller at input loads, $VX=5.1\text{e-}08\text{mm/s}$, $VY=7.5\text{e-}08\text{mm/s}$, $VZ=4.8\text{e-}08\text{mm/s}$ and $VM=1.5\text{e-}07\text{mm/s}$

4.3.5 Cam

The cam is one of the major components that is subjected to high deformation as it rotates in higher speeds. 3 cams are chosen out of the 12 in order to represent the solution more accurately. With simulation being completed the

results are taken and it can be observed that the max normal force that is exerted on the cam is most likely to be in the range of $8e+4$ to $10e+4$

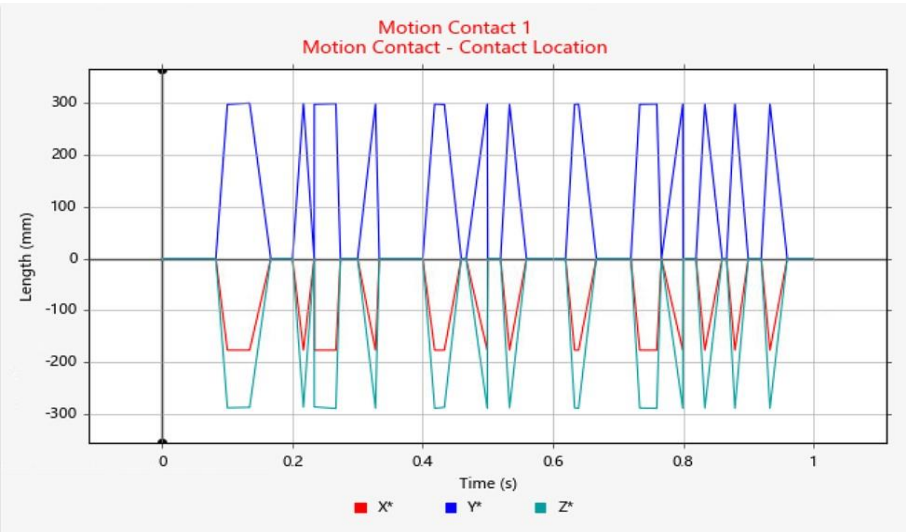


Fig: 4.22: Cam contact location-1

This graph shows time vs. length (displacement) of cam at input loads, $X=184\text{mm}$, $Y=300\text{mm}$, $Z=290\text{mm}$

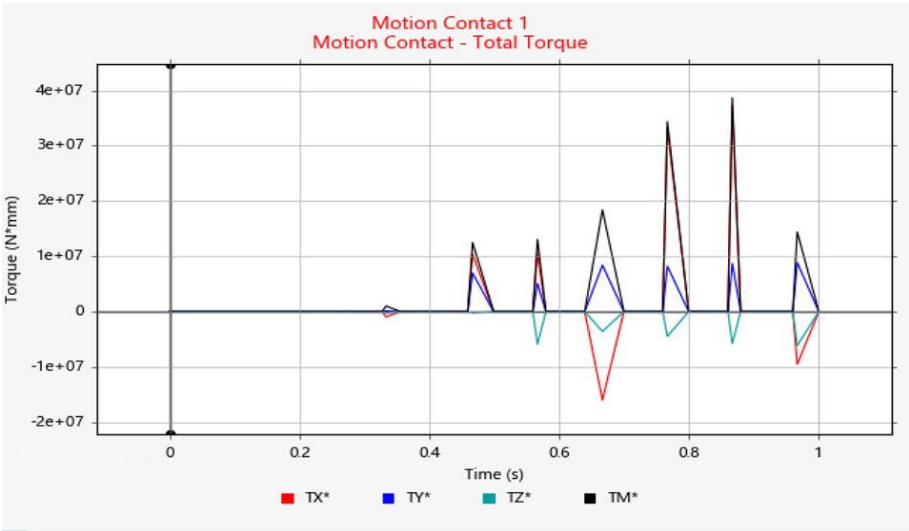


Fig: 4.23: Cam torque-1

This graph shows time vs. torque of roller at input loads, $T_X=1.6e+07\text{N/mm}$, $T_Y=0.8e+07\text{N/mm}$, $T_Z=0.5e+07\text{N/mm}$ and $T_M = 3.9\text{N/mm}$

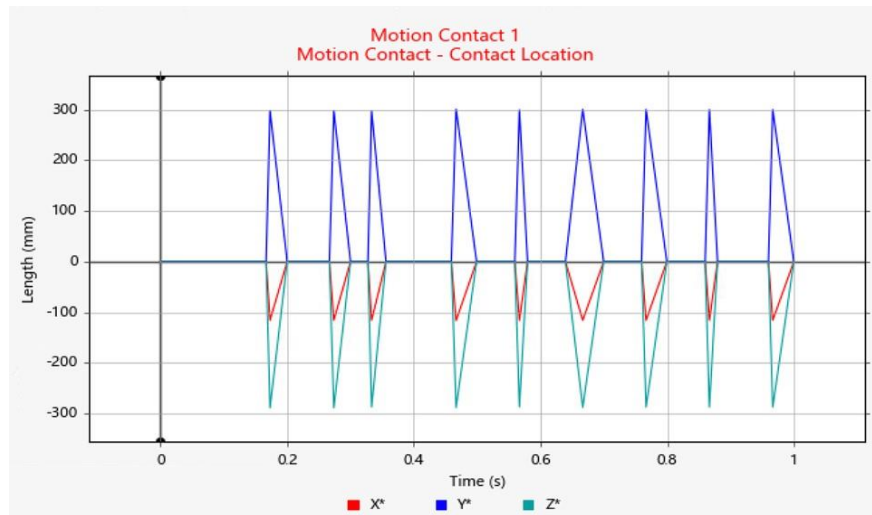


Fig. 4.24: Cam contact location-2

This graph shows time vs. length (displacement) at cam contact at input loads, $X=120\text{mm}$,

$Y=300\text{mm}$, $Z=296\text{mm}$

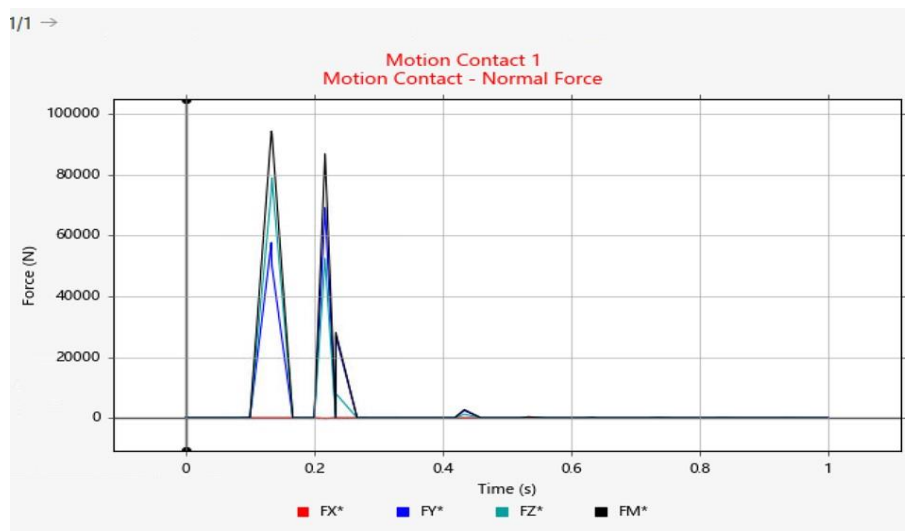


Fig. 4.25: Cam normal force-1

This graph shows time vs. Normal force of cam at input loads, $FX=0\text{mm}$, $FY=57000\text{N}$,

$FZ=80000\text{N}$ and $FM=86000\text{N}$

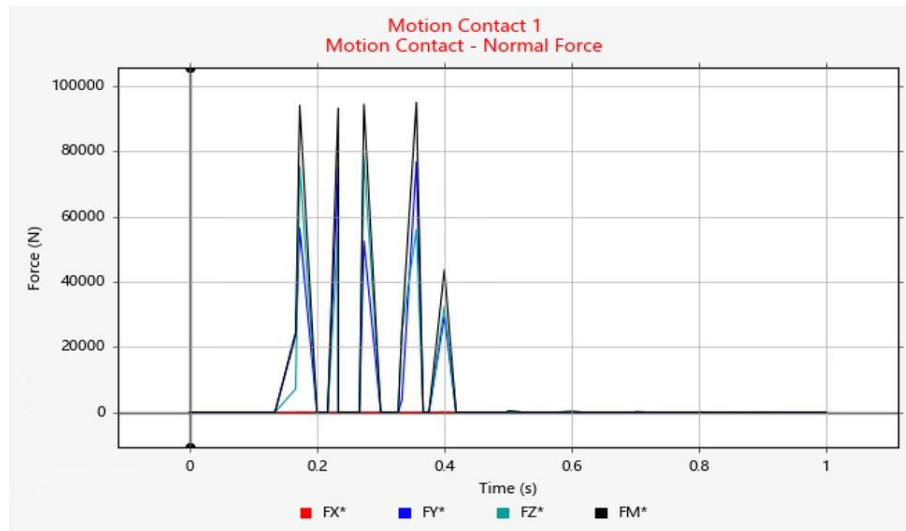


Fig: 4.26: Cam normal force-2

This graph shows time vs. Normal force of cam at input loads, $F_X=0\text{mm}$, $F_Y=78000\text{N}$, $F_Z=76000\text{N}$ and $F_M=86000\text{N}$

4.3.6 Coil spring

When the engine is up and running the coil springs which are placed in between roller of rocker arm and valve are subjected to compressions and it is important. This result can help in validating the design of the spring and examine the forces and other parameters related to the spring

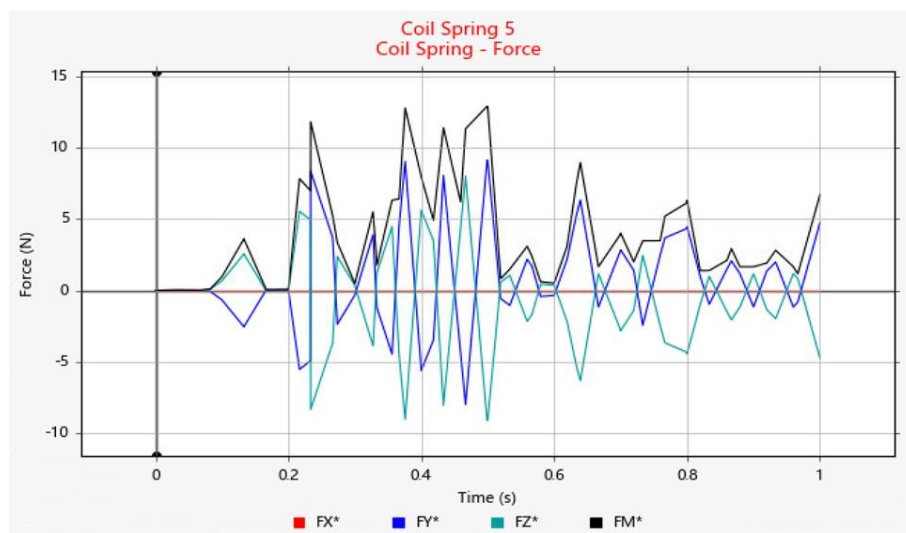


Fig: 4.27: Coil spring Force-1

This graph shows time vs. Normal force of coil spring at input loads, $F_X=0\text{mm}$, $F_Y=9\text{N}$, $F_Z=7\text{N}$ and $F_M=13.7\text{N}$

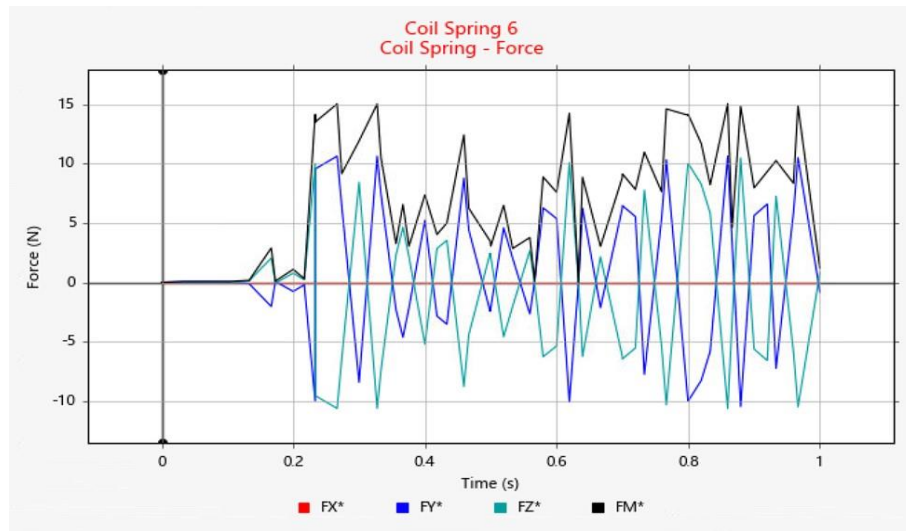


Fig: 4.28: Coil spring Force-2

This graph shows time vs. Normal force of coil spring at input loads, $FX=0\text{mm}$, $FY=10\text{N}$, $FZ=10\text{N}$ and $FM=15\text{N}$

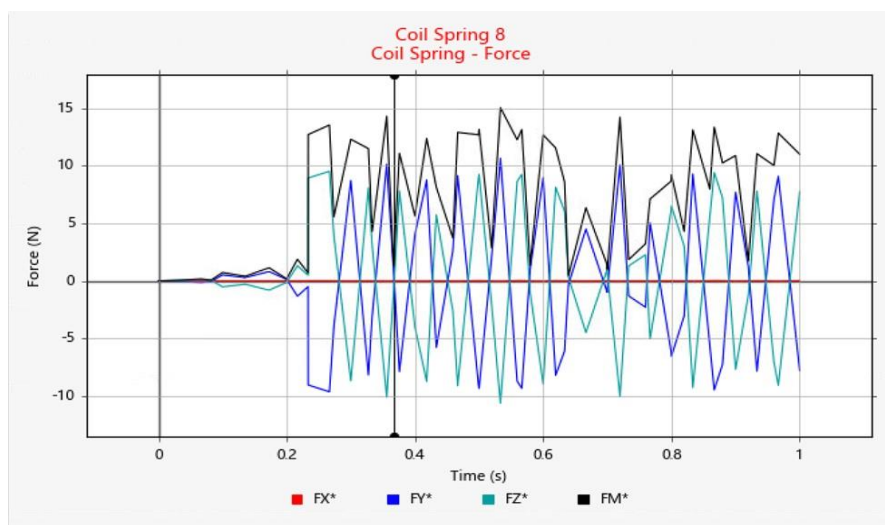


Fig: 4.29: Coil spring Force-3

This graph shows time vs. Normal force of coil spring at input loads, $FX=0\text{mm}$, $FY=10\text{N}$, $FZ=9\text{N}$ and $FM=15\text{N}$

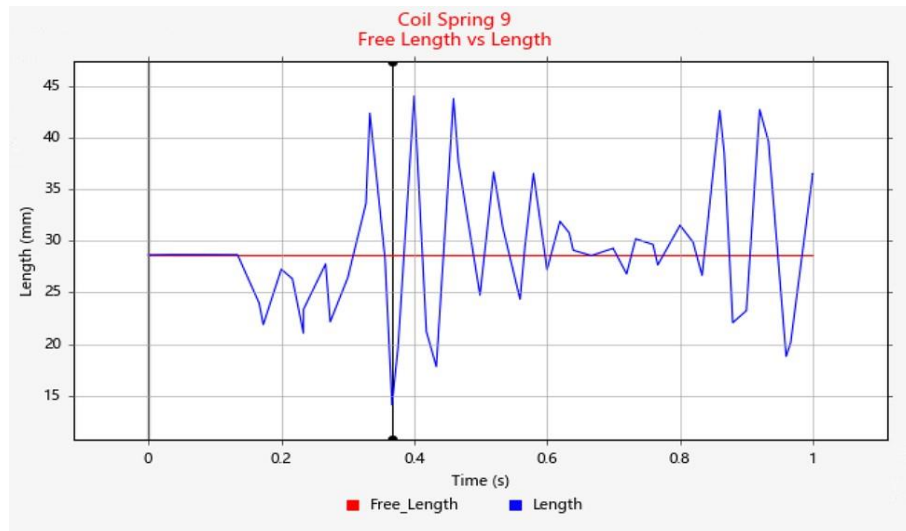


Fig: 4.30: Coil spring free length vs length-1

This graph shows coil spring free length vs. time of coil spring at input loads, and Free length =44mm at0.45s, length=28mm constant.

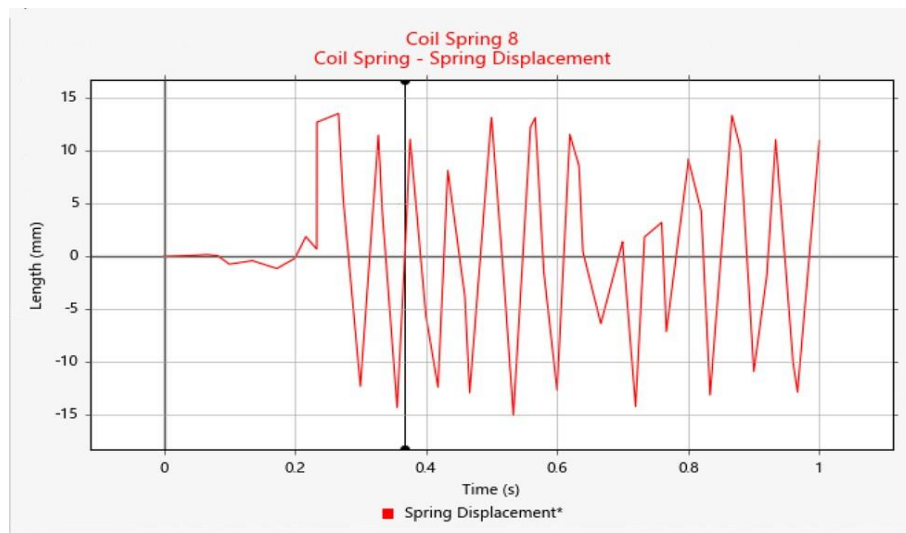


Fig: 4.31: Coil spring displacement-1

This graph shows time vs. displacement of coil spring at input loads, peak value =13mm at 0.27s and Min. value=2mm at 0.69s.

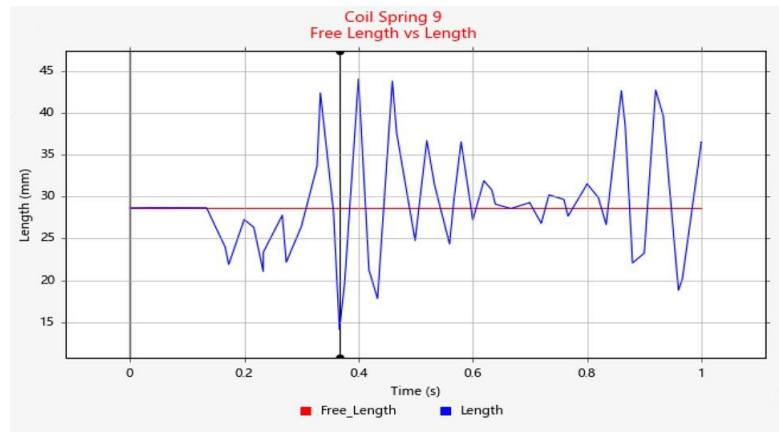


Fig: 4.32: Coil spring free length vs length-2

This graph shows coil spring free length vs. time of coil spring at input loads, and Free length =44mm at0.45s, length=28mm constant.

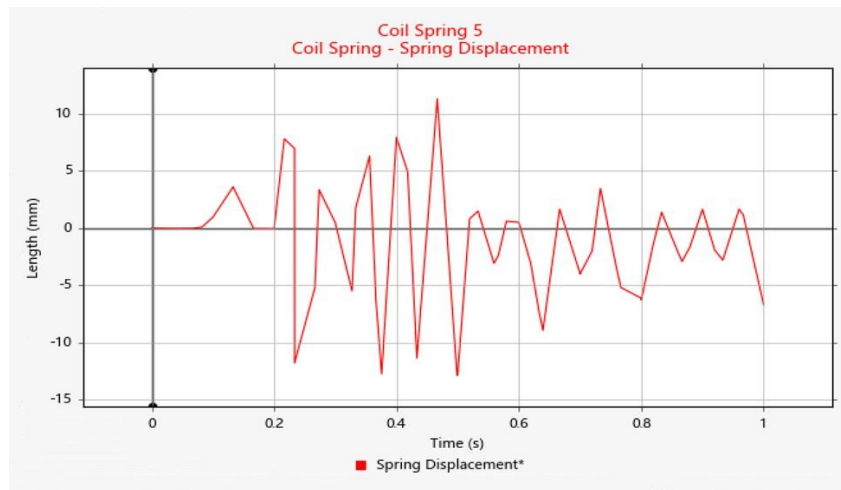


Fig: 4.33: Coil spring displacement -2

This graph shows time vs. displacement of coil spring at input loads, peak value =13mm at

0.47s and Min. value=2mm at 0.58s.

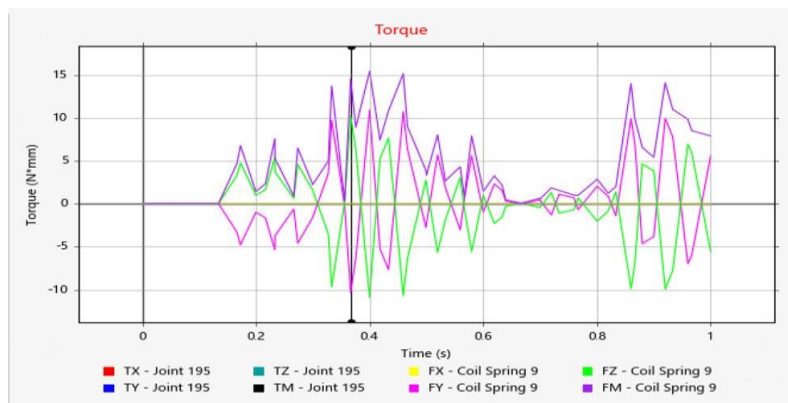


Fig.4.34: Joints type

CHAPTER 5 INNOVATION

We take the idea of SANDWICH Method and implement it to reduce the noise and vibration from the engine.

5.1 SANDWICH method:

A material is placed in between two materials as a single system

This is the simple brief of sandwich method. The two materials used in this method are

- Gray cast iron
- Rubber

5.1.1 GRAY CAST IRON:

Gray cast iron is made by removing pig iron. It is an alloy of carbon and iron. It also contains small amounts of silicon, phosphorus, manganese and sulfur. The reason for its popularity is the ability to construct complex structures and low cost. In addition, the excellent properties of gray cast iron make it one of the most widely used alloys.

Its properties are as follows:

5.1.1.1 HIGH COMPRESSIVE STRENGTH:

The strength is defined by the endurance of any metal to withstand its compressive forces. Gray cast iron has a high compressive strength and, therefore, is widely used in post and columns of buildings. In addition, their compressive strength may be as high as some mild steels.

5.1.1.2 TENSILE STRENGTH:

There are different varieties of gray cast iron and their tensile strength varies accordingly. Some varieties show a strength of 5 tons per square inch, some show 19, but on an average their strength is 7 tons per square inch. However, the addition of vanadium may increase the strength of gray cast iron.

5.1.1.3 LOW MELTING POINT:

Gray cast iron can withstand up to -1140°C to 1200°C

5.1.2 RUBBER:

Rubber is water repellent and resistant to alkalis and weak acids. The rubber's elasticity, toughness, impermeability, adhesion and electrical resistance make it useful as an adhesive, a coating structure, a fiber, a molding compound and an electrical insulator. In general, synthetic rubber has the following advantages over natural rubber: improved aging and weathering, greater resistance to oils, solvents, oxygen, ozone, and some chemicals, and flexibility over a wider temperature range. The advantages of natural rubber are less buildup of heat from flexing and greater resistance to tear when heated.

5.2 WORKING:

Rubber is placed between the two gray cast iron as said in the sandwich method. This helps in absorbing vibration by the rubber and some amount of heat. As rubber melts due to heat we use gray cast iron above and below the material. Gray cast iron helps in absorbing the heat. Due to this the noise and frequency are reduced.

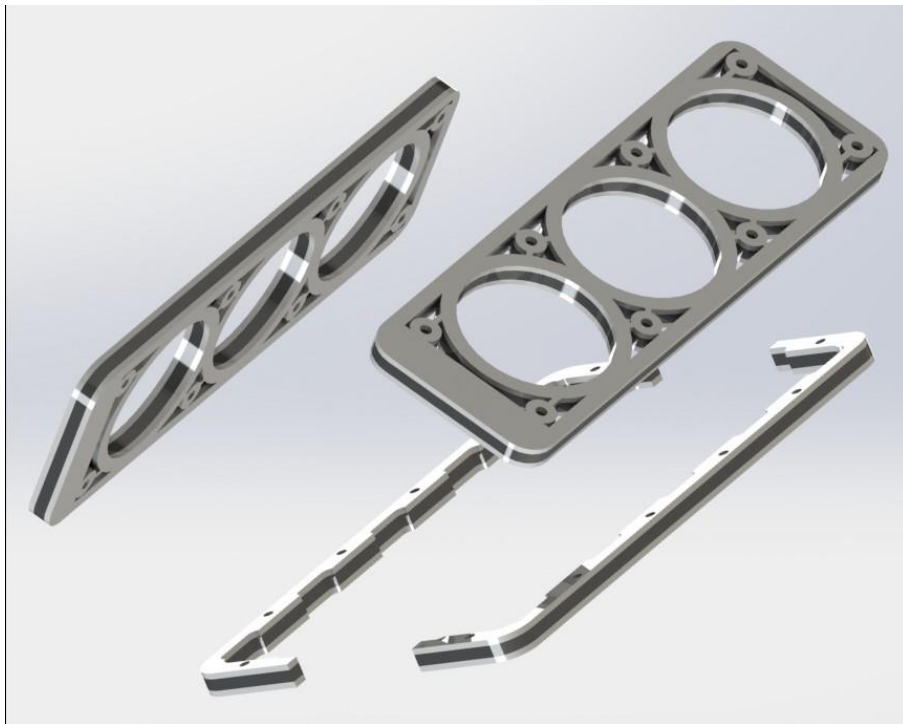


Fig: 5.1: Rubber Pad



Fig: 5.2: Rubber pad placed above & below Engine block

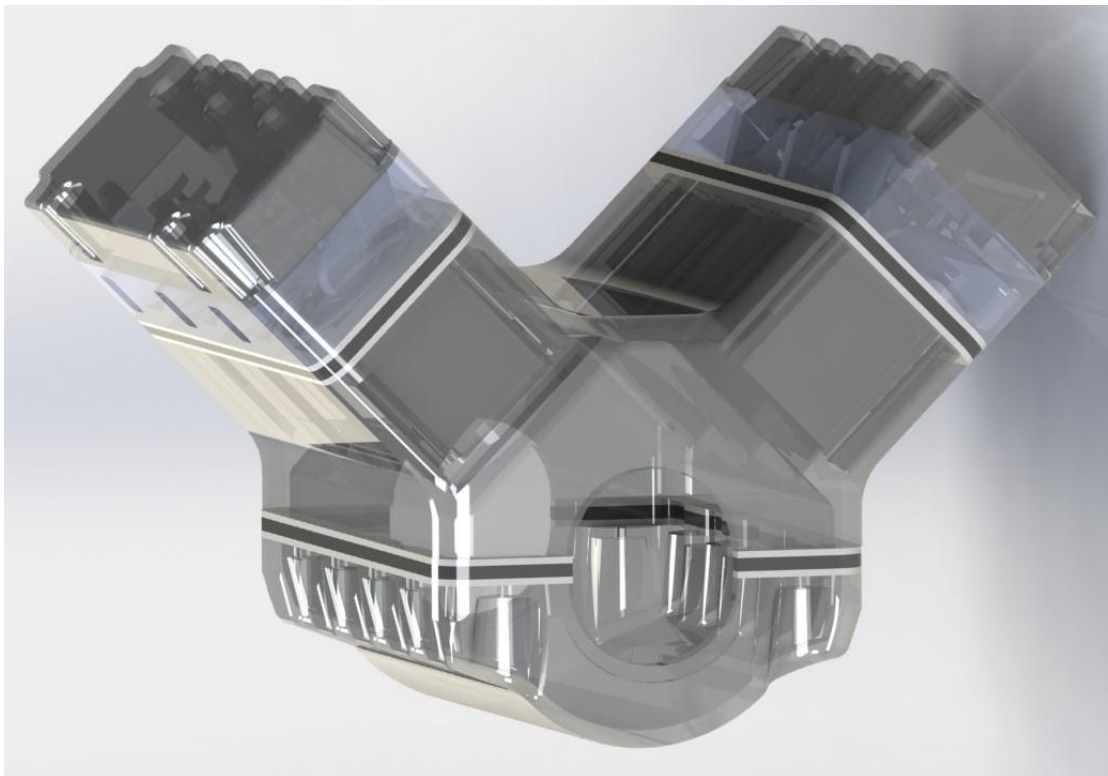


Fig: 5.3: After Engine assembly with rubber pad

CHAPTER 6 NORMAL MODE ANALYSIS

6.1 INTRODUCTION TO NMA

Modal analysis is the study of dynamic properties of system in the frequency domain. Classically it is done with a SIMO (single-input, multiple-output) approach, excitation point, and then the response is measured at several other points. In the past a hammer survey, using a fixed accelerometer and a revolving hammer as an excitation, gave an MISO (multi-input, single-output) analysis, which was mathematically simulated due to the principle of reciprocity. MIMOs (multi-input, multiple-output) have more practical in recent years, partial coherence analysis identifies which part of the response comes from which excitation source. Using multiple shakers has a uniform distribution of energy throughout the structure and results in better consistency in measurement. A single shaker cannot effectively excite all modes of a structure.

Specific excitation signals can be classified as impulse, broadband, swept sign, chirp, possibly others. Each has its advantages and disadvantages.

Signal analysis usually relies on Fourier analysis. The resulting transfer function will show one or more resonances, which can be estimated by characteristic mass, frequency, and measurement.

Model analysis helps determine the vibrational characteristics (natural frequencies and mode shapes) of a mechanical structure or component, reflecting the motion of different parts of the structure under dynamic loading conditions, such as due to lateral forces generated by electrostatic actuators. Natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions.

6.1.1 Modes

In wave theory of physics & engineering a mode in dynamic system is a standing wave state of excitation in which all components of the system will be sinusoidal affected at a certain frequency associated with that mode.

Because no real system can fit perfectly under the standing wave structure the mode concept is taken as specific states of oscillation, thus treating the dynamic system in a linear fashion. In which linear position of states can be performed.

6.1.2 Nodes

Vibration in a dimensional system at a given mode will have nodes or locations where the displacement is always zero. These nodes correspond to points in the mode size where the mode size is zero. Since the vibration of a system is given by the mode size multiplied by a time function, the displacement of the node points remains zero at all times.

When extended into a two-dimensional system, these nodes become lines, where the displacement is always zero. If you look at the animation above, you will see two circles (about half the distance between the edge and centre, and the other on the edge) and a straight line bisecting the disk, displacement is close to zero. In an ideal system these lines are exactly equal to zero, as shown on the right.

6.2 PRE-PROCESSING

The pre-processing with respect to NMA involves providing excitation points, load, contacts and meshing the body.

6.2.1 Meshing

Ansys Meshing Solutions

Meshing is an integral part of the engineering simulation process where complex geometries are divided into simple elements that can be used as discrete local approximations of large domains. The mesh affects the accuracy, convergence, and speed of the simulation.

Mesh details

- Physical preference: Mechanical

- Transition: fast
- Mesh type: Tetrahedral mesh
- Span Angle Centre: fine
- Element size: 1.6^{-002} m
- Boundary box diagonal: 0.97607 m
- Average surface area: $2.5276e-003$ m²
- Minimum edge length: $1.7739e-004$ m
- Transition ratio: 0.272
- Maximum layers: 5
- Proximity and curvature are used to increase the mesh quality
- Nodes: 730897
- Elements: 425909

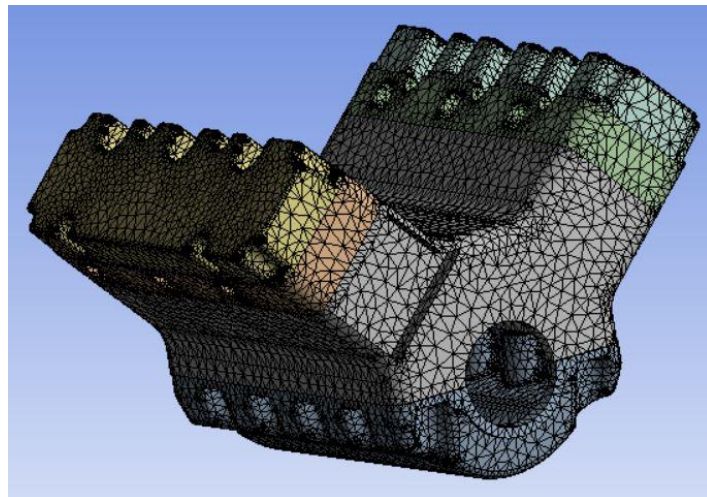


Fig: 6.1: Mesh body of engine

6.2.2 Load

A load should be given to the model in order to know the behaviour of the model based on the various parameters. In this analysis a load of 1000N is given. 1000N is selected because by the literature survey it is observed that 1000N is used as a load by various industry persons as default load.

10 nodes are given as required no. of nodes for the post processor.

6.2.3 Materials Used

In order to resemble a precise physical model materials are assigned to the model.

6.2.3.1 Engine Block: Grey cast iron ASTM 40

6.2.3.2 Cylinder Head: Malleable cast iron ASTM A220

6.2.3.3 Oil pan: Alloy steel SAE 4027

Table: 6.1: Material properties of GCI ASTM40

Grey cast iron ASTM 40				
Chemical composition: C=2.7-4%, Mn=0.8%, Si=1.8-3%, S=0.07% max, P=0.2% max				
Property	Value in metric unit		Value in US unit	
Density	7.06 *10 ³ -7.34 *10 ³	kg/m ³	441-458	lb/ft ³
Modulus of elasticity	124	GA	18000	ksi
Thermal expansion (20 °C)	9.0*10 ⁻⁶	°C ⁻¹	5.0*10 ⁻⁶	in/(in* °F)
Specific heat capacity (25 °C)	490	J/(kg*K)	0.117	BTU/(lb*°F)
Thermal conductivity	53.3	W/(m*K)	370	BTU*in/(hr*ft ² *°F)
Electric resistivity	1.1*10 ⁻⁷	Ohm*m	1.1*10 ⁻⁵	Ohm*cm
Tensile strength	276	MPa	40000	psi
Elongation	1	%	1	%
Shear strength	400	MPa	58000	psi
Compressive yield strength	Min. 827	MPa	Min. 120000	psi
Fatigue strength	138	MPa	20000	psi
Hardness (Brinell)	180-302	HB	180-302	HB
Wear resistance			Low	
Corrosion resistance			Low	
Weldability			Low	
Machinability			Good	
Cast ability			High	

Table: 6.2: Material properties of CI ASTM220

Malleable cast iron ASTM A220				
Chemical composition: C=2-2.7%, Mn=0.25-1.25%, Si=1-1.75%, S=0.03-0.18%, P=0.05%max				
Property	Value in metric unit		Value in US unit	
Density	7.2 *10 ³ -7.45 *10 ³	kg/m ³	450-465	lb/ft ³
Modulus of elasticity	172	GPa	25000	ksi
Thermal expansion (20 °C)	11.9*10 ⁻⁶	°C ⁻¹	6.6*10 ⁻⁶	in/(in* °F)
Electric resistivity	3.9*10 ⁻⁷	Ohm*m	3.9*10 ⁻⁵	Ohm*cm
Tensile strength	586	MPa	85000	psi
Yield strength	483	MPa	70100	psi
Elongation	3	%	3	%
Hardness (Brinell)	217-269	HB	217-269	HB
Wear resistance		Low		
Corrosion resistance		Low		
Weldability		Impossible		
Machinability		Medium		
Castability		Good		
Shock resistance		Good		

Table: 6.3: Material properties of SAE 4027

SAE 4027				
Chemical composition: C=0.40%, Mn=0.8%, Mo=0.25%				
Property	Value in metric unit		Value in US unit	
Density	7.872 *10 ³	kg/m ³	491.4	lb/ft ³
Modulus of elasticity	205	GPa	29700	ksi
Thermal expansion (20 °C)	12.0*10 ⁻⁶	°C ⁻¹	6.60*10 ⁻⁶	in/(in* °F)
Specific heat capacity	477	J/(kg*K)	0.114	BTU/(lb*°F)
Thermal conductivity	44.6	W/(m*K)	309	BTU*in/(hr*ft ² *°F)
Electric resistivity	2.45*10 ⁻⁷	Ohm*m	2.45*10 ⁻⁵	Ohm*cm

Tensile strength (annealed)	515	MPa	74700	psi
Yield strength (annealed)	325	MPa	47100	psi
Elongation (annealed)	30	%	30	%
Hardness (annealed)	78	RB	78	RB
Tensile strength (normalized)	640	MPa	92800	psi
Yield strength (normalized)	420	MPa	60900	psi
Elongation (normalized)	26	%	26	%
Hardness (normalized)	88	RB	88	RB

MODAL ANALYSIS BEFORE RUBBER PAD

6.3 RESULT

Results of various parameters are observed and based on this the design is validated, below are the deformation plots of 10 modes

6.3.1 Mode 1

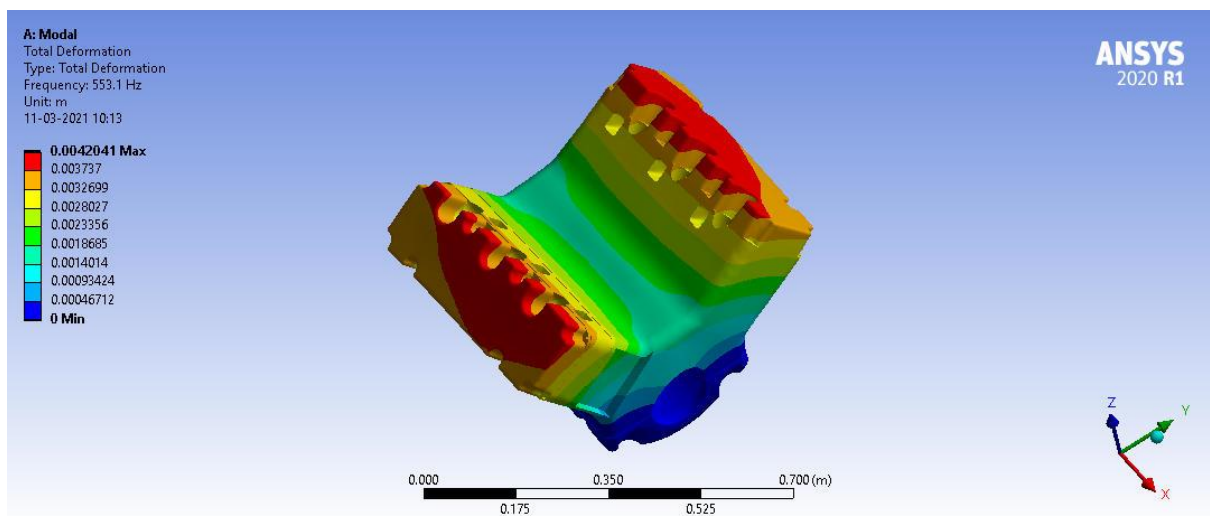


Fig: 6.2: Eigen mode contour plot of mode-1

Normal mode analysis done on the engine at frequency (F) =553.1 Hz at an angle 180, The respective outcome is Max=0.0042041, Min=0.

6.3.2 Mode 2

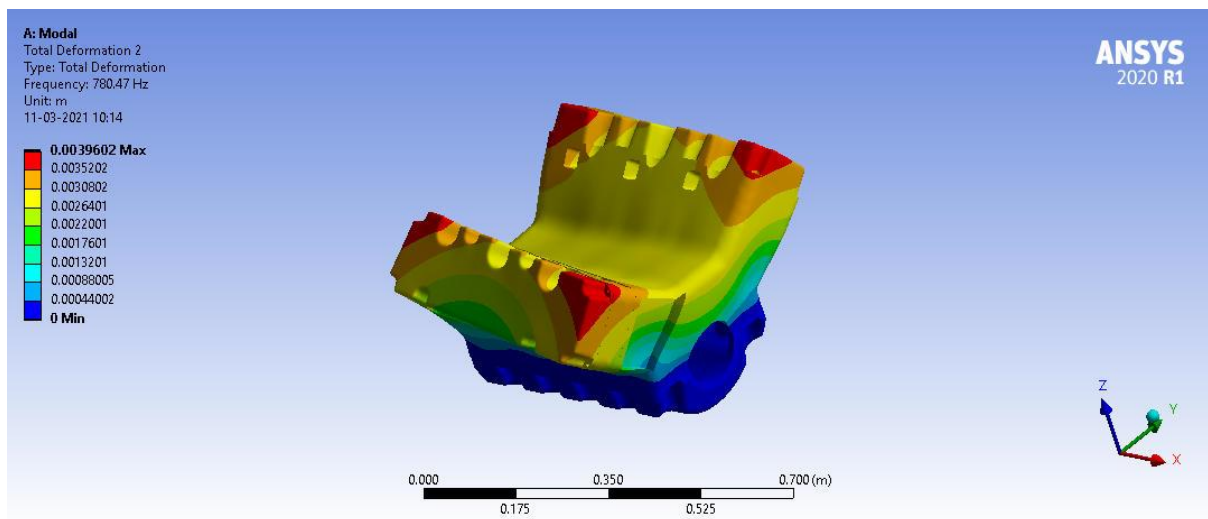


Fig: 6.3: Eigen mode contour plot of mode-2

Normal mode analysis done on the engine at frequency (F) =780.47 Hz at an angle 180, The respective outcome is Max=0.0039602, Min=0

6.3.3 Mode 3

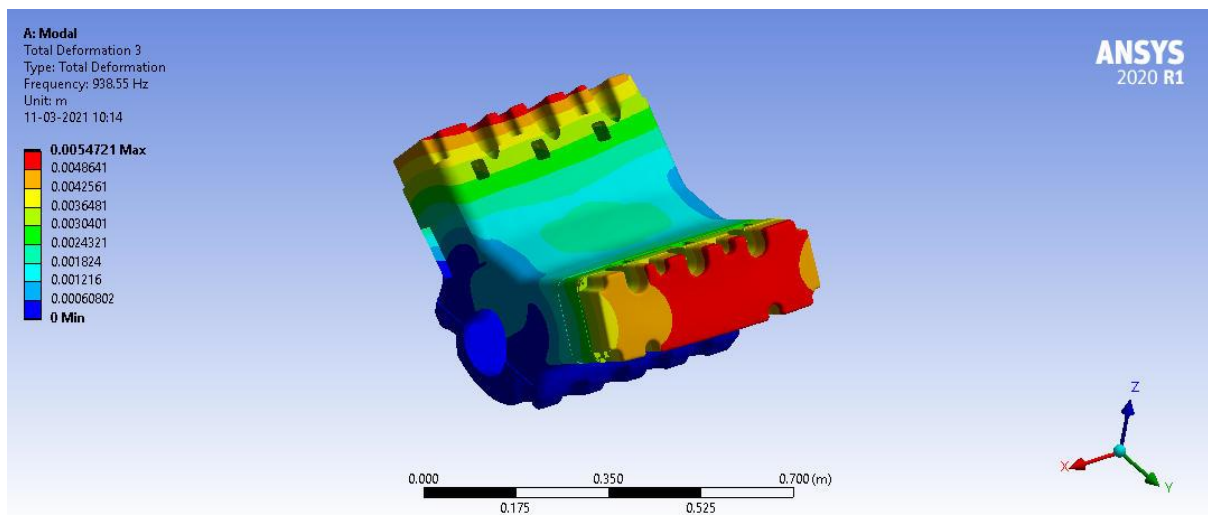


Fig: 6.4: Eigen mode contour plot of mode-3

Normal mode analysis done on the engine at frequency (F) =938.55 Hz at an angle 180, The respective outcome is Max=0.0054721, Min=0

6.3.4 Mode 4

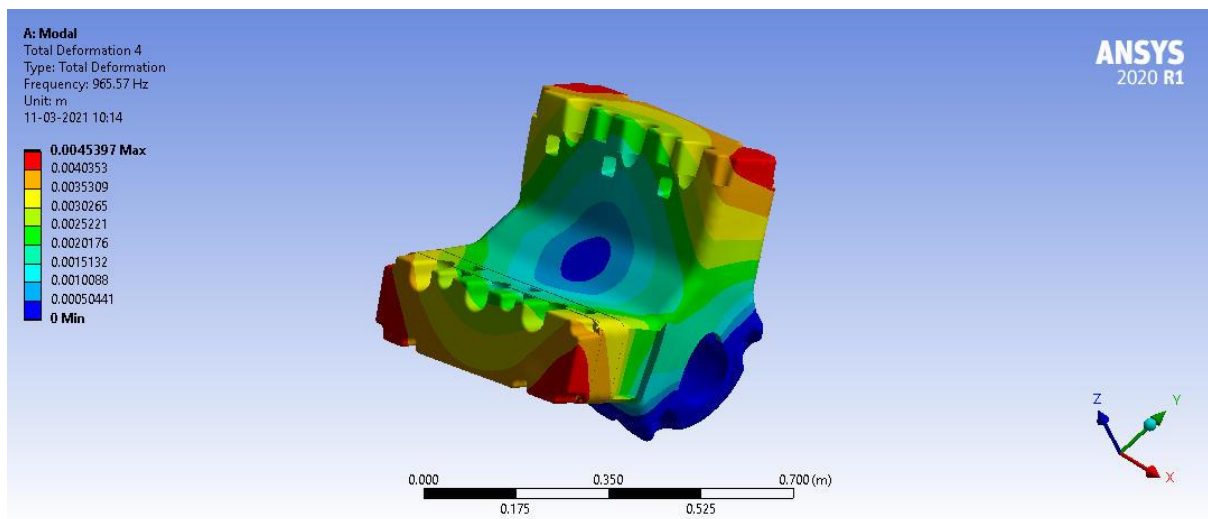


Fig: 6.5: Eigen mode contour plot of mode-4

Normal mode analysis done on the engine at frequency (F) =965.57 Hz at an angle 180, The respective outcome is Max=0.0045397, Min=0

6.3.5 Mode 5

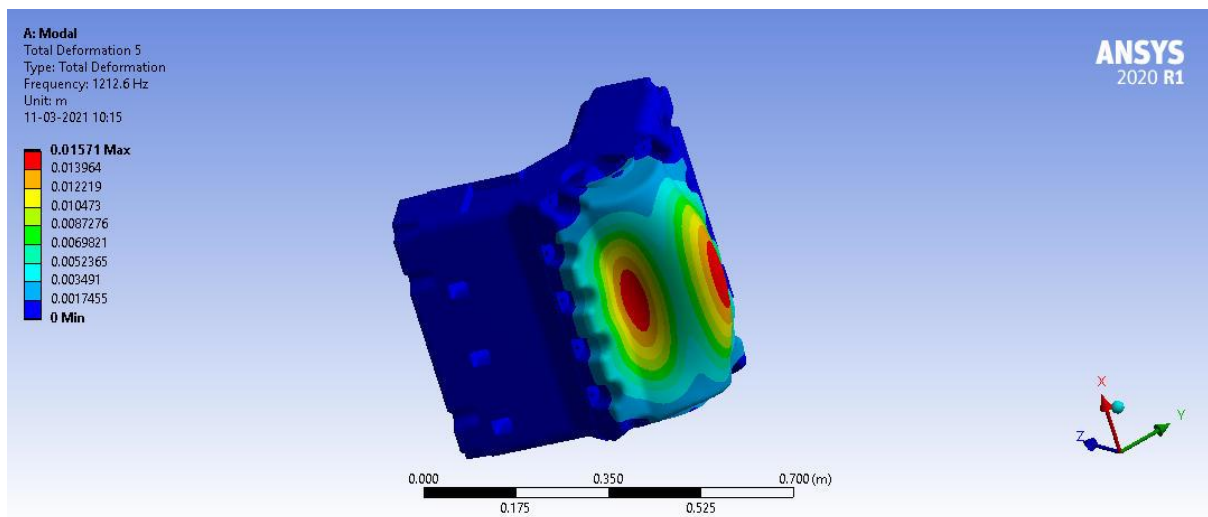


Fig: 6.6: Eigen mode contour plot of mode-5

Normal mode analysis done on the engine at frequency (F) =1212.6 Hz at an angle 180, The respective outcome is Max=0.01571, Min=0

6.3.6 Mode 6

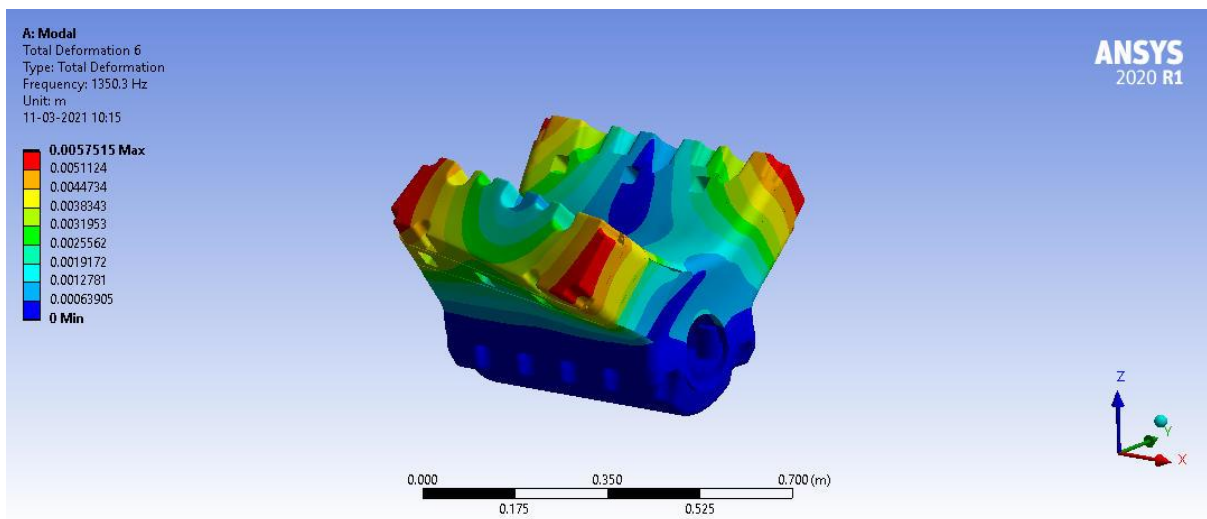


Fig: 6.7: Eigen mode contour plot of mode-6

Normal mode analysis done on the engine at frequency (F) =1350.3 Hz at an angle 180, The respective outcome is Max=0.0057515, Min=0

6.3.7 Mode 7

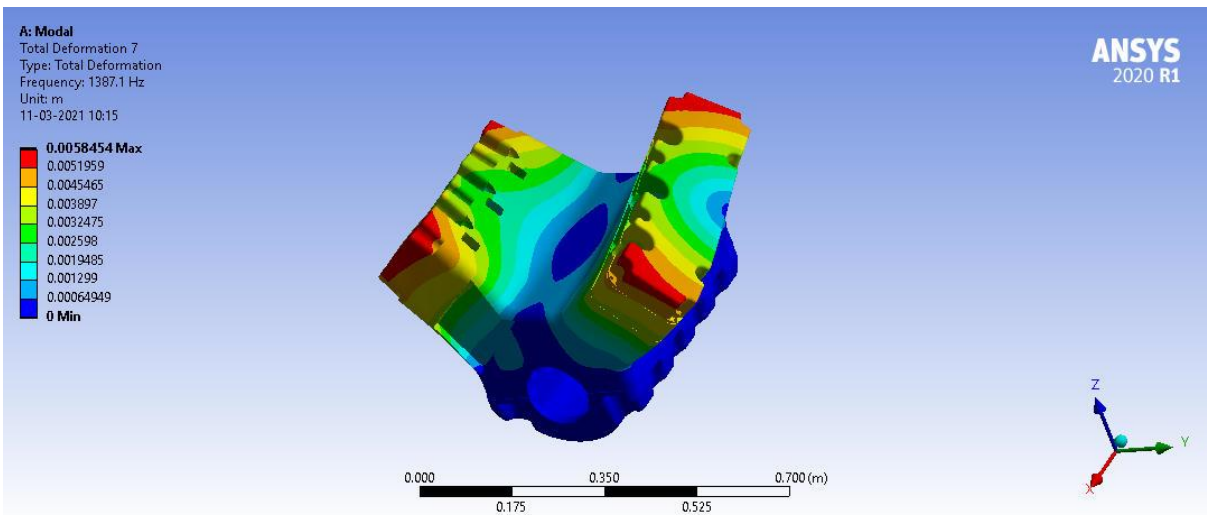


Fig: 6.8: Eigen mode contour plot of mode-7

Normal mode analysis done on the engine at frequency (F) =1387.1 Hz at an angle 180, The respective outcome is Max=0.0058454, Min=0

6.3.8 Mode 8

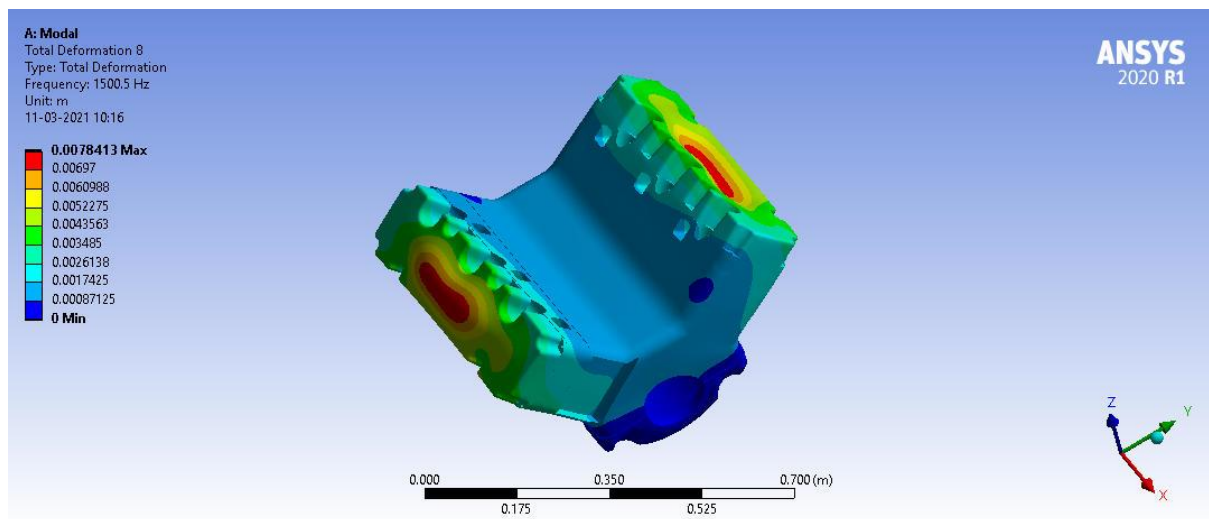


Fig: 6.9: Eigen mode contour plot of mode-8

Normal mode analysis done on the engine at frequency (F) =1500.5 Hz at an angle 180, The respective outcome is Max=0.0078413, Min=0

6.3.9 Mode 9

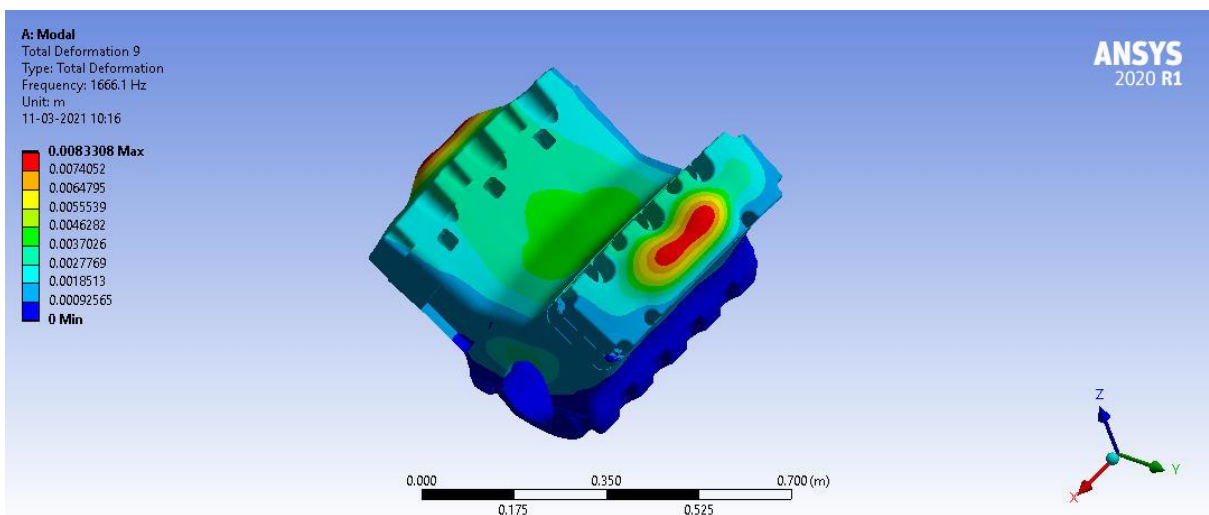


Fig: 6.10: Eigen mode contour plot of mode-9

Normal mode analysis done on the engine at frequency (F) =1666.1 Hz at an angle 180, The respective outcome is Max=0.0083308, Min=0

6.3.10 Mode 10

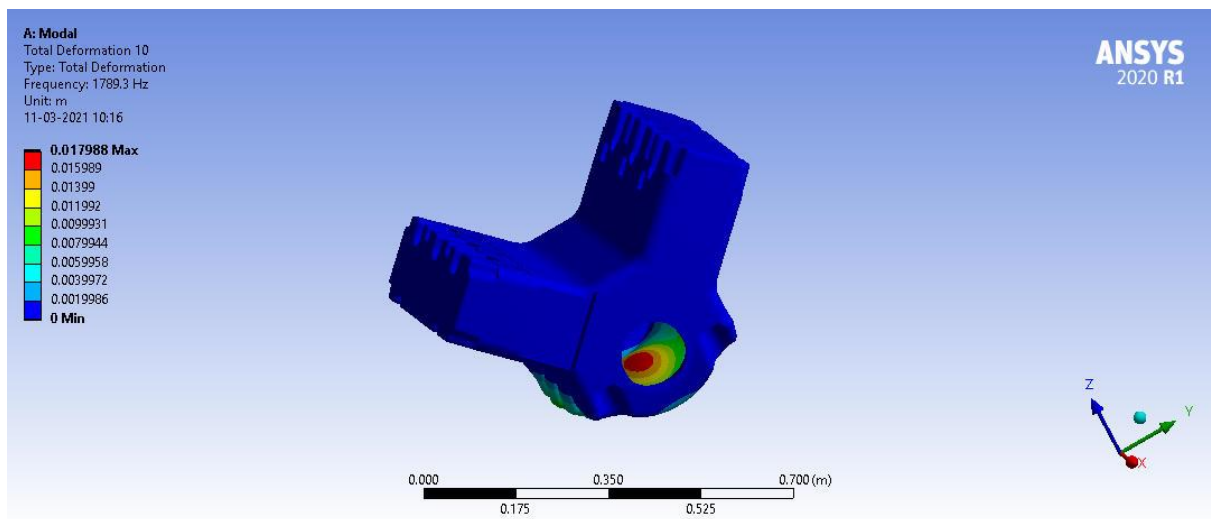


Fig: 6.11: Eigen mode contour plot of mode-10

Normal mode analysis done on the engine at frequency (F) =1789.3 Hz at an angle 180, The respective outcome is Max=0.017988, Min=0

6.3.11 Frequency at modes

Table: 6.4: Frequency at modes

S. No	Mode	Frequency(Hz)
1	1	553.1
2	2	780.47
3	3	938.55
4	4	965.57
5	5	1212.6
6	6	1350.3
7	7	1387.1

8	8	1500.5
9	9	1666.1
10	10	1789.3

6.3.12 Graph

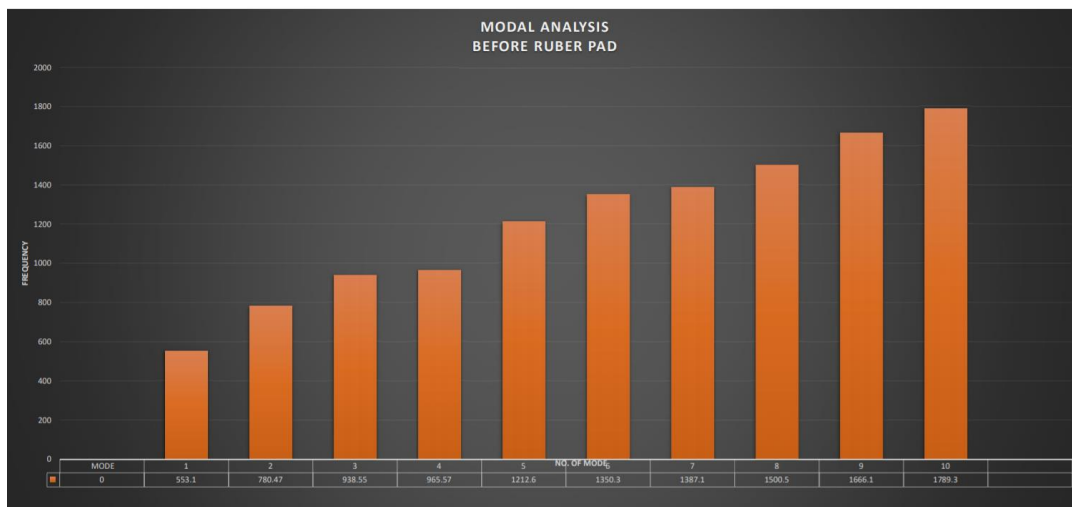


Fig: 6.12: Bar graph

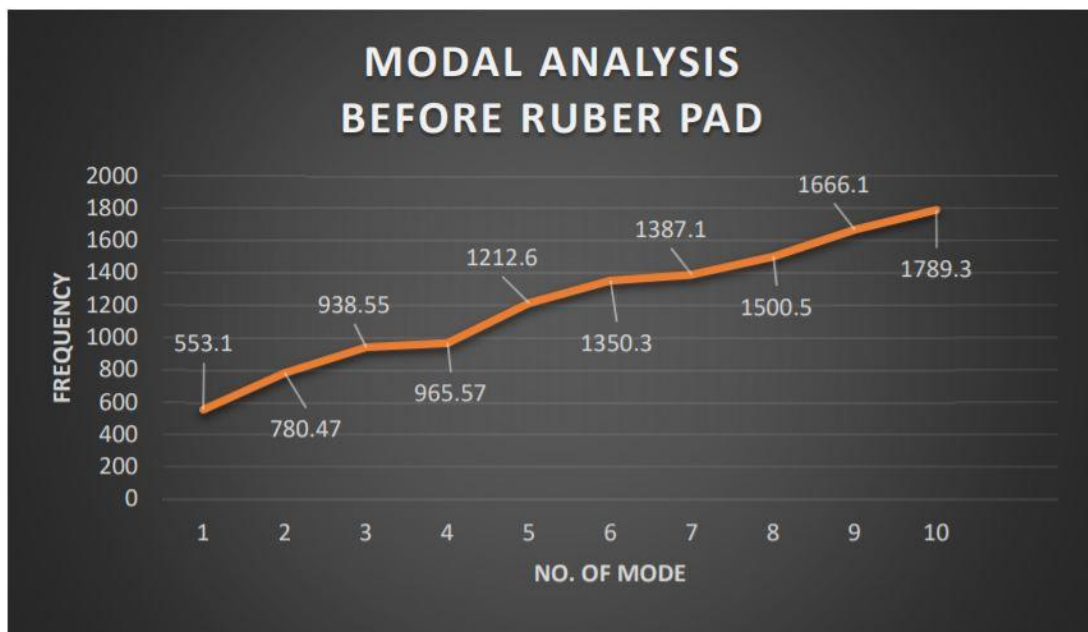


Fig: 6.13: Line graph

MODAL ANALYSIS AFTER RUBBER PAD

6.4 RESULT

Results of various parameters are observed and based on this the design is validated... Below are the deformation plots of 10 modes

6.4.1 Mode 1

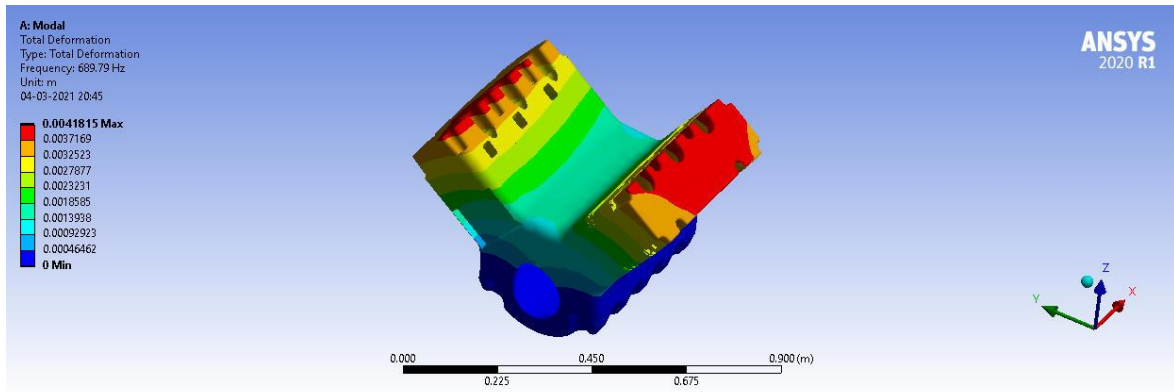


Fig: 6.14: Eigen mode contour plot of mode-1

Normal mode analysis done on the engine at frequency (F) =689.79 Hz at an angle 180, The respective outcome is Max=0.0041815, Min=0.

6.4.2 Mode 2

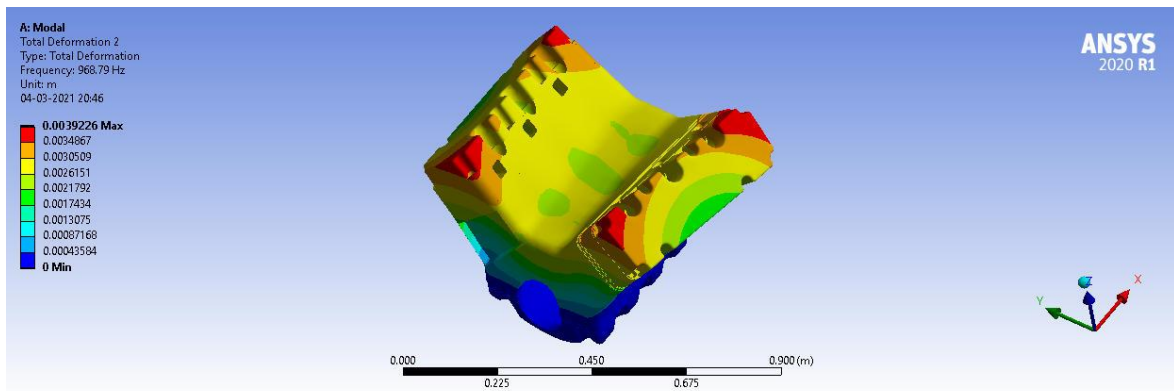


Fig: 6.15: Eigen mode contour plot of mode-2

Normal mode analysis done on the engine at frequency (F) =968.79 Hz at an angle 180, The respective outcome is Max=0.0039226, Min=0.

6.4.3 Mode 3

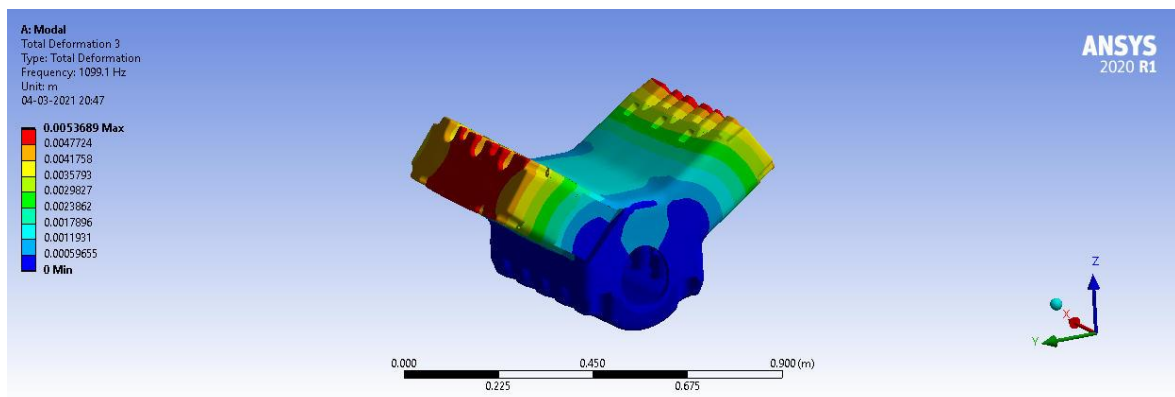


Fig: 6.15: Eigen mode contour plot of mode-3

Normal mode analysis done on the engine at frequency (F) =1099.1 Hz at an angle 180, The respective outcome is Max=0.0053689, Min=0.

6.4.4 Mode 4

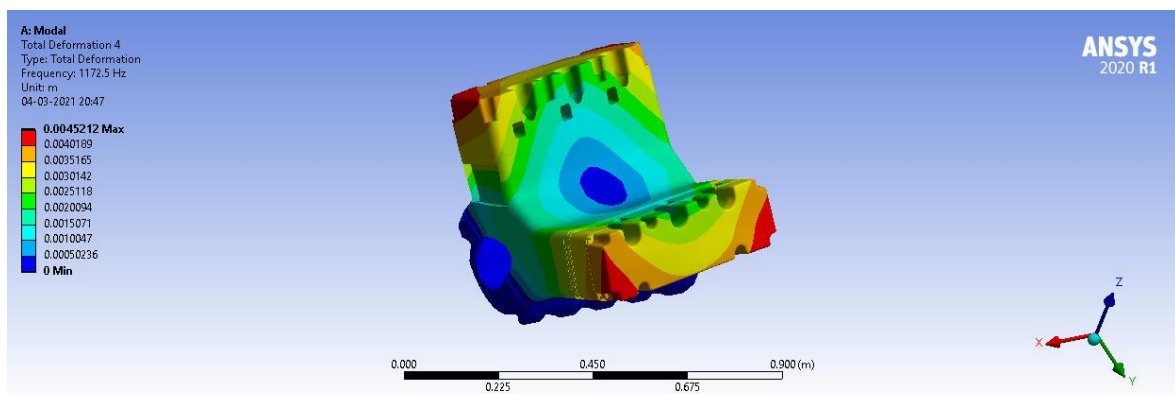


Fig: 6.16: Eigen mode contour plot of mode-4

Normal mode analysis done on the engine at frequency (F) =1172.5 Hz at an angle 180, The respective outcome is Max=0.0045212, Min=0.

6.4.5 Mode 5

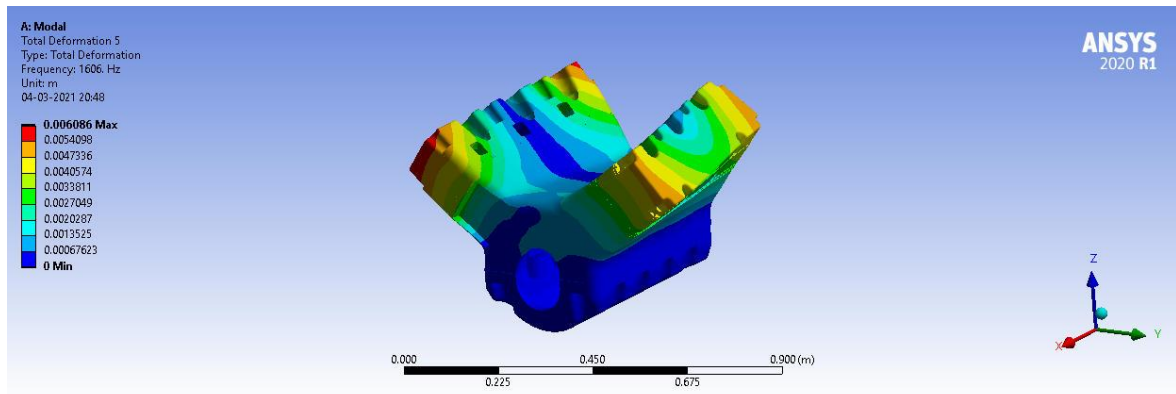


Fig: 6.17: Eigen mode contour plot of mode-5

Normal mode analysis done on the engine at frequency (F) =1606 Hz at an angle 180, The respective outcome is Max=0.006068, Min=0.

6.4.6 Mode 6

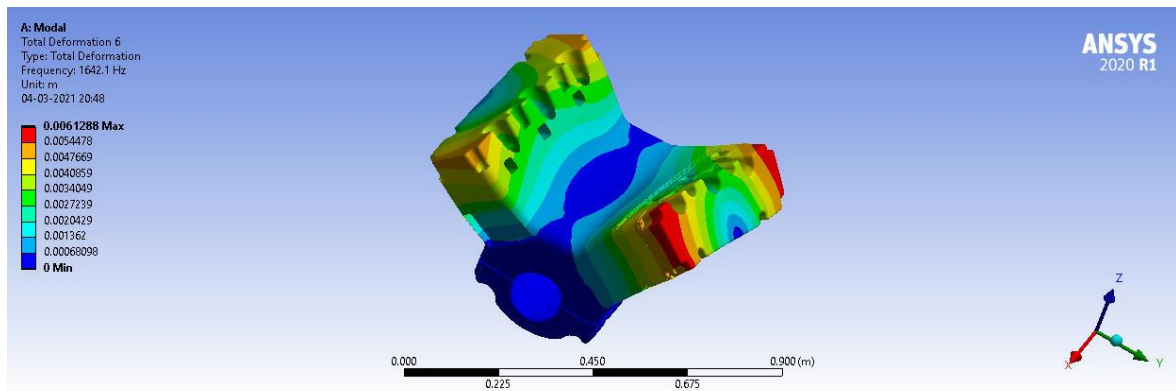


Fig: 6.18: Eigen mode contour plot of mode-6

Normal mode analysis done on the engine at frequency (F) =1642.1 Hz at an angle 180, The respective outcome is Max=0.0061288, Min=0.

6.4.7 Mode 7

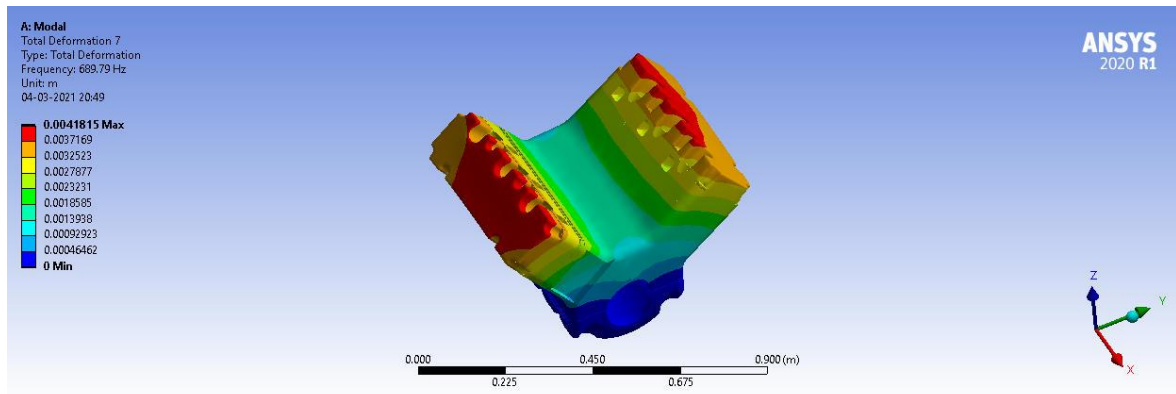


Fig: 6.19: Eigen mode contour plot of mode-7

Normal mode analysis done on the engine at frequency (F) =1707.2 Hz at an angle 180, The respective outcome is Max=0.0061288, Min=0.

6.4.8 Mode 8

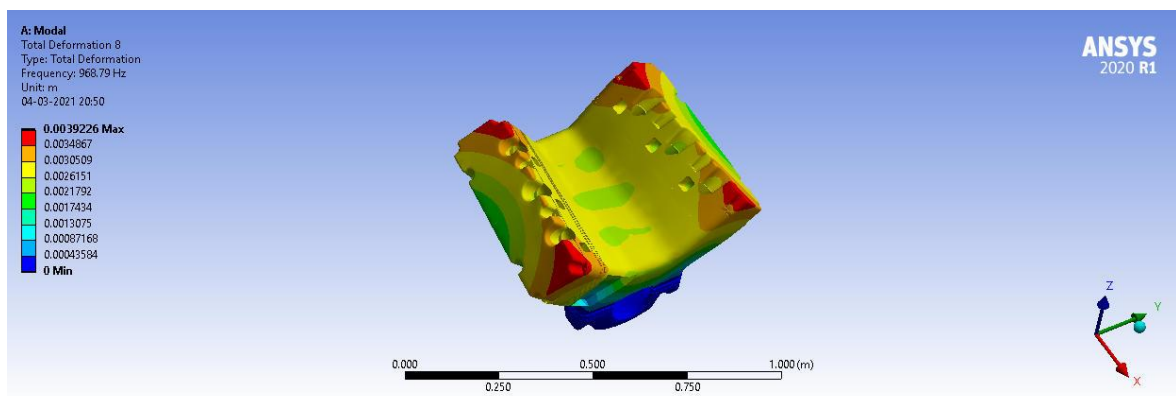


Fig: 6.20: Eigen mode contour plot of mode-8

Normal mode analysis done on the engine at frequency (F) =1846.1 Hz at an angle 180, The respective outcome is Max=0.0039226, Min=0.

6.4.9 Mode 9

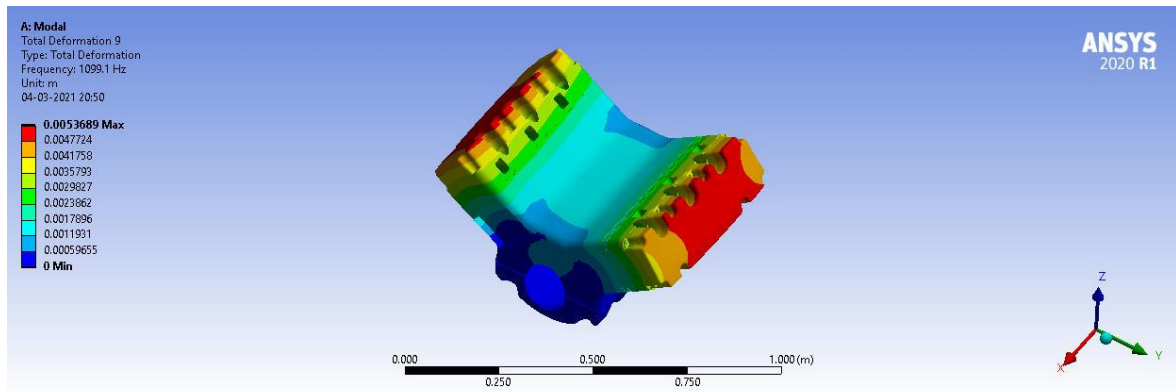


Fig: 6.21: Eigen mode contour plot of mode-9

Normal mode analysis done on the engine at frequency (F) =1846.1 Hz at an angle 180, The respective outcome is Max=0.0053689, Min=0.

6.4.10 Mode 10

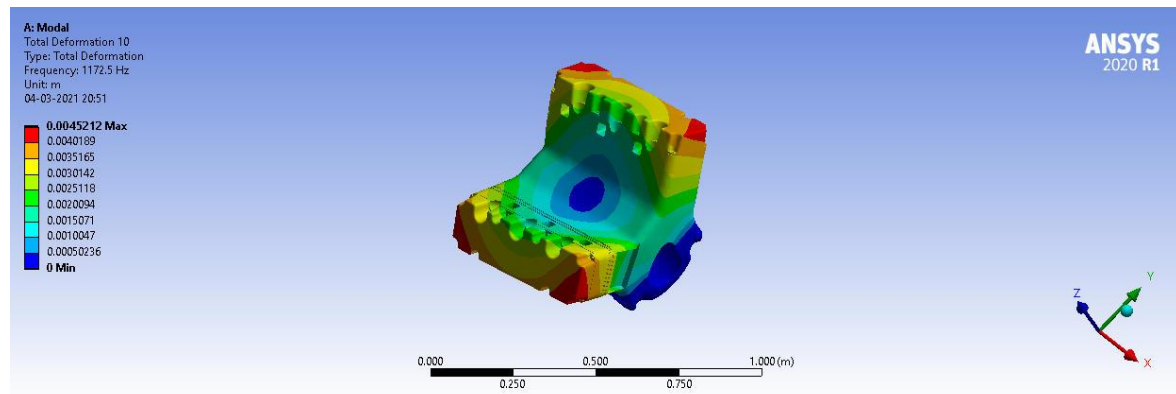


Fig: 6.22: Eigen mode contour plot of mode-10

Normal mode analysis done on the engine at frequency (F) =2356Hz at an angle 180, The respective outcome is Max=0.0045212, Min=0.

6.4.11 Tabular data

Table: 6.5: Frequency at modes

S. No	Mode	Frequency(Hz)
1	1	689.79
2	2	968.79
3	3	1099.1
4	4	1172.5

5	5	1606
6	6	1642.1
7	7	1707.2
8	8	1846.1
9	9	2119
10	10	2356.2

6.4.12 Graph

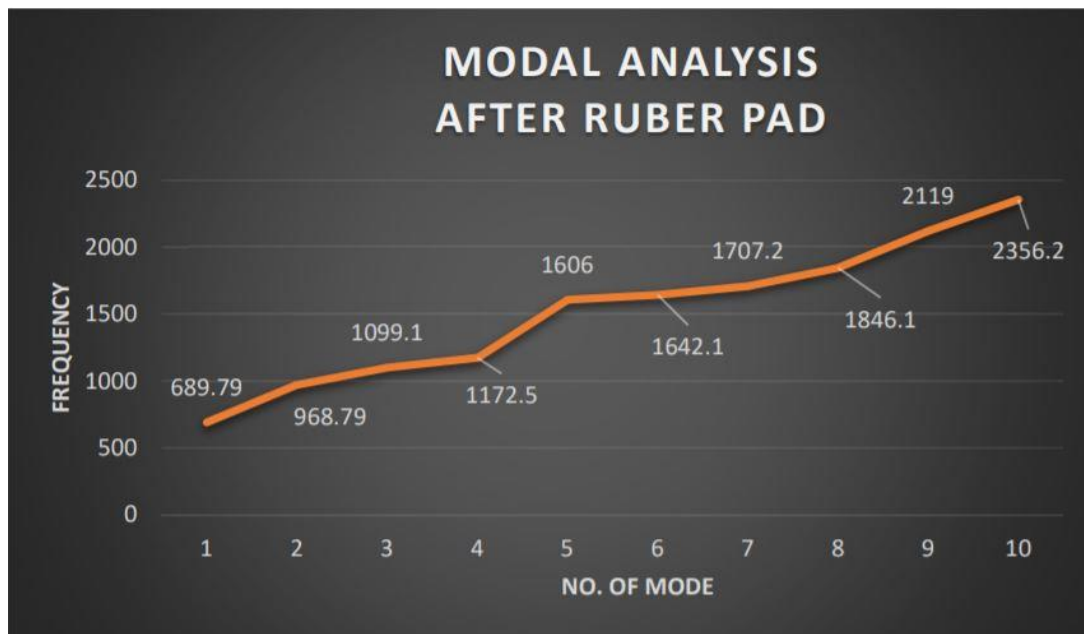


Fig: 6.22: Line graph

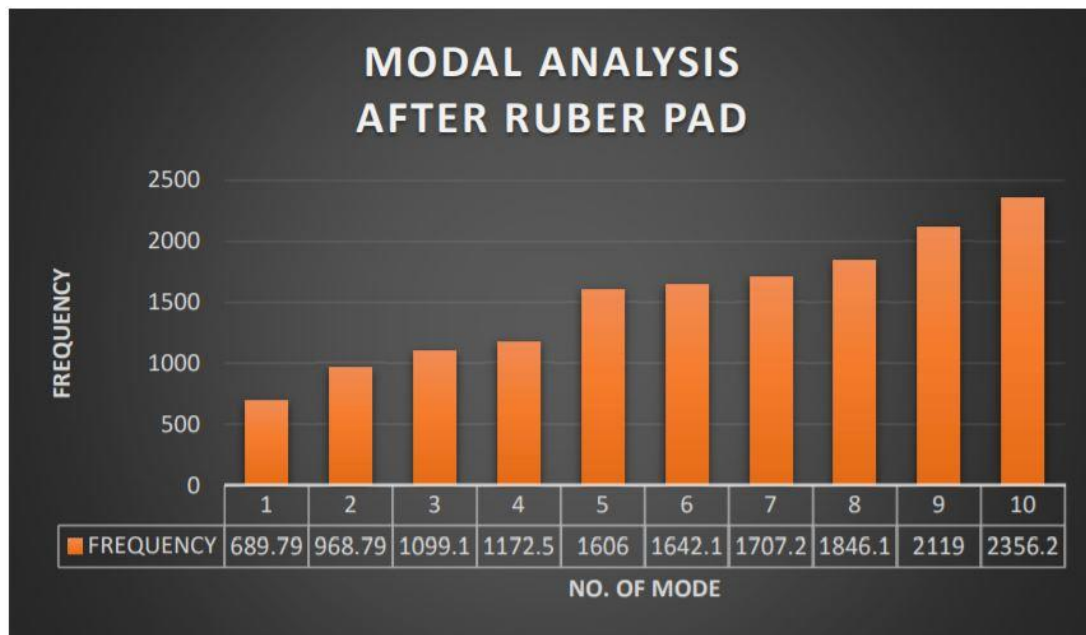


Fig: 6.22: Bar graph

6.5 COMPARING THE RESULTS OF MODAL ANALYSIS BEFORE AND AFTER KEEPING THE RUBBER PAD.

6.5.1 TABULAR DATA

Table: 6.6: Frequency at modes both before and after

MODE	BEFORE	AFTER
1	553.1	689.79
2	780.47	968.79
3	938.55	1099.1
4	965.57	1172.5
5	1212.6	1606
6	1350.3	1642.1
7	1387.1	1707.2
8	1500.5	1846.1
9	1666.1	2119
10	1789.3	2356.2

6.5.2 GRAPH

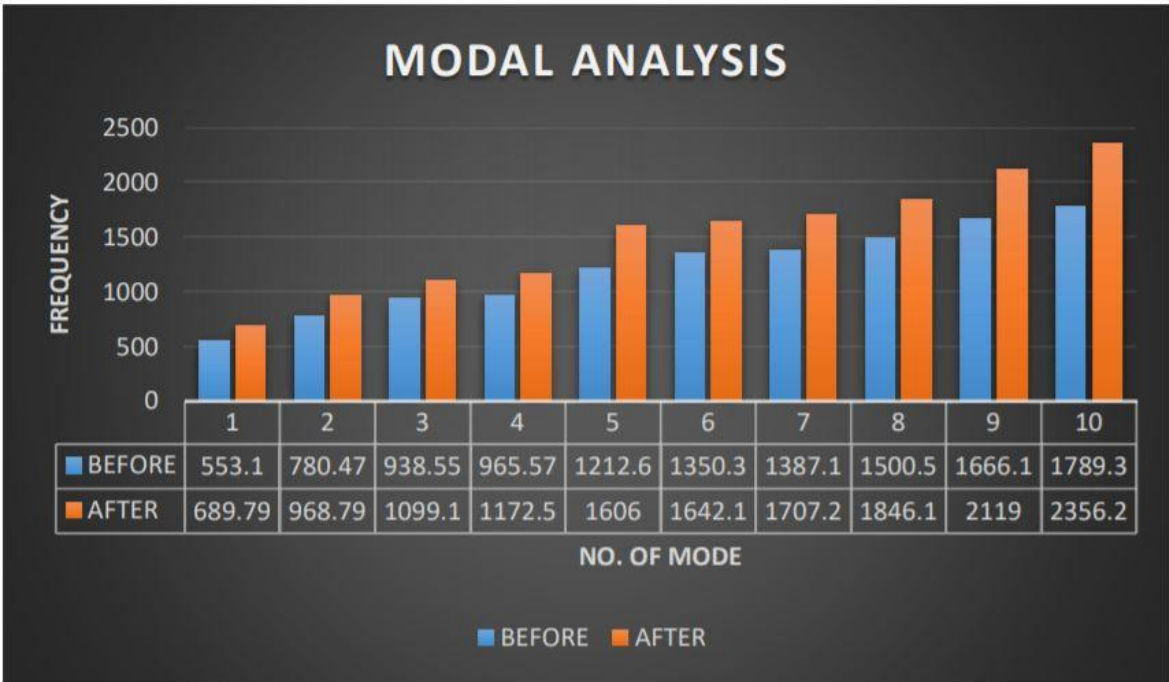


Fig: 6.23: Modal diff. Bar graph

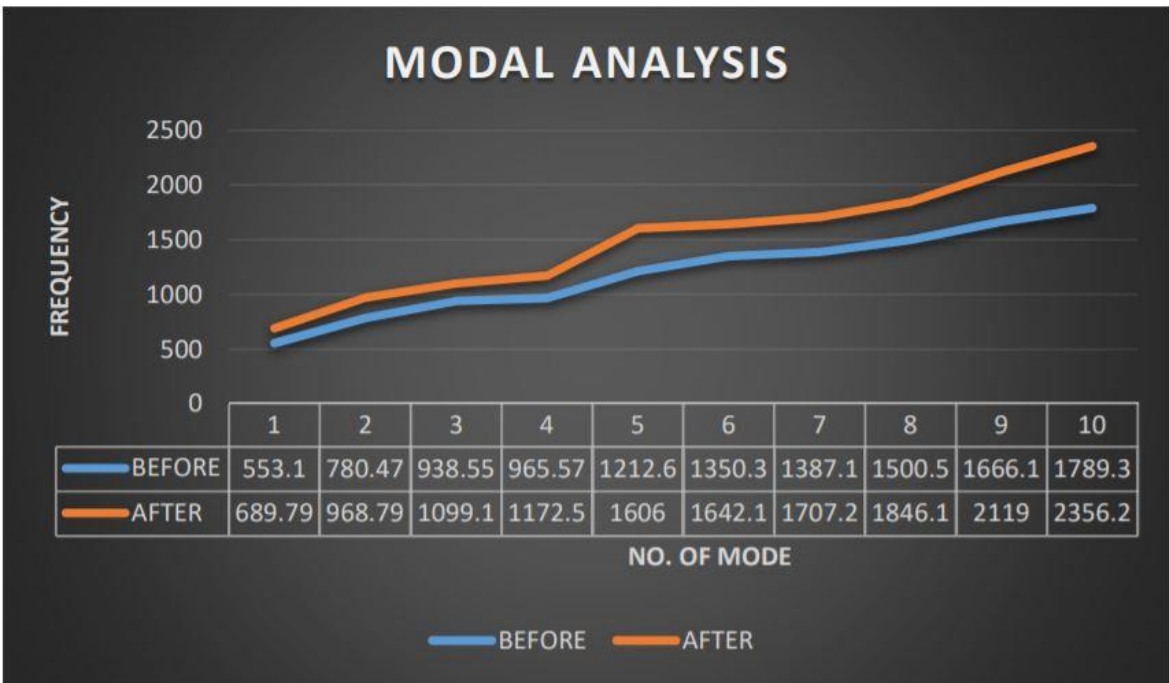


Fig: 6.24: Modal diff. Line graph

CHAPTER 7 FREQUENCY RESPONSE ANALYSIS

7.1 INTRODUCTION TO FRA

Modal analysis helps to determine the modes of vibrations and the frequencies at which those modes are active. But Modal analysis does not give the engineer any info about the real deformation that an excitation of one of those modes will actually cause. So to determine the behaviour of the components at various frequency levels FRA is used. With FRA we can determine more accurate behaviour of the components under various load conditions. This helps in validating the design.

7.2 PRE-PROCESSING

The main aim of this analysis is to see how a body is acting at various frequency levels. So the frequency values are given as an input to the solver and analysis is carried out.

FREQUENCY RESPONSE ANALYSIS BEFORE RUBBER PAD

7.2.1 Inputs

Tabled Frequency range (Hz)

Frequency Spacing: Linear

Range Minimum: 500 Hz

Range Maximum: 2000Hz

Solution Intervals: 50

 Force: (Real) 100., (Imag) 0. N
Components: (Real) 0., 2.e-013, 100. N
Components: (Imag) 0., 0., 0. N

7.2.2 Boundary conditions

As we are trying to see the frequency response w.r.t to the modal analysis is done same setup of boundary conditions are used.

7.3 RESULTS

To validate the results, we have considered 10 load conditions out of 50. They are Load 1,5,10,15,20, 30,35,40,45,50

7.3.1 LOAD 1 (530Hz)

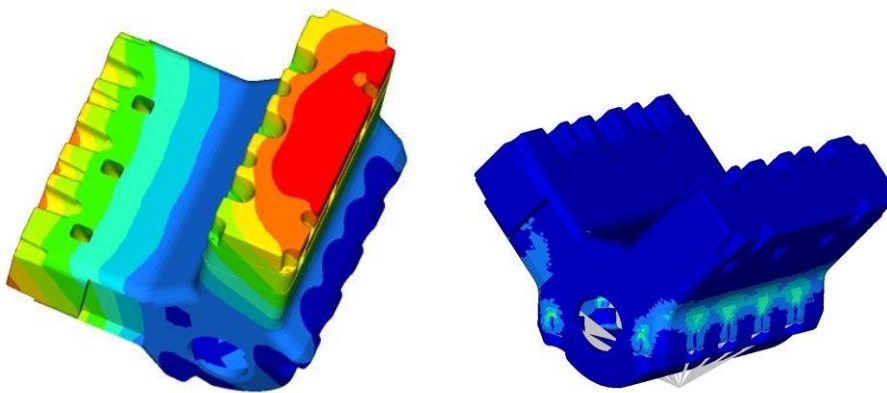


Fig: 7.1: Displacement and Element stresses at load 1

Frequency response analysis done on the engine at frequency (F) =530Hz, Amplitude= 2.1653×10^{-12} at an angle 0.

7.3.2 LOAD 5 (650Hz)

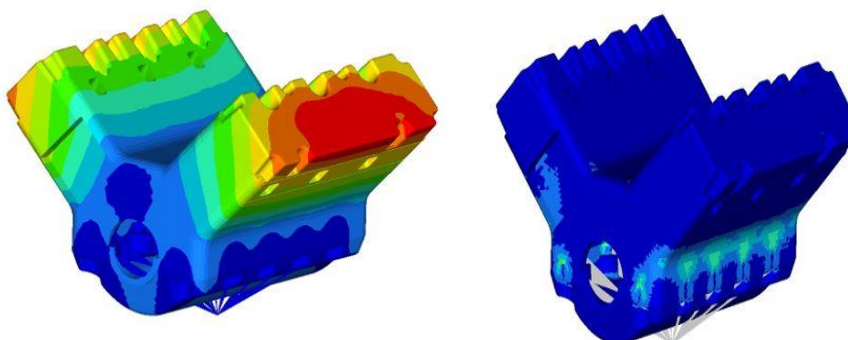


Fig: 7.2: Displacement and Element stresses at load 5

Frequency response analysis done on the engine at frequency (F) =650Hz, Amplitude= 3.7478×10^{-12} at an angle 0.

7.3.3 LOAD 10 (800Hz)

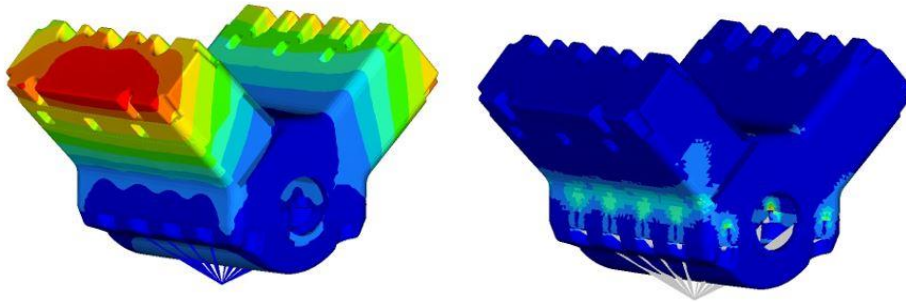


Fig: 7.3: Displacement and Element stresses at load 10

Frequency response analysis done on the engine at frequency (F)
=800Hz, Amplitude= 1.4648×10^{-11} at an angle 180.

7.3.4 LOAD 15 (950Hz)

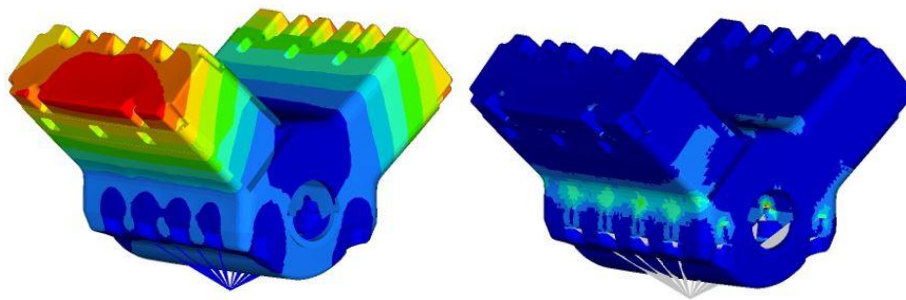


Fig: 7.4: Displacement and Element stresses at load 15

Frequency response analysis done on the engine at frequency (F)
=950Hz, Amplitude= 2.6317×10^{-11} at an angle 180.

7.3.5 LOAD 20 (1100Hz)

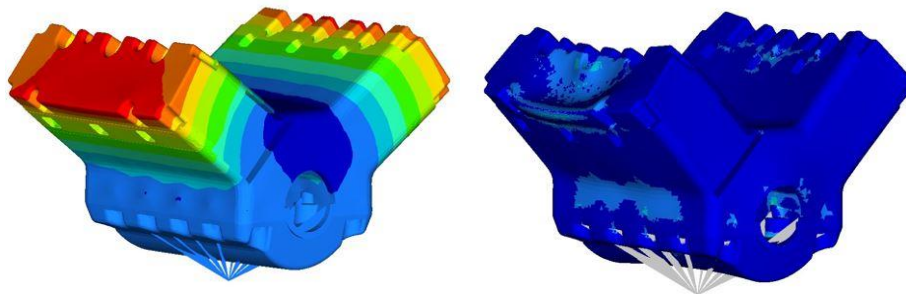


Fig: 7.5: Displacement and Element stresses at load 20

Frequency response analysis done on the engine at frequency (F)
=1100Hz, Amplitude= 2.3725×10^{-12} at an angle 180.

7.3.6 LOAD 30 (1400Hz)

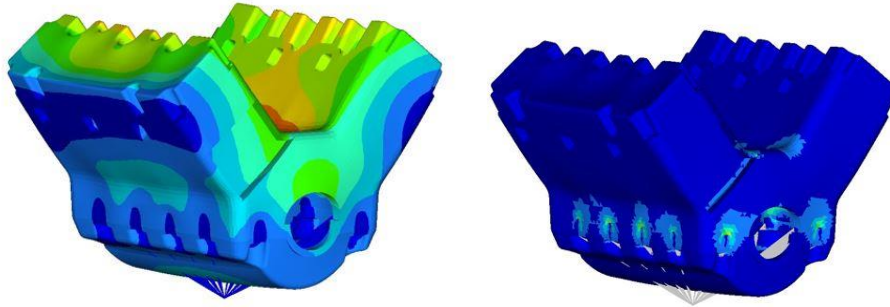


Fig: 7.6: Displacement and Element stresses at load 30

Frequency response analysis done on the engine at frequency (F)
=1400Hz, Amplitude= 5.8662×10^{-12} at an angle 180.

7.3.7 LOAD 35 (1550Hz)

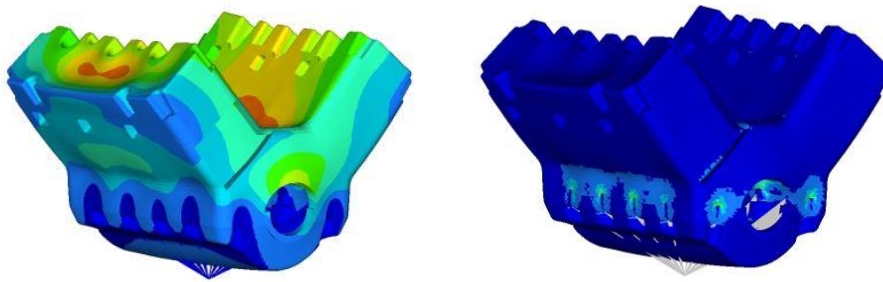


Fig: 7.7: Displacement and Element stresses at load 35

Frequency response analysis done on the engine at frequency (F)
=1550Hz, Amplitude= 7.0931×10^{-12} at an angle 180.

76.3.8 LOAD 40 (1700Hz)

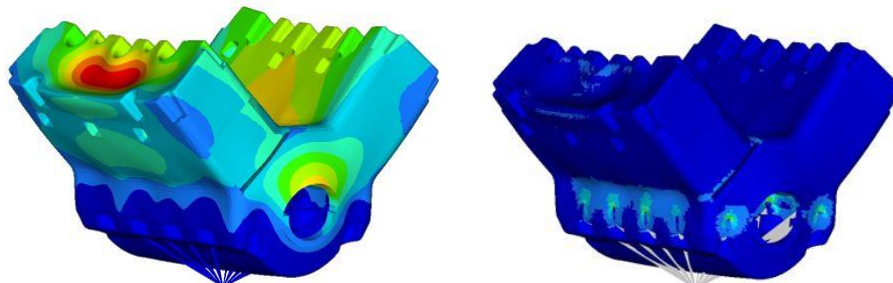


Fig: 7.8: Displacement and Element stresses at load 40

Frequency response analysis done on the engine at frequency (F)
=1700Hz, Amplitude= 1.5387×10^{-11} at an angle 0.

7.3.9 LOAD 45 (1850Hz)

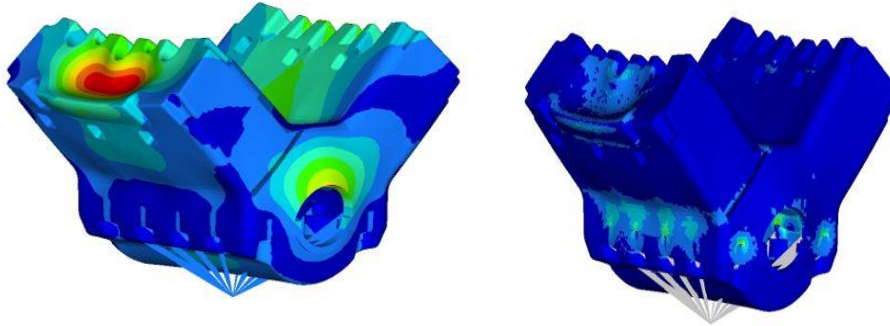


Fig: 7.9: Displacement and Element stresses at load 45

Frequency response analysis done on the engine at frequency (F)
=1850Hz, Amplitude= 2.0386×10^{-12} at an angle 0.

7.3.10 LOAD 50 (2000Hz)

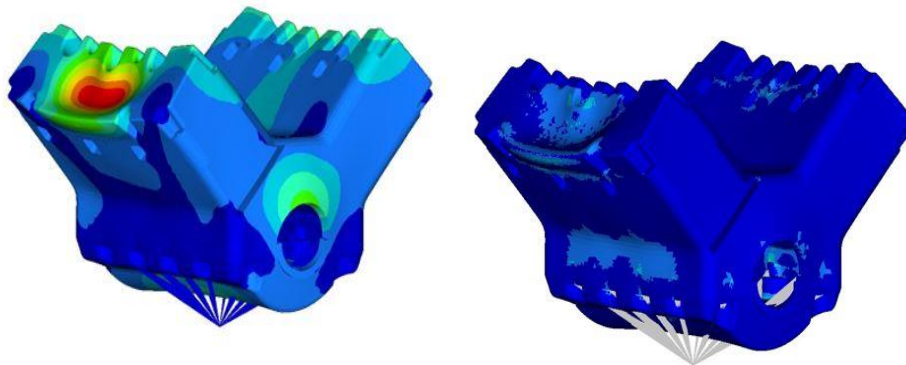


Fig: 7.10: Displacement and Element stresses at load 50

Frequency response analysis done on the engine at frequency (F)
=2000Hz, Amplitude= 8.5569×10^{-13} at an angle 0.

7.3.11 Graph

Table: 7.1: Frequency and amplitude

MODE	FREQUENCY	AMPLITUDE
1	530	2.17E-12
5	650	3.75E-12
10	800	1.46E-11
15	950	2.63E-11
20	1100	2.37E-12
30	1400	5.87E-12
35	1550	7.09E-12
40	1700	1.54E-11
45	1850	2.04E-12
50	2000	8.56E-13

7.3.11.1 Frequency/Amplitude

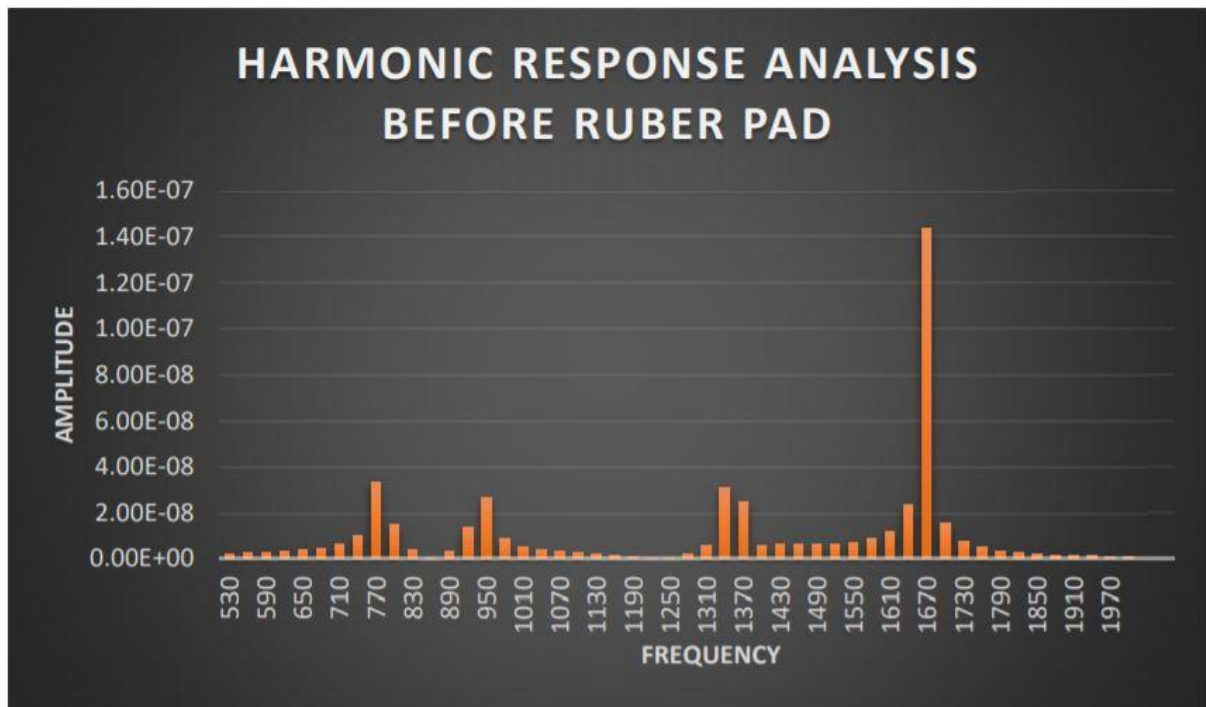


Fig: 7.11: Frequency/Amplitude Bar graph

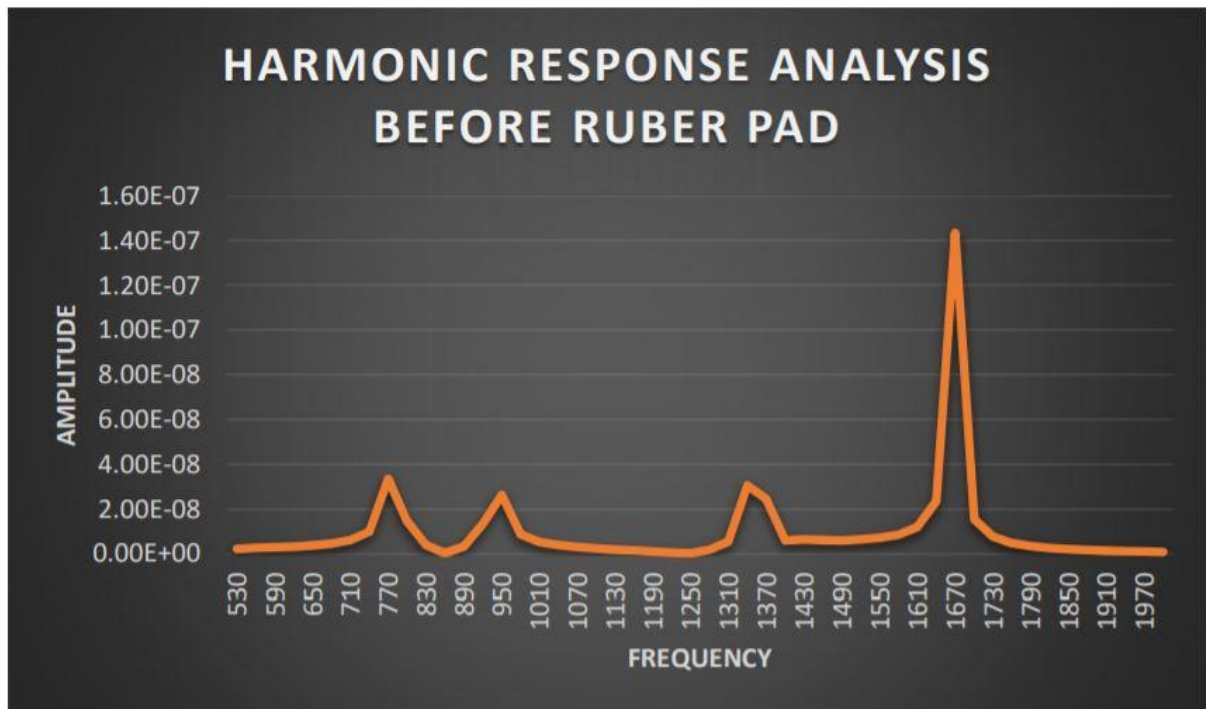


Fig. 7.12: Frequency/Amplitude Line graph

Table: 7.2: Frequency and Phase angle

MODE	FREQUENCY	PHASE ANGLE
1	530	0
5	650	0
10	800	180
15	950	180
20	1100	180
30	1400	180
35	1550	180
40	1700	0
45	1850	0
50	2000	0

7.3.11.2 Frequency/Phase angle

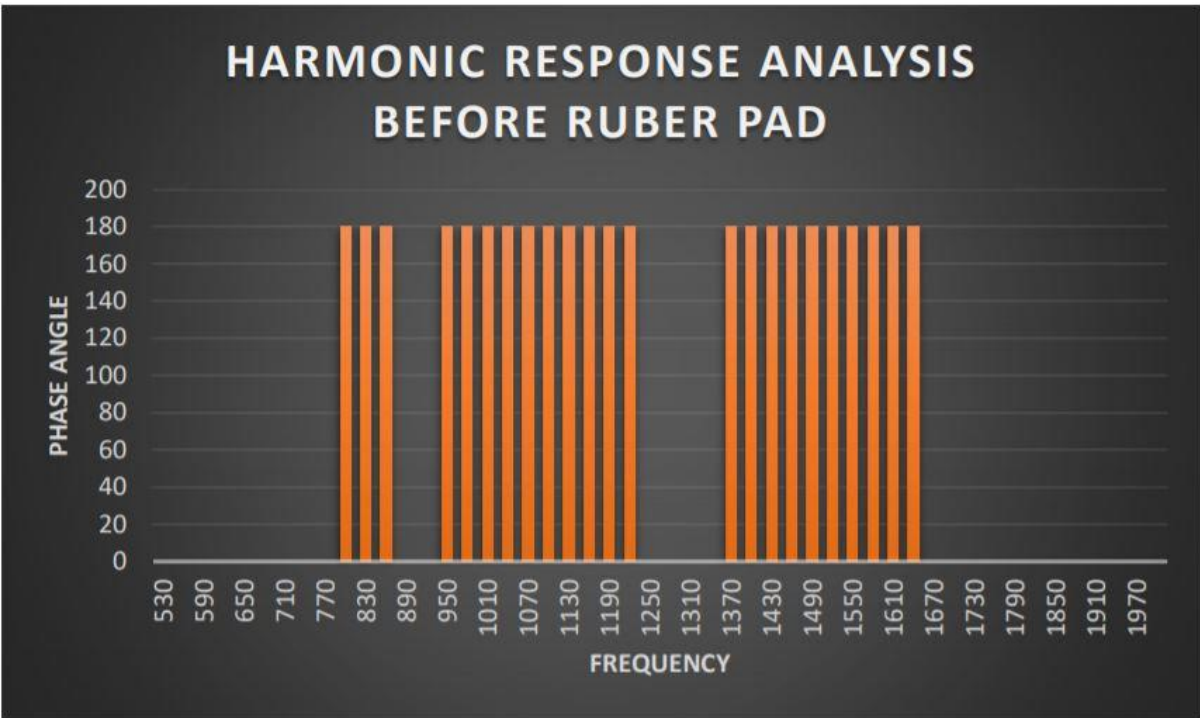


Fig: 7.13: Frequency/Phase angle Bar graph

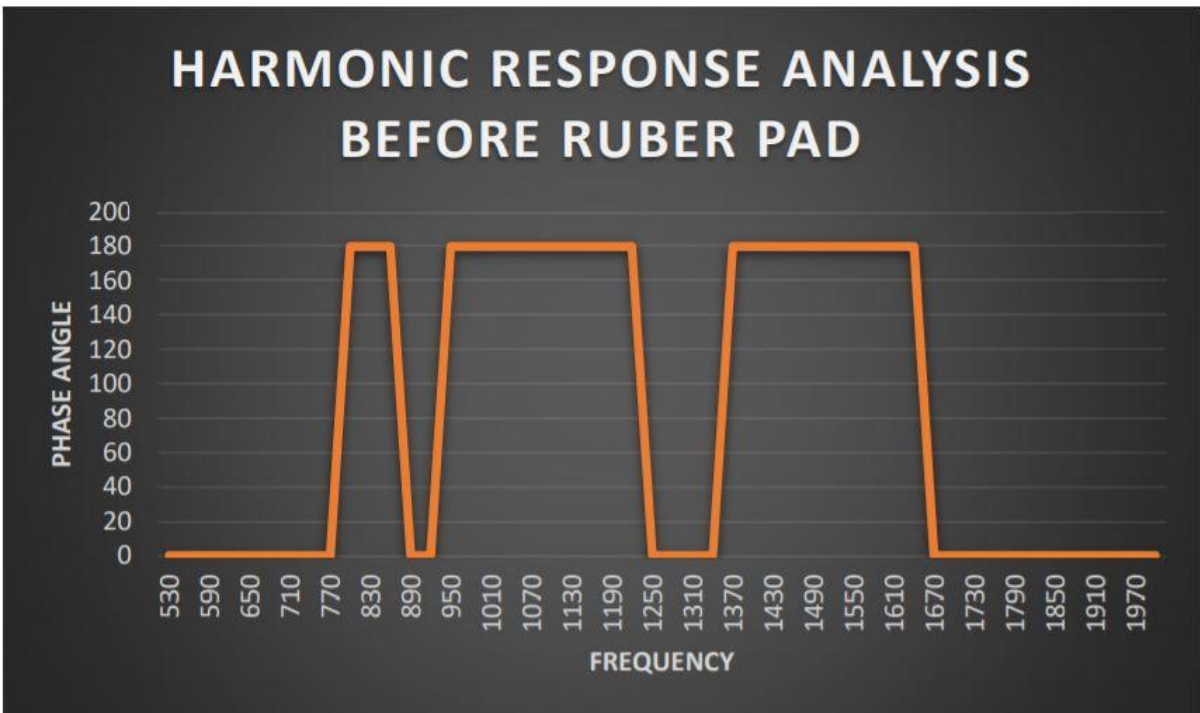


Fig: 7.14: Frequency/Phase angle Line graph

FREQUENCY RESPONSE ANALYSIS AFTER RUBBER PAD

7.4 Inputs

Tabled Frequency range (Hz)

Frequency Spacing: Linear

Range Minimum: 500 Hz

Range Maximum: 2500Hz

Solution Intervals: 50

 Force: (Real) 100., (Imag) 0. N
Components: (Real) 0., 2.e-013, 100. N
Components: (Imag) 0., 0., 0. N

7.4.1 Boundary conditions

As we are trying to see the frequency response w.r.t to the modal analysis is done same setup of boundary conditions are used.

7.5 RESULTS

To validate the results, we have considered 10 load conditions out of 50. They are Load 1,5,10,15,20, 30,35,40,45,50

7.5.1 LOAD 1 (514Hz)

Frequency response analysis done on the engine at frequency (F) =514Hz, Amplitude= 1.2564e-011 at an angle 0.

7.5.2 LOAD 5 (570Hz)

Frequency response analysis done on the engine at frequency (F) =570Hz, Amplitude= 1.3283e-011 at an angle 0.

7.5.3 LOAD 10 (640Hz)

Frequency response analysis done on the engine at frequency (F) =640Hz, Amplitude= 1.6122×10^{-11} at an angle 0.

7.5.4 LOAD 15 (710Hz)

Frequency response analysis done on the engine at frequency (F) =710Hz, Amplitude= 2.5345×10^{-11} at an angle 0.

7.5.5 LOAD 20 (780Hz)

Frequency response analysis done on the engine at frequency (F) =780Hz, Amplitude= 1.4253×10^{-10} at an angle 0.

7.5.6 LOAD 30 (920Hz)

Frequency response analysis done on the engine at frequency (F) =920Hz, Amplitude= 1.1433×10^{-11} at an angle 180.

7.5.7 LOAD 35 (990Hz)

Frequency response analysis done on the engine at frequency (F) =990Hz, Amplitude= 7.0186×10^{-12} at an angle 180.

7.5.8 LOAD 40 (1060Hz)

Frequency response analysis done on the engine at frequency (F) =1060Hz, Amplitude= 5.654×10^{-12} at an angle 180.

7.5.9 LOAD 45 (1130Hz)

Frequency response analysis done on the engine at frequency (F) =1130Hz, Amplitude= 7.8082×10^{-12} at an angle 180.

7.5.10 LOAD 50 (1200Hz)

Frequency response analysis done on the engine at frequency (F) =1200Hz, Amplitude= 3.2271×10^{-10} at an angle 180.

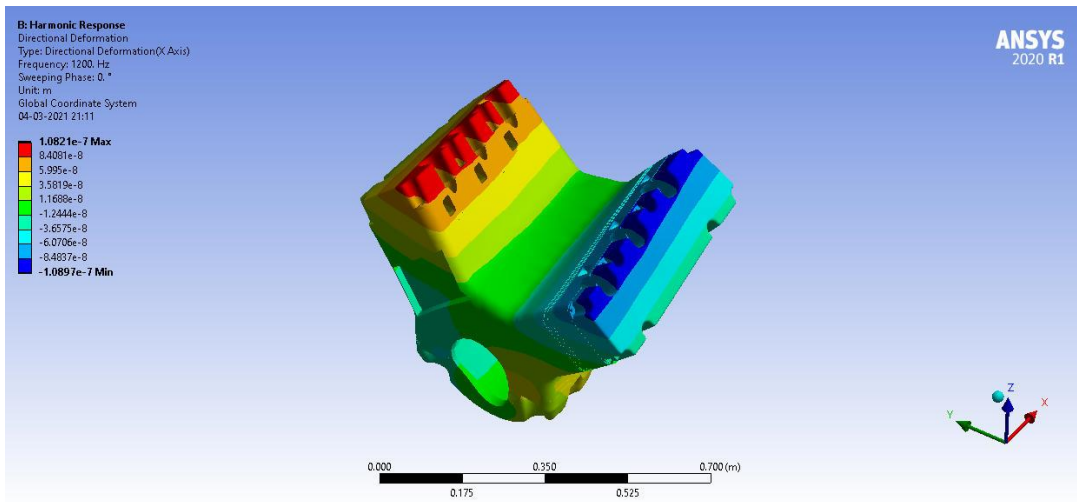


Fig. 7.15: Directional Deformation

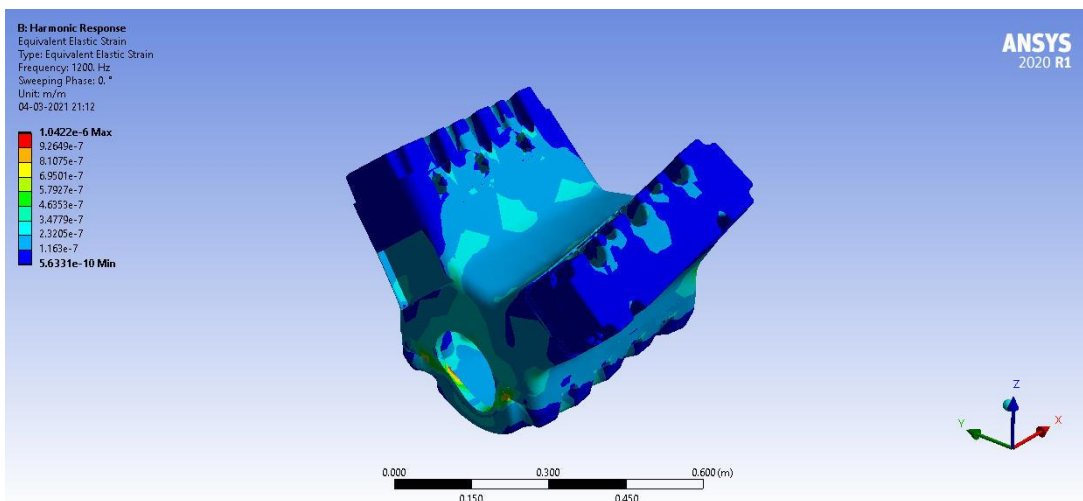


Fig. 7.16: Equivalent Elastic Strain

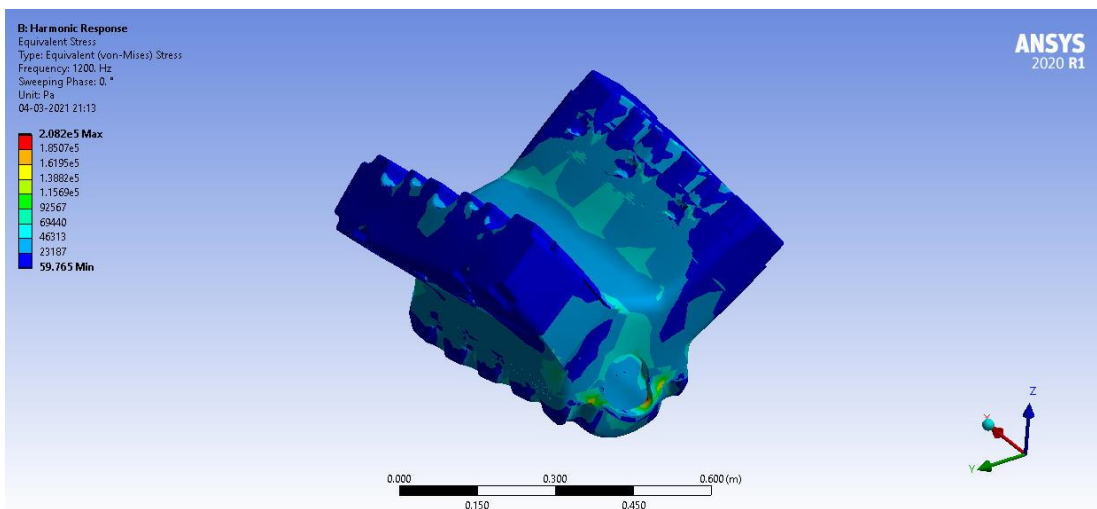


Fig. 7.17: Equivalent Stress

7.5.11 Graph:

Table: 7.3: Frequency and amplitude

MODE	FREQUENCY	AMPLITUDE
1	514	1.26E-11
5	570	1.33E-11
10	640	1.61E-11
15	710	2.53E-11
20	780	1.43E-10
30	920	1.14E-11
35	990	7.02E-12
40	1060	5.65E-12
45	1130	7.81E-12
50	1200	3.23E-10

7.5.11.1 Frequency/Amplitude

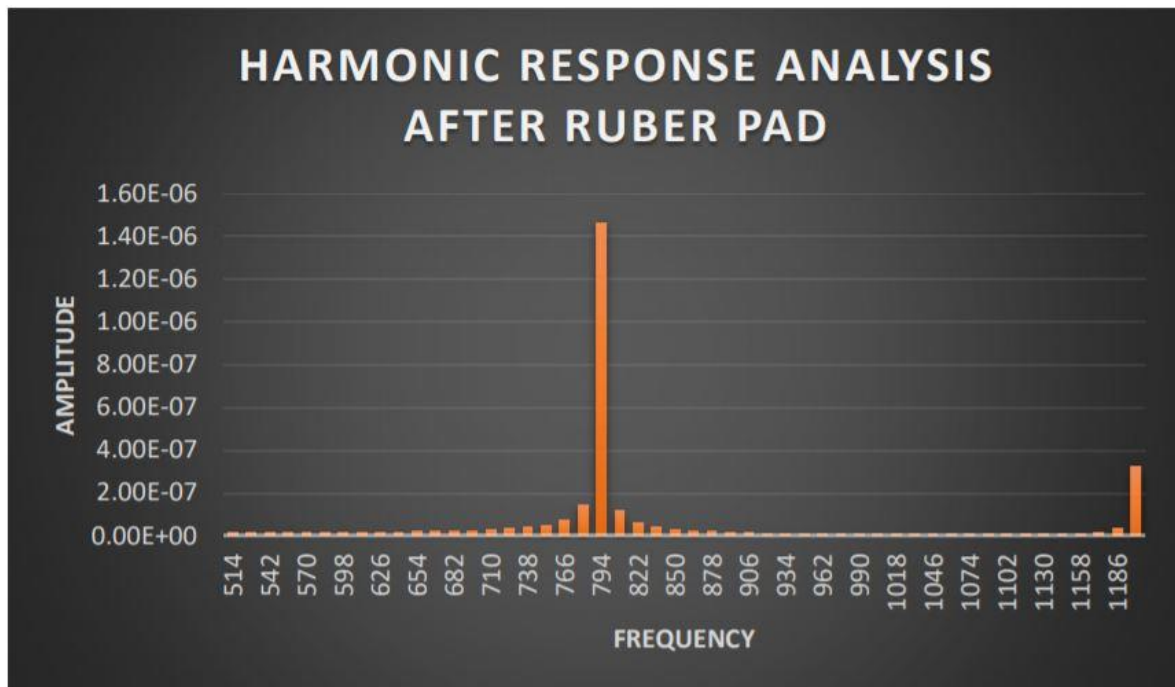


Fig: 7.15: Frequency/Amplitude Bar graph

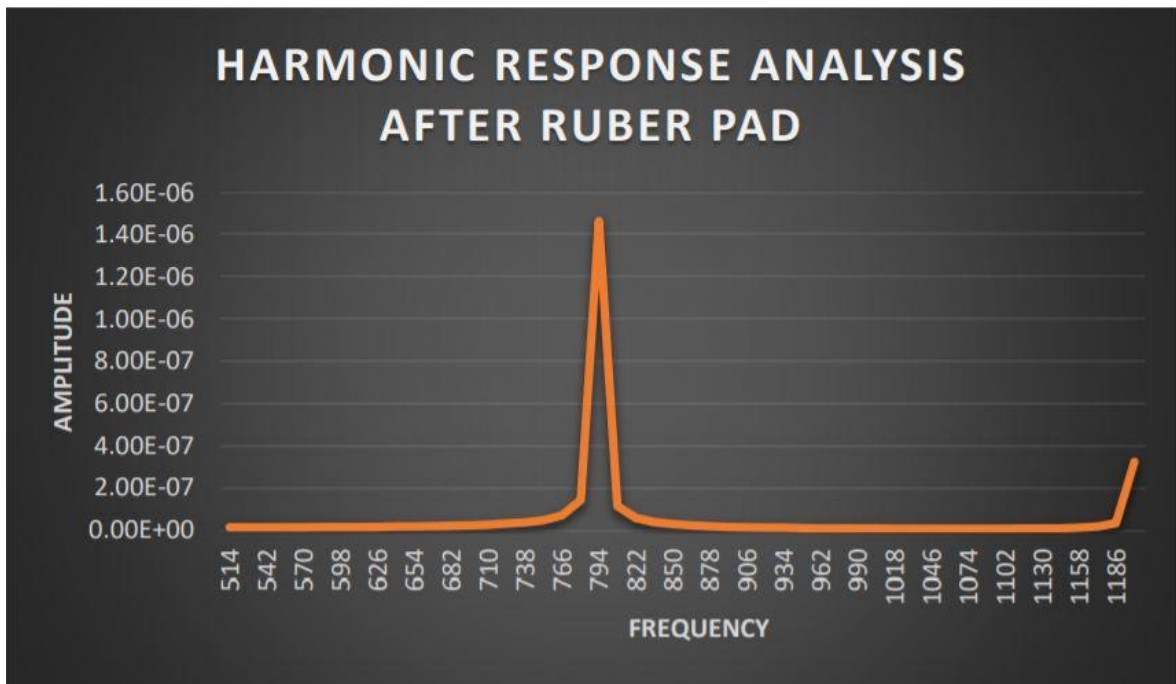


Fig: 7.16: Frequency/Amplitude Line graph

Table: 7.4: Frequency and Phase angle

MODE	FREQUENCY	PHASE ANGLE
1	514	0
5	570	0
10	640	0
15	710	0
20	780	0
30	920	180
35	990	180
40	1060	180
45	1130	180
50	1200	180

7.5.11.2 Frequency/Phase angle

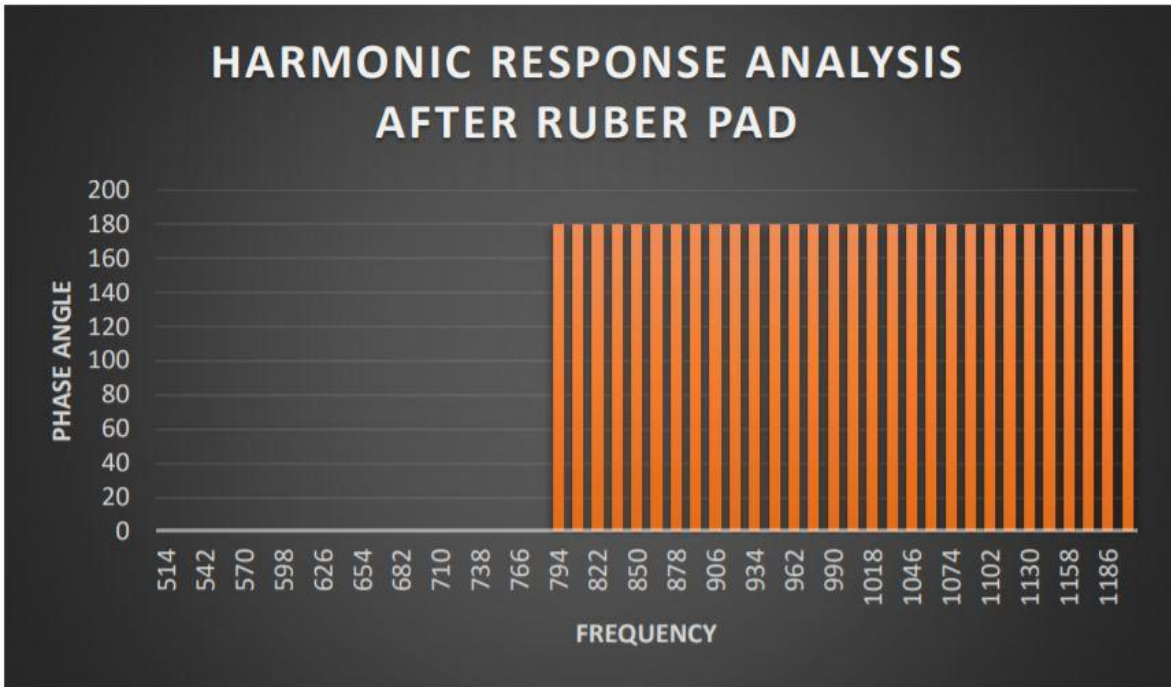


Fig: 7.17: Frequency/Phase angle Bar graph

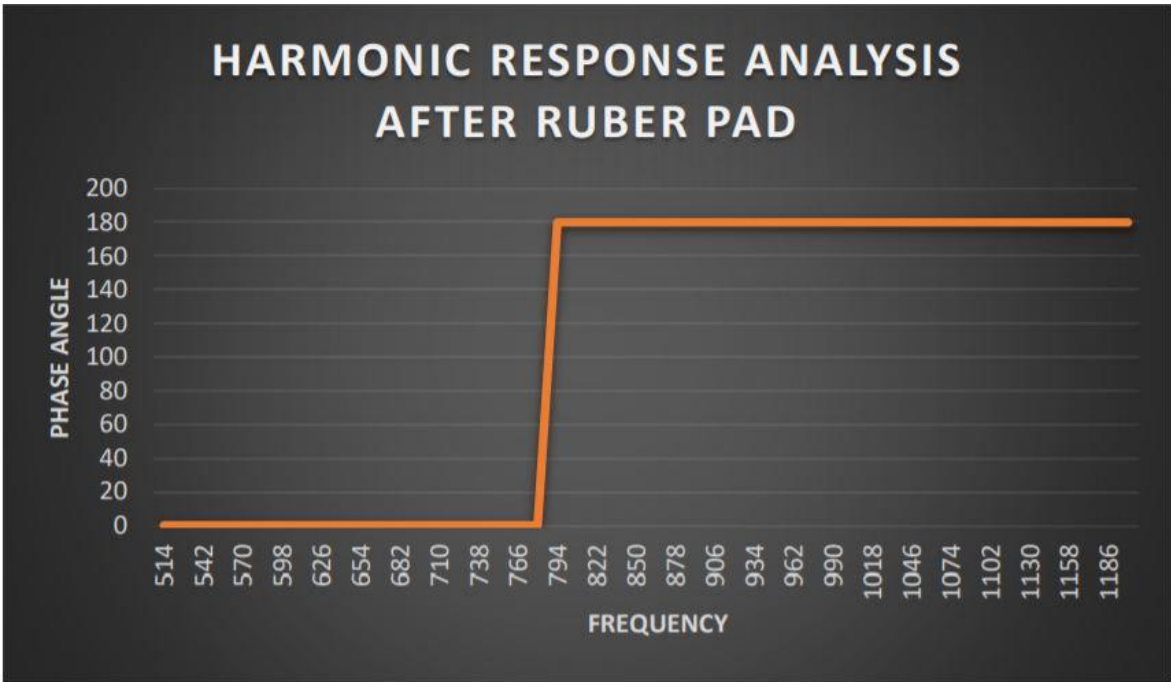


Fig: 7.18: Frequency/Phase angle Line graph

7.6 COMPARING THE RESULTS OF FREQUENCY RESPONSE ANALYSIS BEFORE AND AFTER KEEPING THE RUBBER PAD.

7.6.1 GRAPH

7.6.1.1 FREQUENCY DIFF. GRAPH

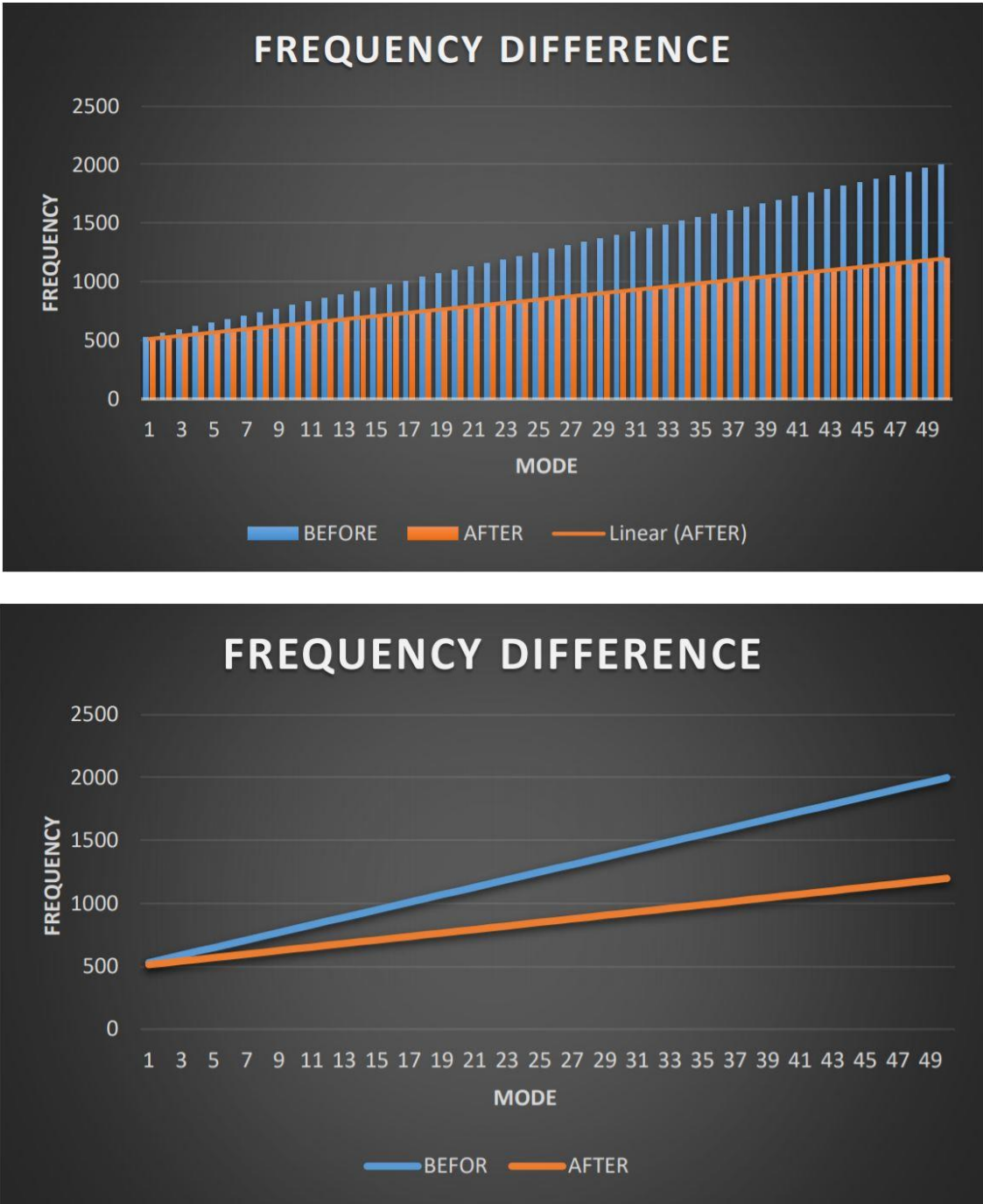


Fig: 7.19: Frequency diff. Bar and Line graph

7.6.1.2 AMPLITUDE DIFF. GRAPH

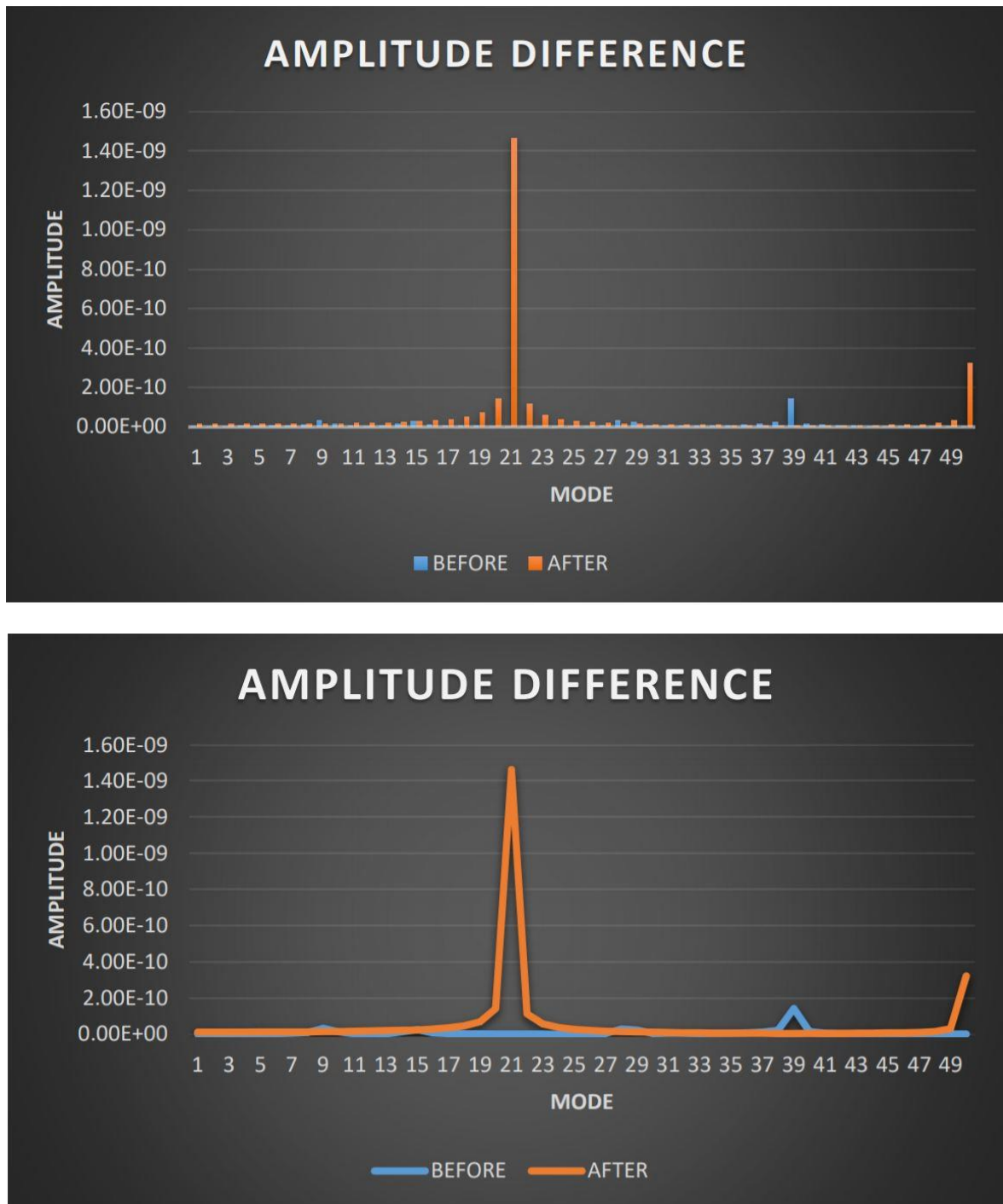


Fig: 7.20: Amplitude diff. Bar and Line graph

7.6.1.3 PHASE ANGLE DIFF. GRAPH

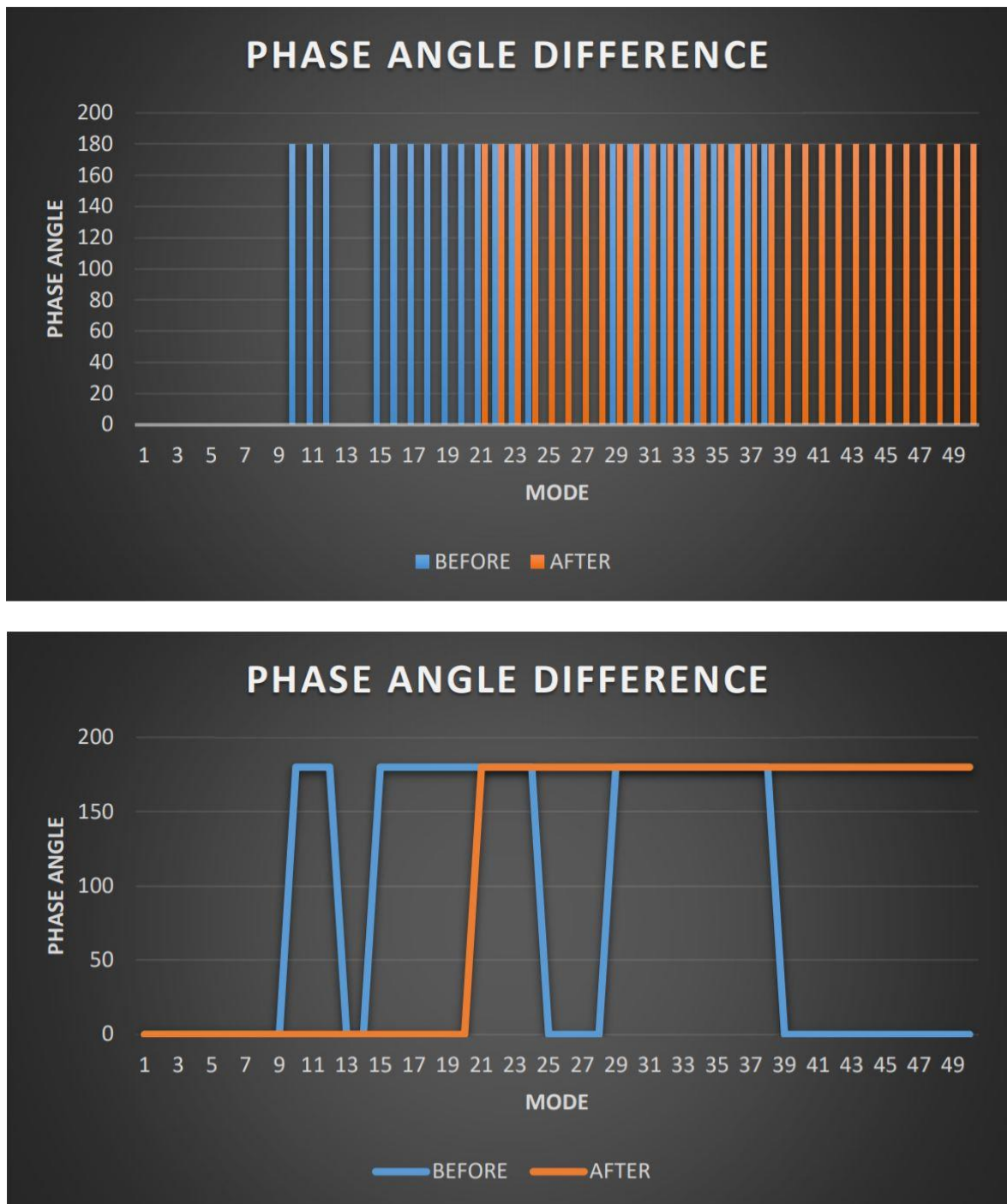


Fig: 7.21: Phase angle diff. Bar and Line graph

CHAPTER 8 MODAL ACOUSTIC ANALYSIS

8.1 INTRODUCTION TO MODAL ACOUSTIC

Random excitations can result from different types of non-deterministic loads such as wind loads, land-weight loads, and other white noise loads. When the structural system is coupled with an enclosed cavity, the structural-acoustic frequency response functions can be obtained using the uncoupled structural modes and the uncoupled acoustic modes, Modal soaking coupled with structural-acoustic coupling involved in the structure. The random response of the coupled structural – acoustic system is then obtained by the sum of structural – acoustic FRFs with auto and cross-spectral random loading applied to the excitation locations.

8.2 PREPROCESSING

In order to carry on the fluid meshing a surface body is designed keeping fluid flow and fluid area in mind. Then the fluid structure is meshed and material is specified as gasoline.

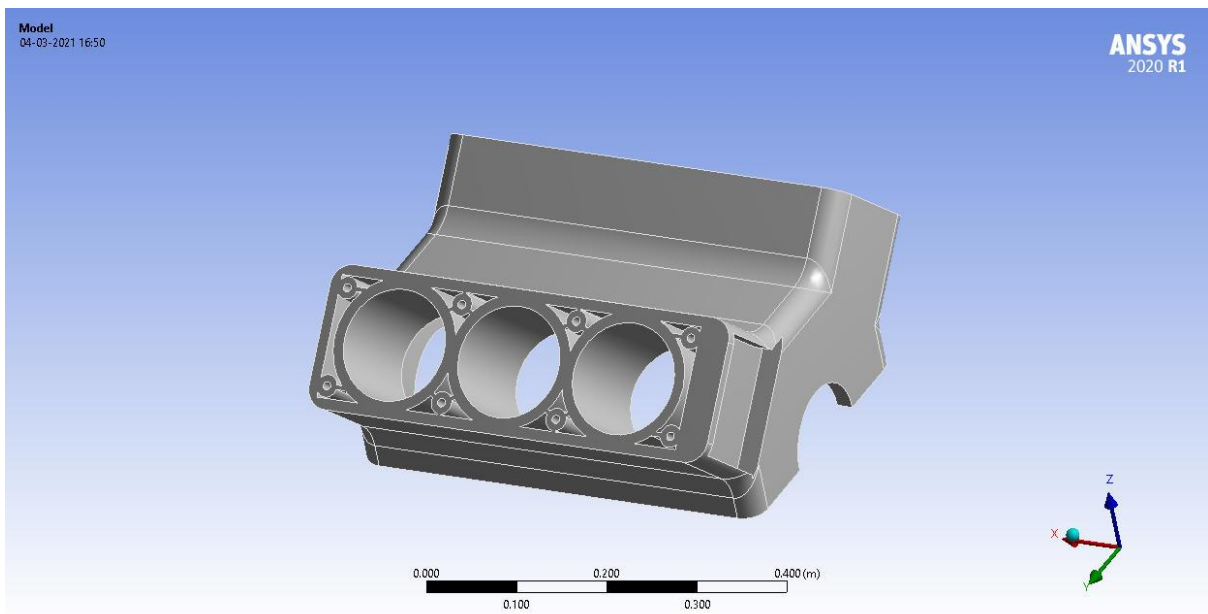


Fig: 8.1: Engine block with fluid structure

8.2.1 Input

Element size: 1.e-002m

Bounding box diagonal: 0.76668 m

Average surface area: 4.8965e-003 m²

Minimum edge length: 1.3888e-003 m

Transition ratio: 0.272

Maximum layers: 5

Nodes: 241968

Elements: 148327

Environment temp: 22°C

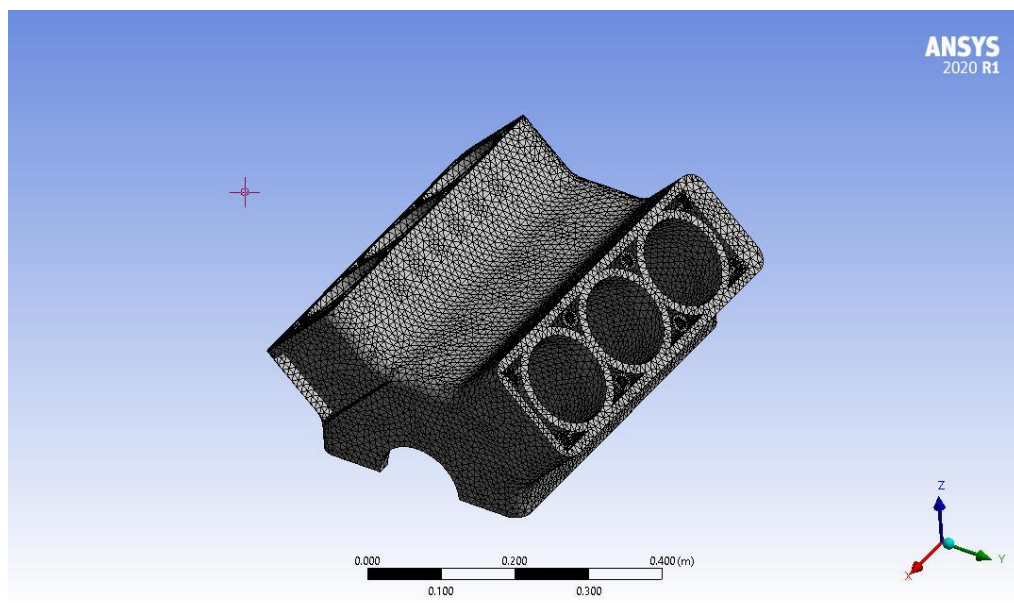


Fig: 8.2: Fluid body Mesh

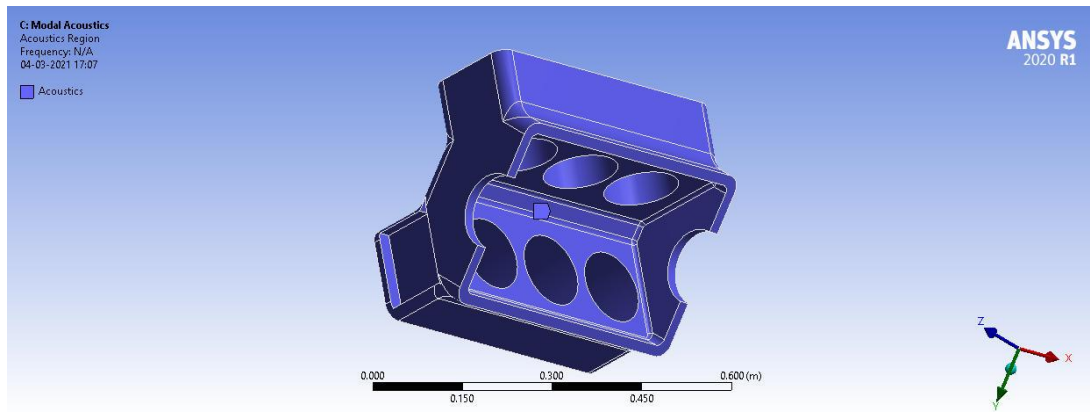


Fig: 8.3: Acoustic Region

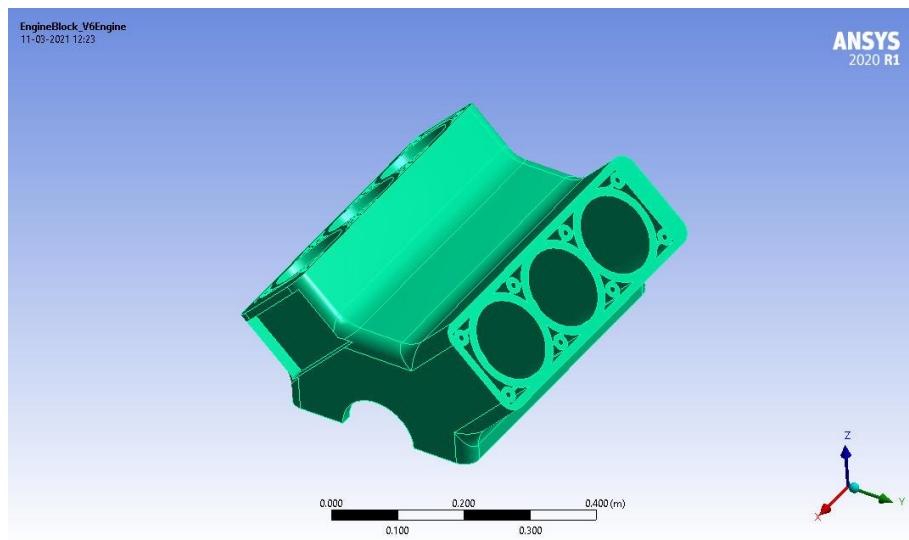


Fig: 8.4: Geometry

8.2.2 Frequency taken as input for analysis

Range minimum: 0.01Hz

Range maximum: 1000000Hz

Reference pressure: 0.00002 Pa

Reference static pressure: 101325 Pa

Fluid behaviour: Compressible

8.2.3 Properties Data

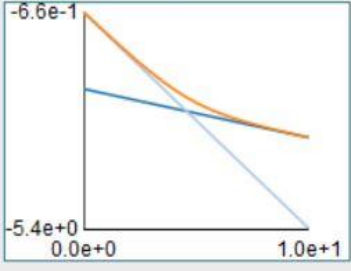
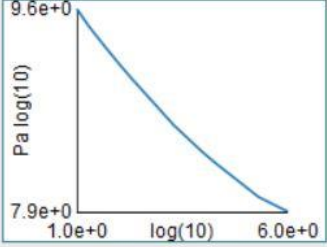
Table:8.1: Properties of air

 Air  	
Density	1.225 kg/m ³
Magnetic 	
Isotropic Relative Permeability	1
Fluid 	
Speed of Sound	346.25 m/s
Other 	
Molecular Weight	28.966 kg/kmol
Lennard Jones Length	3.711 m
Lennard Jones Energy	78.6 J
Thermal Accom Coefficient	0.9137
Velocity Accom Coefficient	0.9137 m/s
Formation Entropy	1.9434e+05 J/°C
Reference Temperature	25 °C
Critical Pressure	3.758e+06 J/m ³
Critical Temperature	-140.85 °C
Acentric Factor	0.033
Critical Volume	0.002857 m ³
Absorption Coefficient	0 1/m

Table: 8.2: Properties of water liquid

Water Liquid	
Density	998.2 kg/m ³
Thermal	
Isotropic Thermal Conductivity	0.6 W/m·°C
Specific Heat Constant Pressure	4182 J/kg·°C
Fluid	
Speed of Sound	1482.1 m/s
Viscosity	0.001003 Pa·s
Other	
Latent Heat	2.2631e+06 J/kg
Vaporization Temperature	10.85 °C
Boiling Point	99.85 °C
Volatile Fraction	1
Binary Diffusivity	3.05e-05
Dpm Surften	0.07194
Vapor Pressure	2658 J/m ³
Molecular Weight	18.015 kg/kmol
Species Phase	1
Formation Enthalpy	-2.8584e+08 J/mol
Reference Temperature	24.85 °C
Lennard Jones Length	1 m

Table: 8.3: Properties of Structural Steel

Structural Steel	
Fatigue Data at zero mean stress comes from 1998 ASME BPV Code, Section 8, Div 2, Table 5-110.1	
Density	7850 kg/m ³
Structural	
Isotropic Elasticity	
Derive from	Young's Modulus and Poisson's Ratio
Young's Modulus	2e+11 Pa
Poisson's Ratio	0.3
Bulk Modulus	1.6667e+11 Pa
Shear Modulus	7.6923e+10 Pa
Isotropic Secant Coefficient of Thermal Expansion	1.2e-05 1/°C
Compressive Ultimate Strength	0 Pa
Compressive Yield Strength	2.5e+08 Pa
Strain-Life Parameters	
S-N Curve	
Tensile Ultimate Strength	4.6e+08 Pa
Tensile Yield Strength	2.5e+08 Pa
Thermal	
Isotropic Thermal Conductivity	60.5 W/m·°C
Specific Heat Constant Pressure	434 J/kg·°C
Electric	
Isotropic Resistivity	1.7e-07 ohm·m
Magnetic	
Isotropic Relative Permeability	10000

8.3 RESULT

The analysis is carried out using “Acoustic Pressure” solver and the results are noted and observed.

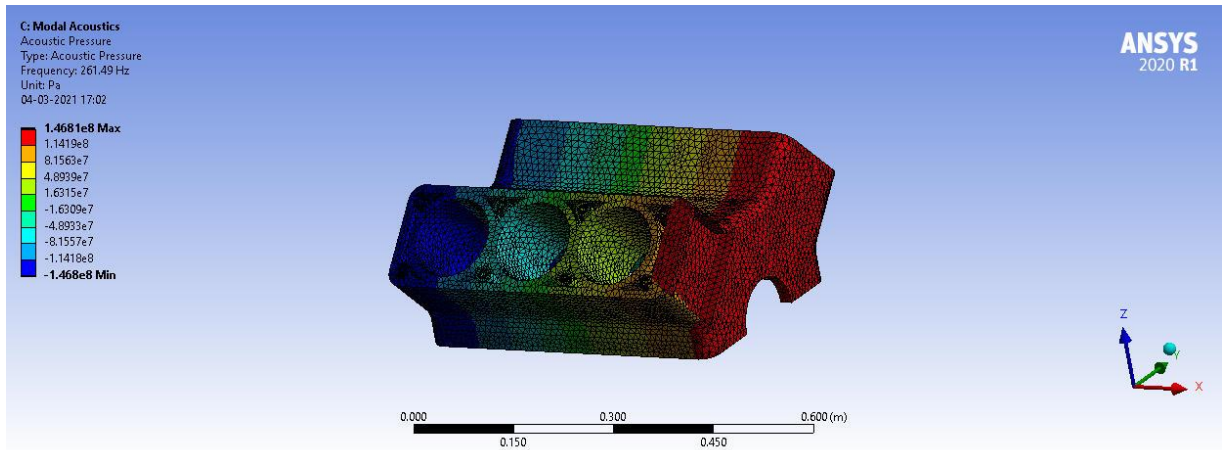


Fig: 8.5: Modal Acoustic is done in engine block at frequency (f)= 261.49Hz, the respective outcome is Max= 1.4681e8, Min= -1.468e8.

8.3.1 Data Table

Table: 8.4: Frequency at modes

Mode	Frequency [Hz]
1.	261.49
2.	267.73
3.	393.52
4.	509.91
5.	534.18
6.	586.77
7.	671.03
8.	683.53
9.	722.98
10.	768.35

8.3.2 Graph

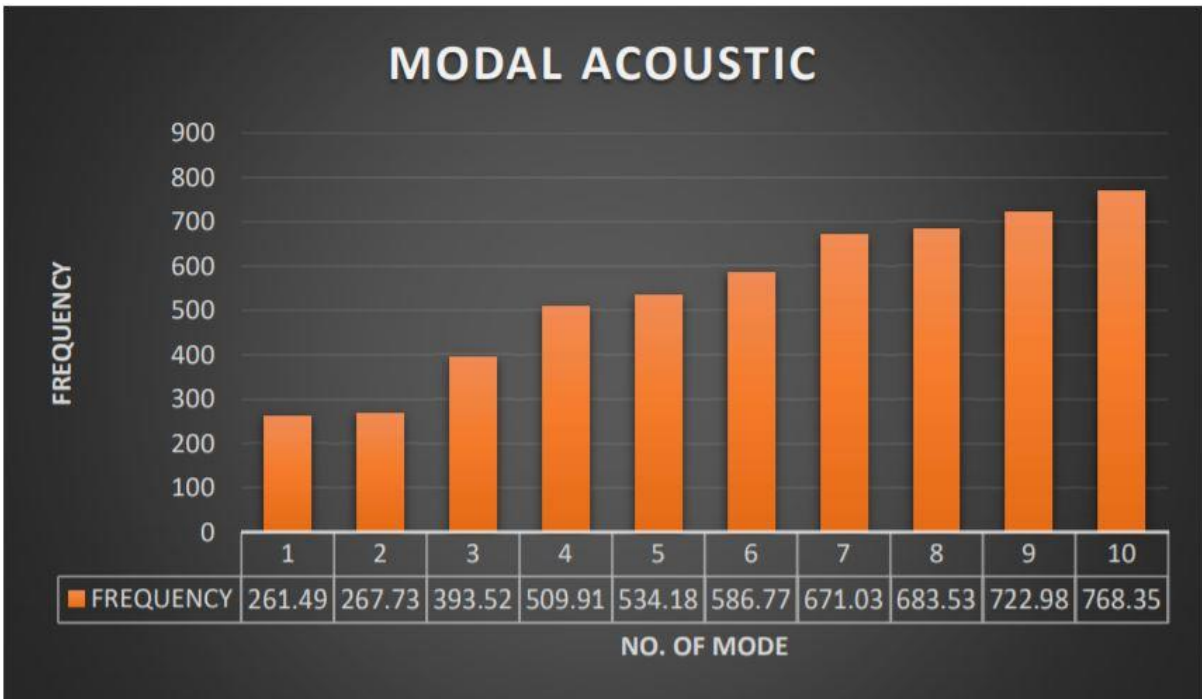


Fig: 8.6: Modal acoustic Bar graph

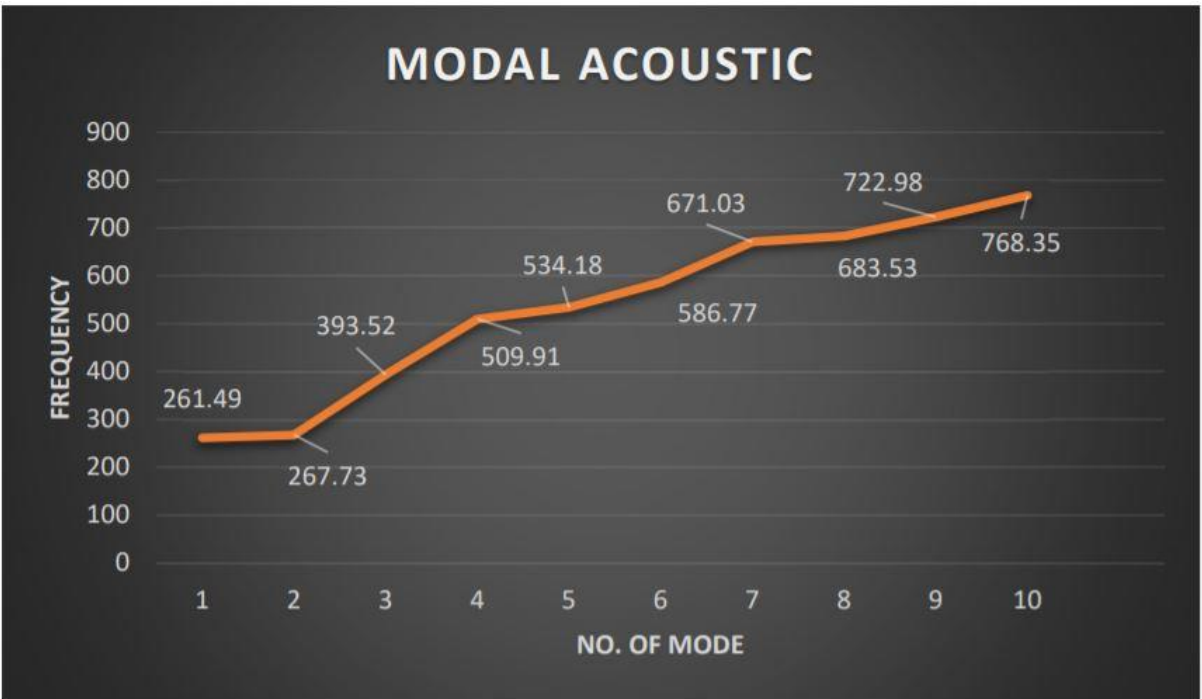


Fig: 8.7: Modal acoustic Line graph

This analysis helped to obtain a noise level vs. frequency graph. Sound proofing can be optimized with these values to increase the ride quality of the passengers.

CHAPTER 9 CONCLUSION

IC engines in current vehicles are subject to high load conditions and are need to work efficiently and accurately, this leads to the need for a strong and efficient design.

With this in mind, we designed the V6 ICE using SolidWorks, again the components were assembled using an assembly workspace. The tool used to verify DFMXpress The components are used in relation to manufacturability. Achieved a result showing us That the actual component complies with standards (ASME) in terms of manufacturability. From this result of SolidWorks, it can be concluded that the design can be constructed.

Designs that follow the rules of the subsequent process are analysed. Commands see behaviour under different circumstances in terms of different parameters. So want to know Behaviour We performed the following analysis.

- MBD
- NMA
- FRA
- MAA

Multi body dynamics is completed so as to understand the reaction forces at various points under variable load conditions. We considered a rpm range of 0-6000 as input for the crankshaft. Post processing is complete and the piston slap force, critical force such as spring Compression forces, contact locations at various points, normal forces, slip velocities at parts like roller. These results helped us to validate the design with respect to reaction forces and dynamics.

One of the main aspect that needed to be concreted is vibration of ICE. To know the behaviour with reference to vibration NMA and FRA are performed. In NMA a constant

load of 1000N is given at excitation point and 10 modes were considered for results.

These results helped us to validate the design under load at the actual nodes. But, the Modes arising in NMA are not tolerably meant to refer to the part that might be

a good with respect to vibration because the frequencies at these 10 default modes are not covered full working frequency of the engine. This gives rise to the need for frequency response analysis. The supported RPM input frequency range is calculated (0-2000Hz).

It has been given with a step size of 50Hz as input at 50Hz. Like the parameters in this analysis deformation element stress, element strains are calculated by ansys. This helps us to validate the design more precisely.

The other important aspect in engines is background level (noise level). To get the noise level with the frequency range modal acoustic analysis is performed. This analysis helped to obtain a noise level vs. frequency graph. Sound proofing can be optimized with these values to increase the ride quality of the passengers.

We now added the innovation to our design. This part is an example of sandwich method. We used two rubber stacked together between two gray cast iron. This is then placed above and below of engine block. After the innovation we repeat the analysis done before to highlight the difference of frequency in both the charts.

As you can see that after the innovation the frequency produced has been reduced. This helps in comfort of the user as the noise and vibration has been reduced which in turn increases ride quality and Engine efficiency.

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