Quasi-Dynamic Stress Analysis on Crank Shaft of Computerized Variable Compression Ratio (VCR) Diesel Engine

B.Nagaraju [#1],J. Sampath Kumar [#2], M.S.S.Srinivasarao[#3],K.Naresh Kumar[#4]

[#1] Professor ,[#2]PG student, [#3] Assistant Professor,[#4] Assistant Professor,
Department of Mechanical Engineering, Anits, Sangivalasa, V.S.P-62.

Abstract *The main objective of this study was to investigate the stresses induced in crankshaft manufactured of AISI E4330 forged steel & an aluminium alloy (7076-T6). A comparative analysis was made to study the behaviour of materials. Crank shaft is one of the most important moving part with a complex geometry in internal combustion Engine. It converts the reciprocating displacement of the piston into a rotary motion. When combustion takes place in the engine, there by high temperature and pressure will be developed inside the engine cylinder. Due to high speed and at high loads, the piston is subjected to large structural stresses, which influences on the crank.*

Experimentation was carried out on a Computerized Variable Compression Ratio (VCR) Diesel Engine Test Rig at Compression ratio of 16.5 for obtaining he results. The obtained results were tabulated for knowing pressure at various crank angles. The results were analysed by drawing the Pressure vs. crank angle variation diagram. The dynamic analysis was carried out by developing the equations of equilibrium from the Free Body Diagrams of individual components of Slider-Crank Mechanism. The forces induced at the pin joints and inertia forces obtained from the dynamic analysis were used as input for further analysis.

A three dimensional model of diesel engine crankshaft is developed by using SOLID WORKS software. And further analysis is carried out by using ANSYS WORKBENCH 15.0 software. Dynamic analysis parameter solving by MAT Lab. These reaction forces are applied along with boundary conditions on the FE model of crank shaft. The stress analysis was performed at critical crank angles of rotation.

Finite element analysis which consists (1) stress analysis (2) Modal analysis and (3) fatigue analysis.

The structural analysis involves determination of induced stresses and deformation for all crank angles at crank pin and bearing supports. The results in the form of von-Mises stresses and deformation were determined for both materials.

Further the crank shaft is also subjected to modal and fatigue analysis for maximum load condition 365^0, from the modal analysis observed different mode shapes of crank shaft and also factor of safety observed from the fatigue analysis.

The results obtained from the structural analysis shows that the stresses induced at crank pin and bearing supports in aluminium alloy (7076-T6) are lesser in comparison with AISI E4330 forged steel for various different crank angles.

The results obtained from modal analysis were inferred that aluminium alloy (7076-T6) exhibits lesser frequency in comparison with AISI E4330 forged steel for different mode shapes.

The fatigue analysis carried out to know the factory of safety of the two materials for at 10^6 cycles. The comparative results shows that the aluminium alloy (7076-T6) exhibits better results in comparison with AISI E4330 forged steel.

Keywords — *Dynamic analysis, SOLID WORKS, ANASYS, Forged steel& Aluminium Alloy.*

I. INTRODUCTION

Crank shaft is a large component with a complex geometry in the I.C engine, which converts the reciprocating displacement of the piston to a rotary motion with a four bar link mechanism. Crankshaft consisting of shaft parts, two journal bearings and one crankpin bearing. The Shaft parts which revolve in the main bearings, the crank pins to which the big end of the connecting rod are connected, the crank arms or webs which connect the crank pins and shaft parts. In addition, the linear displacement of an engine is not smooth; as the displacement is caused by the combustion chamber therefore the displacement has sudden shocks. The concept of using crankshaft is to change these sudden displacements to as smooth rotary output, which is the input to many devices such as generators, pumps and compressors. It should also be stated that the use of a flywheel helps in smoothing the shocks. Crankshaft experiences large forces from gas combustion. This force is applied to the top of the piston and since the connecting rod connects the piston to the crank shaft, the force will be transmitted to the crankshaft. The magnitude of the forces depends on many factors which consist of crank radius, connecting rod dimensions, weight of the connecting rod, piston, piston rings, and pin. Combustion and inertia forces acting on the crankshaft. 1. Torsional load &2. Bending load. Crankshaft must be strong enough to take the downward force of the power stroke without excessive bending so the reliability and life of the internal combustion engine depend on the strength of the crankshaft largely. The crank pin is like a built in beam with a distributed load along its length that varies with crank positions. Each web is like a cantilever beam subjected to bending and twisting. 1. Bending moment which causes tensile and compressive stresses. 2. Twisting moment causes shear stress. There are many sources of failure in the engine one of the most common crankshaft failure is fatigue at the fillet areas due to the bending load causes by the combustion. The moment of combustion the load from the piston is transmitted to the crankpin, causing a large bending moment on the entire geometry of the crankshaft. At the root of the fillet areas stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crank initiation leading to fracture.

An extensive literature review on crankshafts was performed by Zoroufi and Fatemi (2005) [1]. Their study presents a literature survey focused on fatigue performance Evaluation for forged steel. Crankshaft specifications, operation conditions, and various failure sources are discussed. Their survey included a review of the effect of influential parameters such as residual stress on fatigue behaviour and methods of inducing compressive residual stress in crankshafts. The common crankshaft material and manufacturing process technologies in use were compared with regards to their durability performance.

An analytical investigation on bending vibrations was done for a V6 engine by Mourelatos (1995) [4] he used a crankshaft system model (CRANKSYM) to analytically verify a vibration problem related to the flywheel for the mentioned crankshaft. As described in their study, CRANKSYM could perform an analysis considering the crankshaft structural dynamics, main bearing hydrodynamics and engine block flexibility.

Uchida and Hara (1984)[5] used a single throw FEM model. The extrapolation of the experimental equation in their study. In their study the web Thickness of a 60° V-6 crankshaft was reduced while maintaining its fatigue performance and durability under turbo-charged gas pressure. In the study of the 60° V-6 engine crankshaft dimensions, the stress evaluation of the crankpin fillet part was extremely critical since, to reduce the crank's total length, it was necessary to reduce the thickness of the web between main journal and crankpin as much as possible while maintaining its Strength.

II. PROBLEM DEFINITION

A. Statement of problem

The kinematic and dynamic analysis of a crank shaft was carried out using output of a test conducted on computerized VCR kirloskar diesel engine equipment with pressure transducer. Normally, the stress analysis is to be carried out at all analysis of rotation. However since this process is time consuming and requires lot of computational methodology. Hence, the analysis is limited to specific positional of crank angle of 365^0 and such analysis is termed as quasi-dynamic analysis.

B. Materials of Construction

In the present study, the materials considered for the crank shaft are AISI E4330 forged steel & an aluminium alloy (7076-T6). The mechanical properties of these materials are reported in the table I

TABLE I : Material properties

S.No	Property	Material	
		Forged Steel (AISIE4340)	**Aluminium Alloy** (7076-T6)
1	Density	7820 kg/m^3	2810kg/m^3
2	Poisson's ratio	0.3	0.33
3	Young's modules	206.8 Gpa	71.7Gpa
4	Yield strength	800Mpa	503Mpa
5	Ultimate strength	985.7 Mpa	572Mpa

III. SOLUTION METHODOLOGY

A. Derivation of equations for kinematic and dynamic analysis of slider crank mechanism

1. Kinematic analysis

Kinematic Analysis involves determination of linear displacement, linear velocity, linear acceleration of piston and angular displacement, angular velocity and angular acceleration of connecting rod as show in Fig.3.1

It is assumed that the crankshaft rotates at a constant angular velocity.

Fig. 3.1 slider crank mechanism

Kinematics of Piston:

Displacement of piston (x_P)=$r_1[(1-cos\theta)+(sin^2\theta)/2n]$

Velocity of piston (v_p) = $\omega_1 r_1[sin\theta+(sin2\theta)/2n]$

Acceleration of piston (a_p) = $\omega_1^2 r_1[cos\theta + (cos2\theta)/n]$

Kinematics of connecting rod:

Angular velocity of connecting rod (ω_2) =$\omega_1$cos$\theta/\sqrt{(n^2 - sin^2\theta)}$

Angular acceleration of connecting rod (α_2) = {-$\omega^2 sin\theta(n^2 - 1)\}/(n^2 - sin^2\theta)^{3/2}$

Kinematics of Crank:

Angular velocity of Crank (ω_1) = 2πN/60

2. Dynamic analysis

Taking the Kinematic parameters and pressure force acting on piston into consideration as input parameters, the Dynamic analysis of total mechanism was carried out. The first step in this direction is to draw the free body diagram of each of the members and identify all the forces which include the reactive forces of the constraints, inertia forces, weight of members and also external forces acting on them. Next step is to write the Equations of equilibrium for of the members separately as show in Fig 3.2 to3.4.

Dynamics of piston

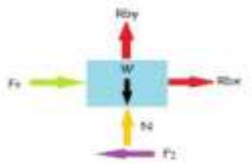

Fig. 3.2 FBD of piston

The equations of equilibrium are

$\sum \boldsymbol{F_x}$= 0, R_{bx}+ F_p - F_I = 0=>R_{bx} = F_i - F_p ⟶

(1)

$\sum \boldsymbol{F_y}$ = 0, N + R_{by} – w =0 =>N=$-R_{by}$ + w ⟶

(2)

Dynamics of crank:

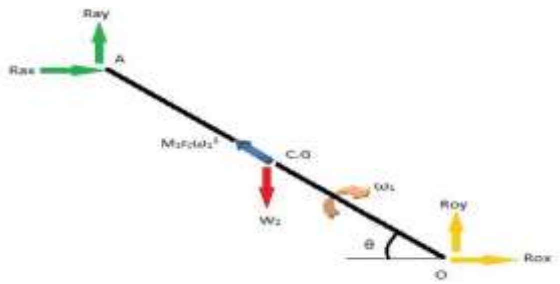

Fig. 3.3 FBD of Crank

The equations of equilibrium are

$\sum \boldsymbol{F_x}$= 0

$R_{ox} + R_{ax} - m_1\omega_1^2 r_c cos\theta$= 0

=>$R_{ox} = m_1\omega_1^2 r_c cos\theta$-$R_{ax}$ ⟶

(3)

$\sum \boldsymbol{F_y}$ = 0

R_{oy}+R_{ay}+$m_1\omega_1^2 r_c sin\theta - w_1$= 0

=>$R_{oy} = w_1$-$m_1\omega_1^2 r_c sin\theta$-$R_{ay}$ ⟶

(4) (4)

At point A:

Axial force= $R_{ax}\ cos\theta - R_{ay}\ sin\theta$

Normal force= $R_{ax}\ sin\theta + R_{ay}\ cos\theta$

At point O:

Axial force= $R_{ox}\ cos\theta - R_{oy}\ sin\theta$

Normal force= $R_{ox}\ sin\theta + R_{oy}\ cos\theta$

Dynamics of connecting rod:

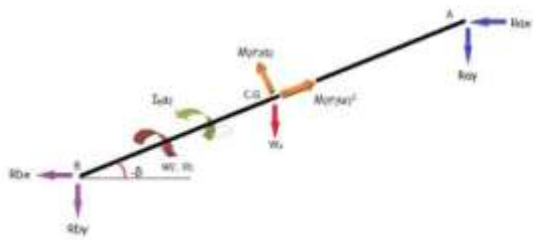

Fig. 3.4 FBD of Connecting Rod

The equations of equilibrium are

$\sum \boldsymbol{F_x}$ = 0

-R_{ax}-R_{bx}-(-$m_2\alpha_2 r_2 sin\beta) + m_2\omega_2^2 r_2 cos\beta = 0$

R_{ax}+$R_{bx} = m_2\alpha_2 r_2 sin\beta + m_2\omega_2^2 r_2 cos\beta$

$R_{ax} = m_2\alpha_2 r_2 sin\beta + m_2\omega_2^2 r_2 cos\beta$-$R_{bx}$ ⟶

(5)

$\sum \boldsymbol{F_y}$ = 0

-$R_{ay} - R_{by} - w_2 - m_2\omega_2^2 r_2 sin\beta + m_2\alpha_2 r_2 cos\beta = 0$

$R_{ay} + R_{by}$ = -

$m_2\omega_2^2 r_2 sin\beta + m_2\alpha_2 r_2 cos\beta - w_2$

$=> R_{by} = m_2\omega_2^2 r_2 sin\beta + m_2\alpha_2 r_2 cos\beta - w_2 - R_{ay}$ (6)

$\sum M\text{-}I\alpha = 0$

Considering moments about B i.e., $\sum M_b\text{-}I_b\alpha = 0$

$$-m_2\alpha_2 r_2^2 - R_{ax} sin\beta l_2 + R_{ay} cos\beta l_2 + w_2 cos\beta r_2 - I_b\alpha_2 = 0$$

$$-m_2\alpha_2 r_2^2 + R_{ay} cos\beta l_2 + w_2 cos\beta r_2 - I_b\alpha_2 + R_{ay} cos\beta l_2 + w_2 cos\beta r_2$$

$=0$ $R_{ay} = (m_2\alpha_2 r_2^2 + I_b\alpha_2 - 2w_2 cos\beta r_2 - R_{ay} cos\beta l_2) / cos\beta l_2$

(7)

At point B:

Axial force = $R_{bx} cos\beta + R_{by} sin\beta$

Normal force = $-R_{bx} sin\beta + R_{by} cos\beta$

At point A:

Axial force = $R_{ax} cos\beta + R_{ay} sin\beta$

Normal force = $-R_{ax} sin\beta + R_{ay} cos\beta$

The above seven equations 1, 5, 7,6,3,4 and 2 are solved simultaneously in the

Specified sequence for the forces R_{bx}, R_{ax}, R_{ay}, R_{by}, R_{ox}, R_{oy} and N.

B. Specifications of Engine Components:

Compression Ratio = 16.5
Mass of piston (m) = 0.728 Kg
Mass of connecting rod (m_2) = 1.961 Kg
Mass of crank (m_1) = 13.5 Kg
Weight of piston (W) = 7.14168 N
Weight of connecting rod (W_2) = 19.23741 N
Weight of crank (W_1) = 132.435 N
Radius of crank (r_1) = 0.055 m
Radius of crank from C.G (r_c) = 0.011776 m
Angular acceleration of crank (ω_1) = 157.079 rad/s
Length of connecting rod (l_2) = 0.2304 m
Slenderness ratio (n) = 4.1891
Radius of connecting rod (r_2) = 0.134627
Weight of fly wheel (W_f) = 686.7 N
Inertia force of the crank shaft(I_c)=3922.57 N

C. Variable Compression Ratio Diesel Engine:

The setup consists of single cylinder, four stroke Variable compression ratio diesel engine connected to eddy current dynamometer for loading the compression ratio change without stopping the engine and without alerting the combustion chamber geometry by specially designed tilting cylinder block arrangement. Setup is provided with necessary instruments for combusting pressure and crank-angle measurement. These signals interfaced to computer through engine indicator for p-v diagrams. The set up enables to study for VCR engine performance for brake power, indicated power, frictional power, BMEP, IMEP, brake thermal efficiency, indicated thermal efficiency, mechanical efficiency, volumetric efficiency, specific fuel consumption, heat balance. The test rig specifications are reported in the Table II.

TABLE II : Test rig specifications

Features	Specifications
Load	23.86 N-M
Speed	1500 rpm
Fuel rate	1.46-2.06 kg/hr
Air rate	16.20-17.08 m^3/hr
Water Flow	40.80 cc/sec
Cooling Water inlet Temp	26.70 ^{0}C
Cooling Water outlet Temp	30.80^{0}C

D. Experimental Results of Test Conducted VCR Diesel Engine at Compression Ratio 16.5:

TABLE III: VCR RESULTS

Crank angle(degrees)	Pressure(bar)	Volume(cc)
365	51.3	36.98

Using the above values the PV-diagram is plotted

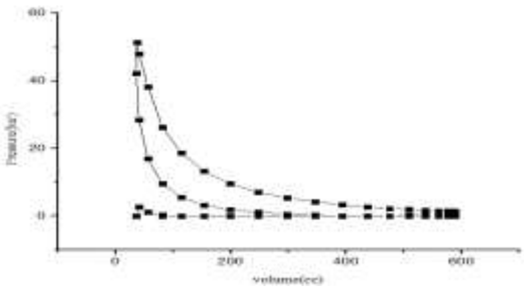

Fig 3.6 PV-diagram at Compression Ratio 16.5

Dynamic results are obtained from MAT Lab are shown in table3.2

TABLE IV: Results of dynamic analysis

crank angles (Degree)	axial force @ bearing (N)	normal force@ bearing (N)	axial force@ crank pin(N)	normal force@ crank pin(N)
365	-20957	-1879.5	24871	2011.4
490	172.95809	-5796.4794	3651.6183	5711.0031
540	5704.0810	-106.8108	-1777.8448	-25.62320209
590	-318.04713	5090.4744	4345.5840	-5175.18078

IV MODELLING OF CRANK SHAFT

A three dimensional model of diesel engine crankshaft is developed by using SOLID WORKS software as shown in Fig. 4.1.

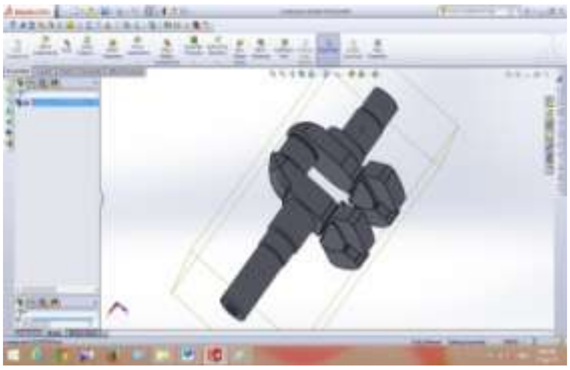

Fig. 4.1 solid works model of a CRANK SHAFT

V FINITE ELEMENT METHOD

The finite element method is numerical analysis technique for obtaining approximate solutions to a wide variety of engineering problems. Because of its diversity and flexibility as an analysis tool, it is receiving much attention in engineering colleges and industries.

A. Results of Analysis

The stress analysis is carried at various critical angles of 365^0, 490^0 , 540^0 and 590^0 by considering the Rax, Ray, Rox & Roy forces at crank pin and bearings supports.

The results obtained for von-Mises stresses, total deformation and equivalent elastic strain of forged steel and aluminium are reported in the Fig 5.1 to 5.3 and 5.4 to 5.6 by considering the loads applied at crank pin and bearing supports are constrained. Similarly the same results are reported in the Fig 5.7 to 5.9 and 5.15 to 5.17 for forged steel and aluminium alloy by considering the loads applied at bearing supports and crank pin is constrained.

The modal analysis is carried out for different mode shapes at crank angle of 365^0 and the loads applied at bearing supports and crank pin is constrained. At the same condition, the fatigue analysis was carried out and determined the factor of safety for both materials. The results obtained from the modal analysis was reported in the Fig 5.10 to 5.13 for forged steel, whereas from Fig from 5.18 to 5.21 for aluminium alloy. The results obtained for fatigue analysis for both materials are reported in the figures from 5.14 and 5.2.

1. Crank angle 365^0 :

1.1Forces applied at crank pin and bearings are constrained

Forged steel

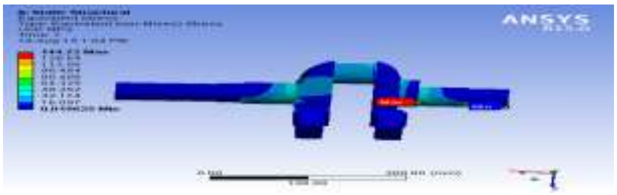

Fig. 5.1: Equivalent Von-Mises stress

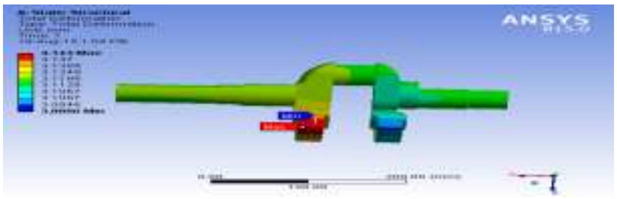

Fig.5.2: Total Deformation.

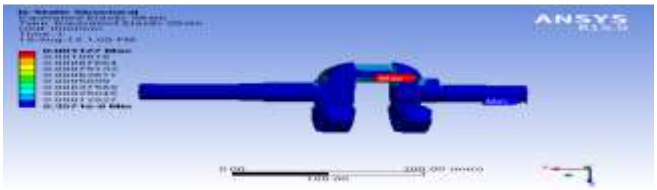

Fig 5.3: Equivalent elastic strain

Aluminium alloy

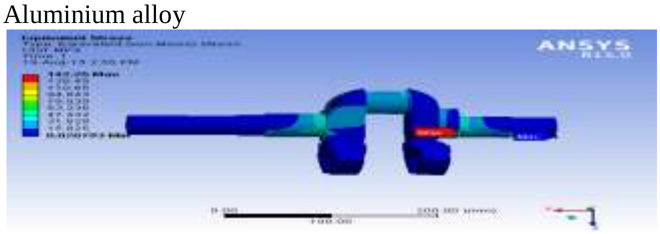

Fig 5.4: Equivalent Von-Misses stress

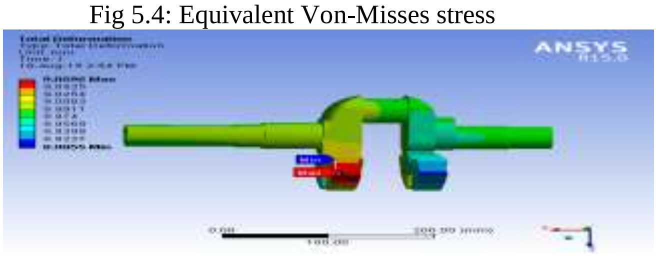

Fig 5.5: Total Deformation

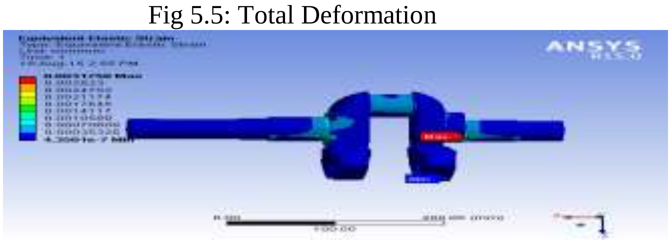

Fig 5.6: Equivalent elastic strain

2.Forces are applied at bearings and crank pin is constrained

Forged steel

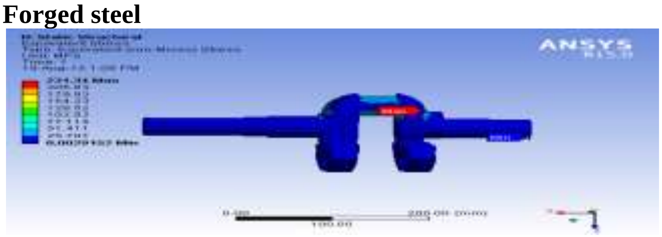

Fig 5.7: Equivalent Von-Mises stress

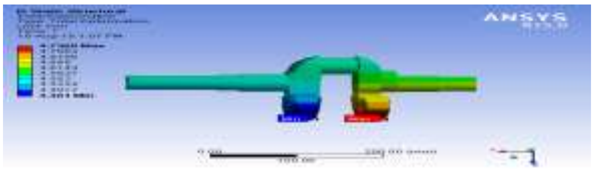

Fig 5.8: Total Deformation.

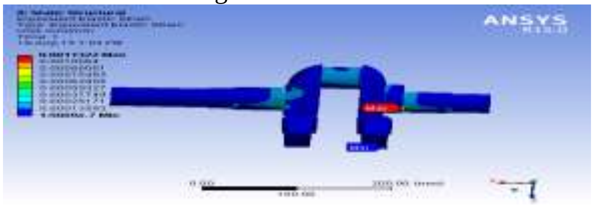

Fig 5.9: Equivalent Von-Misses stress

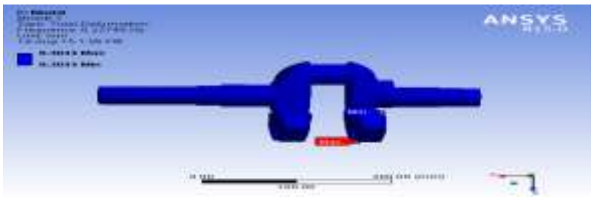

Fig 5.10: Mode shape1

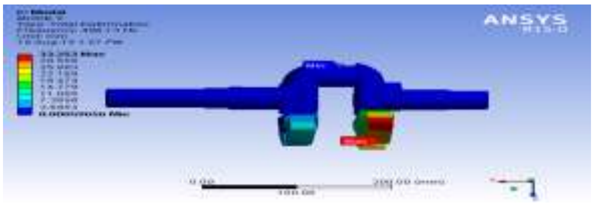

Fig 5.11: Mode shape 2

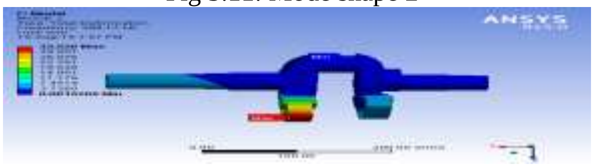

Fig 5.12: Mode shape 3

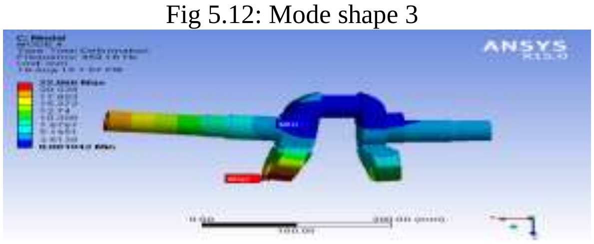

Fig5.13: Mode shape 4

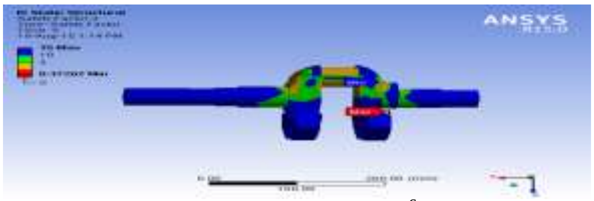

Fig 5.14: Factory of Safety(10^6 cycles)

Aluminium alloy

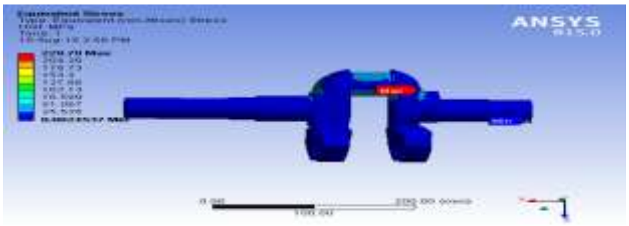

Fig 5.15: Equivalent stress

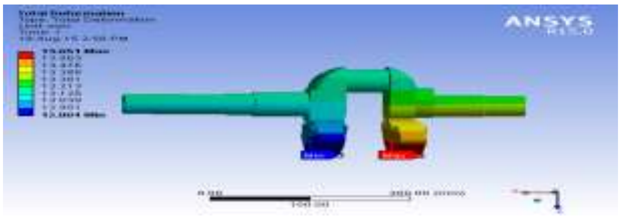

Fig 5.16: Total Deformation

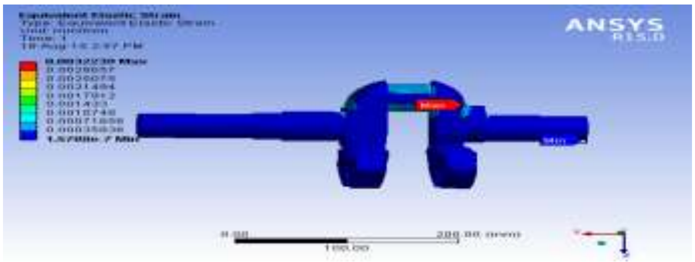

Fig 5.17: Equivalent elastic strain

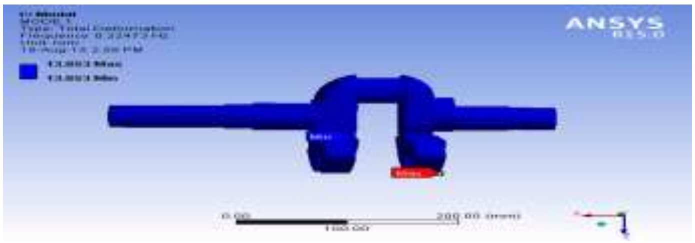

Fig 5.18: Mode shape 1

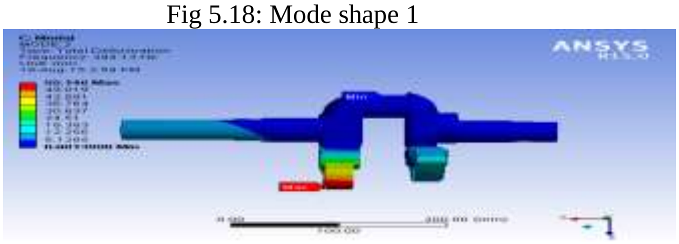

Fig 5.19: Mode shape 2

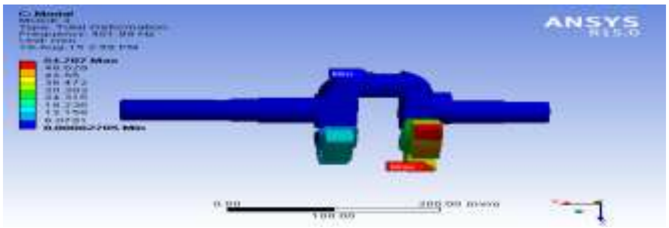

Fig 5.20: Mode shape 3

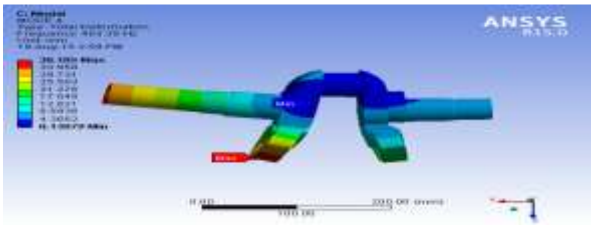

Fig 5.21: Mode shape 4

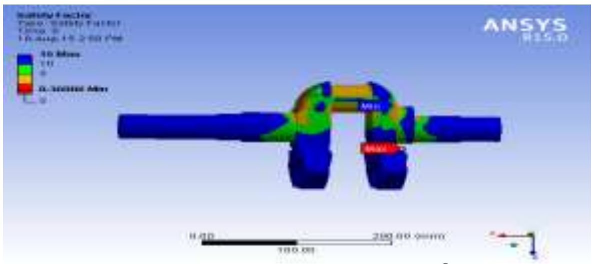

Fig 5.22: Factory of Safety(10^6 cycles)

In all case (Fig 5.23 and 5.24) Compared with parameters for both materials , Equivalent Stress are decreases but Equivalent Elastic Strain &Todal Deformation are higher in Aluminium alloy

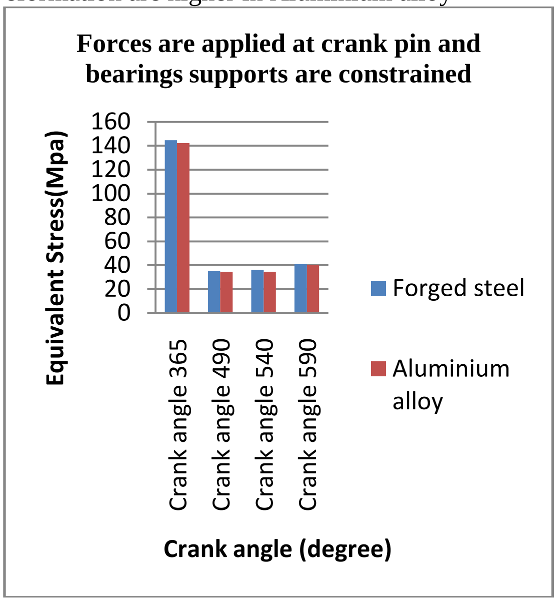

Fig 5.23 Equivalent stress

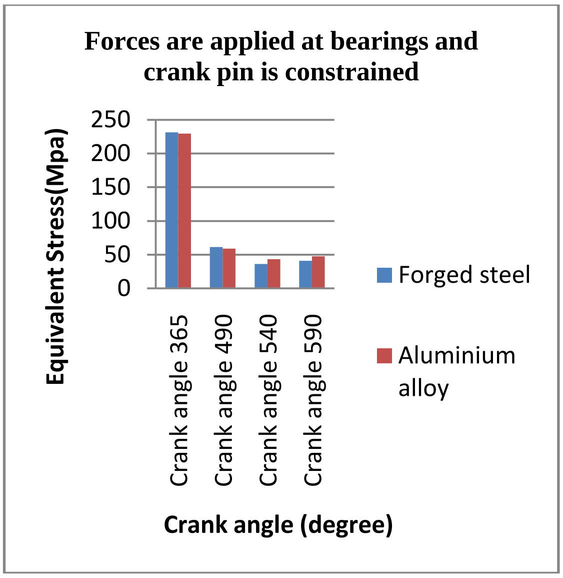

Fig 5.24 Equivalent stress

TABLE V: Factory Of Safety (10^6 cycles)

Materials	Factory Of Safety (10^6 cycles)
Forged steel	0.37
Aluminium alloy	0.36

It was inferred from the Table V, Factory Of Safety for both materials is almost equal

V. CONCLUSIONS

The present work of stress analysis was carried out at compression ratio of 16.5. The following conclusions can be drawn from the results obtained.

1. Ideally, the stress analysis has been carried out at various angles of crankshaft rotation. However, it was found that this was required after studying the variation of axial and normal force on crank shaft. Hence, the analysis was restricted to few positions and therefore the title named as—QUASI-DYNAMIC STRESS ANALYSIS.
2. The effect of mesh on the results was also included in the study. The analysis shows that fine mesh with advanced features is in better option for stress analysis as the crank shaft very intricate in shape.
3. The stress analysis was carried at critical angles of 365^0, 490^0, 540^0 and 590^0 and with different loading conditions at constant compression raio of 16.5. The results obtained from the structural analysis shows that the stresses induced at crank pin and bearing supports in aluminium alloy (7076-T6) are lesser in comparison with AISI E4330 forged steel for various different crank angles.
4. The results obtained from modal analysis was inferred that aluminum alloy (7076-T6)

exhibits lesser frequency in comparison with AISI E4330 forged steel for different mode shapes.

5. The fatigue analysis of two different materials is conducted for 10^6 cycles. The results show that the factor of safety of two different materials is found to be almost same with negligible change.
6. The results conclude that Aluminum alloy (7076-T6) exhibits better results in comparison with AISI E4330 forged steel.

REFERENCES

[1] Zoroufi, M. and Fatemi, A., "Fatigue Performance Evaluation of Forged versus Competing Process Technologies: A Comparative Study of Forged Steel versus Austempered Ductile Iron Crankshafts," A report prepared for: Forging Industry Educational and Research Foundation (FIERF) and American Iron and Steel Institute (AISI), Toledo, OH, USA, 2005.

[2] Silva, F. S., "An Investigation into the Mechanism of a Crankshaft Failure," Key Engineering Materials, Vols. 245-246, pp. 351-358, 2003.

[3] Jensen, E. J. , "Crankshaft Strength Through Laboratory Testing," SAE Technical Paper No. 700526, Society of Automotive Engineers, Warrendale, PA, USA,1970.

[4] Uchida, S. and Hara, K., "The Development of the DCI Crankshaft for the Nissan 60°-V6 Engine," SAE Technical Paper No. 841220, Society of Automotive Engineers, Warrendale, PA, USA, 1984.

[5] Mourelatos, Z. P., "An Analytical Investigation of the Crankshaft-Flywheel Bending Vibrations for a V6 Engine," SAE Technical Paper No. 951276, Society of Automotive Engineers, Warrendale, PA, USA1995.

[6] Payer, E., Kainz, A., and Fiedler, G. A., 1995, "Fatigue Analysis of Crankshafts Using Nonlinear Transient Simulation Techniques," SAE Technical Paper No. 950709, Society of Automotive Engineers, Warrendale, PA, USA,1995.

[7] Borges, A. C., Oliveira, L. C., and Neto, P. S.,"Stress Distribution in a Crankshaft Crank Using a Geometrically Restricted Finite Element Model," SAE Technical Paper No. 2002-01-2183, Society of Automotive Engineers, Warrendale, PA, USA,2002.

[8] C.M.Balamurugan, R.Krishnaraj, Dr.M.Sakthivel, K.Kanthavel, Deepan Marudachalam M.G and R.Pala," Computer Aided Modeling and Optimization of Crankshaft" ,IJSER, Volume 2, Issue 8, ISSN 2229-5518.