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LIFE PREDICTION OF OPTIMAL SINGLE CYLINDER ENGINE CRANKSHAFT USING FINITE ELEMENT ANALYSIS

Shimelis Mekuria Edo[*1], Mukesh Kumar[2] & Vishant Kumar Gulia[3]
[*1&2]Lecturer, Department of Mechanical Engineering, Assosa University, Ethopia
[3]Ganga Technical Campus

ABSTRACT
Crankshaft is a large volume production component with a complex geometry in the Internal Combustion Engine (ICE). The reciprocating displacement of the piston converts into a rotary motion by crank. An attempt is made in this paper to study life prediction of single cylinder engine crankshaft with geometry optimization. The modeling of the original and optimized crankshaft is created using SOLIDWORK Software. Finite element analysis (FEA) is performed to obtain maximum stress point or dangerous area, and life of original and optimized crank shaft using the ANSYS software with applying the boundary conditions. Then the results Von-misses stress and life of crankshaft is done using ANSYS software results. From result of geometry optimization parameter like increasing crankpin inner diameter and decreasing crank web thickness are changes in model of crankshaft used to predict its life. As crankpin inner diameter increases life of shaft decreases at first iteration and life of shaft increase as crank web thickness decreases in second iteration of ANSYS result.

Keywords: *Crankshaft, Life, Finite Element Analysis (FEA), Optimization.*

I. INTRODUCTION

Crankshaft is one of the most important moving parts in internal combustion engine and it is a large component with a complex geometry in the engine. In general it converts reciprocating motion of the piston is linear and is converted into rotary motion and vice versa with a four link mechanism [1].

The most common application of a crankshaft is in an automobile engine. However, there are many other applications of a crankshaft which range from small one cylinder lawnmower engines to very large multi cylinder marine crankshafts and everything in between [2]. A crankshaft consists of cylinders as bearings, plates as the crank webs and crank-pin.

Large forces from gas combustion applied on the piston in internal combustion engine. 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. Crankshaft must be strong enough to withstand 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.

Section geometry changes in the crankshaft cause stress concentration at fillet areas where bearings are connected to the webs of the crank. In addition, these component experiences both bending and torsional load during its life service. Therefore, areas of filleted portion are locations that experience the most critical stresses during the service life of the crankshaft [3].

II. METHODS AND CONDITIONS

Material
Generally, crankshafts materials should be readily shaped, machined and heat-treated, and have adequate strength, toughness, hardness, and high fatigue strength. The crankshaft used in this study is AISI1045 forged steel mostly used in agricultural machines that have single cylinder diesel engine.

Table 1 Material property of Crankshaft

Material Type	AISI1045steel
Density	7.85g/cm^3
Young's modulus	221GPa
Poisson's ratio	0.3
Yield stress	370Mpa
Tensile strength	235Mpa

Table 2 Specification of desired engine

Capacity	395cc
Number of cylinder	1
Bore x stroke	86mmx68mm
Compression ratio	18.1
Maximum power	8.1hp at 3600rpm
Maximum gas pressure	25bar

Table 3 Dimension of single cylinder engine crankshaft

parameter	symbol	Value
Web Thickness Left and Right	Wt	20.5mm
Web width Left and Right	Ww	65mm
Diameter of the shaft/journal	ds	35
Crankpin oil hole diameter	Coh	18mm
Length of the Crank Pin	Lc	32mm
Diameter of the Crank Pin	dc	39
Length of the Crank shaft	L	341mm
Crankpin fillet radius	Rh	3mm

Table 4 Parameters used in crankshaft life prediction

Parameters	Dimension
Crankpin inner hole diameter	18mm
Crank web thickness	20.5mm
Journal hole length	34mm

Force acting on crankshaft

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The crankshaft, connecting rod, and piston constitute a four bar slider-crank mechanism, which converts the sliding motion of the piston (slider in the mechanism) to a rotary motion. Since the rotation output is more practical and applicable for input to other devices, the concept design of an engine is that the output would be rotation.

Gas load from combustion chamber applied on the piston to rotate crankshaft by the help of connecting rod.

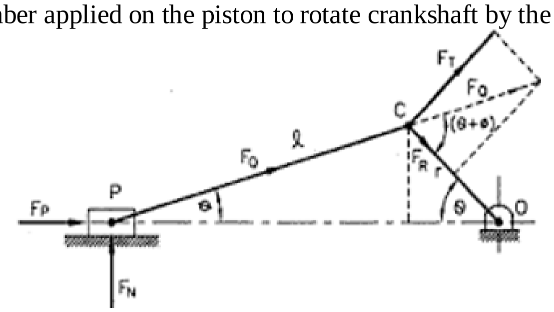

Fig 1 assembly of piston connecting rod and crank

Force on piston is calculated using the equation below

$$F_p = \frac{\pi}{4} D^2 P_{max} \qquad (1)$$

From the above equation F_p represents force of piston, D is the diameter of the piston in mm and P_{max} Maximum intensity of pressure on piston in N/mm^2. Maximum piston force is 14.52KN.
Thrust force in the connecting rod is given by

$$F_Q = \frac{F_P}{cos\emptyset} \qquad (2)$$

The maximum value of trust force on connecting rod is 14.68KN
Thrust on the crank shaft can be split into tangential component and the radial component. The tangential force or the rotating effort on the crank

$$\mathrm{FT} = \mathrm{FQ} \sin(\theta + \emptyset) \qquad (3)$$

The radial force along the crank

$$\mathrm{FR} = \mathrm{FQ} \cos(\theta + \emptyset) \qquad (4)$$

Maximum values of tangential and radial force on the crankshaft are 10.05KN and 10.68NK respectively.

Design of Crankpin

The equivalent twisting moment on the crank pin is given by

$$T_e = \sqrt{Mc^2 + Tc^2} \qquad (1)$$

Where Mc bending moment and Tc twisting moment at the center of crankshaft. The equivalent twisting moment is 360.7KN-mm.
Torsional shear stress (τ) is also calculated as equivalent torsional momentT_e.

$$\tau = \frac{16T_e}{\pi {d_o}^3 (1 - k^3)} \qquad (2)$$

From equation d_o is outer diameter of crankpin and k is the ratio of outer to inner diameter of crankpin. Shear stress (τ) becomes 44N/mm^2 at inner diameter of crankpin 18.75mm.
Equation of equivalent bending moment is written as follows

$$M_{ev} = \sqrt{(kbxMc)^2 + \frac{3}{4}(ktxTc)^2} \qquad (3)$$

Where Kb is combined shock and fatigue factor for bending (Take Kb=1) and K_t shows combined shock and fatigue factor for torsion (Take Kt =1). Finally equivalent bending moment becomes 382.2 KN-mm.
Equivalent (Von-Misses) stress, σ_v is derived from equation of equivalent bending moment is written as follows

$$\sigma_v = \frac{32Mev}{\pi {d_o}^3 (1 - k^3)} \qquad (4)$$

(Von-Misses) stress, σ_v on the crankpin is 73.8N-mm^2 when inner diameter of crankpin increased to 18.75mm.

Crank web design

Maximum bending moment on the crank web calculated as

$$M_b = H_2[\, b_2 - \left(\frac{l_c}{2} + \frac{t}{2}\right)] \qquad (1)$$

Where
Mb = Maximum bending moment on the crank web
H_2= Horizontal reactions at bearings 2 which is half of piston force
b_2 = Distance of bearing 2 from the center of crankpin
Finally bending moment on the crankweb is 297660N-mm.
And Section Modulus on the crankweb is

$$\boldsymbol{z} = \frac{w\ x\ t^2}{6} \qquad (2)$$

Where w and t shows width and thickness of crankweb.
Bending stress induced in the crank web is,

$$\sigma_b = \frac{M_b}{Z} \qquad (3)$$

When the thickness of crankweb is 19.2, bending stress induced in the crank web is 73.3N/mm^2.

The geometry changes on desired parameters can reduce the total weight of crankshaft. The weight reductions achieved by individual component, i.e., crankweb, journal and crankpin 4.15%.

From design of crankpin and crankweb the induced stress is less than design stress of material which is 75Mpa. So the design is safe.

III. CRANKSHAFT MODEL AND FINITE ELEMENT ANALYSIS

Crankshaft Model
The model of crankshaft is created by solidwork2014 software.
The dimension used to draw a model was taken from engine specification chart.
The figure 3.1 below shows the original single cylinder crankshaft model before modification.

Finite Element Analysis of Single Cylinder Crankshaft
For complex geometry of a given part it is difficult to analyze the finite element analysis manually. Hence, for this research work ANSYS Workbench was used. The CAD model of a single cylinder crankshaft created by solidwork and was imported to ANSYS Workbench to start the simulation step by step.

Meshed model of Crankshaft

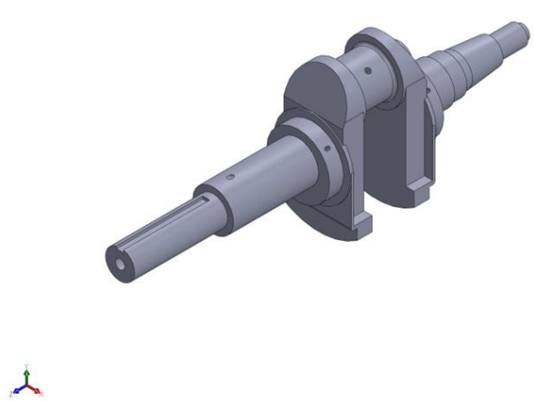

Figure 2 crankshaft model 1

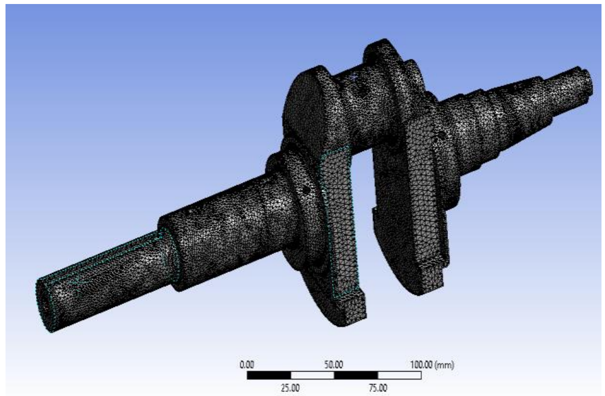

Figure 3 Meshed model of Crankshaft

As number of element increases (or element size decreases) the stress values predicted by FEM approach to exact value. The effect of force on each portion of the component is not same. The purpose of meshing is to perform the analysis on each small division separately. In this research, static structure analysis is used. The type of element used is tetrahedron and no of elements are 75196 when element size is 1.2mm.

Loading and Boundary Conditions

Boundary Conditions

Crankshaft is a constrained with a ball bearing from one side and with a journal bearing on the other side. The ball bearing is press fitted to the crankshaft and does not allow the crankshaft to have any motion other than rotation about its main axis. Since only 180 degrees of the bearing surfaces facing the load direction constraint the motion of the crankshaft, this constraint is defined as a fixed semicircular surface equal to ball bearing's width. The other side of the crankshaft is constrained with a journal bearing. Therefore, this side was modeled as a semicircular edge facing the load at the bottom of the fillet radius fixed in a plane perpendicular to the central axis and free to move along central axis direction.

The distribution of load over the connecting rod bearing is uniform pressure on 120 degree of contact area. Since the crankshaft is in interaction with the connecting rod, the same loading distribution will be transmitted to the crankshaft.

Loading

In combustion chamber maximum load is applied to crankshaft when a piston is at top dead center. It must be strong enough to take the downward force during power stroke without excessive bending. So the reliability and life of internal combustion engine depend on the strength of the crankshaft largely. In ANSYS the load is applied perpendicular to the surface of crankpin to represent a piston load at top dead center.

In this study a pressure of 2.5 MPa from Table 2.2 is applied on the crankpin at top dead center position of the piston. From the figure below red color arrow on crankpin is assumed to be maximum load acting on the crankshaft at top dead center.

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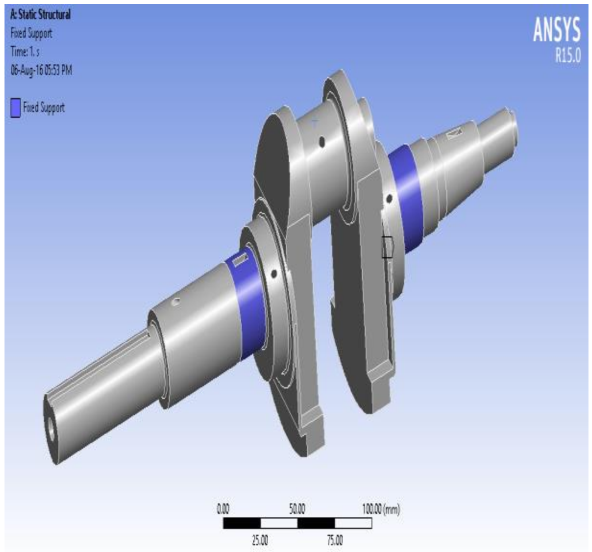

Figure 4 Boundary conditions of Crankshaft

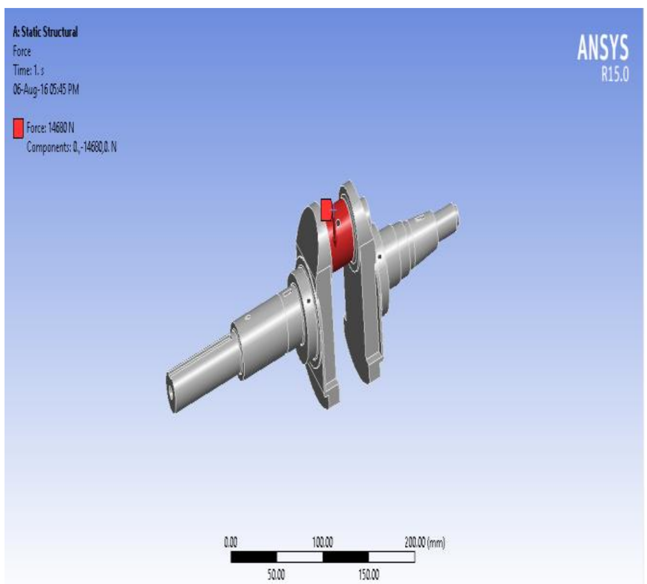

Figure 5 14680N force applied on Crankpin of Shaft

Stress Analysis at Variable Crankpin inner diameter

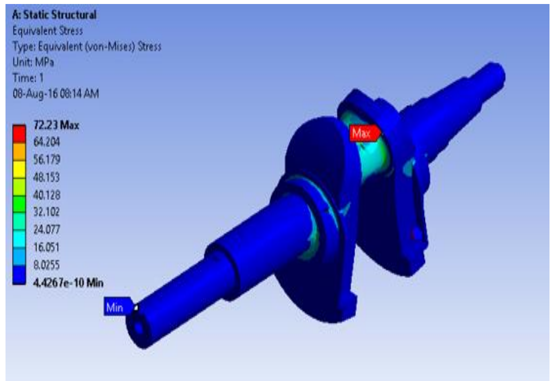

Figure 6 Von-mises stress at Crankpin diameter 18mm

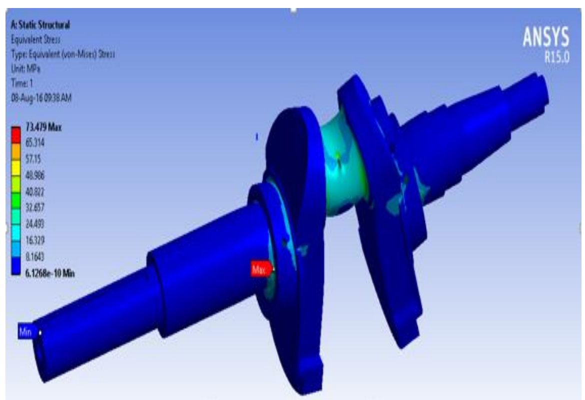

Figure 7 Von-mises stress at Crankpin diameter 18.25mm

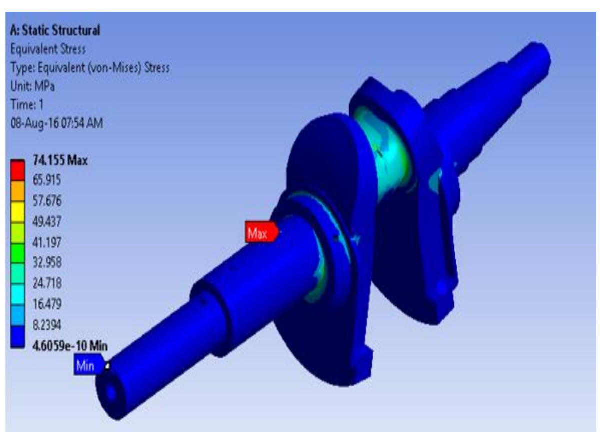

Figure 8 Von-mises stress at Crankpin diameter 18.75mm

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Life Prediction on Variable Crankpin inner diameter

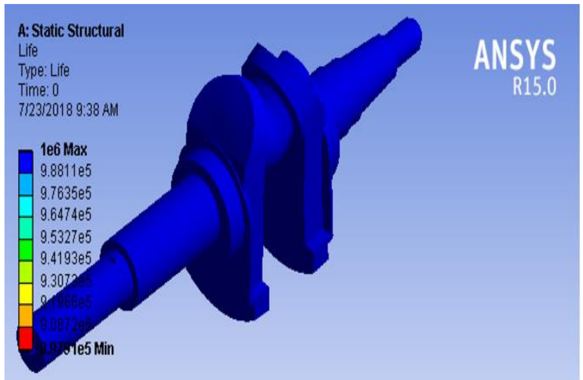

Figure 9 Life at Crankpin diameter 18.1

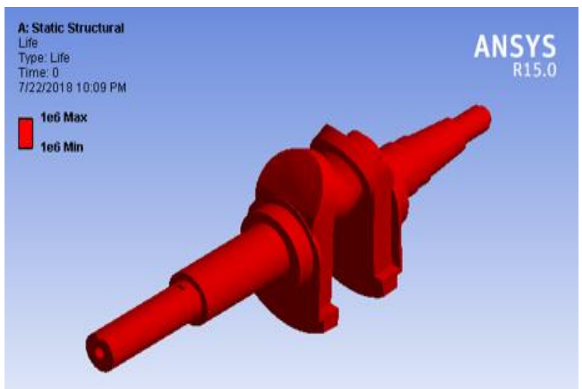

Figure 10 Life at Crankpin diameter 18.1

Stress Analysis at Variable Crankweb Thickness

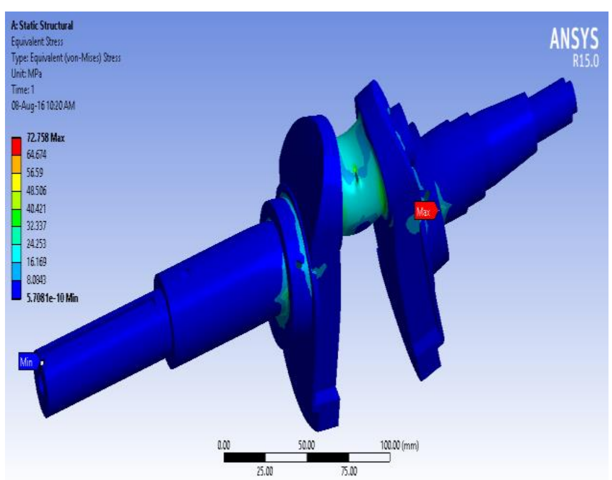

Figure 11 Life at Crankpin diameter 18. 1

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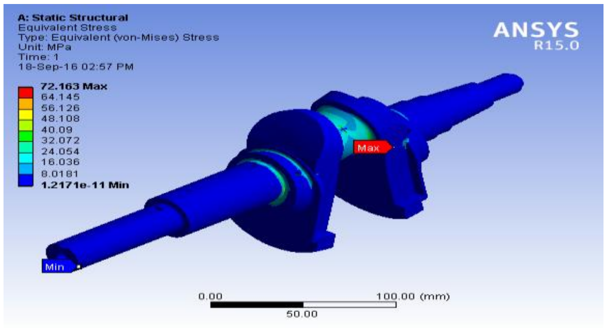

Figure 12 Von-mises stress at Crankweb thickness 19.5mm

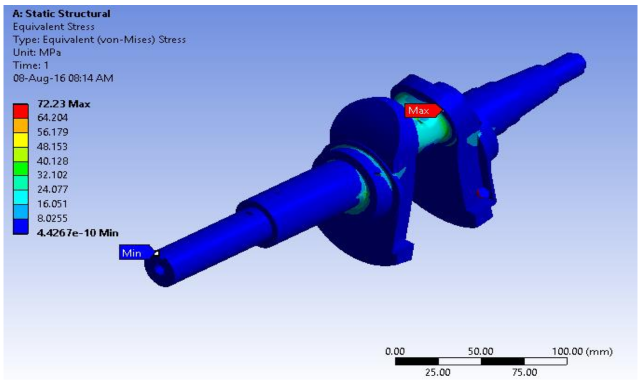

Figure 3.13 Von-mises stress at crankweb thickness 19.2mm

Life Prediction on Variable Crankweb

The iteration of life prediction on crankweb start from web thickness of 20.5mm. But the result is equal with life at crankpin inner diameter of 18.75mm because it is the starting point of last step.

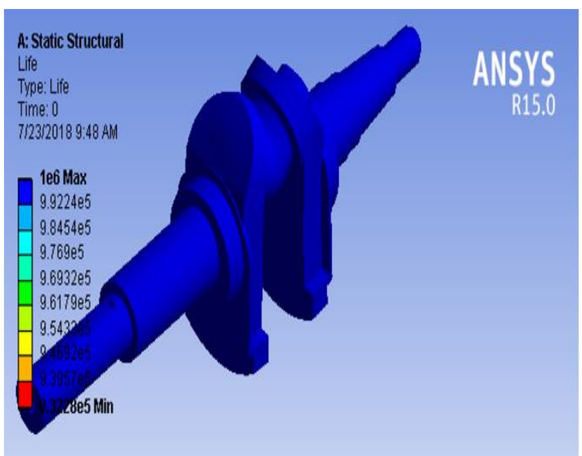

Figure 14 Life at Crankweb thickness 19.5mm

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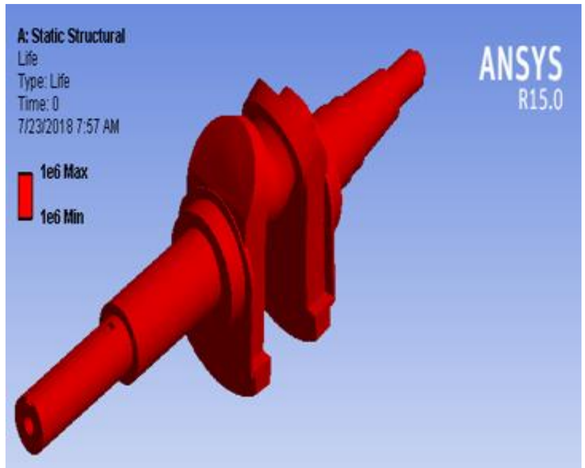
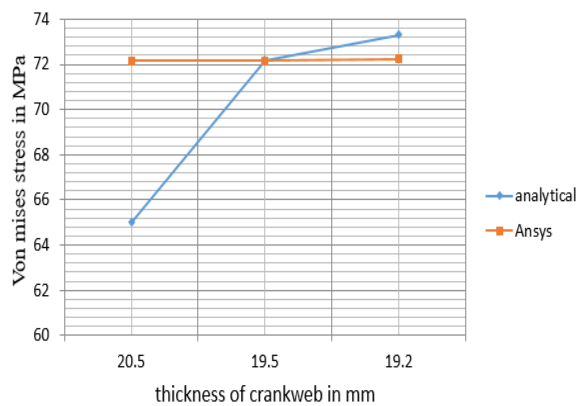


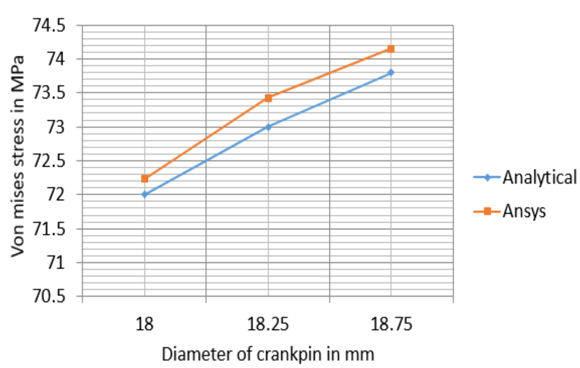
Figure 15 life at crankweb thickness 19.2mm

IV. RESULT AND DISCUSION



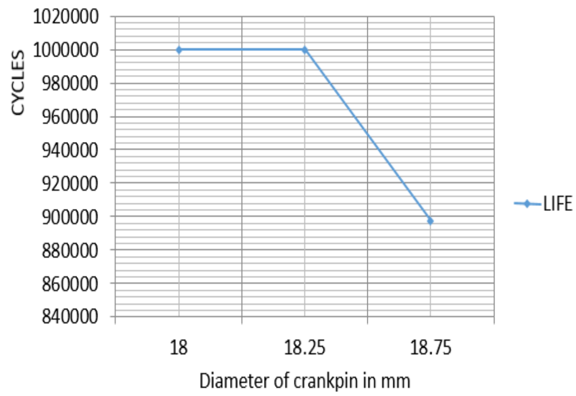
Graph 1 Bending stress of analytical and ANSYS result

As crankweb thickness decreases from 20.5mm to 19.2mm, the calculated stress increases from 65MPa to 73.3MPa. On the other hand the stress from ANSYS also increases from 72.158MPa to 72.23MPa.



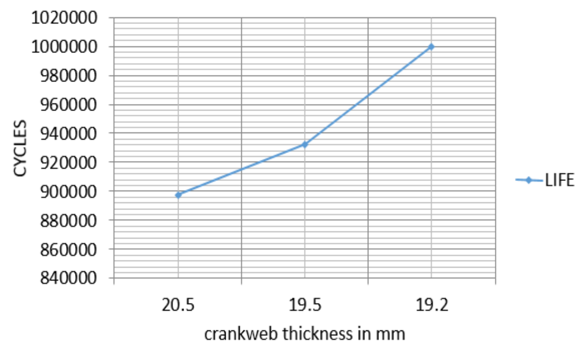
Graph 2 Bending stress of analytical and ANSYS result

As crankpin inner diameter increases 18mm to 18.75mm, the calculated stress increases from 72MPa to 73.8MPa. On the other hand stress by ANSYS result also increases from 72.23Mpa to 74.155Mpa.



Graph 3 Life of crankshaft at variable inner diameter

From Graph 4.3 when crankpin inner diameter increases from 18.25mm to 18.75mm the life of crankshaft decreases. As thickness of crankpin decreases life also decreases.



Graph 4 Life of crankshaft at variable crankweb thickness

Graph 4.4 shows last iteration in optimization of crankshaft. As crankweb thickness reduced weight of shaft decreases and life increase.

V. CONCLUSION AND FUTURE WORK

Conclusion

In this research paper finite element analysis (FEA) and Analytical methods have done. The maximum stress appears at the crankpin. FEA analysis using ANSYS workbench from fatigue tool is very efficient and simple method for predicting life according to force applied to the crankshaft. The use of numerical method such as Finite Element Method is a good tool to reduce the time consuming theoretical work. Model of original and optimized crankshaft has completed using SOLIDWORK software. FEA performed with various parameters of crank shaft such as crankweb and crankpin inner diameter. From optimization of crankpin inner diameter iteration, Von misses stress increases and number of cycles to failure decreases. On the other hand optimization of cranweb thickness iteration, Von misses stress increases and number of cycles to failure increases. The stress of Analytical and FEA results shows close agreement but not exceeded allowable stress. Optimization process of crankshaft refers reliable according to this research paper because weight of shaft reduces as material remove from given parameters. Final result of FEA shows life of modified and original crankshaft has no difference.

Future Work

Crankshaft transmits motion generated by the linear displacement of the piston by a working fluid to rotational motion of a shaft.

There are many applications of crankshafts and the most common are in engines. Life prediction analysis of this paper work has completed using software on geometry size optimization. The best way of analyzing life is in ANSYS result. So future work of this research will be written as follows.

- Since the analysis of crankshaft life prediction completed using with software only, it is necessary to test with experimental setup.
- Additionally by using better strength material property the life of the crankshaft will be improved.
- Crankshaft has to be dynamically balanced, optimization process fulfill removing or adding material in the crank web.

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