Dynamic & Static Analyses of Model of Engine Crankshaft

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Abstract:

This study investigates dynamic & static behaviour of automotive engine crankshaft with the specific conditions. This work investigates the coupled modes, including, couple torsional flexural vibration and coupled longitudinal flexural vibration for non rotating engine crankshaft. The finite element models of this generally used are in two categories beam elements and solid elements and in this study the solid element category is being used. By using this model the natural frequencies and mode shapes of the engine crankshafts are determined by the FEM. by applying the specific load, the stress analysis of engine crankshaft is carried out by Finite element method. This study is useful to predict the safe values of dynamic as well as static behaviour of engine crankshaft.

Keywords: finite element model, natural frequency, mode shapes, modal analysis, stress analysis

I. Introduction:

After years of steady, conventional model changes, the automobile engine industry is in the midst of the most powerful product changeover in its history. To accomplish the need to design a sensible engine, the structural engineer will need to use imaginative concepts. The demands on the automobile designer increased and changed rapidly, in every respect of design and engineering. The main and foremost part of the engine is a crankshaft which is used to propel the vehicle ahead which must be able to accomplish all the design requirements whether structural or vibration. The crankshaft which we are going to accommodate here is a part of single cylinder SI Engine as shown in figure 1.

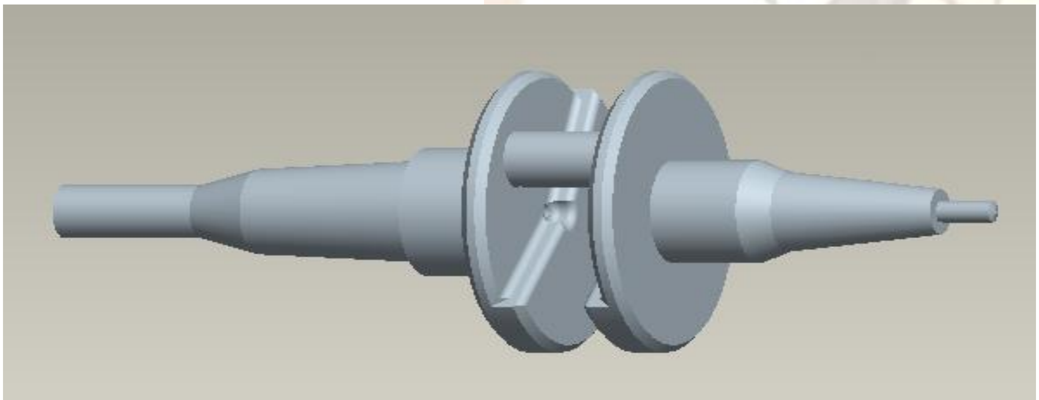

Figure 1, Geometric model of single cylinder SI engine Crankshaft

This model is being modelled in considering all the designing parameters of the crankshaft.

Presently, the rotor-bearing systems utilized for modelling rotating machinery and their mounting structures, for example, electric motors, turbo machinery, transmission shafts, propellers, etc., are commonly analyzed by the finite element method. Computations of natural frequencies, mode shapes, critical speeds, steady state responses, and transient responses play important roles in the design, identification, diagnosis, control of rotor-bearing 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 finite element procedures developed for rotor-bearing systems are directed toward generalizing and improving the shaft model proposed by Ruhl and Booker .1]. Nelson and McVaugh.2], Zorzi and Nelson .3] utilized finite beam element models to formulate the dynamic equation for a linear rotor system and determine the stability and steady state responses. The beam formulations based primarily on Timoshenko's assumptions have been given by Thomas *et al*. .4] Ozguven and Ozkan .5] and Nelson .6] further developed the finite element model by including the effects of rotary inertia, gyroscopic moments, shear deformation, and internal damping. Tapered beam elements have been developed by Rouch and Kao .7] and Greenhill *et al*..8] to model a linearly varying diameter along the beam length. Stability and steady state responses of asymmetric rotors with a flexible shaft have been studied by Genta .9]. The effects of both deviatoric inertia and stiffness due to an asymmetric shaft and disk have been studied by the finite element method in research by Kang *et al*..10].

Moreover, many mechanical systems such as turbine blades, aircraft propellers, robot arms and engine crankshafts, which are perpendicular to a rotating axis, can be modelled by a radial rotating beam element. Nagaraj and Shanthakumar .11] used the Galerkin method, together with an eighth!order polynomial, for solving a problem concerning a rotating beam without a hub. Putter and Manor .12] presented a six-degree-of-freedom element for a rotating radial cantilever beam to allow inclusion of a shroud mass. Hoa.13] presented a finite element formulation for a uniform rotating beam with a tip mass. Khulief and Yi .14] developed a finite element formulation representing the vibrational response of a uniform rotating beam with a tip mass during flapping and lead-lag motion. Their formulation accounts for the centrifugal force field and centripetal acceleration effects. Yokoyama .15] developed a finite element procedure for determining the free vibration characteristics of rotating uniform Timoshenko beams. The effects of hub radius, setting angle, shear deformation and rotary inertia on the natural frequencies of the rotation beams have been examined. Magari *et al*. .16] developed a rotating blade finite element with coupled bending and torsion. Khulief .17] derived explicit expressions for the finite element mass and stiffness matrices using a consistent mass formulation for the vibration of a rotating tapered beam. Bazoune and Khulief .18] developed a finite element for vibration analysis of rotating tapered Timoshenko beams.

Bagci and Rajavenkateswaran .19] utilized a spatial finite line element method in an analysis of rotors, including crankshafts. Smaili and Khetawat .20] proposed a spatial four-node beam element based on Timoshenko's theory for the modelling of crankshafts.

Geradin and Kill .21] derived a three-dimensional finite element for modelling flexible rotors, because it produces good estimation in stepped and tapered shaft analyses. Stephenson *et al*. .22] showed an excellent agreement between the measured frequencies and the frequencies calculated by using an axisymmetric solid finite element model. Stephenson and Rouch .23] presented axisymmetric solid finite elements with matrix reduction in modelling a rotating shaft.

III. Methodology

The complex spatial nature of engine crankshaft makes the solid element approach an attractive one for determining their coupled flexural, torsional and longitudinal vibration modes. This study models practical crankshaft by using a tetrahedron elements for static and dynamic testing. For preparing the results of static and dynamic analysis, a crankshaft from a single cylinder SI engine is used. The model of the crankshaft has been modelled in pro e wildfire 4.0 with suitable dimensions then after dynamic and static analysis has been performed at the platform of ANSYS 14.0. the result has been drawn from various modes of crankshaft for dynamic analysis and considering specific loads and boundary condition for static analysis.

Computational Dynamic Analysis

Here we have carried out the dynamic analysis for the natural frequency of the crankshaft using the block lanczos method in Ansys platform. The value of the natural frequency is represented in the below mentioned figure.

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	186.75	1	1	1
2	296.75	1	2	2
3	330.31	1	3	3
4	471.44	1	4	4
5	667.67	1	5	5

The different mode shapes representing the vibration is presented below

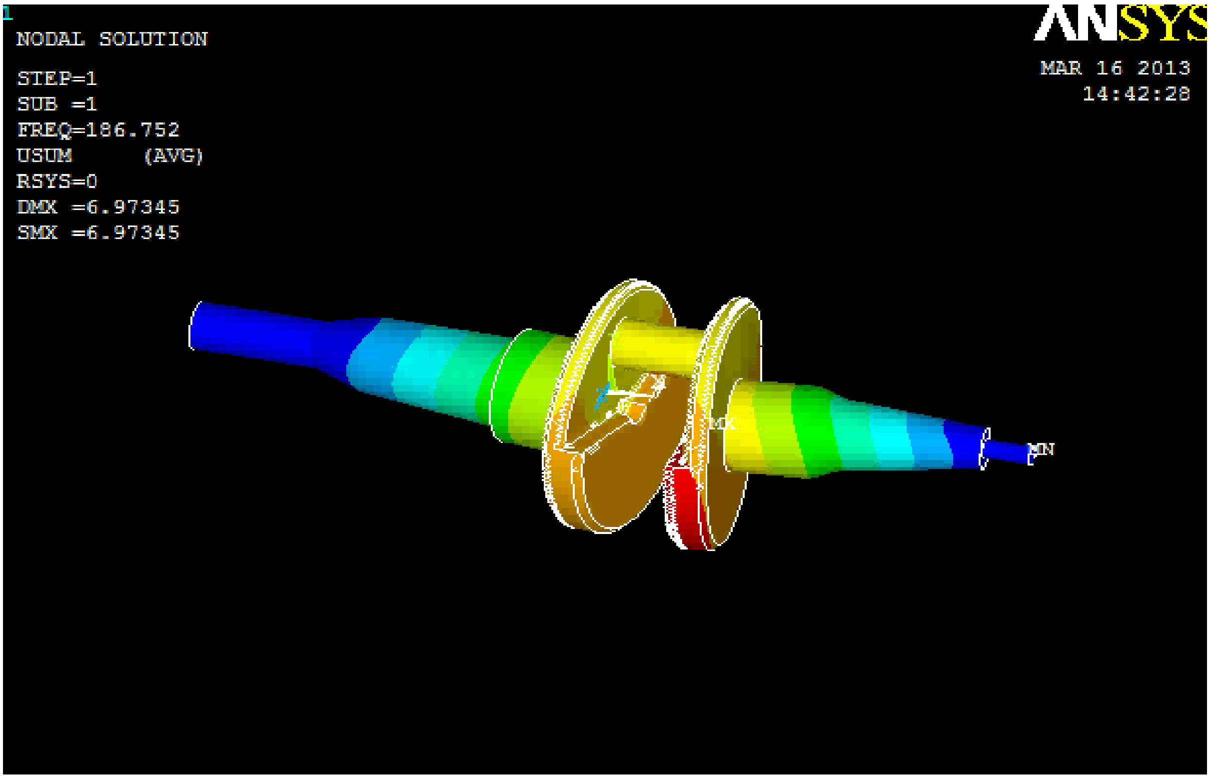

Figure 2, First Mode

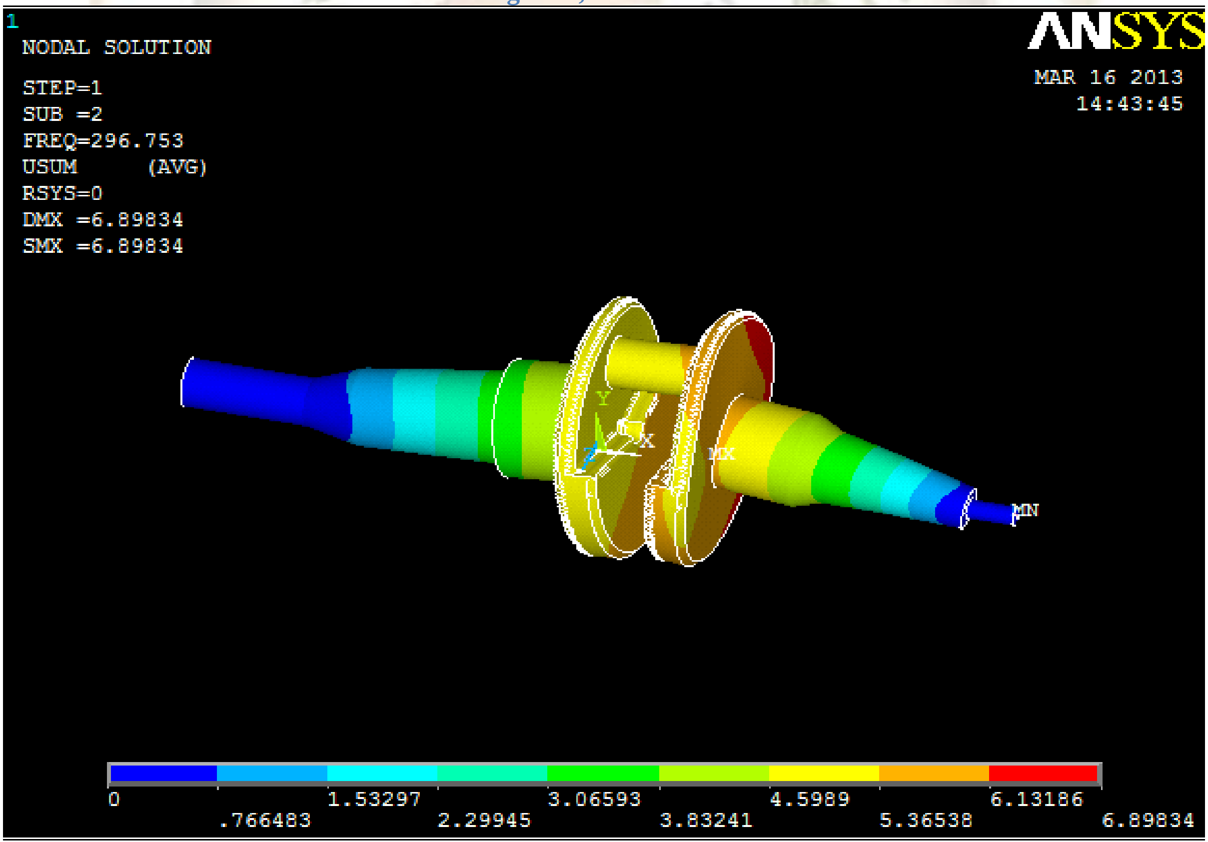

Figure 3, Second Mode

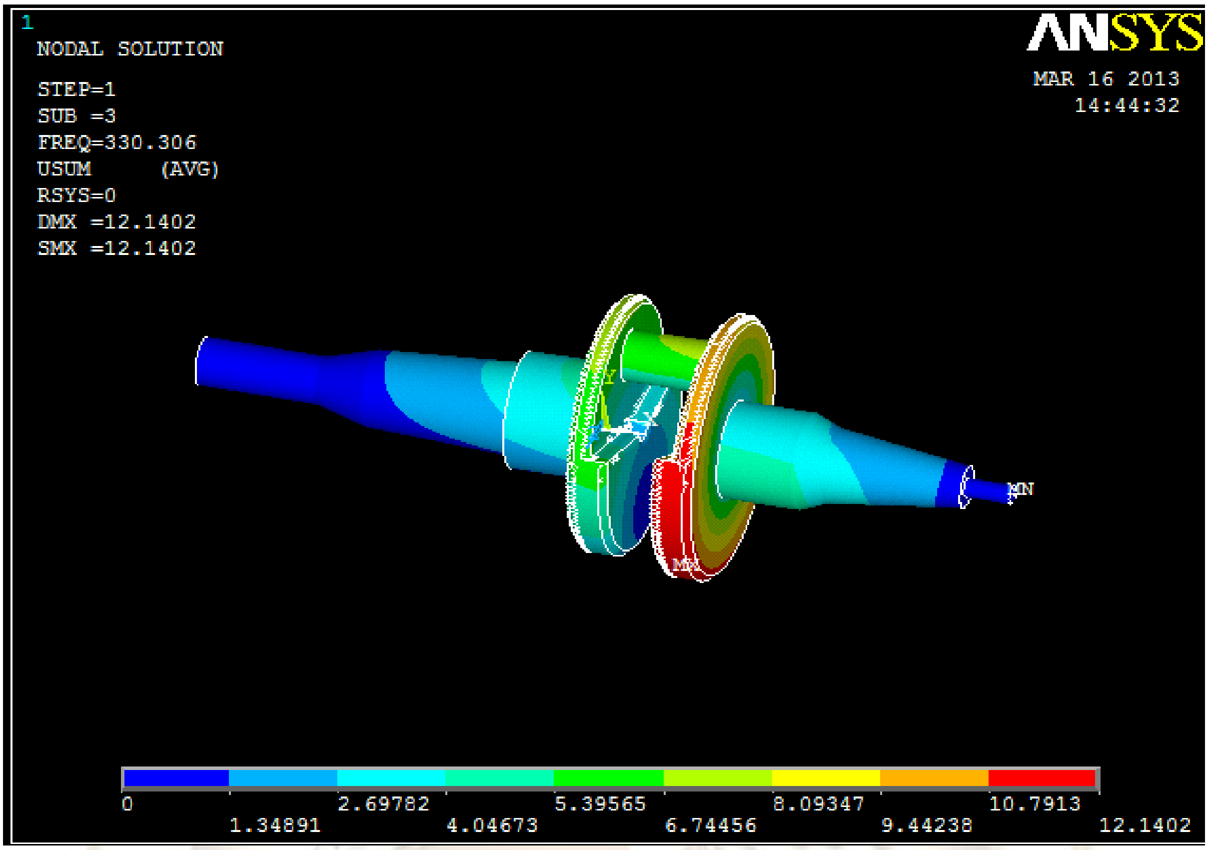

Figure 4, Third Mode

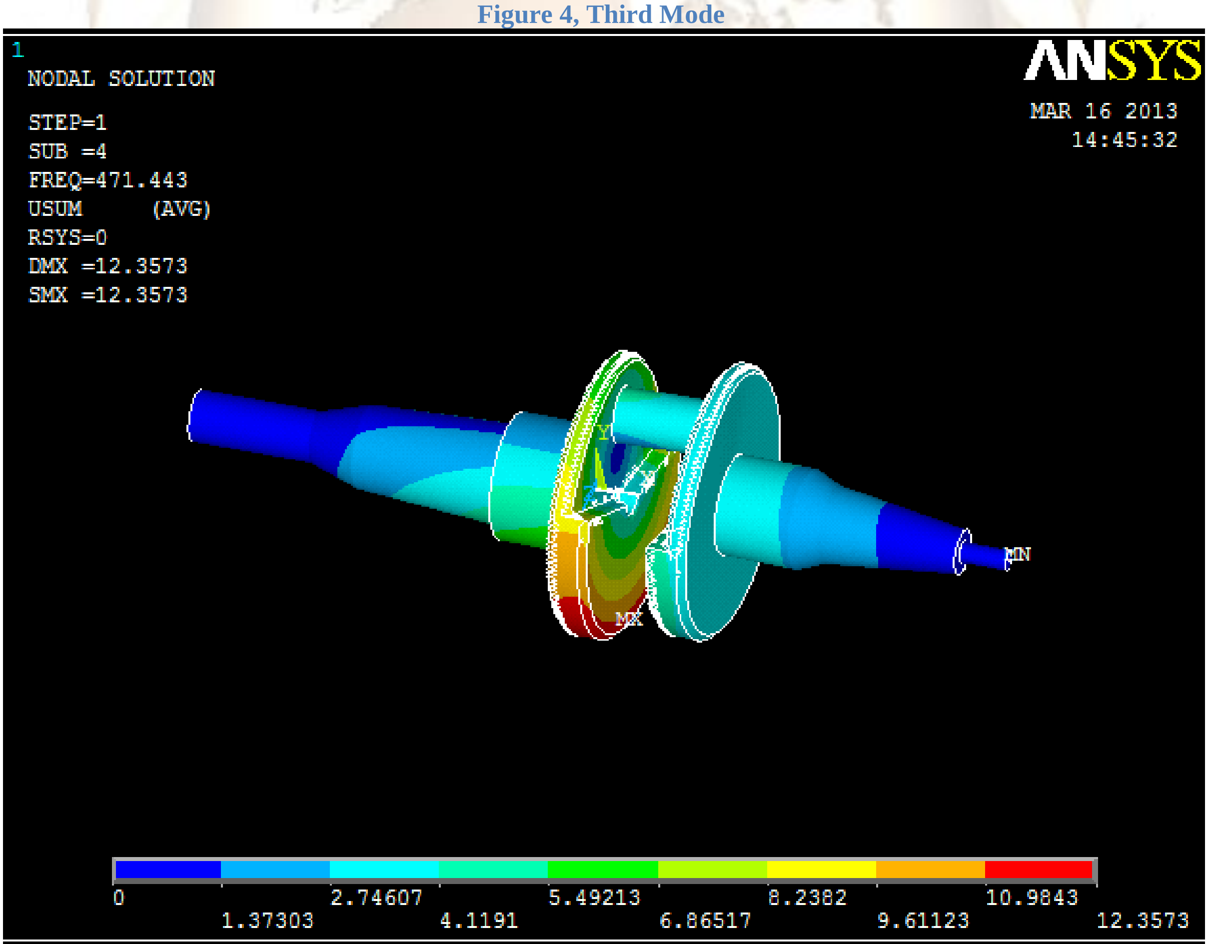

Figure 5, Fourth Mode

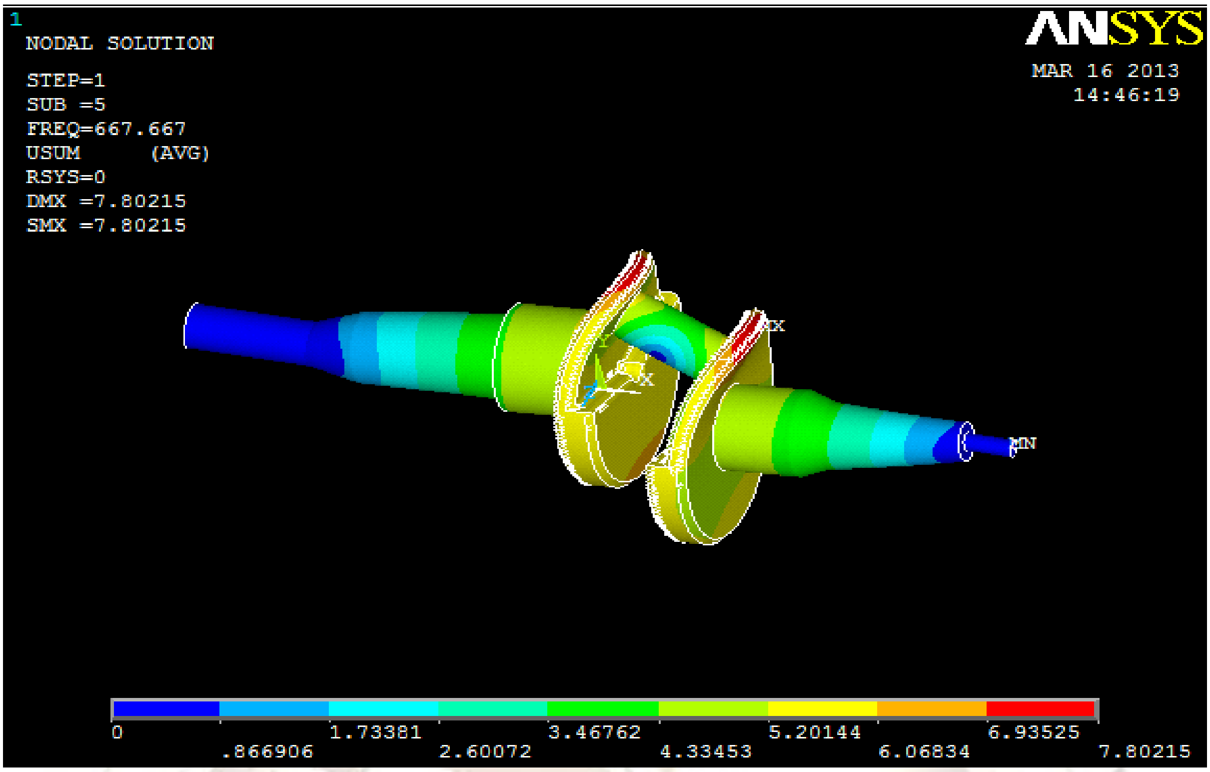

Figure 6, Fifth mode

Computational Static Analysis

The static analysis represents the loading conditions on the engine crankshaft along with the displacement and deformation diagrams. For which the geometrically modelled model has to be turned in the meshed model by performing meshing operation under the finite element methodology.

The meshing has always been the key of the finite element model and for the exact solution of any object; it should be properly meshed with relevant element shape and size. Generally for the solid bodies the hexahedron mesh is preferred due to high degree of accuracy. We introduced the element division in each line for the sufficient meshing, i.e., number of divisions 5. For boundary conditions of the crankshaft, it is to be suspended from the both sides or we can say that the joints of bearing and crankshaft will bear zero degree of freedom, and the crank pin will come across the applied load. The load will be distributed among the crank pin and its is opposed by adjoining members in the opposite direction of the crankshaft, i.e. in Y axis as shown in figure 7.

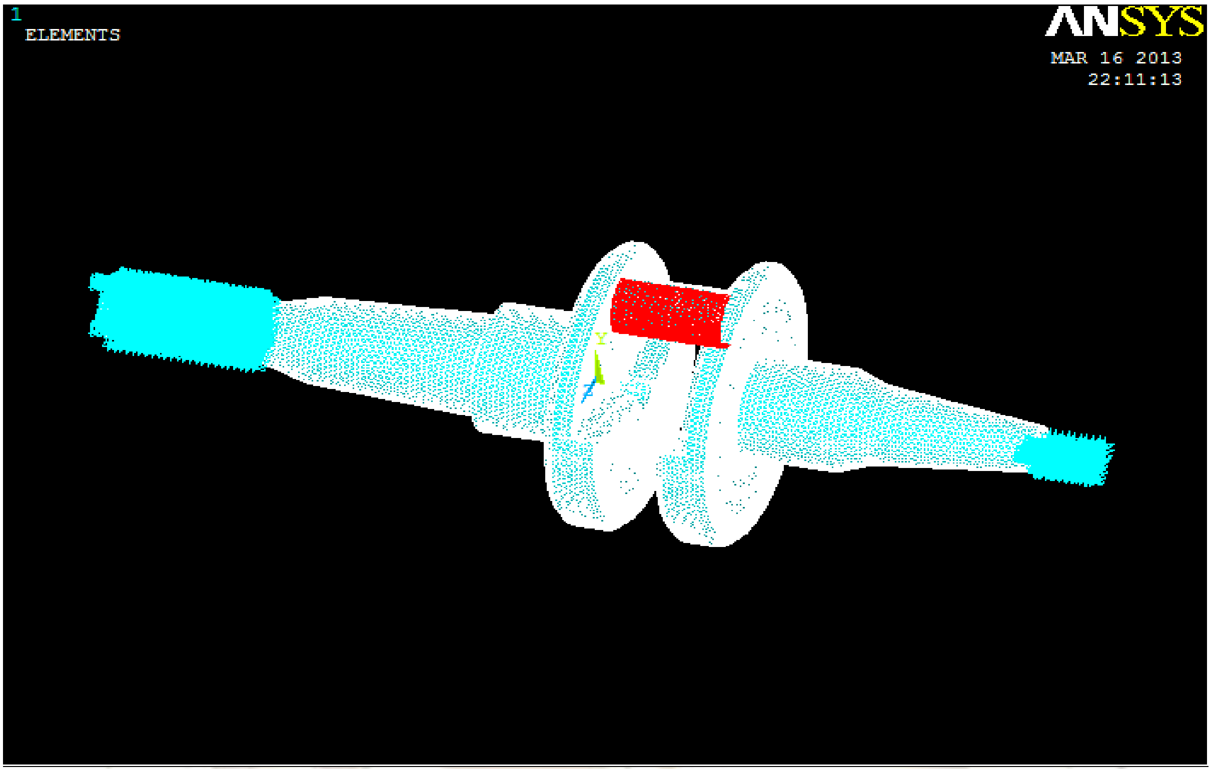

Figure 7, Meshed model with loads and constraints

The load data processed for this crankshaft is 120 MPa for the crankshaft of material having E= 200GPa and 1/m = 0.3. The density of the material is taken as7.856Kg/m^3. This pressure force is applied by the gases of product of combustion and been transferred to the crank pin via piston and gudgeon pin through connecting rod. Afterwards the model is solved for various parameters like displacement diagram, von mises stress diagram and von mises strain diagram. The results is represented in the following figures below

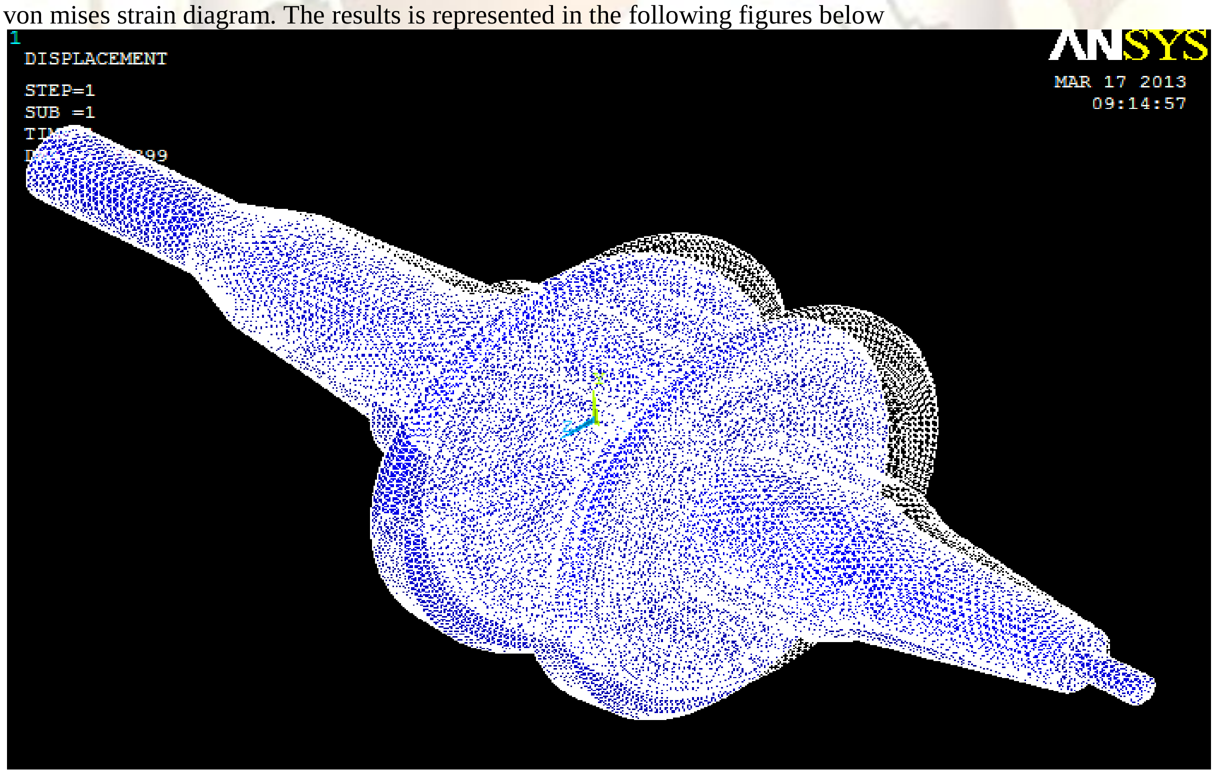

Figure 8, Deformation diagram

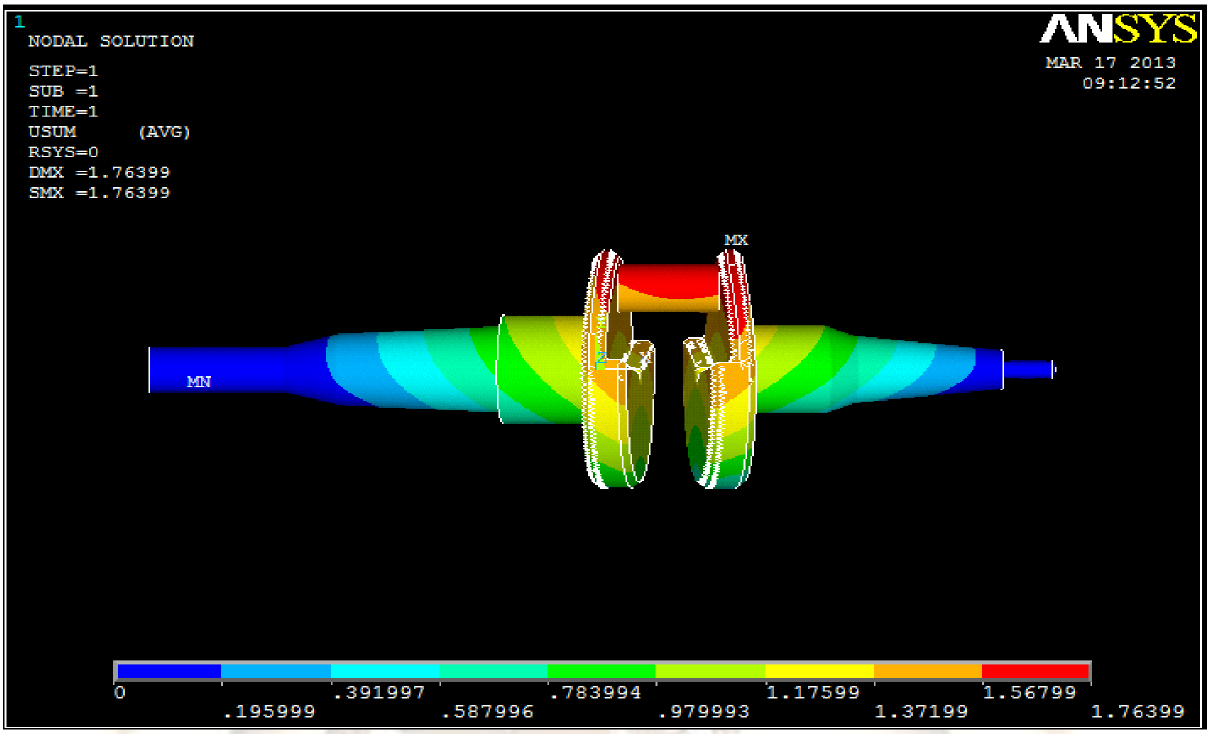

Figure 9, Displacement vector sum

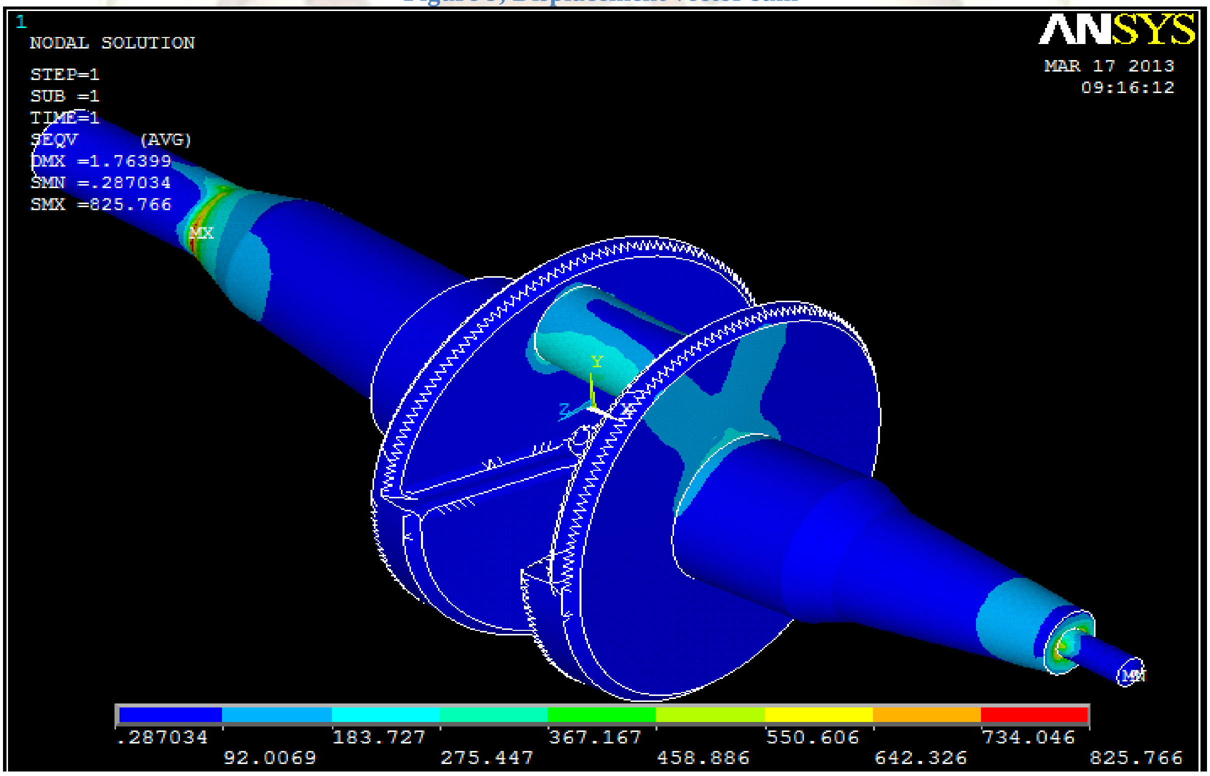

Figure 10, Von Mises Stress diagram

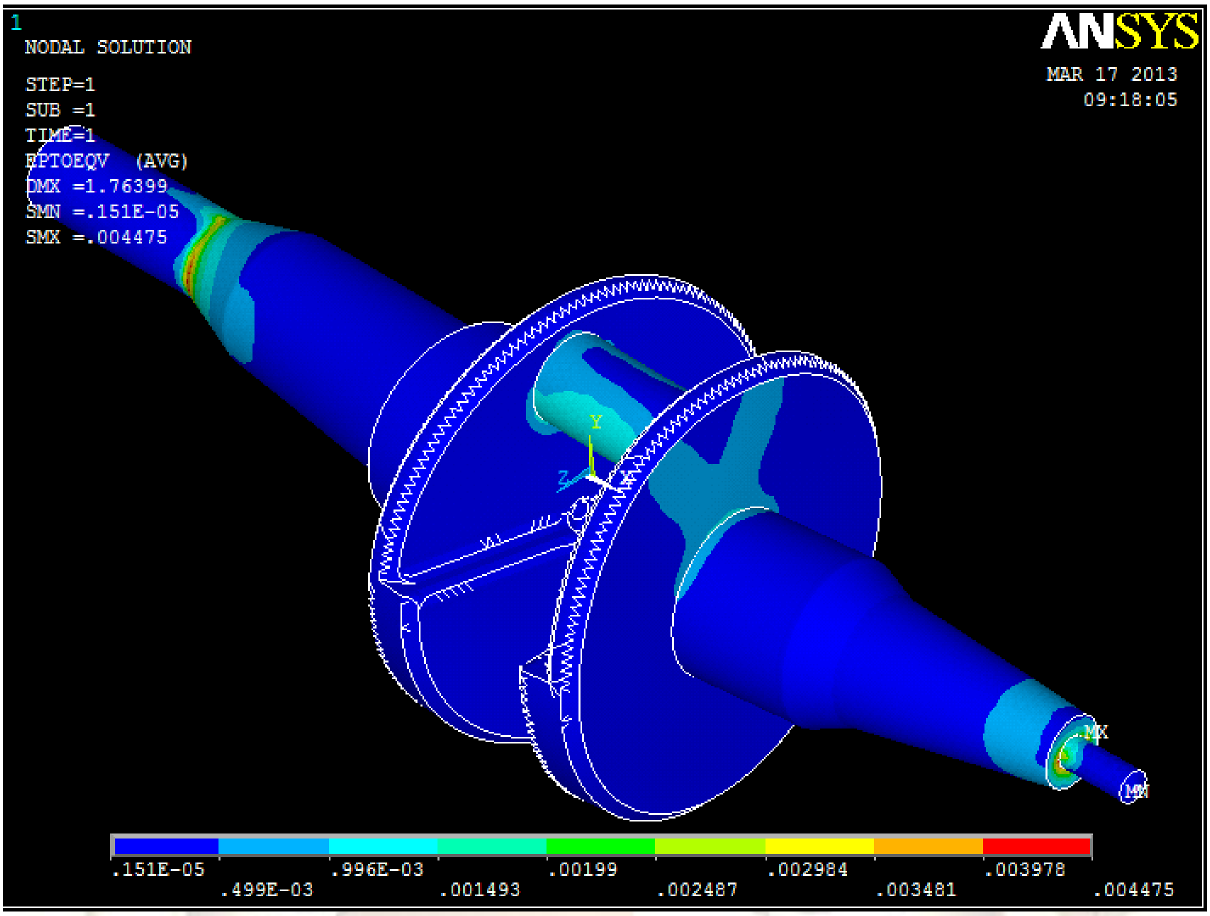

Figure 11, Total Von Mises mechanical strain

IV. Conclusion

This study entails the dynamic behaviour and the static behaviour of the engine crankshaft under the different respective conditions. The first case is of about the dynamic analysis, precisely saying that it was a natural frequency analysis to know the range of the natural frequency, which lies in between 186.75Hz to 667.67Hz for various modes.

The stress strain evaluation under the static analysis is executed at the load of 12MPa, and has performed with the solid elements. The value of static analysis using FEM is DMX= 1.76 mm, SMN= 0.28MPa, SMX= 825MPa as deformation and stresses respectively, and SMN = 0.15 x 10^{-05}, SMX = 0.0045 as the strain values. The failure in terms of deformation and the stress concentrations has been predicted at the locations of changing section, bearing end, crank pin. Overall the model is safe as the induced stress is within the permissible limits, because the ultimate strength of the material is quite high. This concept value implies the safe value of applied loads vibrations and deformations

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