International Journal for Ignited Minds (IJIMIINDS)

Damping Assessment for Crankshaft Design to Reduce the High Vibrations

Darshak T R[a], Shivappa H A[b] & Preethi K[c]

[a]*PG Student, Dept of Mechanical Engineering, Dr. AIT, Bangalore, India.*
[b]*Asst. Prof, Dept of Mechanical Engineering, Dr. AIT, Bangalore, India.*
[b]*Asst. Prof, Dept of Mechanical Engineering, Dr. AIT, Bangalore, India.*

ABSTRACT

This research focuses on the identification and quantification of dynamic damping characteristics for the actual 4-stroke diesel engine with the aim of predicting and assessing the dynamic damping using finite element analysis. The torsional and bending vibration output from finite element analysis results are correlated with the results obtained from the testing and empirical relations. Every system containing individual mass and stiffness distribution is susceptible to vibrate. These vibrations can cause either by single impulse of load or a periodical load. In the first case a free vibration occurs and in the second case a forced vibration is achieved. A free vibration is not significance in technical applications because there is always a periodical load, which causes a force vibration. However, free vibration analysis is very important in determining the natural frequencies of the system. The outcome of the research will be a more realistic in achieving the highly accurate results which will be tested and correlated with the experimental results.

Keywords *– Crankshaft, Vibration, Harmonic, Modal, FEA.*

1. INTRODUCTION

Damping has an effect on restricting or eliminating oscillations. Normally damping is caused by processes which dissipate the energy stored in oscillation. Each and every time if the cylinders fire torque is taken to the crankshaft. The crankshaft deflects under this torque which causes the vibrations when torque is released. At certain value of engine speeds the torque imparted to cylinder is in sync with the vibrations in crankshaft, which in turn results in phenomenon called resonance. This resonance causes stress beyond what the crankshaft can accommodate resulting in crankshaft failure.

Many works and also research have been done based on the torsional vibrations problem of the reciprocating internal combustion engine from as early as 20th century. Therefore damping characteristics of the crankshaft vibration problems are not as clear as required by design, field or system designers because of complexity and the ambiguity of the vibration damping phenomenon. So this research concentrates on identifying and determining the dynamic damping characteristics of a single cylinder four-stroke diesel engine with the focus on predicting and inspecting the dynamic damping.

Each system has an individual mass and the stiffness distribution is subjected to vibration. These vibrations are caused either by the single impulse of the load or a periodical load. In the first case free vibration is generated and forced vibration in the second case. A free vibration doesn’t have any significance in the technical applications because of the periodical type of load, which causes force vibration. However, the free vibration analysis is very important in determining natural frequencies of system.

Torsional vibration is one of most prominent complex problem in the crack shaft design. Hence the resultant moment of inertia is increased due to the big mass of the piston assembly and the connecting rod of the crank mechanism thereby decreasing the natural frequency of the first and the second mode to range rotational speed of the engine. It results in torsional resonance along with low order harmonic of the tangential force which is acting on the crankpin, which is undesirable in connection with the occurring of significant additional torsional moment.

The damping caused during torsional vibration is occurred by the damping from externally caused piston friction. Other means of the damping are the combined friction of the crankshaft material and the viscous friction into the crankshaft bearings but their effect is negligible and hence not added in the recent study. Ansys is preferred for carrying out the dynamic analysis.

Half power point method based on the experimental modal damping of the modes from the measured response curve which is being validated with respect to the damping calculated from analysis. The study is being made which establishes reasonable dynamic damping model which helps in design the efficient and the optimum crankshaft design for the loading of torsional vibration. The outcome of the research is being more actual in acquiring highly accurate results.

2. DAMPING

4.1 Damping

The process by which amplitude in vibration steadily diminishes is called damping. Damping is an energy–dissipating mechanism. Damping is an influence within or upon an oscillatory system that has the effect of reducing, restricting or preventing its oscillations.

The damping of a system can be described as being one of the following:

- Overdamped: The system returns (exponentially decays) to equilibrium without oscillating.
- Critically damped: The system returns to equilibrium as quickly as possible without oscillating.
- Underdamped: The system oscillates (at reduced frequency compared to the undamped case) with the amplitude gradually decreasing to zero.
- Undamped: The system oscillates at its natural resonant frequency (ωo).

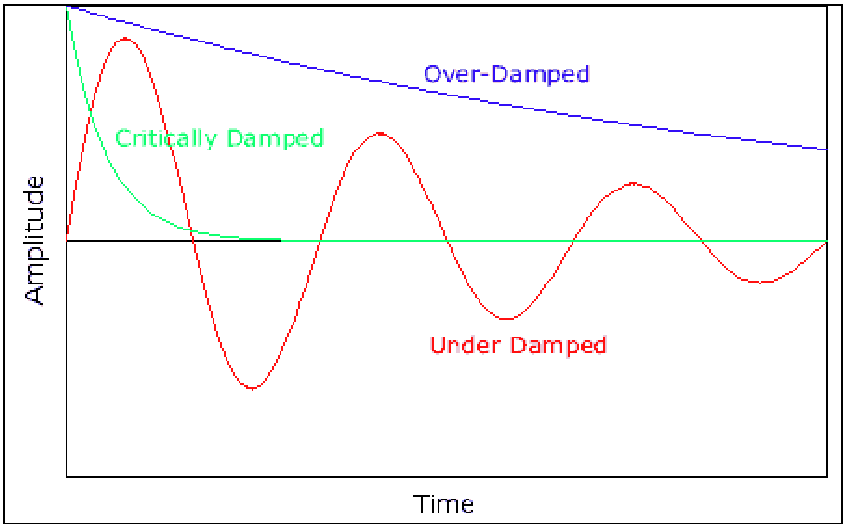

Fig 1: Amplitude v/s Time Plot Showing Various Forms of Damping.

The damping ratio is a parameter, usually denoted by ζ (zeta) that characterizes the frequency response of a second order ordinary differential equation. The damping ratio provides a mathematical means of expressing the level of damping in a system relative to critical damping.

Mathematical representation:

Damping ratio (ζ)

$\zeta = C/Ccr$

Where,

C is the damping coefficient

$Ccr = 2m\omega n$ is the critical damping coefficient

The damping ratio is presented as a fraction of critical damping or percentage of critical damping.

Based on damping a system is classified as

- Underdamped if $\zeta < 1$
- Critically damped if $\zeta = 1$
- Overdamped if $\zeta > 1$

Quality factor / Amplification factor (Q)

$Q = 1/2\zeta$

Loss factor (η)

$\eta = 1/Q$

4.2 Damping Estimation (Half – Power Bandwidth Method)

Half – Power bandwidth method is one of the methods of estimating damping. This research adopts the same method to determine the damping of the crankshaft. Damping in mechanical systems may be represented in numerous formats. The most common forms are Q and ζ where,

Q is the amplification or quality factor

ζ is the viscous damping ratio or fraction of critical damping.

These two variables are related by the formula

$$Q = 1/2\zeta \quad (1)$$

An amplification factor of Q =10 is thus equivalent to 5% damping.

The Q value is equal to the peak transfer function magnitude for a single-degree-of freedom subjected to base excitation at its natural frequency. This simple equivalency does not necessarily apply if the system is a multi-degree-of-freedom system, however. Another damping parameter is the frequency width Δf between the -3 dB points on the transfer magnitude curve. The conversion formula is

$$Q = f_n/\Delta f \quad (2)$$

Where f_n is the natural frequency.

The -3 dB points are also referred to as the half power points on the transfer magnitude curve. Equation (2) is useful for determining the Q values for a multi-degree-of-freedom system as long as the modal frequencies are well separated.

3. GEOMETRY AND FE MODELING

3.1 Geometry

In this section the geometric creation, computer aided modeling and assembly of the intermediate crankshaft and piston of a four stroke diesel engine are discussed. The model consists of a crankshaft and it is created as per drawing shown in the fig 2 using CATIA V5R20. It is then assembled with the piston model created as per the given dimensions and the final assembled model is shown in fig 3.

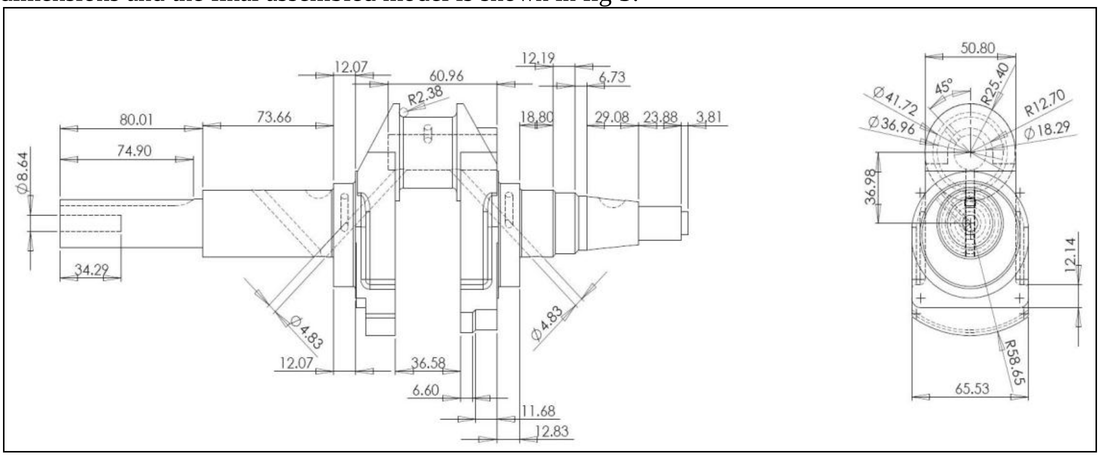

Fig 2: Standard Dimensions of the Crankshaft.

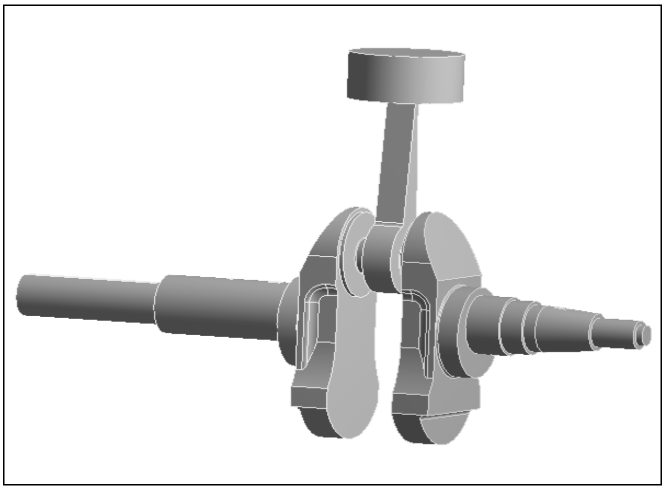

Fig 3: Model of the Crankshaft as per Standard Dimensions.

6.2 Meshing

In limited component examination essential idea is just to break down the structure, which is a gathering of distinct pieces commonly called components that are associated collectively at the limited number of focuses called as Nodes. Stacking limit conditions, which are then connected to components and hubs. Hence system of these components is defined as Mesh. For volume meshing, a tetrahedral mesh generally provides a more automatic solution with the ability to add mesh controls to improve the accuracy in critical regions. The assembled crankshaft model is built using tetrahedral mesh. Complete model is meshed with total no of 305289 elements and 678432 no of nodes.

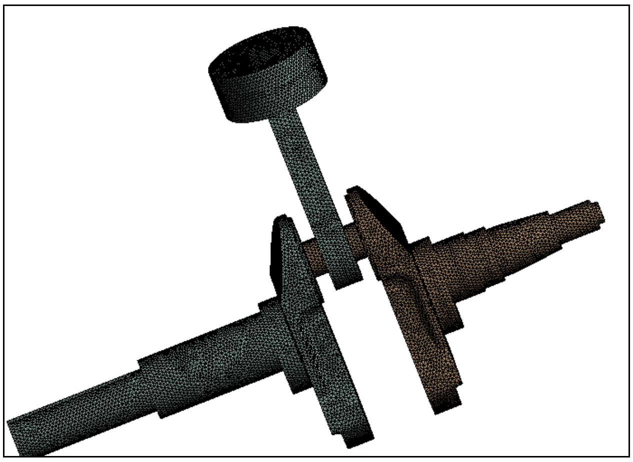

Fig 4: Tetrahedral Mesh for the Model.

6.3 Boundary Condition

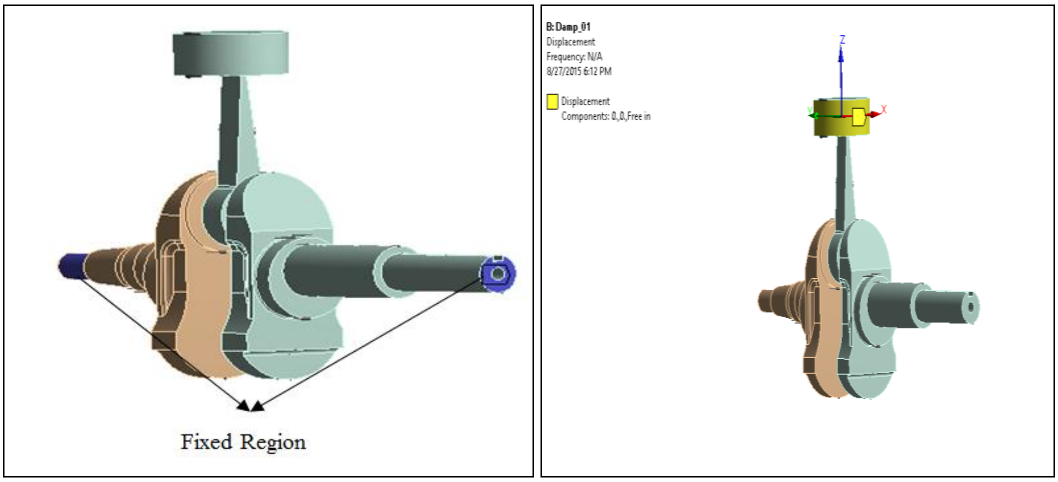

Fig 5: Plot Showing Boundary Condition used in the Analysis.

4. RESULTS AND DISCUSSIONS

4.1 Modal analysis

Modal analysis was performed on the crankshaft model and various modal shapes and respective frequencies were determined.

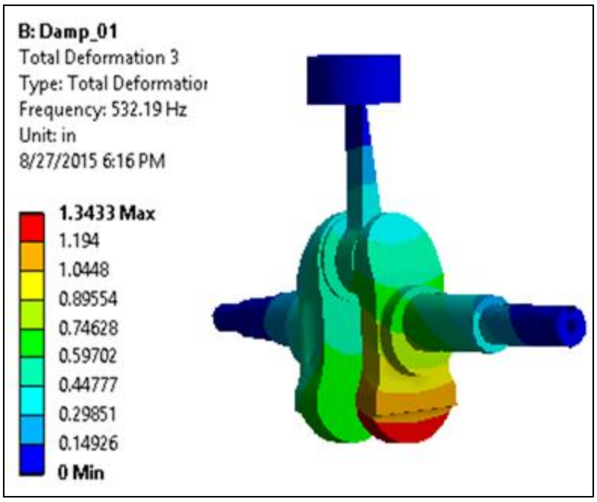

Fig 6: Mode shape for first mode.

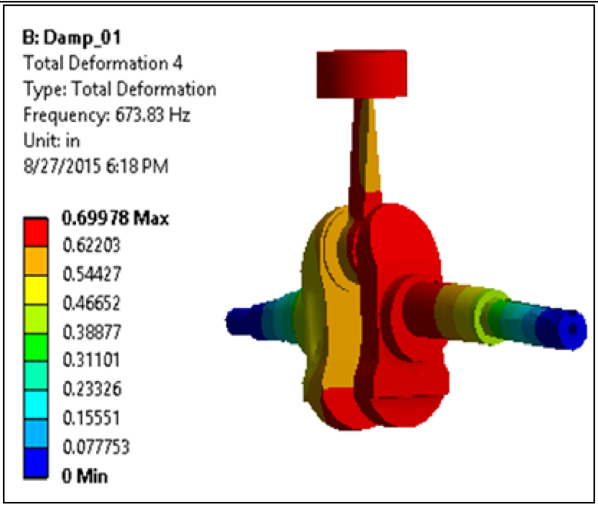

Fig 7: Mode shape for second mode.

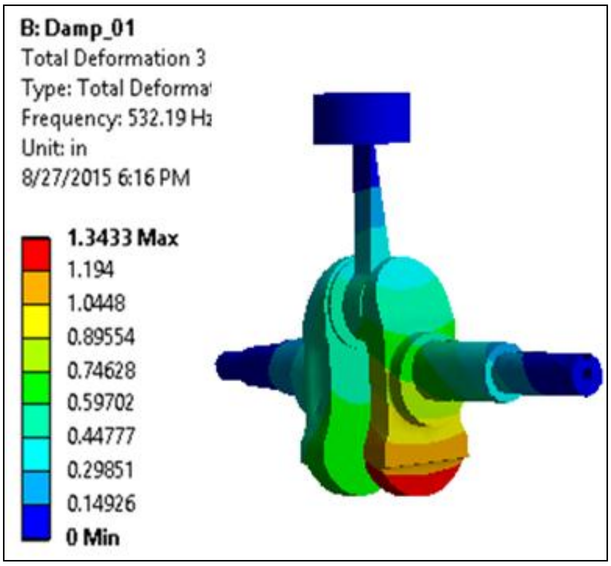

Fig 8: Mode shape for third mode.

4.2 Harmonic Analysis

Based on the various modal frequencies obtained from the modal analysis, harmonic analysis is performed.

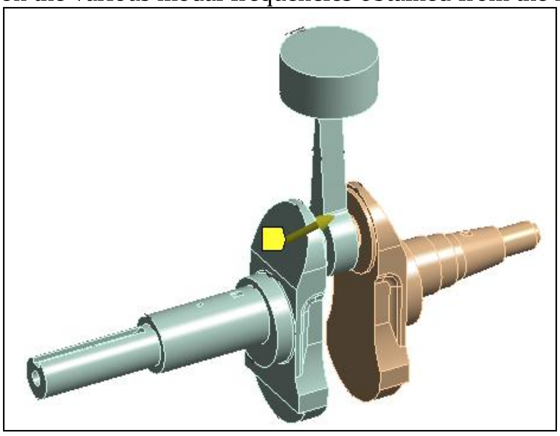
Fig 9: Acceleration applied in harmonic analysis.

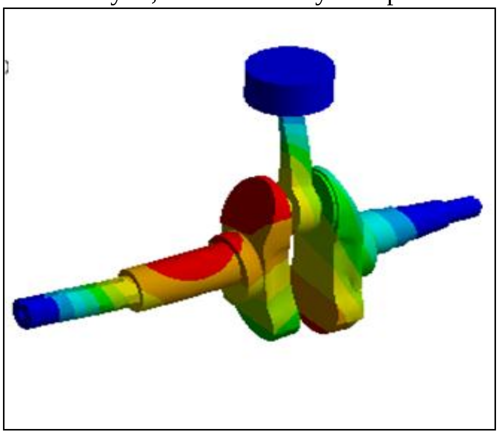
Fig 10: Deformation Plot.

Acceleration load of 5G is applied as shown in the above Fig 9.

$=5 \times 9.81\ m/s^2$

$= 49.05\ m/s^2$

Since the deformations along Y and Z axes are negligible, they are not considered for the analysis. Once the directional deformation is obtained damping ratios are applied in the range of 1–4% and corresponding frequency responses are determined.

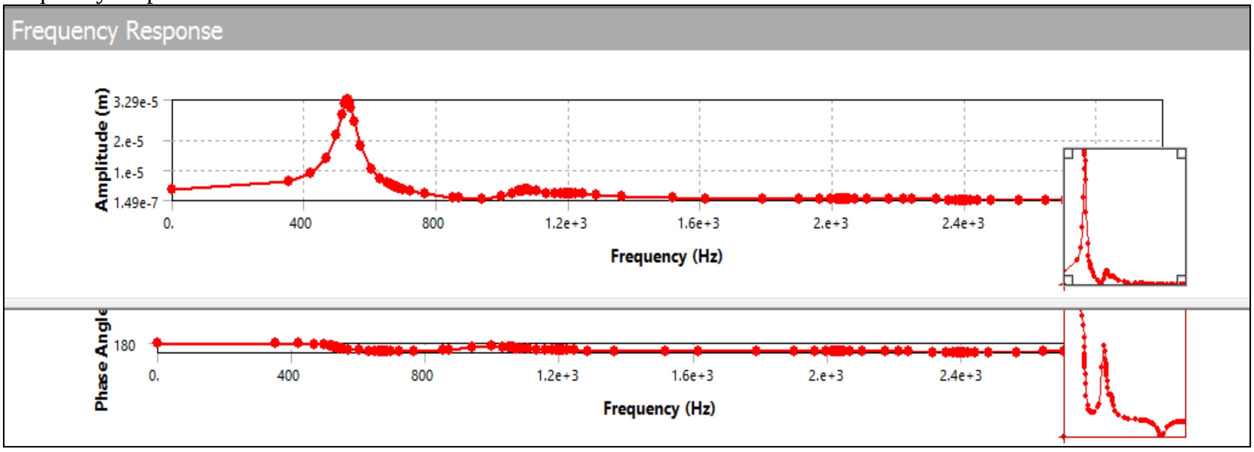

Fig 11: Frequency response for damping ratio of 1%.

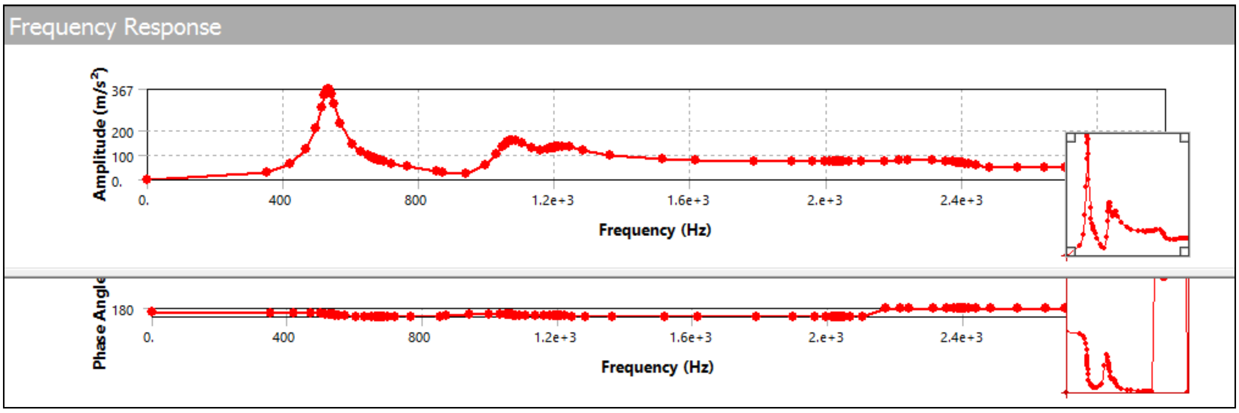

Fig 12: Frequency response for damping ratio of 2%.

Similarly, analysis is conducted for 3 and 4% Damping ratios.

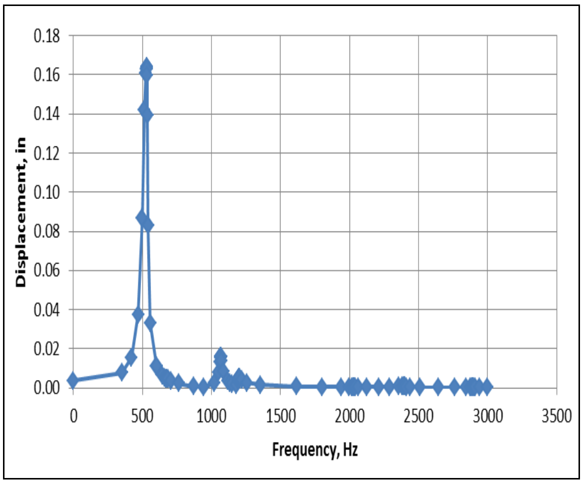

Fig 13: Displacement v/s Frequency (1% damping).

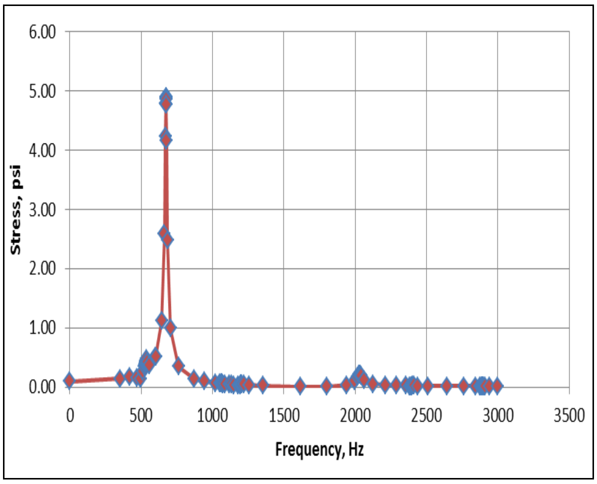

Fig 14: Stress v/s Frequency (1% damping).

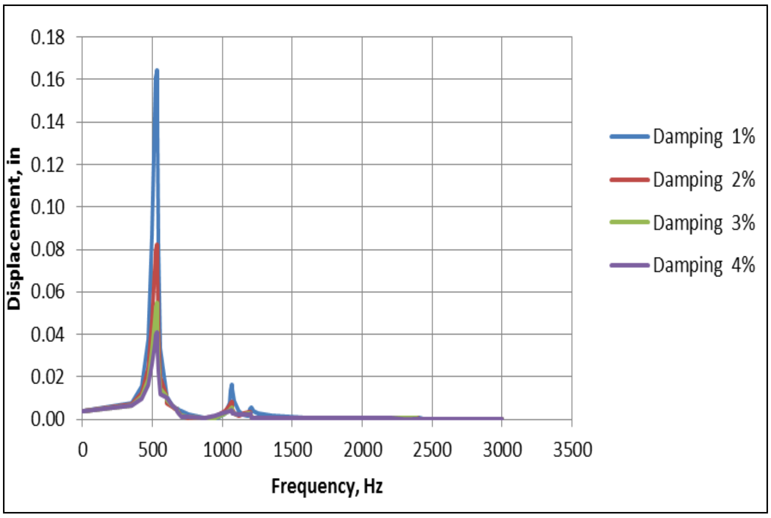

Fig 15: Displacement v/s Frequency for all Damping Ratios.

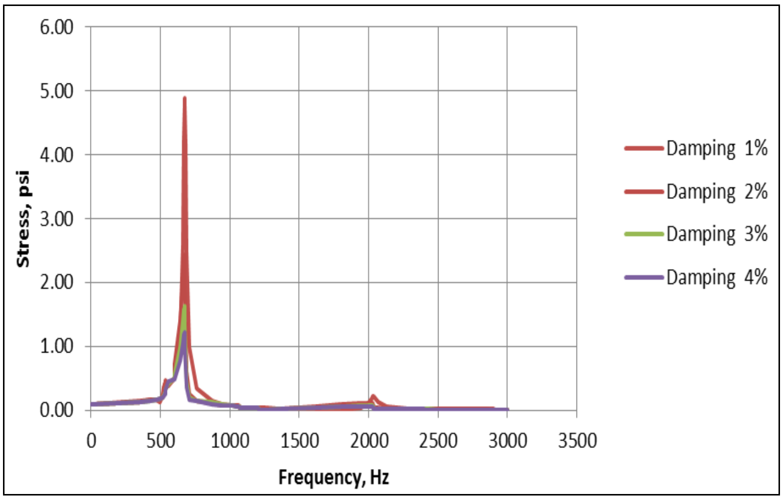

Fig 16: Stress v/s Frequency for all Damping Ratios.

According to the *half power bandwidth method* for damping calculation,

$$Q = \frac{1}{2\zeta} \tag{3}$$

The conversion formula is

$$Q = \frac{f_n}{\Delta f} \tag{4}$$

Where, Q is the amplification factor, f_n is the natural frequency
Δf is the frequency width in the graph.
From the analysis we can see that the minimum and maximum frequency is 510 Hz and 560 Hz respectively for damping ration 1% load case. The mean frequency is 535 Hz (natural frequency).
$\Delta f = 560 - 510 = 50$ Hz
From (3) and (4)

$$\frac{1}{2\zeta} = \frac{f_n}{\Delta f}$$

$$\zeta = \frac{\Delta f}{2 * f_n} = \frac{560 - 510}{2(535)}$$

$= 0.04673 = 4.673\%$

Similarly, for other damping ratio calculations are shown below.

Damping ration 2%,

$$\zeta = \frac{\Delta f}{2*f_n} = \frac{550-520}{2(535)}$$

= 0.02804 = 2.804%

Damping ration 3%,

$$\zeta = \frac{\Delta f}{2*f_n} = \frac{545-525}{2(535)}$$

= 0.01869 = 1.869%

Damping ration 4%,

$$\zeta = \frac{\Delta f}{2*f_n} = \frac{545-525}{2(535)}$$

= 0.01869 = 1.869%

The mean of all the four ζ values is given by,

$$= \frac{0.04673 + 0.02804 + 0.01869 + 0.01869}{4}$$

= 0.03271 = 3.271%

From the half power bandwidth calculation we can conclude that the crankshaft requires 3.271% damping. This level of damping to the crankshaft will protect it from torsional vibrations and it will not fail. So we can say that the crankshaft of a four stroke single cylinder diesel engine requires a damping of 3.271%.

5. CONCLUSION

Analysis of the crankshaft of a four stroke diesel engine made of forged steel resulted in different displacement, acceleration and stress values for different damping ratios. To determine which viscous damping ratio or fraction of critical damping is ideal for the crankshaft, the half power bandwidth method was applied. Based on this method, graphs were plotted and calculations were made which resulted in the damping ratio of 3.217%.

This value can be used a standard for further manufacturing of forged steel crankshaft of a four stroke diesel engine.

Although one can fine tune this percentage by considering the other sources of damping such as integral friction of crankshaft material and viscous friction into crankshaft bearings (their influence are negligible and hence was not included in the present study).

REFERENCES

1. Farzin H. Montazersadgh and Ali Fatemi, "*Dynamic Load and Stress Analysis of a Crankshaft*", SAE International, 2007.
2. Jian Meng and Yongqi Liu, Ruixiang Liu, "*Finite Element Analysis of 4-Cylinder Diesel Crankshaft*", MECS, Aug-2011.
3. Momin Muhammad Zia Muhammad Idris, "*Crankshaft Strength Analysis Using Finite Element Method*", IJERA, Vol-03, Issue-01, Jan-Feb 2013.
4. Jaimin Brahmbhatt and Prof. Abhishek choubey, "*Design and Analysis of Crankshaft for Single Cylinder 4-Stroke Diesel Engine*", *IJAERS, Vol*-01, Issue-IV, July-Sep 2012.
5. Bhumesh J. Bagde and Laukik P. Raut, "*Finite Element Analysis of Single Cylinder Engine Crank Shaft*", IJAET, May-2001.
6. B.Varun, "*Stress Analysis and Optimization of Crankshafts Subject to Static Loading*", IJECS, Vol-03, ISSUe-02, May-2014.
7. R. J. Deshbhratar and Y. R. Suple, "*Analysis & Optimization of Crankshaft Using Fem*", IJMER, Vol-02, Issue-05, Sep-Oct, 2012.