Harmonic Analysis Of Engine Crankshaft As A Fixed Beam

R. B. Sharma[1], Vikas Sharma[2] & Yogendra S. Rajput[2]

Asst. Prof. & Head, Mech. Engg., Dept., RustamJi Institute of Technology, Tekanpur, Gwalior, Madhya Pradesh, India[1]

Research Scholar, M.Tech. Automobile Engineering, RustamJi Institute of Technology, BSF Academy, Tekanpur, Gwalior, Madhya Pradesh, India[2]

Abstract

This study examines the behaviour of the crankshaft under the complex load functions i.e., the real and the imaginary loads of a force vector. Crankshaft being a very crucial part of the engine for providing the adequate brake power by sustaining all the pressure and various loads of the product of combustion. Here in this paper, the harmonic behaviour of the single cylinder SI engine crankshaft is investigated under the dynamic force of piston transferred to crankshaft through connecting rod and gudgeon pin. The results has been shown in the term of graphs between the amplitude and the frequency of the various quantities like displacement and stress levels using finite element method in ANSYS WB platform.

Keywords: complex load functions, harmonic behaviour, dynamic loads, amplitude, frequency, finite element method.

I. Introduction

In current times, the rotor bearing systems utilized for modelling rotating machinery and their supporting structures are of very high importance, as excessive vibration could harm the mounting points or may produce high were and tear thereby producing sound and wearing out of material. The bearing material is generally of very soft type, like Babbitt, chrome steel or in conjunction with copper bushes. The rotor bearing is a very important system to absorb thrust and vibrations during continuous run for long operations. The various types of rotor bearing system are exampled as, electric motors, turbo machinery, transmission shafts, propellers, etc. Now, out these types of rotor bearing, crankshaft engaged with the bearing is one of the important types. In such type of lower pair the one part has to be remaining fix as the bearing where as the crankshaft turns inside it. The speed of the crankshaft depends on the throttling opening and the amount of the charge burned inside the chamber. For a general class single cylinder SI engine, it varies from 0 rpm to 10,000 rpm respectively.

When running at such large speeds, severe vibrations are observed which could be very severe for the engine performance. Hence to predict the behaviour of such class of vibration effects majorly are commonly analyzed by the finite element method. The forces applied by the pistons are periodic as they repeats after every 2nd revolution or we can say after 720 revolution of crankshaft. So, if we consider a motion of the type

$$X1 = A1\, sin\omega t,$$

Here, ω is the natural frequency and the motion will be repeated after 2π/ω time, and *X1* & *A1* be the displacement. The harmonic motion is represented in terms of circular sine and cosine functions.

The velocity and the acceleration are $x`_1 = \frac{dx1}{dt}$ =A1 ω cos ωt and $x_1`` = -\omega^2 x_1$. Thus, the acceleration in a simple harmonic motion is always proportional to its displacement and directed towards a fixed particular point. Computations of harmonic amplitude and frequencies for deformations and stresses, responses play important roles in the design, identification, diagnosis, and control of crankshaft systems. Thus, an accurate prediction for the dynamic characteristics of a rotor bearing system using FEM is essential for modern equipment.

II. Literature Review

The crankshaft is very important in high-power drilling pump. The information on the vibration mode and frequency response is critical for the crankshaft to work normally in drilling pump. ANSYS finite element analysis software was used to implement the modal analysis of the crankshaft thus obtaining characteristic frequency and the corresponding vibration mode and then by the harmonic response analysis which was carried out based on the modal analysis, dynamic response of the crankshaft was obtained. The result shows that the working speed of crankshaft can avoid the region of sympathetic vibration effectively, no resonance will happen. The strength and fatigue life of crankshaft are also checked according to the stress results. The analysis also makes a base for the optimized design of dynamic performance of the crankshaft [1].

The development and application of a technique for the steady-state and transient analyses of diesel engine crankshaft torsional vibrations is presented in this paper. Crankshafts in emergency diesel generators undergo torsional vibrations due to the effect of cylinder firing pressure and the inertia of the reciprocating parts. A diesel engine crankshaft is subjected to steady-state loads during normal engine operation (constant speed and constant load) and to transients during start-ups, coast downs, and load changes. Often the transients may result in torsional stresses that far exceed those normally designed for at constant speed and constant load [2]. Crankshafts developed wide applications even before methods to design them got evolved. Initially, they were assumed to be made up of beam segments. Since, the length to width ratios of the segments are of the order of one, these methods were not effective. Therefore, till recently they were designed based mostly on in-house experience and by using empirical formulae. Of late, with the advent of powerful computers, analysis based on finite element method has come into use for crankshaft design. In this work, we generate, from fundamentals, a systematic procedure to design crankshafts for finite life [3].

III. Methodology

For the present work on the crankshaft, first of all the crankshaft has to be modelled in the design modeller tool of the Ansys Workbench. The stiffness of the material of the crankshaft is considered as flexible. This crankshaft modelled here is of single throw type also. The maximum permissible pressure depends on gas pressure, journal velocity, amount and method of lubrication. Most crankshaft failures are caused by progressive fracture due to repeated loading. Since the failure of the crankshaft is likely to cause a serious engine destruction. Hence, high priority analyses of all types are used. Considering this crankshaft is developed after making an allowance for all the parameters of centre type crankshaft with applied load of pressure of 12MPa which acts during the whole cycle operation known as the mean effective pressure during the cycle. Figure 1 represents the geometric model of crankshaft of single cylinder SI Engine.

Later on the model is meshed as shown in figure 2 using the solid element mesh of tetrahedron type with physical preference of Mechanical and relevance of 29. The smoothing is kept medium and transition fast with coarse span angle centre and minimum edge length of 2.4770mm respectively.

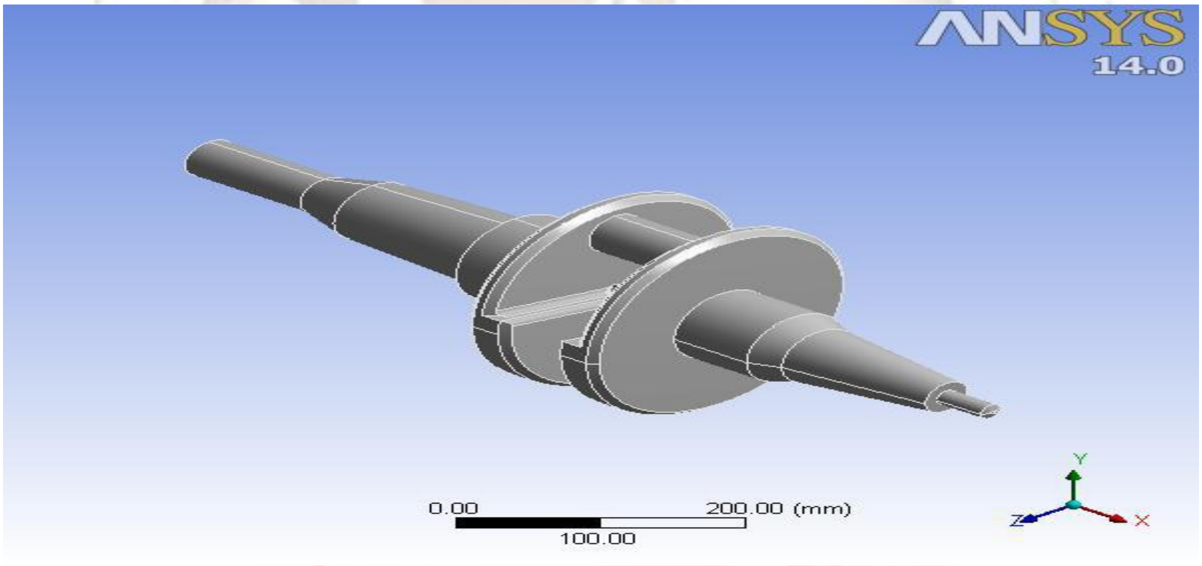

Figure 1, Geometric model of crankshaft of single cylinder SI Engine

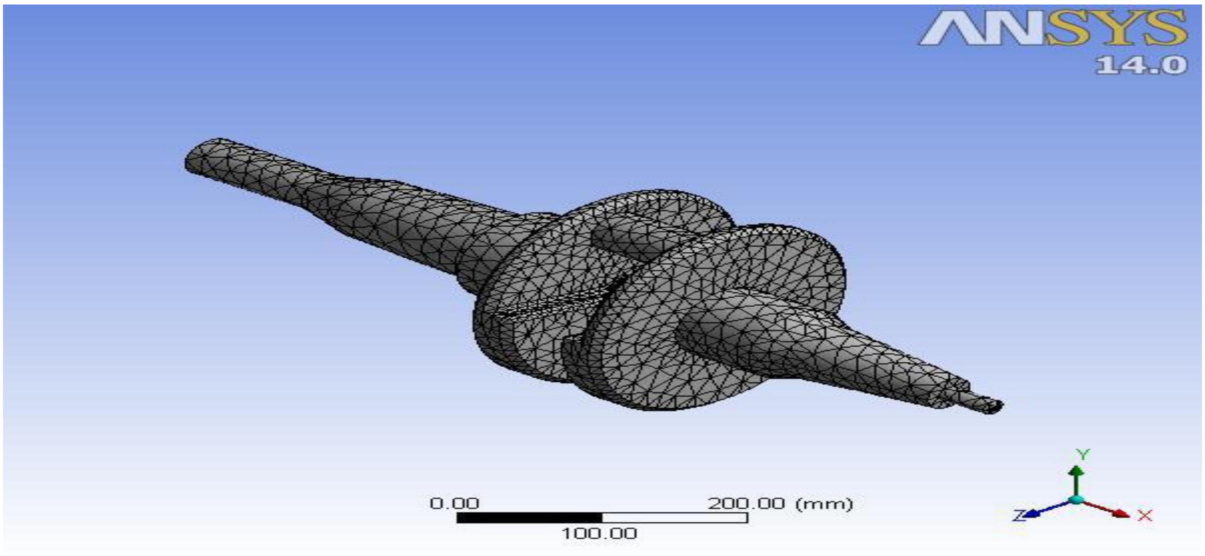

Figure 2, FE Model of Crankshaft of Single Cylinder SI Engine

After the meshing of the model, the boundary conditions and the constraint have to apply in order to get the outcome. So for the boundary conditions the crankshaft has to be made fix or we can say by keeping all DOF=0; so that analogically, it will behave like a fixed beam conditions and pressure has been applied to the crankpin end of the crankshaft as shown in figure 3. The applied load is of pressure of 12 MPa.

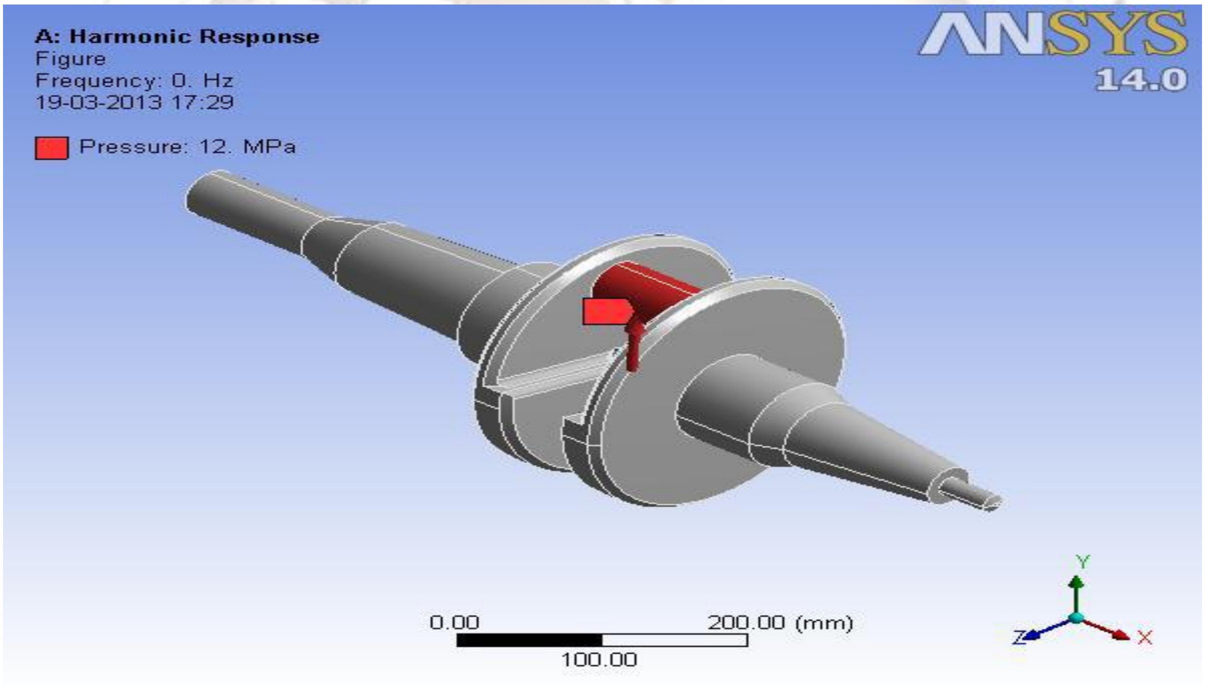

Figure 3, Point of application of Pressure

IV. Results

The results represent the various contours under the applied load of the pressure for the harmonic analysis. The first figure represents the all values of deformation in various parts of crankshaft as projected in figure4. The output in this diagram small intervals of frequency. The range of frequency starts from 100Hz and lasts to 1000 Hz. figures the maximum deformation of 7.5548e-004 mm at a frequency of 1000 Hz with 0° phase angle. Amplitude and the frequency diagram depict the various peak points of amplitude at different frequency points. The magnitude of amplitude of harmonic vibrations goes on changing at small The amplitude hikes at 300 Hz and 600 Hz becomes lowers at 700 Hz.

Type	Total Deformation	Directional Deformation	Equivalent Elastic Strain	Equivalent (von-Mises) Stress
By			Frequency	
Frequency			Last	
Phase Angle	0. °			
Orientation		X Axis		
Coordinate System		Global Coordinate System		
Results				
Minimum	0. mm	-7.4555e-004 mm	5.6877e-009 mm/mm	7.3424e-004 MPa
Maximum	7.5548e-004 mm	5.3853e-004 mm	6.8439e-006 mm/mm	0.87406 MPa

Table 1 Results generated in Harmonic Analysis

Type	Normal Stress	Normal Elastic Strain	Directional Acceleration	Normal Stress	Normal Elastic Strain
Orientation	X Axis				
Suppressed	No				
Options					
Frequency Range	Use Parent				
Minimum Frequency	0. Hz				
Maximum Frequency	1000. Hz				
Display	Bode				
Frequency				10. Hz	
Duration				720. °	
Results					
Maximum Amplitude	0.71356 MPa	3.3145e-006 mm/mm	19680 mm/s²		
Frequency	300. Hz		600. Hz		
Phase Angle	180. °		0. °		
Real	-0.71356 MPa	-3.3145e-006 mm/mm	19680 mm/s²	0.25073 MPa	1.1597e-006 mm/mm
Imaginary	0. MPa	0. mm/mm	0. mm/s²	0. MPa	0. mm/mm
Amplitude				0.25073 MPa	1.1597e-006 mm/mm
Reported Frequency				100. Hz	

Table 2, Results displayed in graphs

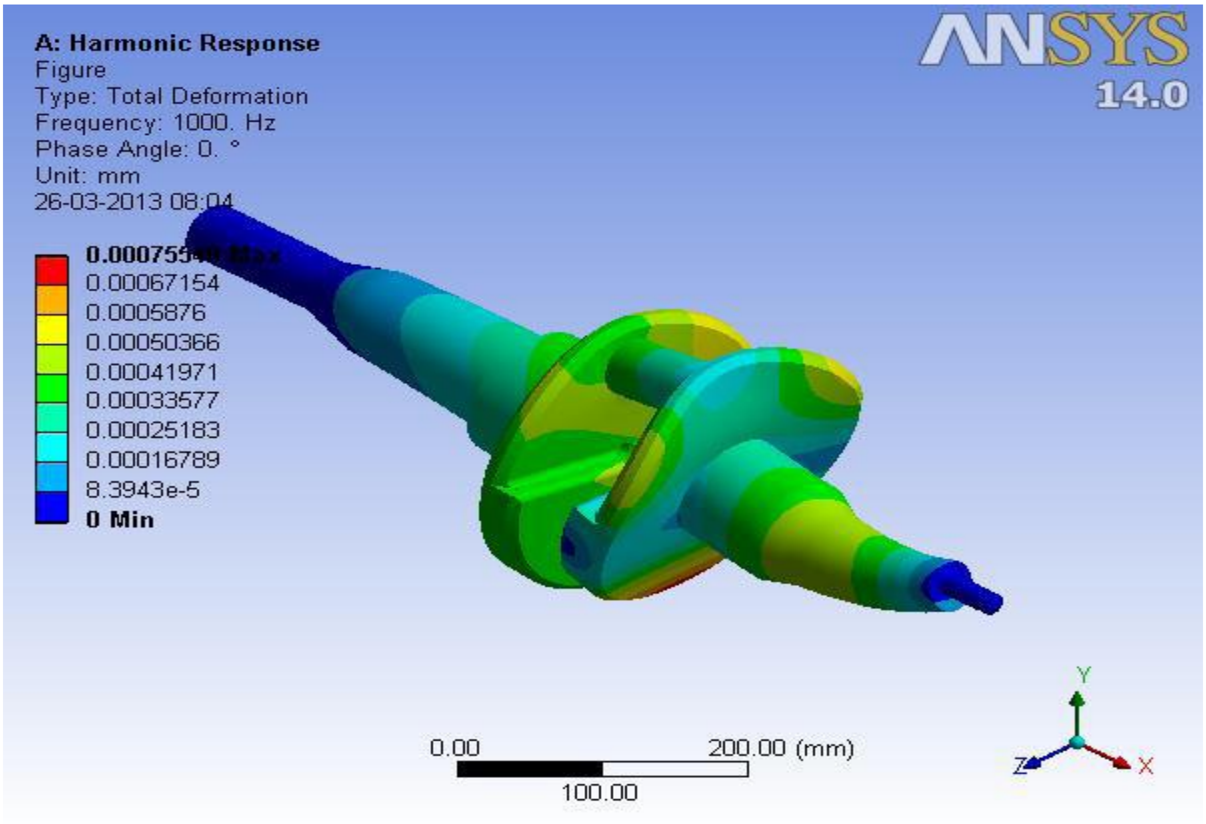

Figure 4, Total deformation

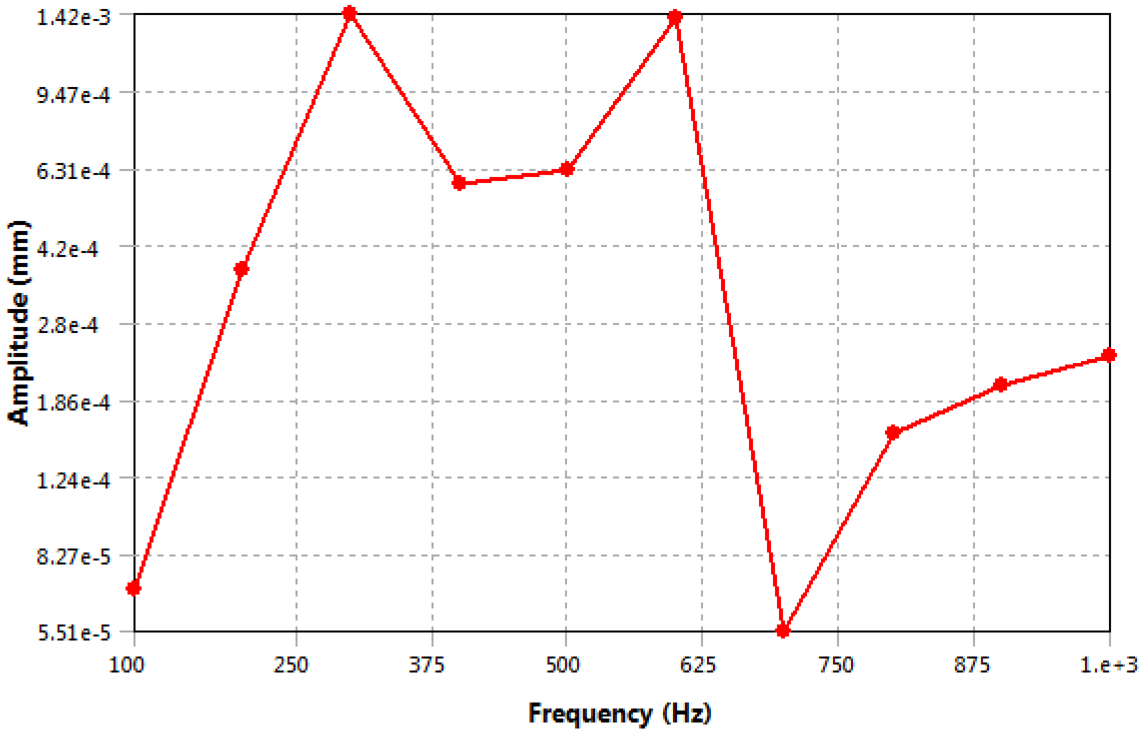

Figure 5, Amplitude vs. Frequency for deformation

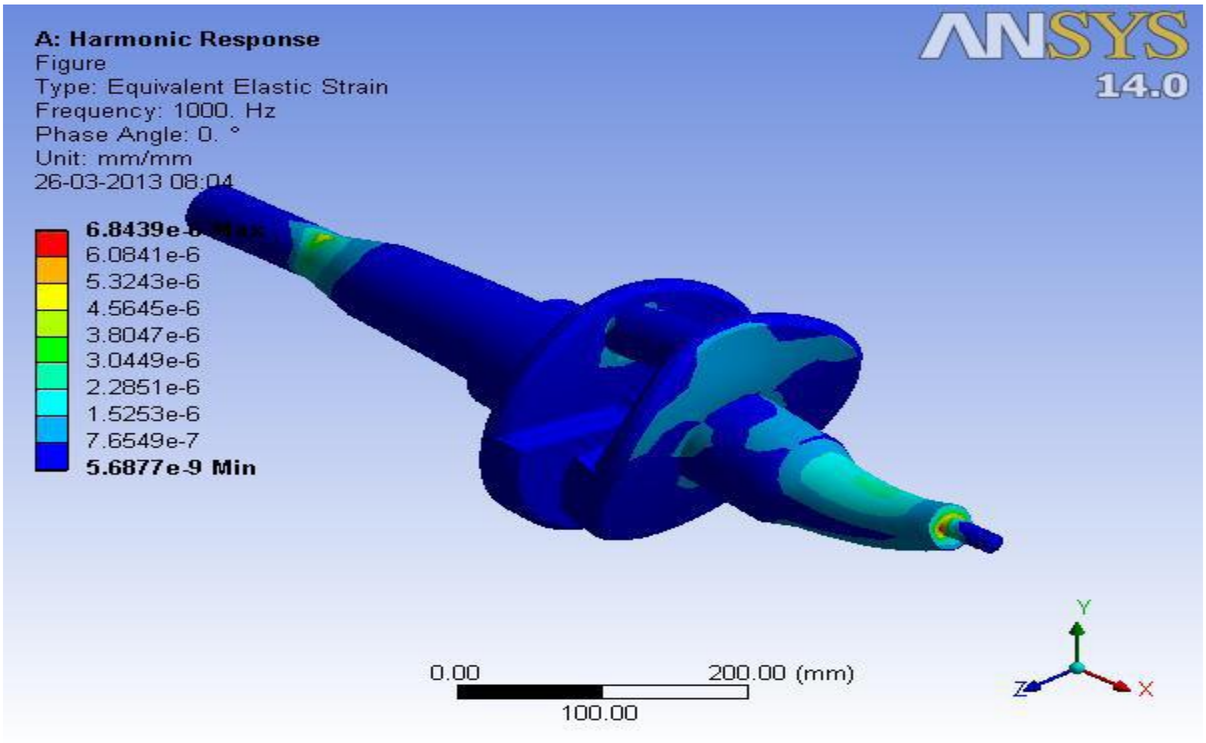

Figure 6, Equivalent Elastic Strain

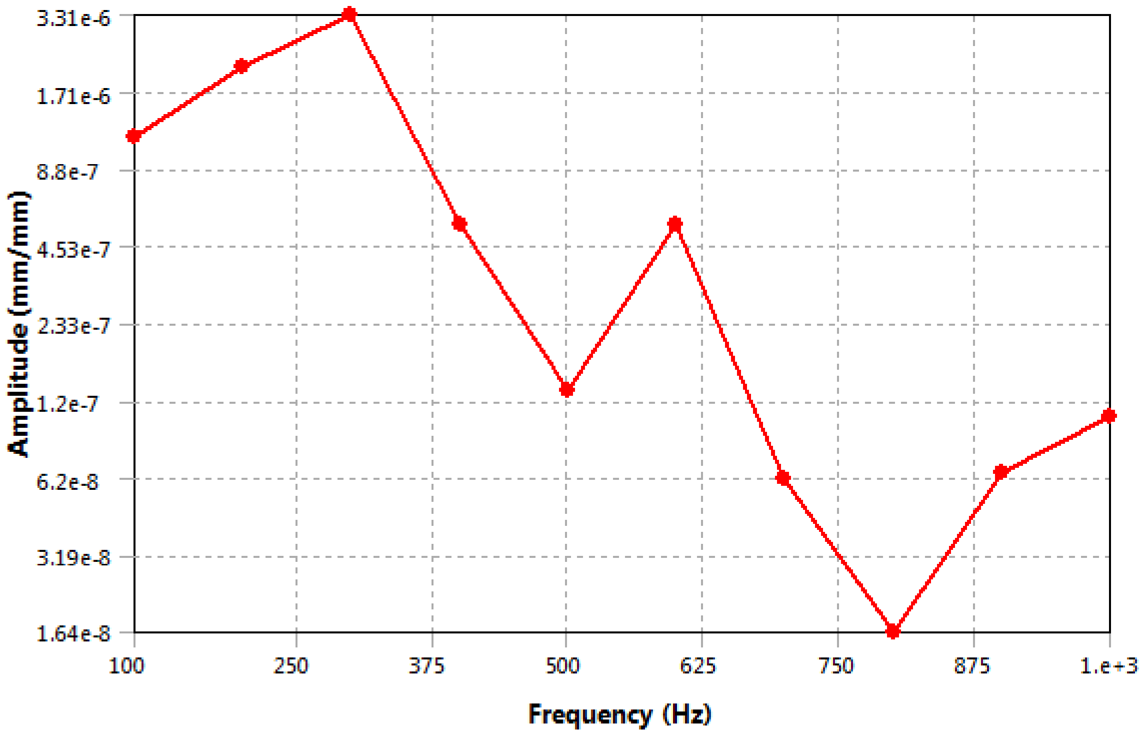

Figure 7, Amplitude vs. Frequency for Strain

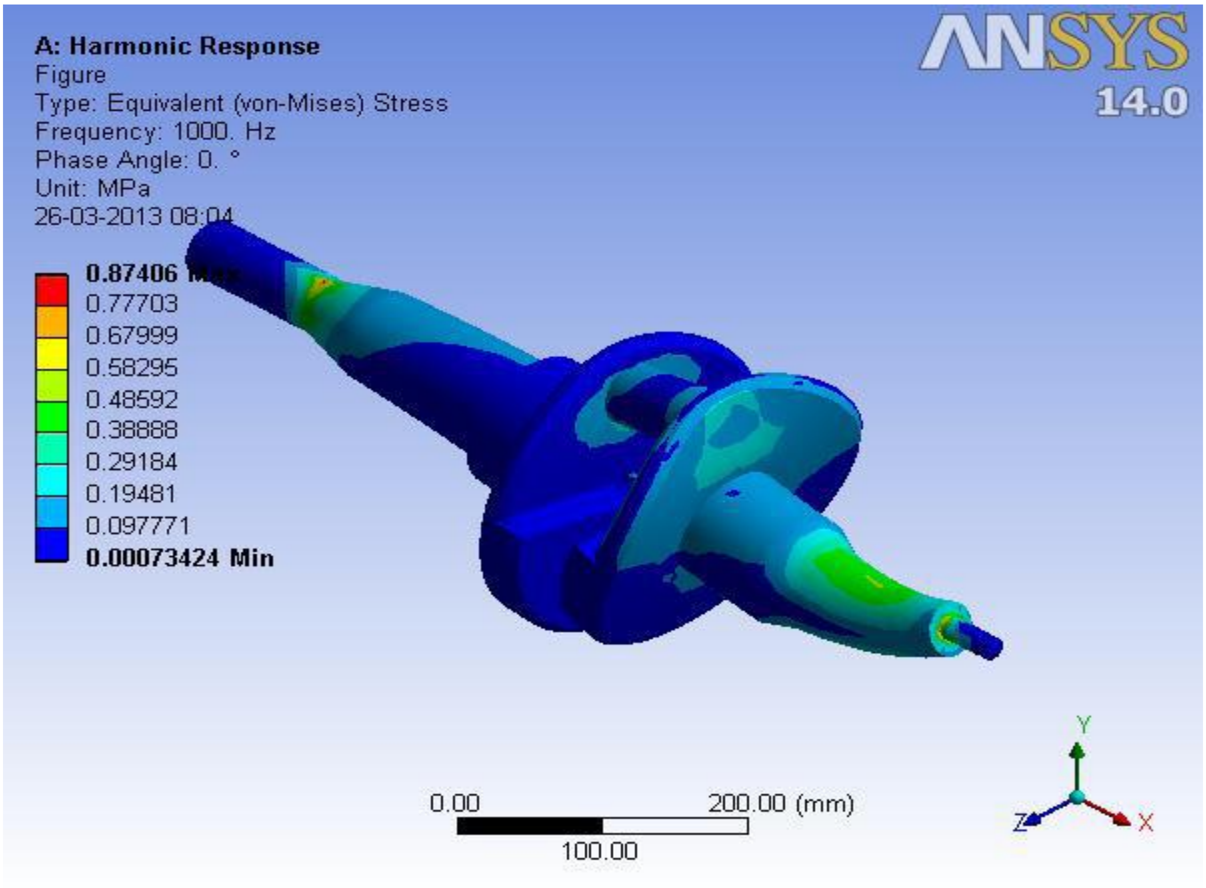

Figure 8, Equivalent Von-mises stress

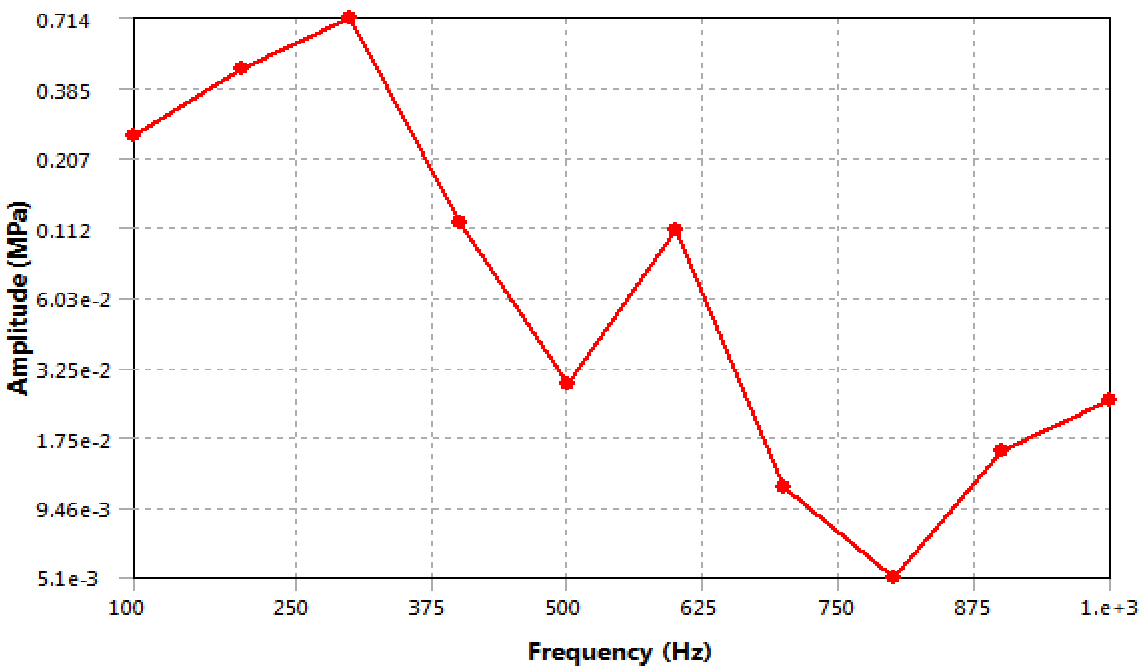

Figure 9, Amplitude vs. Frequency for Stress

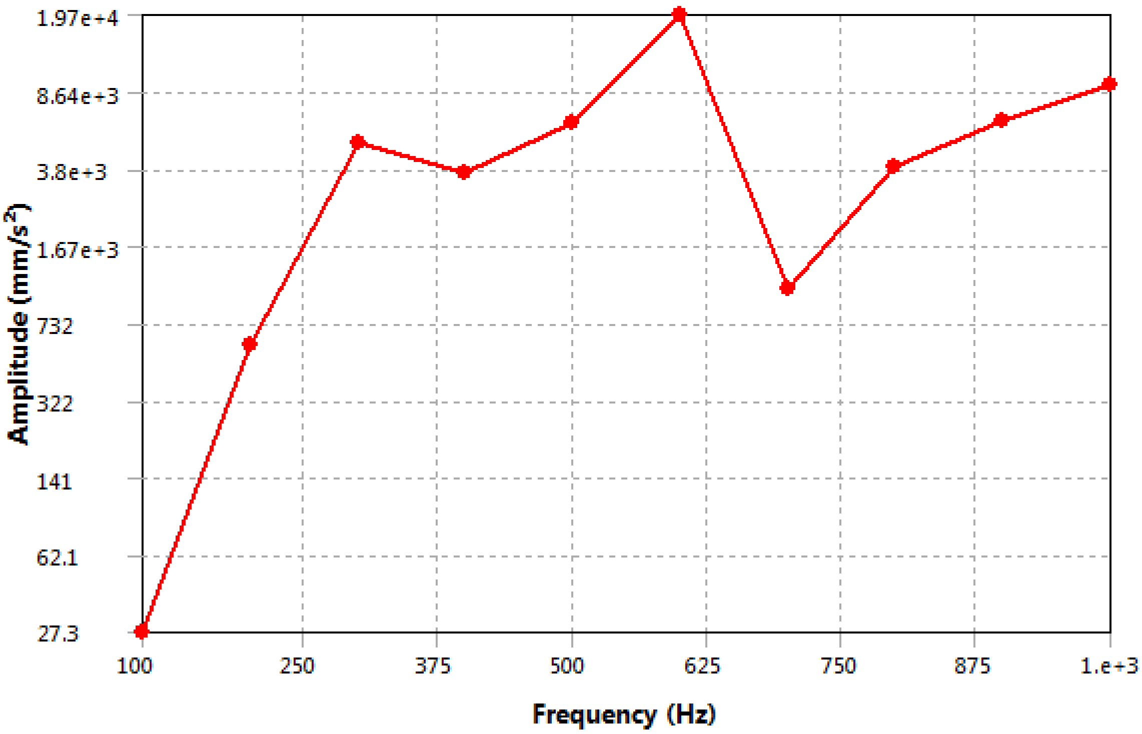

Figure 10, Amplitude vs. Frequency for Acceleration

V. Conclusion

The marks characterize the various form of the under the applied load of the pressure in the case of harmonic analysis. With study of harmonic analysis we identified the major and critical locations of a crankshaft where failure can arise. The various values of deformation, stress, strain for the crankshaft have been shown in the results with contours. The various graphs plotted represent the behaviour of the crankshaft under a speed of 9550 rpm of the engine. this work is helpful to find out various critical positions involved in crankshaft so that after this study some suitable amendments may be performed on the design of crankshaft according to constraints and boundary conditions.

References

[1] Yifei, Y. and Bing, S.; Modal and Harmonic Analysis of Crankshaft for High-Power Drilling Pump; MAXWELL Science Publications; 2013, Vol. 5, Issue: 01.
[2] Johnston, P. and Shusto, L., "Analysis of Diesel Engine Crankshaft Torsional Vibrations," SAE Technical Paper 872540, 1987.
[3] Prakash, V., Aprameyan, K., and Shrinivasa, U., "An FEM Based Approach to Crankshaft Dynamics and Life Estimation," SAE Technical Paper 980565, 1998.
[4] http://en.wikibooks.org/wiki/Mechanical_Vibration
[5] Jacob Pieter Den Hartog, " Mechanical Vibration", Fourth Edition, Courier Dover Publications, 1956
[6] Lalane, C.; Mechanical Vibration & Shock Analysis, Volume 2, John Wiley & Sons Publication, USA 2009.
[7] Schmitz, L. T. and Smith, S. K.; Mechanical Vibrations: Modelling & Measurement
[8] G. GENTA 0877 *Journal of Sound and Vibration* 124, 27-53. Whirling of unsymmetrical rotors a finite element approach based on complex co-ordinates.
[9] Singh, V. P..; Mechanical Vibrations, Third Edition, Dhanpat Rai & Co. (Pvt). Ltd., India 2012.
[10] Khurmi, R. S. and Gupta, J. K,; Machine Design, Eurasia Publishing House, New Delhi, 2011
[11] Chandrupatla, T.R. and Belegundu, A. D.,; Introduction to Finite Element in Engineering", Third Edition, Prentice-Hall of India Private Limited, New-Delhi, 2008.
[12] Ansys element reference library, Ansys14