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FATIGUE STRENGTH AND DYNAMIC VIBRATIONAL ANALYSIS OF V8 ENGINE CRANK SHAFT USING FINITE ELEMENT METHOD

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Abstract - ***The objective of this study was to develop a model for V8 engine and perform fatigue analysis on it. In this study a dynamic simulation was conducted on forged steel crankshaft of V8 engine. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. Load calculations were performed analytically and were verified by simulations in ANSYS. This load was then applied to the FE model in ANSYS and boundary conditions were applied according to the engine mounting conditions. The analysis was done for engine speed 14000 rpm and as a result, critical engine speed and critical region on the crankshafts were obtained. A static structural analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. Steady loading and response conditions are assumed; that is, the loads and the structure's response are assumed to vary slowly with respect to time. Following results shows stress and deformations of v8 engine crankshaft. The stress obtained by static structural analysis of the forged steel crankshaft is 138.91 M Pa by applying a force of 25 KN which is well below the allowable limit of 144 M Pa. The Fatigue results thus is used to determine life, damage, and factor of safety of the crankshaft using a stress-life or strain-life approach. The Fatigue Tool is available only for static structural and flexible dynamic analyses. In this case, for forged steel, stress life approach is adopted and the SN curve for the material as per ANSYS.***

Key Words: *V8 engine, Load calculations, SN curve, Static And Dynamic Analysis etc…*

1. INTRODUCTION

The crankshaft plays an important role in all Internal Combustion Engine. It is a large component, which converts the reciprocating motion of the piston in to rotary motion of the crank shaft. Presently V8 engine is used in **MOTOSPORT** vehicle. It has complex geometry. The crankshaft experiences cyclic load due to which fatigue failure occurs over a period. The fatigue analysis has to be considered in the design stage itself. In production industry, design and development of crankshaft plays an role in order to reduce manufacturing cost of the product and minimize weight resulting in lighter and compact engine to give higher output with better fuel efficiency. Since the engine is V type, loading is quite complicated, which makes multi-body dynamic analysis more inevitable to have accurate stress values of torsional vibration. The study of multi cylinder engine crank shaft can be treated as equivalent to that of a single cylinder engine. For modeling, two journal bearings and a crank pin are considered. But the dynamic analysis for multi cylinder engine crank shaft differs from that of single cylinder engine. Since fatigue is the major cause of failure, this work majorly tries to locate the critical regions of failure of the crank shaft due to fatigue and determine the values stresses at these locations. Some researchers have made an attempt in this direction.

Alex et al [1] in their work on Dynamic analysis of Crankshaft analyzed the average von-misses stress and principle shear stress over the crankshaft using ANSYS WORKBENCH software, the created model using 3D modeling software CATIA. They used medium carbon steel En-9 as a material of crank shaft. They inferred that the static analysis of crank shaft yields over estimated values while dynamic analysis yields realistic values. They noticed that at the critical locations, the dynamic analysis is not affected by torsional load. In view of this they simplified the crankshaft analysis by applying only bending load. Further, they found that fillet areas of the crankshaft were critical and stress concentration factors were unduly high. Mahesh et al [2] in their work used

finite element method for estimating life of crankshaft. They performed dynamic force analysis using MATLAB. The resulting values were used as input to the model. They analysed the model using ANSYS. To obtain critical locations on the crank shaft, they performed analysis with different engine speeds and fillet radius to crank pin diameter ratios and found that crankshaft failed due to formation of cracks in fillet area. They concluded that the crank shaft design should consider fatigue as major factor. In their work on Finite Element Analysis of the crankshaft using Ansys workbench, Ashwani Kumar Singh et al [3] used nickel chrome steel and structural steel for the material of crank shaft, modelled the crankshaft using pro/E 5 software and analyzed the model using Ansys workbench. They found that the analytical results conformed to the software results. They noticed that the deformation is maximum at the centre of the neck of the crank pin. Also they found that the high stresses were developed at the edge of main journal. They concluded that nickel chrome is better material for crank shaft over the structural steel. During Multi-body dynamic Simulation on Flexible Crankshaft System, Ma Xingguo et al [4] created a virtual prototype of flexible multi-body crankshaft system of engine and subjected it to dynamic simulation. They inferred that mechanical behaviour of crankshaft of engine in real working condition can be simulated with the virtual prototype. They found that the centralized force of joint can be dispersed to the joint 'nodes of finite element model by the flexible multi body dynamic analysis and hence can determine the real-time dynamic loads exactly to provide the loads and boundary condition for the analysis in ANSYS. Further, they concluded that the transient response achieved in ANSYS with dynamic loads can be used to determine the dangerous area of the crankshaft in a working cycle and to evaluate the mechanical property of the crankshaft.
Shin Han et al [5] in their work on multi-body dynamic stress analysis of a crankshaft for V8 engine modeled hydrodynamic journal bearing efficiently using contact interaction with exponential pressure-over closure relationship which reduced computing time compared to hard contact approach. They observed the highest torsional vibration and eccentricity in the farthest crank journal from flywheel and the journal fillet in the vicinity of flywheel was subjected to the highest stress. Zissimos P. Mourelatos [6] predicted dynamic response of the crank shaft with the help of model using sub structuring technique. According to his predictions, misalignments generated due to large bending moments are also large. Boris B. Kosenok and Valeriy B. Balyakin [7] in their study of the Dynamic Characteristics of a Two-Cylinder Internal Combustion Engine using Vector Models concluded that it is better to arrange the cylinders in a vertical plane than on a horizontal plane in order to reduce horizontal vibrations. Further , during design of a multi-cylinder ICE, it is necessary to carry out similar dynamic analyzes of the kinematic scheme of the main mechanism at the preliminary stages of the design process in order to choose optimum geometric parameters which lead to reduction of dynamic loads on elements of the mechanism and increase in strength and fatigue life cycle characteristics. Priya.D.Shah and Kiran.K.Bhabhor [8] during parametric optimization of four cylinder engine Crankshafts concluded that higher the frequency of vibrations higher is the stiffness as frequency is directly proportional to stiffness and inversely proportional to mass. Further by reducing mass and increasing stiffness, maximum stiffness can be achieved.
Abhishek choubey et al [9] used SOLID WORKS in creating a 3-D model of crankshaft and used ANSYS software for analysis. Their observations were that centre of crankpin neck had maximum deformation. They also observed that the fillets of crank shaft were subjected to maximum stress

2. FINITE ELEMENT MODELING OF CRANK SHAFT FOR MULTI BODY DYNAMIC ANALYSIS

Finite element modeling of any solid component consists of geometry generation, applying material properties, meshing the component, defining the boundary constraints, and applying the proper load type. The finite element analysis was performed on crankshaft. Stresses from these analyses were used for superposition with regards to dynamic load applied to the crankshaft.

2.1 Geometry of Crankshaft

The dimensions of the crankshaft shown in Figure 2.1 were taken from 2d drawing taken from forging industry. Then solid model the forged steel crankshaft was generated using ANSYS. The solid model generated for the forged steel crankshaft is shown in Figure 2.2. It consists of eight cylinders. The picture of the forged steel crankshaft from which the geometry was generated is shown in Figure 2.3.

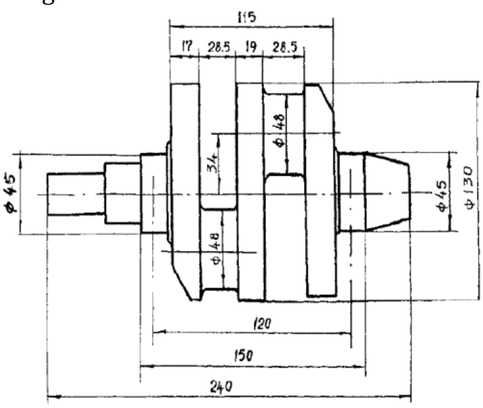

Figure -2.1: 2d drawing showing the dimensions of the forged steel crankshaft

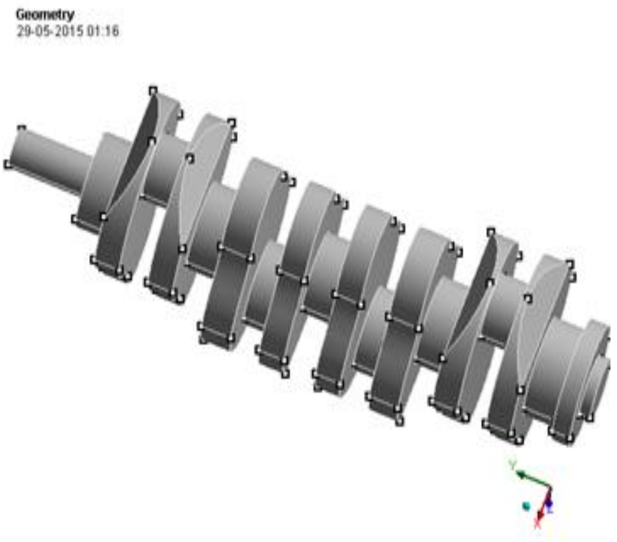

Figure-2.2: Finite Element Model of forged steel crankshaft

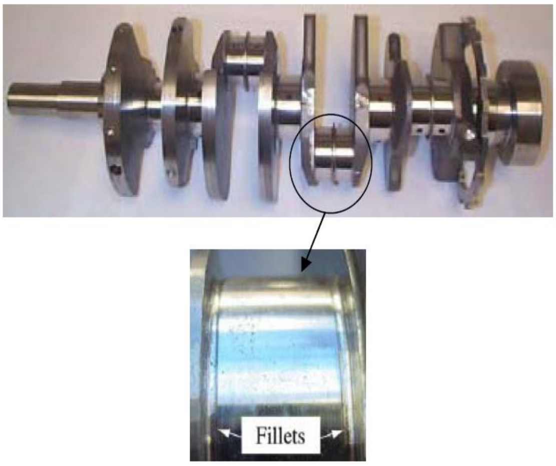

Figure-2.3: Picture of forged steel crankshaft

As can be seen in the picture of the crankshaft the gear on the rear side of the main bearing is not included in the digitized model, instead the plane next to it is extruded to cover the section which the gear covers. This simplification is reasonable since the gear is press fit at this location and the gear tooth does not have any effect on stresses at fillet areas where high stress gradient exists. It should be mentioned that simplification on the gear will not affect the stiffness of the model because the boundary condition used on this side of the crankshaft is a sliding edge and moving its location along the main bearing shaft will not change the stress results. In addition, the significant effect of bending load was observed to be in the cross section of the crankpin bearing and crank web. Therefore, simplifications on the main bearings will not affect stresses at critical areas. The threaded front shaft is not threaded in the model, since this part is out of loading and boundary conditions and has no effect on stresses at different locations. Drilled holes on the counter weights in order to balance the crankshafts and inside threads on holes at the back of the crankshafts were not included in the models since their presence makes the geometry complicated but they do not affect stresses at critical locations. The detailed drawing of the v8 crank shaft indicating all the parts and the critical sections are shown in Figure 2.4.

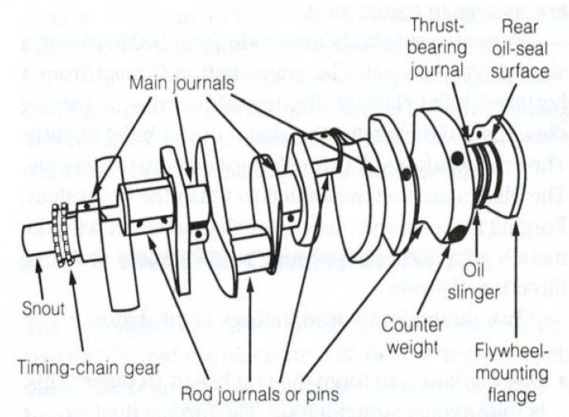

Figure-2.4: Parts of v8 crankshaft

2.2 Mesh Generation

Figure-2.5: Meshed Model of forged steel crankshaft

Tetra mesh was used to increase the stiffness and free meshing was used. It consists of 4630 elements and 6030 nodes. The element type used is solid 10 node quadratic tetrahedral. The finite element meshed model of forged steel crankshaft is shown in Figure 2.5.The global mesh size is 5mm.Convergence of stress at different locations was considered as the criterion for mesh size and number of elements selection. The material properties of the crank shaft are given in Table 1.

Table-1: Material properties for forged steel crank shaft

Sl No	Properties	Value
1	Modulus Of Elasticity	2.1×10^5 N/Mm^2
2	Poissons Ratio	0.30
3	Density	7850 Kg/ Mm^3
4	Yield Strength	250 N/Mm^2
5	Tensile Strength	460 N/Mm^2
6	Elongation (Minimum%)	18%
7	Hardness	201-255

3. LOADING AND BOUNDARY CONDITIONS

3.1 Boundary Conditions

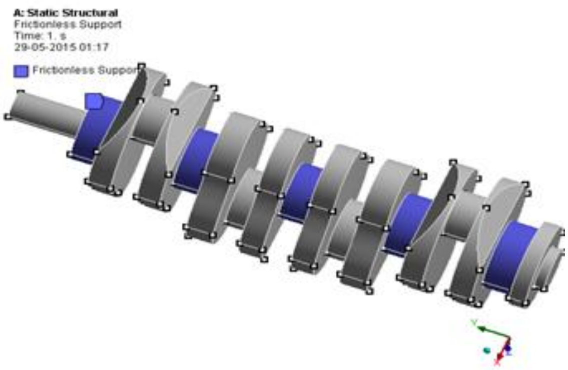

Figure-3.1: FEA Model of forged steel crankshaft showing frictionless support

The supports are frictionless. It prevents one or more flat or curved faces from moving or deforming in the normal direction relative to the face. No portion of the surface of the body can move, rotate, or deform normal to the face as shown in Figure 3.1. The surface body is free to move, rotate, and deform tangential to the face. The crankshaft is constrained with a ball bearing on both sides. The ball bearing is press fit to the crankshaft and does not allow the crankshaft to have any motion other than rotation about its main central axis.

3.2 Loads

A. Force

LOAD (FORCE) CALCULATION

Force= pressure x area
Pressure=39.3bar = 0.393N/mm^2 (standard for forged steel v8 crankshaft)
Area of crankshaft= $(\pi/4)^* d^2$
Where d= diameter of crankshaft
Since d=90mm
Force = $0.393 \text{x} (\pi /4)\ 90^2$
Force (f) = 25.01KN
Therefore total load acting on v8 crankshaft is 25KN

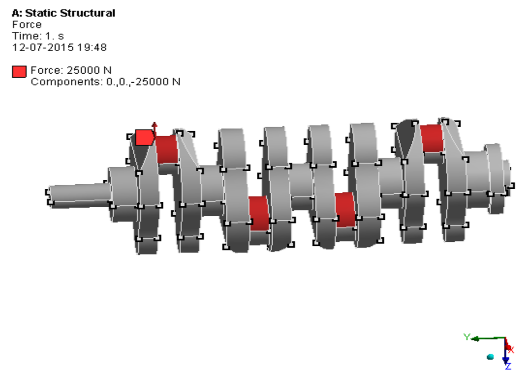

Figure-3.2: FEA Model of forged steel crankshaft showing Loads

The force vector is distributed across one or more flat or curved faces. The force vector is represented both in magnitude and direction. The components of the Load vector are as shown in Figure 3.2 after application of load.

B. Rotational Velocity

V8 engine crankshaft is mainly used in racing bikes where it rotates at high speed varying from 8000rpm to 20000rpm. The FEA model of forged steel crankshaft on which rotational velocity is applied is shown in Figure 3.3. The rotational velocity applied is 14000 rpm.

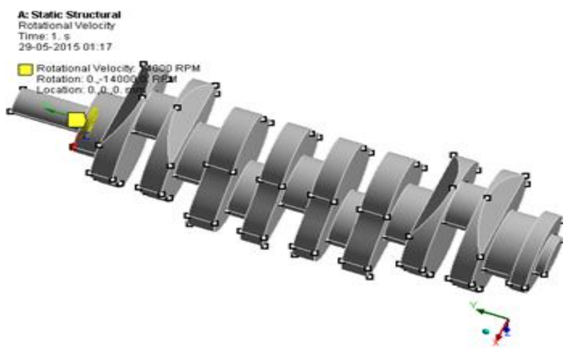

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Figure-3.3: FEA Model of forged steel crankshaft showing rotational velocity

4. RESULTS AND DISCUSSIONS

4.1 Structural Analysis Results (Static Stress Result)

A static structural analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. Steady loading and response conditions are assumed; that is, the loads and the structure's response are assumed to vary slowly with respect to time. Following results shows stress and deformations of v8 engine crankshaft. The stress obtained by static structural analysis of the forged steel crankshaft is 138.91 M Pa by applying a force of 25 KN which is well below the allowable limit of 144 M Pa. This can be seen in table 2. Hence the Crank shaft is safe under the applied static load.

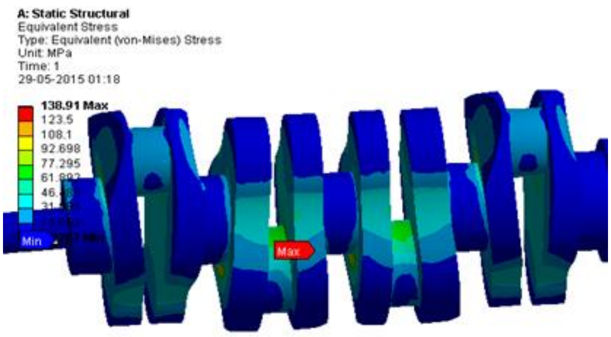

Figure-4.1: Von-mises (equivalent) stress

4.2 Fatigue Results

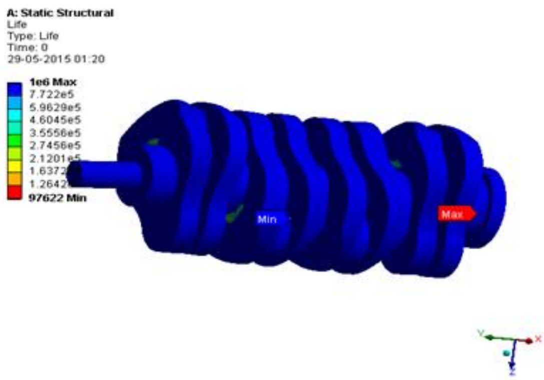

Figure-4.2: Geometry showing Fatigue life in Cycles

It is used to determine life, damage, and factor of safety of the crankshaft using a stress-life or strain-life approach. The Fatigue Tool is available only for static structural and flexible dynamic analyses. In this case, for forged steel, stress life approach is adopted and the SN curve for the material as per ANSYS reference handbook is as shown in Figure 4.2

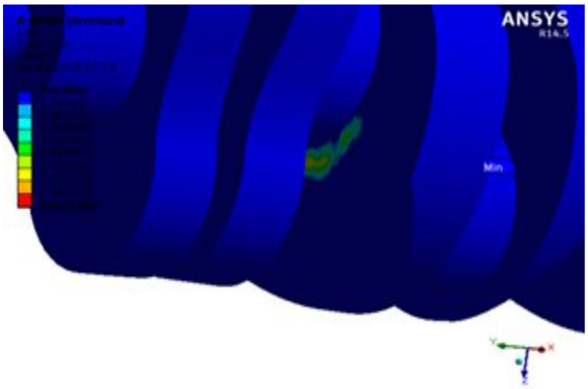

Figure-4.3: Geometry showing damage at the dislocation of crankshaft

The fatigue life was found to be 97622 cycles. It can be observed that failure takes place at the dislocation as shown in Figure 4.3 due to stress concentration. This agrees with the results of other researchers.

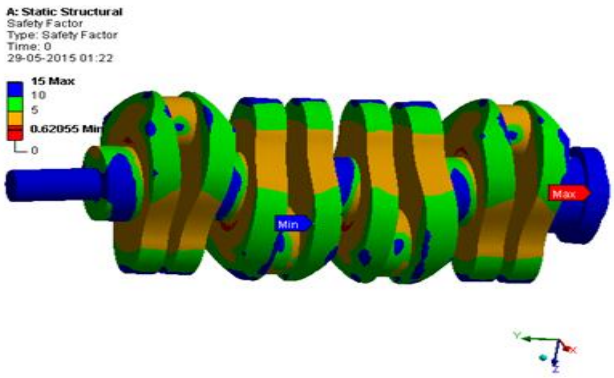

Figure-4.4: Fatigue safety factor

The portions in red colour shown in Figure 4.4 indicate the locations at which the crankshaft can fail due to factor of safety less than 1.5. In order to safe guard these sensitive locations and increase the life of the crankshaft, thickness of these portions have to be increased.

4.3 Modal Analysis

Mode shapes are dimensionless representations of the shape that the structure undergoes when vibrating at a specific frequency (corresponding natural frequency of the mode shape).Modal analysis determines the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component. It can also serve as a starting point for another, more detailed dynamic analysis The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions. The natural frequencies for modes 1 to 6 and the corresponding mode shapes are presented in the Figures 4.5 to 4.10 respectively. The maximum natural frequency obtained is 2337.5 Hz while the resonant frequency of forged steel is 2800Hz.

Table-2: Mode shapes and Natural Frequencies

Mode Shape	Natural Frequency
MODE1	148.08HZ
MODE2	148.08HZ
MODE3	1175.9HZ
MODE4	1570.1HZ
MODE5	2233.8HZ
MODE6	2337.5HZ

Hence the crankshaft is safe under the prevailing conditions of loads and boundary conditions. It was found that after 6modes there is no much variations in the natural frequency of the crankshaft. Hence analysis was stopped at 6th mode

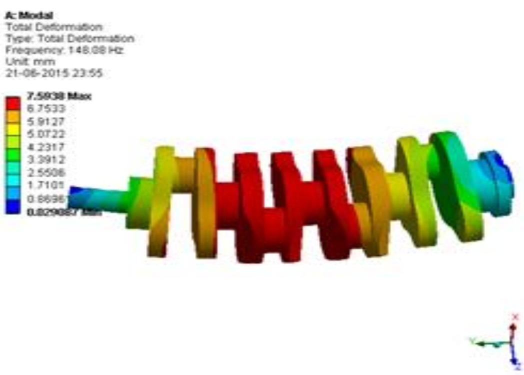

Figure-4.5: Mode 1

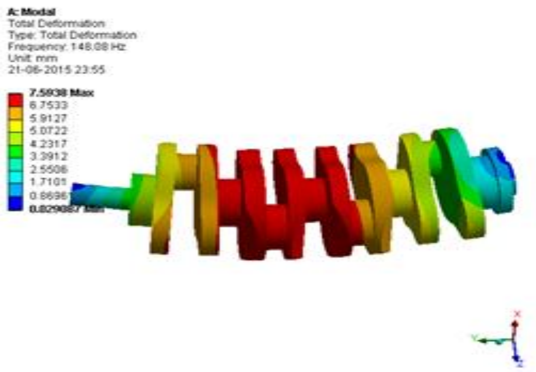

Figure-4.6: Mode 2

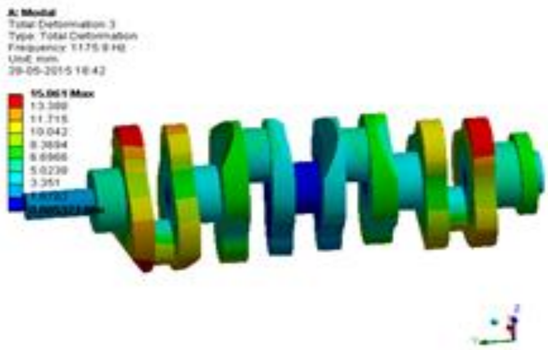

Figure-4.7: Mode 3

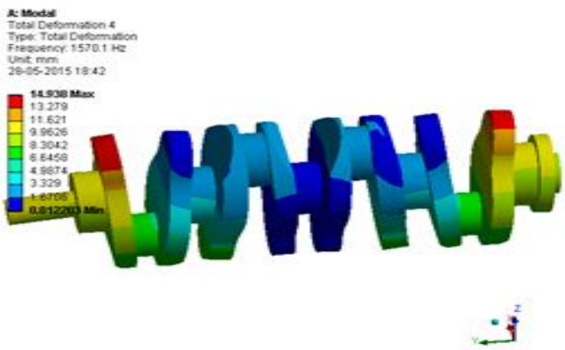

Figure-4.8: Mode 4

Figure-4.9: Mode 5

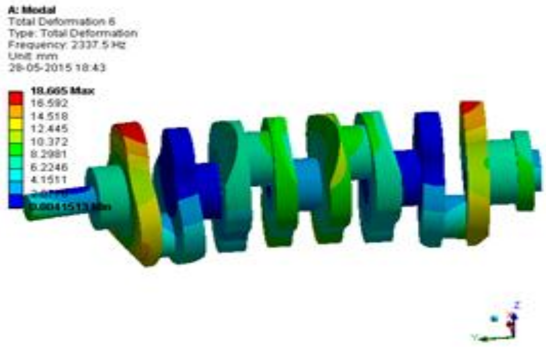

Figure-4.10: Mode 6

5. CONCLUSIONS

1. From static analysis it was observed that maximum stress is located at tip region which is around 139Mpa which is within permissible limits.
2. Critical (i.e. failure) locations on the crankshaft geometry are all located on the fillet areas because of high stress gradients in these locations, which result in high stress concentration factor.
3. Adding fillet increases the fatigue strength of the crankshaft and increases the life of the component significantly.
4. Dynamic loading analysis of the crankshaft results in more realistic and is used to determine the natural frequencies at transverse vibrations.
5. From modal analysis, the maximum natural frequency obtained is 2337.5 Hz while the resonant frequency of forged steel is 2800Hz . Hence the crank shaft is safe under given loading and boundary conditions
6. The fatigue life was found to be 97622 cycles and the failure takes place at the dislocation due to stress concentration.

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