MODELING AND STRESS ANALYSIS OF CRANKSHAFT USING FEM PACKAGE ANSYS

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Abstract- Crankshaft is one of the critical components for the effective and precise working of the internal combustion engine. It has a complex shape of geometry. In an arbitrary position of the crank, due to tangential force, the crank arm will be subjected to transverse shear, bending and twisting, while due to radial component it is subjected to direct stress and bending. It will be laborious to consider all these straining actions in several positions of the crank. Generally, the crank is designed for two positions; those are maximum twisting moment and maximum bending moment.In this project, an attempt has been made to analyze the crankshaft in several positions of the crank, by using Finite element software ANSYS. The static analysis is conducted on the crankshaft with three different materials in different orientations. The results are validated with theoretical calculations for two crank positions for all materials.

Key Words:*Crankshaft, Solidworks, hypermesh, ANSYS Workbench, FEM, Al6061, Inconel x750*

1. INTRODUCTION

The crankshaft plays a vital role in all Internal Combustion Engines. It is a large component, which converts the reciprocating displacement of the piston into rotary motion with a four link mechanism. It has complex shape of geometry. The crankshaft experiences a cyclic load, due to the cyclic load fatigue failure occur over a period. The fatigue analysis has to be considered in the design stage itself. The design and development of crankshaft is always been an important task for the production industry, in order to reduce the manufacturing cost of the product, minimum weight possible and proper fatigue strength and other functional requirements. These improvements result in lighter and smaller engines with better fuel efficiency and higher power output. This study was conducted on a four cylinder four stroke cycle engine. Three different crankshafts from similar engines were studied in this research. The finite element analysis was performed for each crankshaft.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. At 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.

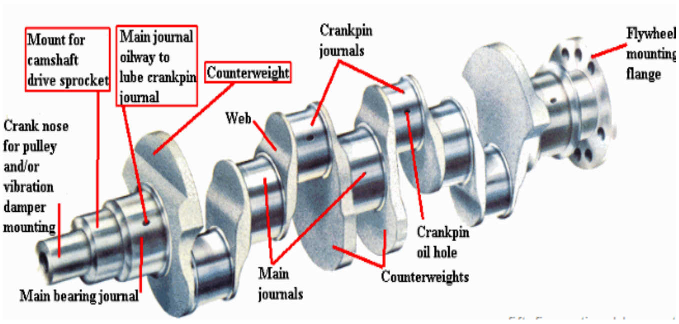

Figure 1.1: Multicylinder crankshaft

Rinkle garg and Sunil Baghl. [1] have been analyzed crankshaft model and crank throw were created by Pro/E Software and then imported to ANSYS software. The result shows that the improvement in the strength of the crankshaft as the maximum limits of stress, total deformation, and the strain is reduced. The weight of the crankshaft is reduced .There by, reduces the inertia force. As the weight of the crankshaft is decreased this will decrease the cost of the crankshaft and increase the I.C engine performance.

C.M. Balamurugan et al [2] has been studied the Computer aided Modeling and Optimization of crankshaft and compare the fatigue performance of two competing manufacturing technologies for automotive crankshafts, namely forged steel and ductile cast iron. The Three dimensional model of crankshaft were created by solid edge software and then imported to ANSYS software. The optimization process included geometry changes compatible with the current engine, fillet rolling and results in increased fatigue strength and reduced cost of the crankshaft, without changing connecting rod and engine block.

Gu Yingkui, Zhou Zhibo. [3] have been discussed a three-Dimensional model of a diesel engine crankshaft created by using PRO/E software and analytical ANSYS Software tool, it shows that the high stress region mainly concentrates in the knuckles of the crank arm & the main journal and the crank arm & connecting rod journal ,which is the area most easily broken.

Abhishekchoubey, and Jamin Brahmbhatt.[4] have been analyzed crankshaft model and 3-dimentional model of t he crankshaft were created by SOLID WORKS Software and imported to ANSYS software. The crankshaft maximum deformation appears at the centre of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journals and crank cheeks and near the central point journal. The edge of main journal is high stress area.

R. J. Deshbhratar, and Y.R Suple.[5] have been analyzed 4-cylinder crankshaft and model of the crankshaft were created by Pro/E Software and then imported to ANSYS software The maximum deformation appears at the centre of crankshaft surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks, and near the central point. The edge of main journal is high stress area. The crankshaft deformation was mainly bending deformation under the lower frequency. And the maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks.

An extensive literature review on crankshafts was performed by Zoroufi and Fatemi (2005) (6). Their study presents a literature survey focused on fatigue performance evaluation and comparisons of forged steel and ductile cast iron crankshafts. In their study, crankshaft specifications, operation conditions, and various failure sources are discussed. The common crankshaft material and manufacturing process technologies in use were compared with regards to their durability performance. In their literature review, geometry optimization of crankshafts, cost analysis and potential cost saving opportunities are also briefly discussed.

Solanki et al. [7] presented literature review on crankshaft design and optimization. The materials, manufacturing process, failure analysis, design consideration etc. were reviewed. The design of the crankshaft considers the dynamic loading and the optimization can lead to a shaft diameter satisfying the requirements of the automobile specifications with cost and size effectiveness. They concluded that crack grows faster on the free surface while the central part of the crack front becomes straighter. Fatigue is the dominant mechanism of failure of the crankshaft.

Meng et al. [8] discussed the stress analysis and modal analysis of a 4 cylinder crankshaft. FEM software ANSYS was used to analyze the vibration modal and distortion and stress status of crank throw. The relationship between frequency and the vibration modal was explained by the modal analysis of crankshaft. This provides a valuable theoretical foundation for the optimization and improvement of engine design. Maximum deformation appears at the centre of the crankpin neck surface. The maximum stress appears at the fillet between the crankshaft journal and crank cheeks, and near the central point journal. The crankshaft deformation was mainly bending deformation was mainly bending deformation under the lower frequency. Maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks. So, the area prone to appear the bending fatigue crack.

Montazersadgh and Fatemi [9] choose forged steel and a cast iron crankshaft of a single cylinder four stroke engine. Both crankshafts were digitized using a CMM machine. Load analysis was performed and verification of results by ADAMS modeling of the engine. At the next step, geometry and manufacturing cost optimization was performed. Considering torsional load in the overall dynamic loading conditions has no effect on von misses stress at the critically stressed location. Experimental stress and FEA results showed close agreement, within 7% difference. Critical locations on the crankshaft are all located on the fillet areas because of high stress gradients in these locations. Geometry optimization results in 18% weight reduction of the forged steel. Fillet rolling induces compressive residual stress in the fillet areas, which results in 165% increase in fatigue strength of the crankshaft.

2. OBJECTIVES

i) To model the crankshaft using SOLIDWORKS software
ii) To mesh the model of crankshaft using HYPERWORKS software
iii) Static analysis by using ANSYS WORKBENCH software

3. MODELING OF CRANKSHAFT

Configuration of the Engine to which the crankshaft belongs

Table 1: Engine configuration

PARAMETER	VALUE
Crank pin radius	60mm

Shaft diameter	90mm
Thickness of the crank web	50mm
Length of the crank pin	197mm
Maximum pressure	35bar

First, Prepare Assembly in Solid works for crankshaft and Save as this part as .x_t for Exporting into Hypermesh Environment. Import .x_t Model in Hypermesh Module

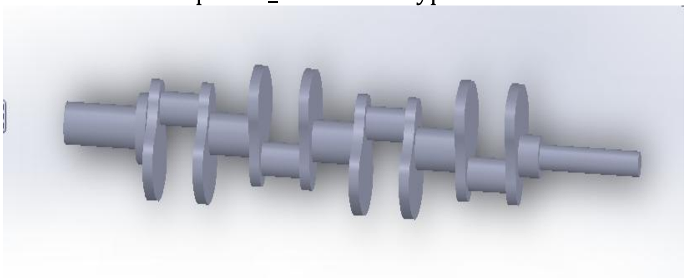

Figure 1: Model created in solidworks

4. MESHING THE MODEL:

Mesh Statics:
Type of Element: Tetrahedrons
Number of Nodes: 753229
Number of Elements: 686492

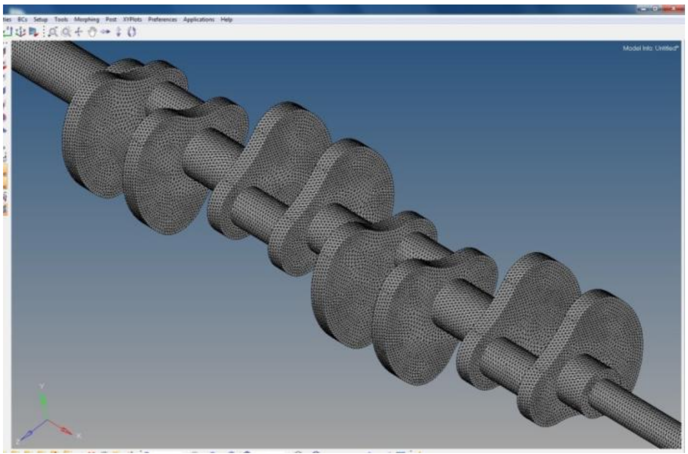

Figure 2: Meshed model of crankshaft

Save the file in .cdb format and export it into ANSYS WORKBENCH

5.MATHEMATICAL MODEL FOR CRANKSHAFT

i)Crank is at dead centre:
Bore diameter (D) =200mm
F_Q=Area of the bore ×Max.Combustion pressure
$=\pi/4\times D^2\times Pmax=109.95KN$
$\Phi=0$
$H1= F_Q/2=54.97X10^3N=H2$
Design of crankpin:
$b_1= b_2=100+50+197.2/2=248.6mm$
$M_C= H_1Xb_2=13.66x10^6$ N-mm
$M= \pi/32xd^3x\sigma_v$, d=120mm
$\sigma_v=80.56N/mm^2$
ii) Crank is at an angle of maximum twisting moment
Force on the piston:
Bore diameter (D) =200mm,
F_Q= Area of the bore ×Max.Combustion pressure
$= \pi/4\times D^2\times P_{max}=109.95KN$
In order to find the Thrust Force acting on the connecting rod (F_Q), and the angle of inclination of the connecting rod with the line of stroke (i.e. angle Ø).
Which implies
$\sin\varphi=\sin30/5=0.1$
$\varphi=5.73$
We know that thrust Force in the connecting rod,
$F_Q =F_P/\cos Ø$
From we have
Thrust on the connecting rod, F_Q =110.50KN
Thrust on the crankshaft can be split into tangential component and radial component
1. Tangential force on the crankshaft,
$F_T = F_Q \sin(\theta+ Ø) =64.5KN$
2 .Radial force on the crankshaft,
$F_R = F_Q \cos(\theta+ Ø) =89.69KN$
Reactions at bearings (1&2) due to tangential force is given by
$H_{T1}=H_{T2} = F_T/2=32.25KN$
Similarly, reactions at bearings (1&2) due to radial force is given by
$H_{R1} = H_{R2}= F_R/2=44.84KN$
Design of crankpin:
Let d= diameter of crankpin in mm
We know that bending moment at the centre of the crankshaft
$M_C=H_{R1}\times b_2=44.84X1000X248.6$
$=11.14X 10^6$ N-mm
Twisting moment on the crankpin
$(T_C)=32.25X1000X130=4.192X 10^6$ N-mm
From this we have equivalent twisting moment
$T=\sqrt{Mc^2 + Tc^2}=11.902X10^6$N-mm
Von-misses stress induced in the crankpin
$M=\sqrt{(Kb + Mc)2 + \frac{3}{4}(Kt \times Tc)2}$
$= 22.93X10^6$ N-mm
$M= \pi/32xd^3x\sigma_v$
$\sigma_v=135.19N/mm^2$
Shear stress:
$T= \pi/16d^3x\, \boldsymbol{\tau}$
$\boldsymbol{\tau}=35N/mm^2$

6. ANALYSIS

ANSYS is general-purpose Finite Element Analysis (FEA) software package. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces (of user designed size) called elements. The software implements equations that govern the behavior of these elements and solves them all; creating a comprehensive explanation of how the system acts as a whole. The ANSYS Workbench environment is an intuitive up-front finite element analysis tool that is used in conjunction with CAD systems and/or Design Model. ANSYS Workbench is a software environment for performing structural, thermal, and electromagnetic analyses. The Workbench focuses on attaching existing geometry, setting up the finite element model, solving, and reviewing results

6.1 Static Structural Analysis

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. The types of loading that can be applied in a static analysis include externally applied forces and pressures, Steady-state inertial forces (such as gravity or rotational velocity), Imposed (nonzero) displacements, Temperatures (for thermal strain).

6.2 Materials used for Crankshaft

- Structural steel
- Inconel x750
- Al6061

6.3. IMPORT INTO ANSYS WORKBENCH

After meshing in hyperworks save the file in .cdb format and export it into ANSYS WORKBENCH

Applying material for crankshaft

Material I details:

Material type:- Structural steel
Poisson ratio: - 0.3
Young's modulus:-2x10^5 MPa

Define boundary condition for analysis:

Boundary conditions play an important role in Finite Element Analysis. Here we have taken both remote displacements for bearing supports are fixed.
Then apply pressure on the crankpin as shown in the figure

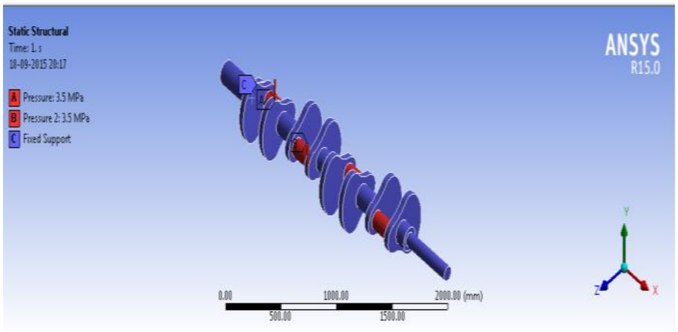

Figure 3: Apply Boundary conditions on the crankshaft

The pressure 3.5 MPa is applied on the top of the crankpin surface

Run the analysis and Get Results:

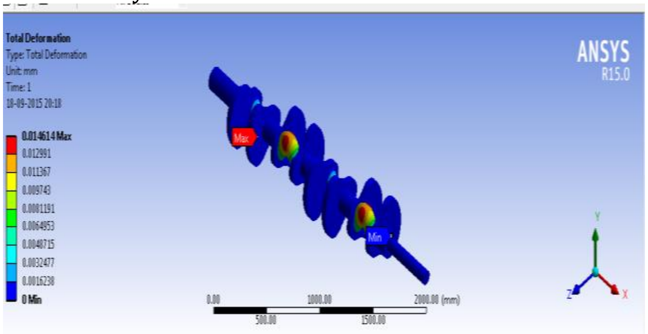

Figure 4:- Total deformation (mm)

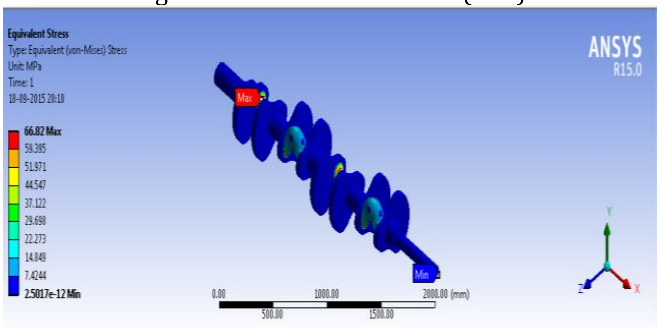

Figure 5:- crankshaft von-misses stress (MPa)

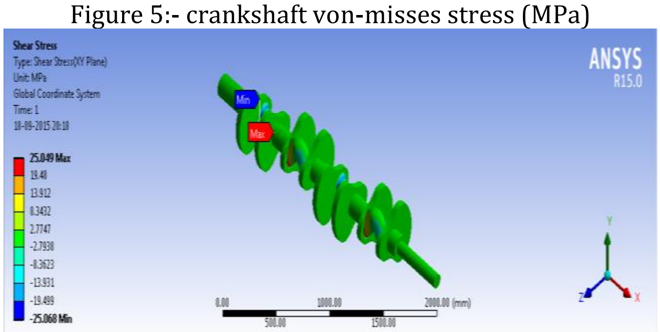

Figure 6:- crankshaft shear stress (MPa)

Material II details:

Material type: - Inconel X750
Poisson ratio: - 0.29
Young's modulus:-213.7X10^3 MPa

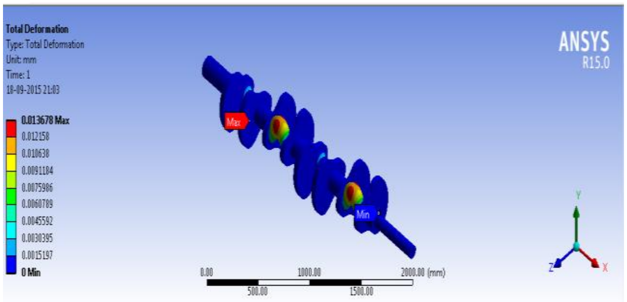

Figure 7:- Total deformation (mm)

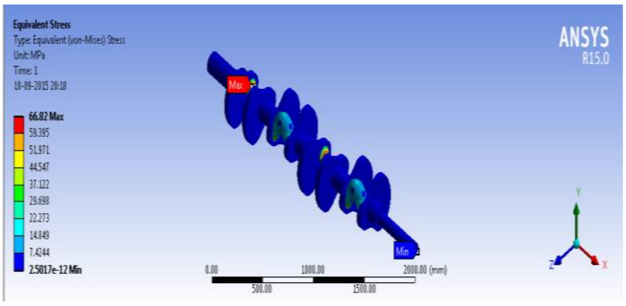

Figure 8:- crankshaft von-misses stress(MPa)

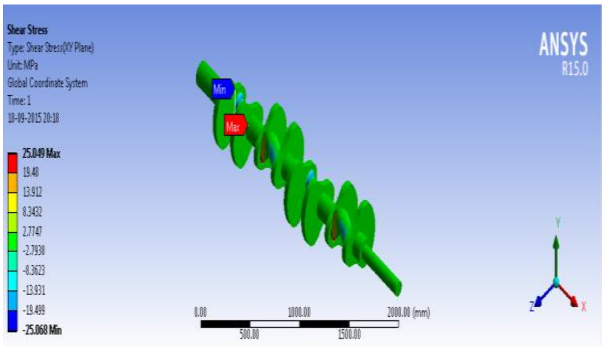

Figure 9:- crankshaft shear stress(MPa)

Material III details:
Material type: - Al 6061
Poisson ratio: - 0.33
Young's modulus:-70X10^3 MPa

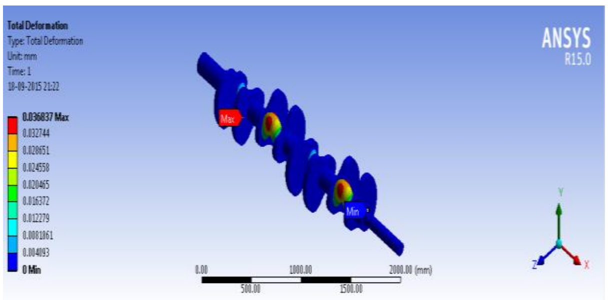

Figure 10:- Total deformation(mm)

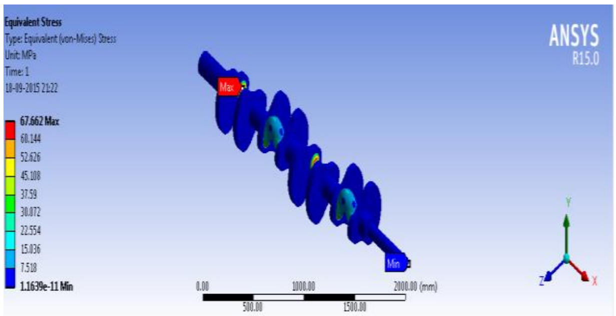

Figure 11:- crankshaft von-misses stress(MPa)

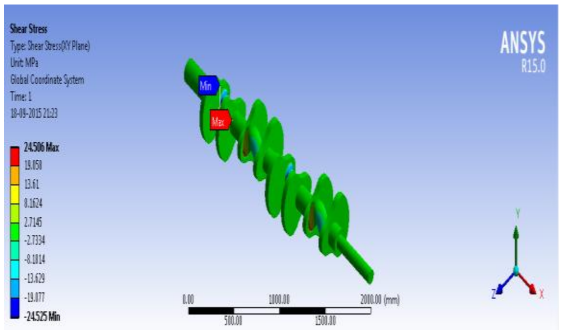

Figure 12:- crankshaft shear stress (MPa)

7. RESULTS OF STRUCTURAL ANALYSIS FOR THREE MATERIALS WHEN CRANK IS AT DEAD CENTRE

Table 2: Results of structural analysis for three materials when crank is at dead centre

Material	Total deformation(mm)	Equivalent stress(MPa)	Shear stress(MPa)
Structural steel	0.0146	66.821	25.041
Al 6061	0.0368	67.662	24.502
Inconel x750	0.0136	66.823	25.042

PRESSURE APPLIED ON THE CRANKSHAFT AT VARIOUS ANGLES:

Table 3: Pressure values at various crank angles

Crank Angle(deg)	Pressure(Bar)
0	35
30	30.3
60	17.5
90	0
120	-17.5
150	-30.3
180	-35
210	-30.3
240	-17.5
270	0
300	17.5

330	30.3
360	35

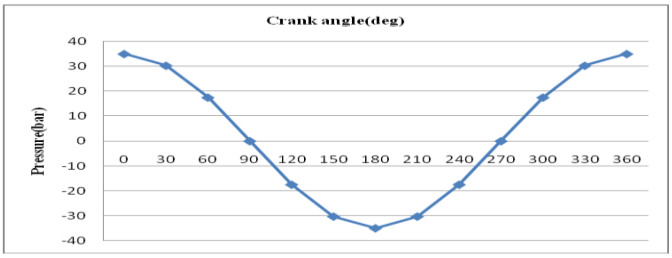

Figure 13: Variation of pressure with crank angle

8.RESULTS OF STRUCTURAL ANALYSIS FOR THREE MATERIALS WHEN CRANK IS AT VARIOUS ANGLES:

Material I**:** STRUCTURAL STEEL

Table 4: Results for structural steel at various angles

Crank Angle (Deg)	Pressure (bar)	Max Deformation (mm)	Max Vonmis ses Stress (Mpa)	Shear Stress (Mpa)
0	35	0.0146	66.82	25.06
30	30.3	0.0125	57.27	21.48
60	17.5	0.0073	33.41	12.53
90	0	0	0	0
120	-17.5	0.0073	33.41	12.53
150	-30.3	0.0125	57.27	21.48
180	-35	0.0146	66.82	25.06
210	-30.3	0.0125	57.27	21.48
240	-17.5	0.0073	33.41	12.53
270	0	0	0	0
300	17.5	0.0073	33.41	12.53
330	30.3	0.0125	57.27	21.48
360	35	0.0146	66.82	25.06

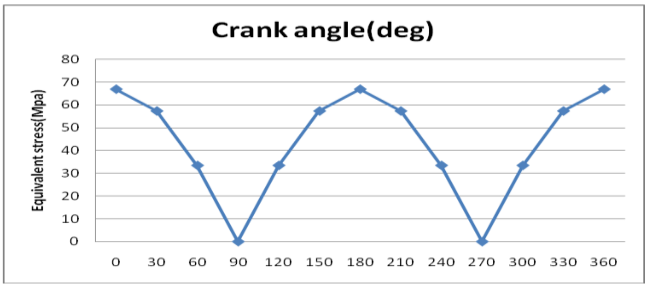

Figure 14: Variation of Equivalent stress with crank angle

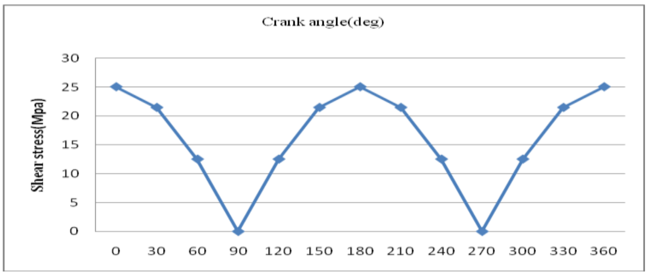

Figure 15: Variation of Shear stress with crank angle

Material II: Al 6061

Table 5: Results for Al6061 at various angles

Crank angle	Pressure	Max deformation	Max vonmis ses stress	Shear stress
		(mm)	(mpa)	(mpa)
0	35	0.025	67.662	24.525
30	30.3	0.015	57.99	21.022
60	17.5	0.012	33.831	12.263
90	0	0	0	0
120	-17.5	0.012	33.831	12.263
150	-30.3	0.015	57.99	21.022
180	-35	0.025	67.662	24.525
210	-30.3	0.015	57.99	21.022
240	-17.5	0.012	33.831	12.263
270	0	0	0	0
300	17.5	0.012	33.831	12.263
330	30.3	0.012	57.99	21.022
360	35	0.025	67.662	24.525

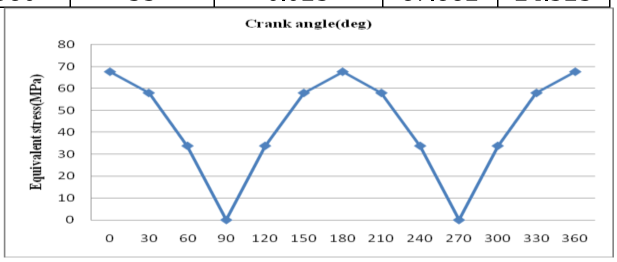

Figure 16: Variation of Equivalent stress with crank angle

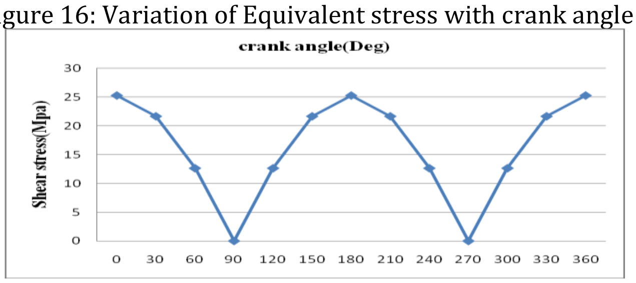

Figure 17: Variation of Shear stress with crank angle

Material III: INCONEL X750

Table 6: Results for inconelx750 at various angles

Crank angle	Pressure (bar)	Max deformation	Max vonmises	Shear stress

(Deg)		(mm)	stress (Mpa)	(Mpa)
0	35	0.0136	66.6	25.24
30	30.3	0.0116	57.093	21.63
60	17.5	0.0068	33.3	12.62
90	0	0	0	0
120	-17.5	0.0068	33.3	12.62
150	-30.3	0.0116	57.093	21.63
180	-35	0.0136	66.6	25.24
210	-30.3	0.0116	57.093	21.63
240	-17.5	0.0068	33.3	12.62
270	0	0	0	0
300	17.5	0.0068	33.3	12.62
330	30.3	0.0116	57.093	21.63
360	35	0.0136	66.6	25.24

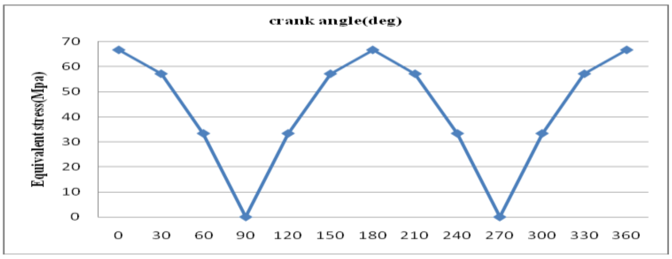

Figure 18: Variation of Equivalent stress with crank angle

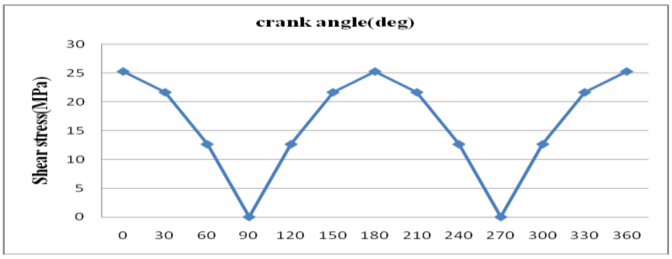

Figure 19: Variation of Shear stress with crank angle

9. VALIDATED RESULTS

Table 7: Comparison between the Theoretical and practical results

S.No	Type of stress	Theoretical Results	ANSYS Results
1	Von-misses stress(N/mm^2)	80.56	66.82
2	Shear stresses (N/mm^2)	35	25.04

10. CONCLUSION

From the above results, Al6061 is subjected to high von-misses stresses compared to remaining two materials.
Inconel x750 is subjected to little deformation when compared to remaining two materials.The maximum deformation appears at the center of crankpin neck surface.From the results it is concluded that the crankshaft design is safe since the von-misses stresses are within the limits.The maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks.

REFERENCES

[1.] Rincle Garg, Sunil Baghla, "Finite element analysis and optimization of crankshaft", International Journal of Engineering and Management Reaserch, vol-2, Issue-6, ISSN: 2250-0758, Pages: 26-31, December 2012.
2.] C.M Balamurugan, R. Krishnaraj, Dr.M.sakhivel, K.kanthavel, Deepan Marudachalam M.G, R.Palani, "Computer Aided modeling and optimization of Crankshaft", International Journal of scientific and Engineering Research, Vol-2, issue-8, ISSN:2229-5518, August-2011.
[3.] Go Lingui, Zhou Zibo, "Strength Analysis of Diesel Engine Crankshaft Based on PRO/E and ANSYS", Third International Conference on Measuring Technology and Mechatronics Automation, 2011.
[4.] Abhishek choubey, Jamin Brahmbhatt, "Design and Analysis of Crankshaft for single cylinder 4-stroke engine", International Journal of Advanced Engineering Research and studies, vol-1, issue-4, ISSN:2249-8974, pages: 88-90, July-sept 2012.
[5.] R.J Deshbhratar, Y.R Suple, "Analysis and optimization of Crankshaft using FEM", International Journal of Modern Engineering Research, vol-2, issue-5, ISSN: 2249-6645, pages: 3086-3088, Sept-Oct 2012.
[6.] Farzin H. Montazersadgh and Ali Fatemi "Stress Analysis and Optimization of Crankshafts Subjected to Dynamic Loading", AISI, August 2007.
[7]. A. Solanki, K. Tamboli, M.J, M.J, M.J, M.J. Zinjuwadia, (2011), "Crankshaft Design and Optimization- A Review" National Conference on Recent Trends in Engineering & Technology
[8]. J. Meng, Y. Liu, R. Liu, (2011), "Finite Element analysis of 4-Cylinder Diesel Crankshaft" I.J. Image, Graphics and Signal Processing, vol 5, pp. 22-29
[9]Shenoy, P. S. and Fatemi, A., 2006, "Dynamic analysis of loads and stresses in connecting rods,"IMechE, Journal of Mechanical Engineering Science, Vol. 220, No. 5, pp. 615-624