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Optimization and Finite Element Analysis of Single Cylinder Engine Crankshaft for Improving Fatigue Life

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Abstract: Crankshaft is large volume production component with a complex geometry in internal combustion Engine (ICE), which converts the reciprocating displacement of the piston into a rotary motion of the crank. An effort was done in this paper to improve fatigue life for single cylinder engine crankshaft with geometric optimization. The modeling of the original and optimized crankshaft is created using SOLIDWORK Software and imported to ANSYS software for analysis. Finite element analysis (FEA) was performed to obtain maximum stress point or concentrated stress, to optimize the life of crank shaft by applying the boundary conditions. The maximum stress appears at the fillet areas between the crankshaft journal and crank web. The FE model of the crankshaft geometry is meshed with tetrahedral elements. Mesh refinement are done on the crank pin fillet and journal fillet, so that fine mesh is obtained on fillet areas, which are generally critical locations on crankshaft. The failure in the crankshaft initiated at the fillet region of the journal, and fatigue is the dominant mechanism of failure. Geometry optimization resulted in 15% stress reduction and life is optimized 62.55% crankshaft which was achieved by changing crankpin fillet radius and 25.88% stress reduction and life is optimized 70.63% of crankpin diameter change. Then the results Von-misses stress, shear stress and life of crankshaft is done using ANSYS software results. It was concluded from that the result of geometric optimization parameter; like changing crankpin fillet radius and crankpin diameter were changes in model of crankshaft to improve fatigue life of crankshaft.

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

1. 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 into rotary motion and vice versa with a four link mechanism [1]. The most common application of a crankshaft takes place 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. 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. It 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 [3]. The objective of this study was to analyze stress in critical location for improving fatigue life with geometric optimization of a single cylinder engine typical to that used in a riding lawnmower. Rate failure of crankshaft is not limited to selecting a material, such as steel or iron, a process, such as forging or casting, and surface treatment. Farzin H. Montazersadgh et al [4] suggested the modifications for improvement in fatigue life such as changing the main bearing radius, crank pin radius, fillet radius of crankpin pin and changing the type of material in crankshaft, which are very common modifications usually done in crankshaft geometry. This paper was proposed to analyze the stresses acting on the crank shaft and to improve fatigue life using geometric optimization identically increasing crankpin fillet radius, increasing crankpin diameter will improve crankshaft life.

2. Material Optimization

An extensive study was performed by Nallicheri et al. (1991) [5] on material alternatives for the automotive crankshaft based on manufacturing economics. They considered steel forging, nodular cast iron, micro-alloy forging, and a tempered ductile iron casting as manufacturing options to evaluate the cost effectiveness of using these alternatives for crankshafts. Ashwani Kumar Singh, et al. [6] the modeling of crankshaft was done in Pro-E and simulation in ANSYS. They used nickel chrome steel and structural steel for the material of crank shaft, and concluded that nickel chrome is more reliable than structural steel. In a literature survey by Zoroufi and Fatemi [7], they have discussed the fatigue performance and compared forged steel and cast iron crankshafts. In their study failure sources of crankshaft were discussed and also different methods of crack forming in fillet region. They also compared nodular cast iron, forged steel, a tempered ductile iron, which concluded that fatigue properties of forged steel are better than cast iron. They also added the cost analysis and geometry optimization of crankshaft. Adding fillet rolling was considered in the manufacturing process. Fillet rolling induces compressive residual stress in the fillet areas, which results in 165% increase in fatigue strength of the crankshaft and increases the life of the component significantly [8]. Since fatigue fracture initiated near the fillets is one of the primary failure mechanisms of automotive crankshafts, fillet rolling process has been used to improve the fatigue lives of crankshafts in many applications. Fillet rolling manufacturing process is good in cost and making save time. Bhumesh J. Bagde et al [9] carried out finite element analysis of single cylinder engine crank shaft. In this paper, the crankshaft model was created by Pro-E Wildfire 4.0 software. Then, the model created by Pro-E Wildfire 4.0 was imported to ANSYS software. The analysis of the crank shaft will be done using five different materials. These materials are EN9, SAE 1046, SAE 1137, SAE 3140 & Nickel Cast iron. The comparison of analysis results of all five materials will show the effect of stresses on different materials and this will help to select suitable material. Material optimization is concluded that considered based on manufacturing economics, application, and fatigue properties of material. During using alternative material they have used only software analysis not only this is satisfactory the other issue like mechanical properties and carbon content of material should be considered. Generally optimizations like geometry, material and manufacturing of this component will result in high cost saving increase the fuel efficiency of the engine and improve fatigue life.

3. Materials and Methods

In this study we revealed that, the crankshaft should have enough strength to withstand the forces to which it is subjected i.e. the bending and twisting moments; enough rigidity to keep the distortion minimally. Stiffness to minimize, and strength to resist, the stresses due to torsional vibrations of the Minimum weight, especially in aero engines. Generally, the crankshafts materials must be readily shaped, machined and heat-treated, and have adequate strength, toughness, hardness, and high fatigue strength while production. So far, the crankshaft material used in this study is forged steel (AISI 1045) steel which is medium carbon steel and the type of engine is Honda engine and the typical Uses of Medium Carbon Steel:-

a. 0.3 - 0.4: Lead screws, Gears, Worms, Spindles, Shafts, and Machine parts.
b. 0.4 - 0.5: Crankshafts, Gears, Axles, Mandrels, Tool shanks, and Heat-treated machine parts

Since the carbon content 0.4% - 0.5% is better for crankshaft the material AISI 1045 steel is between this percent and the best one for crankshaft production. This study mainly focuses on the single cylinder diesel engine crankshaft used in agricultural sector. Where this single cylinder engine is used for off roads in rural areas for agricultural purpose. Due to its application is off road or on heavy duty its rate of failure is maximum rather than on road vehicles, so it is necessary to improve its fatigue life.

The following materials shown below in the table (Tables 2 and 3) are used in existing model

Table 1. *Material properties of the crankshaft.*

Material Type	Forged Steel (AISI 1045 steel)	Unit
Density	7833	kg/m^3
Young's modulus	221	GPa
Poisson's ratio	0.3	-
Yield stress	625	MPa
Ultimate tensile strength	827	MPa

Table 2. *Dimension of single cylinder engine crankshaft.*

S. No	Parameters	Symbol	Values
1	Diameter of the Crank Pin	d_c	37mm
2	Length of the Crank Pin	l_c	32mm
3	Crankpin oil hole diameter	c_{oh}	18mm
4	Diameter of the shaft/journal	d_s	35mm
5	Web Thickness (Both Left and Right Hand)	w_t	18mm
6	Web Width (Both Left and Right Hand)	w_w	65mm
7	Length of the Crank shaft	L	341mm
8	Crankpin fillet radius	R_h	3mm

Table 3. *Parameters for optimization geometry.*

S. No	Parameters	Original crankshaft	Optimized crankshaft		
1	Crankpin fillet radius	3	3.5	4	4.5
2	Crankpin diameter	37	38	39	40

Crankpin fillet radius increment is depending on the r/d ratio (fillet radius to crankpin diameter) which exists between

$0.03 \leq r/d \leq 0.13$; $0.03 \leq 3/37$, $3.5/37$, $4/37$, $4.5/37 \leq 0.13$ or $0.03 \leq 0.08$, 0.09, 0.011, 0.012, ≤ 0.13 [40]. It is impossible to optimize when fillet radius to crankpin diameter is greater than 4.5mm for the reason that it does not exist between this ranges. If the diameter of crankpin is 25mm to 50mm has 1mm increment will be added [11]. Since the original crankpin diameter is 37mm 1mm is increased due to exists between this ranges.

Forged steel AISI 1045 material is selected for our experiment which has a density of 7833 Kg/m^3 with 221Gpa Young's Modulus, 0.3 Poison Ratio, 625MP Yield Stress, 827 MPa Ultimate Tensile Strength. Using these parameters a model of Crankshaft in SOLID WORK software was formed and later imported to ANSYS WORKBENCH 16 as shown in figure 1.

The FE model of the crankshaft geometry is meshed with tetrahedral elements shown in Figure 2 where the mesh refinement are done on the crank pin fillet and journal fillet, so that fine mesh is obtained on fillet areas, which are generally critical locations on crankshaft. Tetrahedral shape of element is used for meshing the imported complex geometries to the ANSYSWORKBENCH software. The 3D crankshafts is performed on Solid work and exported to the ANSYS that the profile is subdivided into nodes and elements. Mesh optimization was done to get more accuracy results that optimization is carried out until the *FEA* results and analytical solutions are close to each other. Depending upon the requirement of the accuracy of results the fineness of meshing varies. This meshing varies used is 1.5mm, 2mm, 2.5mm and 3mm. Finer is the meshing more we are closer to the actual results and when mesh size increases maximum stress on the component become decreased.

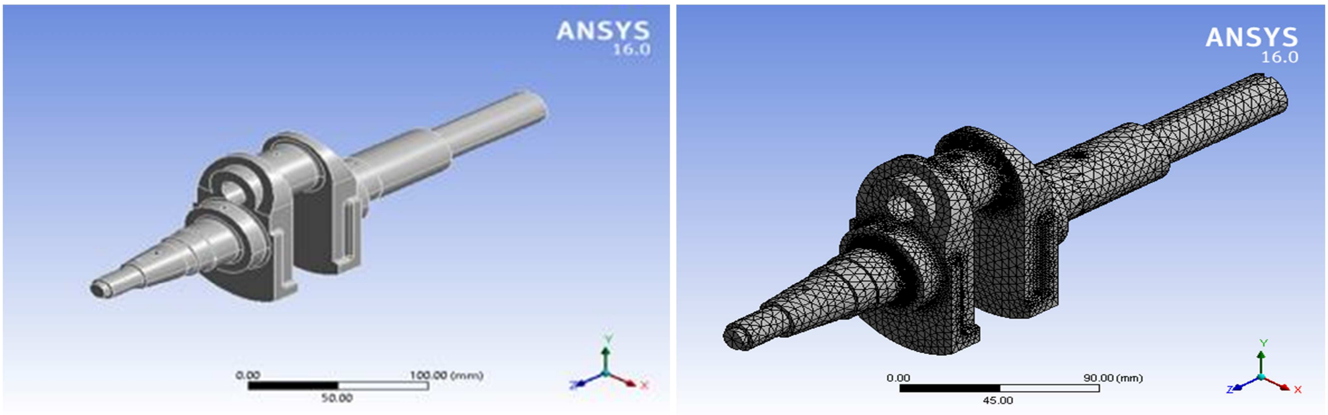

Figure 1. *(a) Single cylinder engine crankshaft model with sold work and (b) Single cylinder engine crankshaft meshing with triangular shape of elements.*

Load applied on the component shown Figure 2, is at the position of maximum bending moment or is at the dead center and the force also applied with red color highlighted which is represented with letter 'A' & letter 'B' shows a fixed support for the structural of the crankshaft. Boundary condition is based on under supporting condition of crankshaft [12].

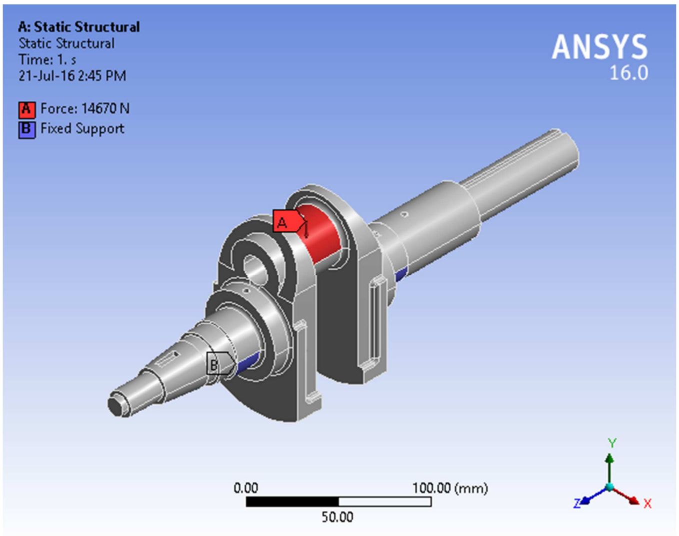

Figure 2. *Input data and boundary condition for single cylinder engine crankshaft.*

The figure 3 shows the detail design optimization from live to dead design for the crank shaft toward improving fatigue life of crank shaft. Also the initial design modelling, optimization, geometry optimization till optimized was done in this discussion below.

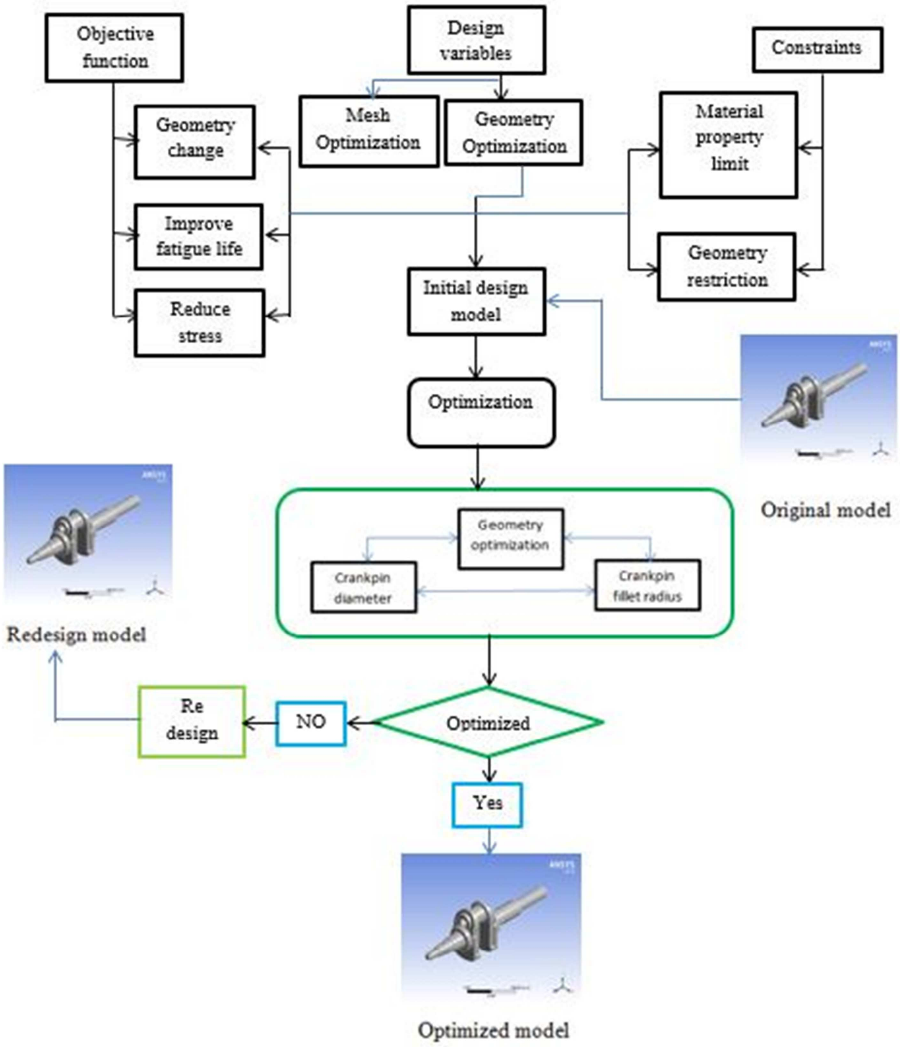

Figure 3. *General flowchart of Crankshaft Optimization procedure.*

4. Results and Discussions

4.1. Stress at Different Fillet Radius

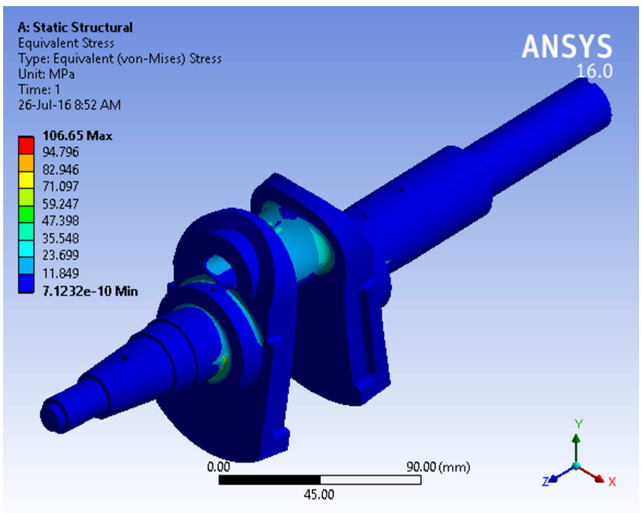

Figure 4. *Von misses stress at 3 mm fillet radius.*

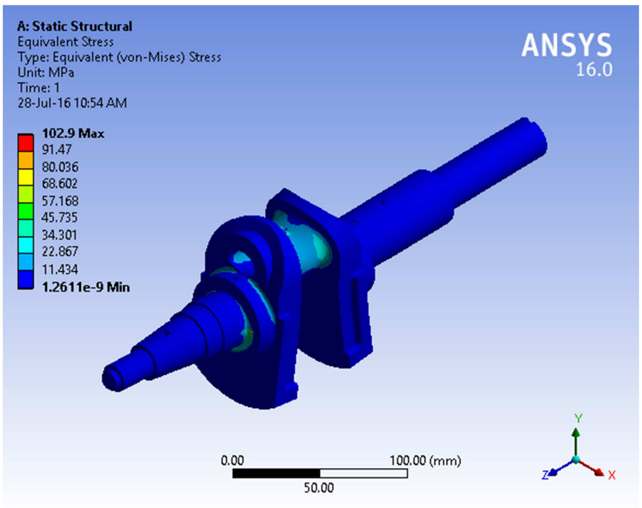

Figure 5. *Von misses stress at 3.5 mm fillet radius.*

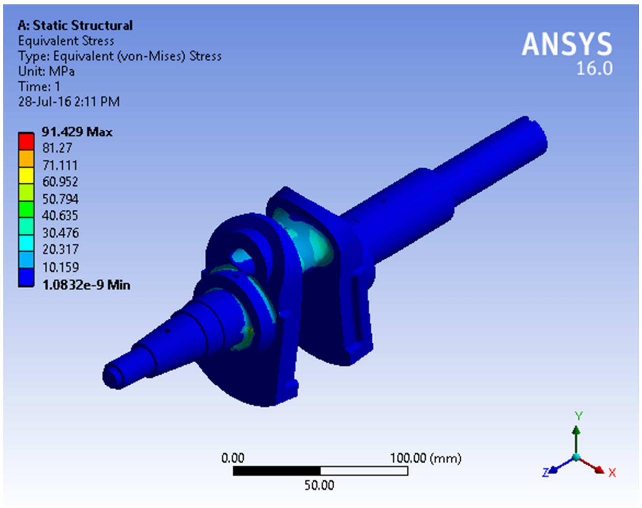

Figure 6. *Von misses stress at 4 mm fillet radius.*

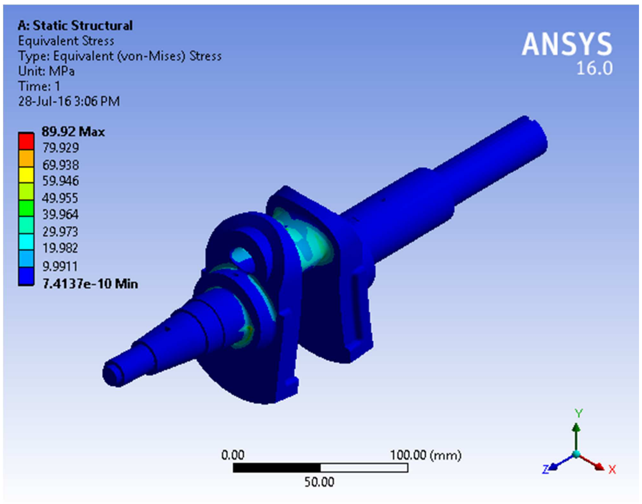

Figure 7. *Von misses stress at 4.5 mm fillet radius.*

4.2. Shear Stress at Different Fillet Radius

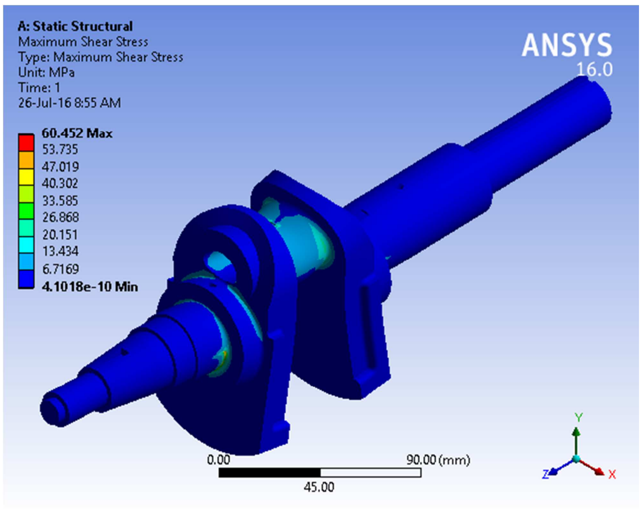

Figure 8. *Shear stress at 3 mm fillet radius.*

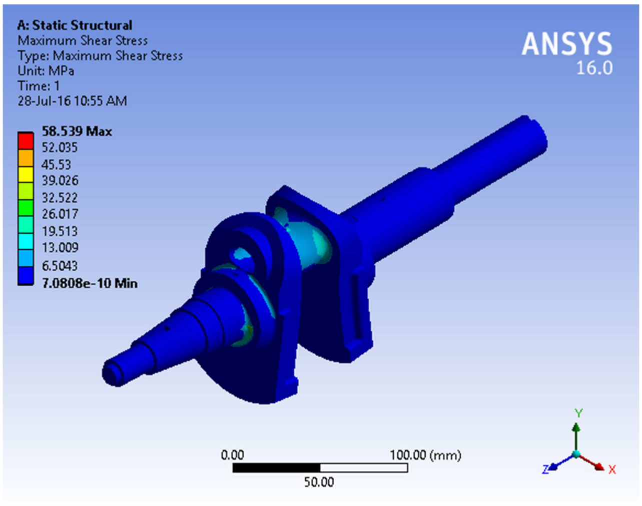

Figure 9. *Shear stress at 3.5 mm fillet radius.*

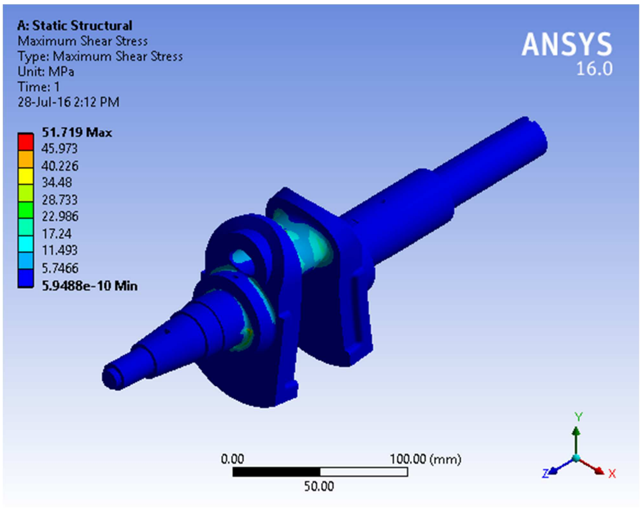

Figure 10. *Shear stress at 4 mm fillet radius.*

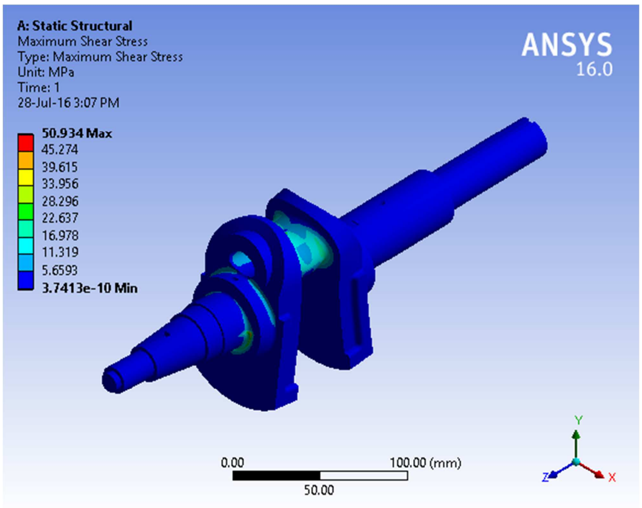

Figure 11. *Shear stress at 4.5 mm fillet radius.*

4.3. Stress at Different Crankpin Diameter

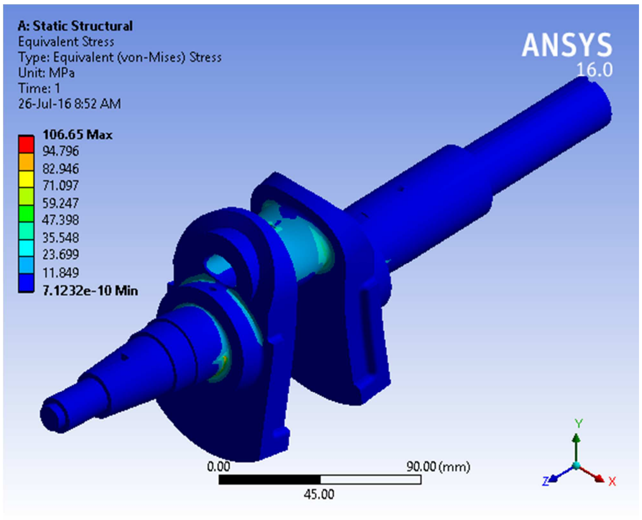

Figure 12. *Von misses stress at 37mm d_c.*

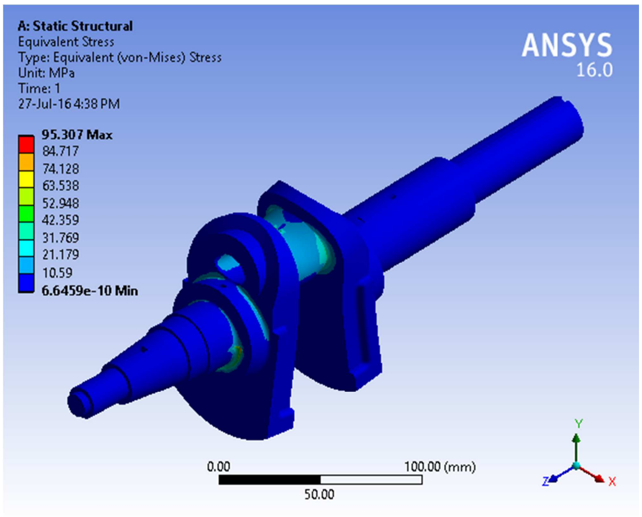

Figure 13. *Von misses stress at 38 mm d_c.*

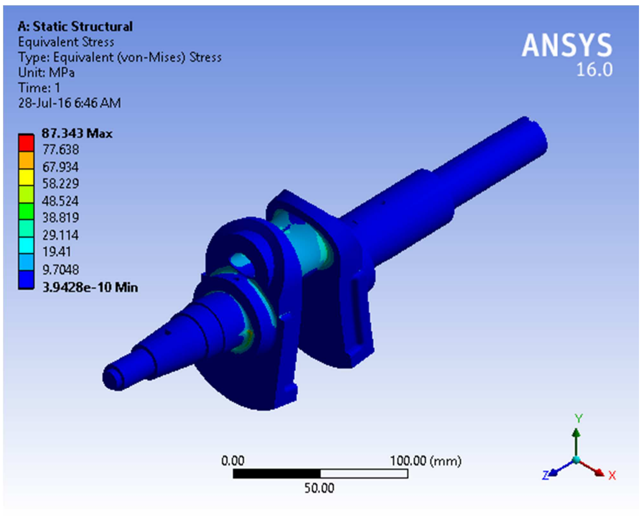

Figure 14. *Von misses stress at 39 mm d_c.*

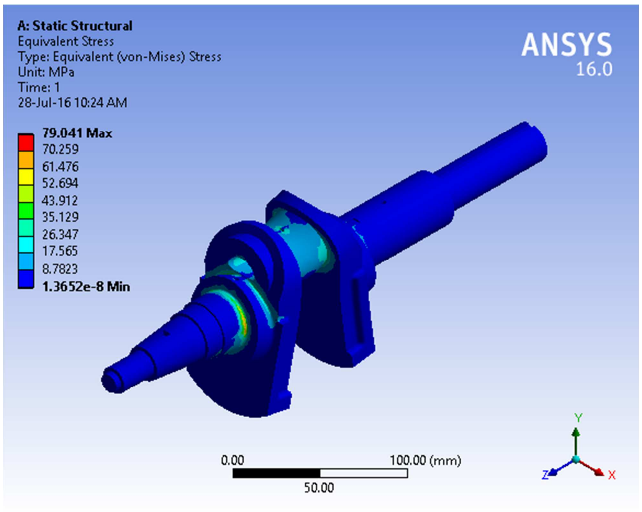

Figure 15. *Von misses stress at 40 mm d_c.*

4.4. Shear stress at Different Crankpin Diameter

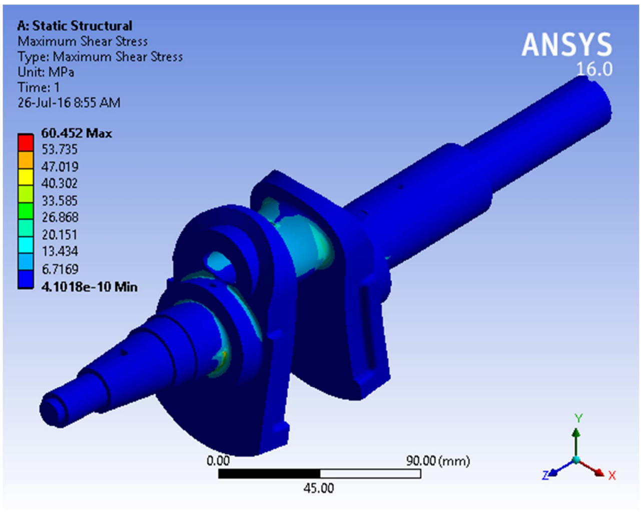

Figure 16. *Shear stress at 37 mm d_c.*

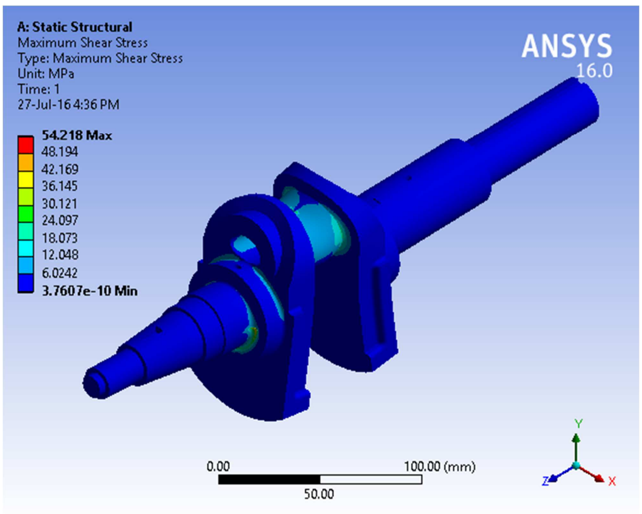

Figure 17. *Shear stress at 38 mm d_c.*

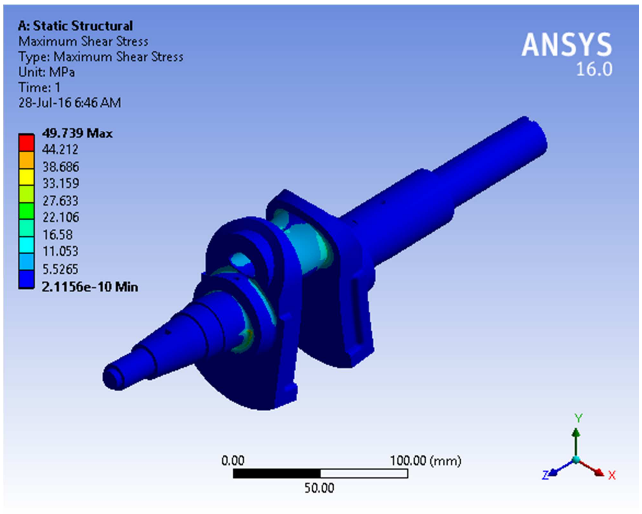

Figure 18. *Shear stress at 39 mm d_c.*

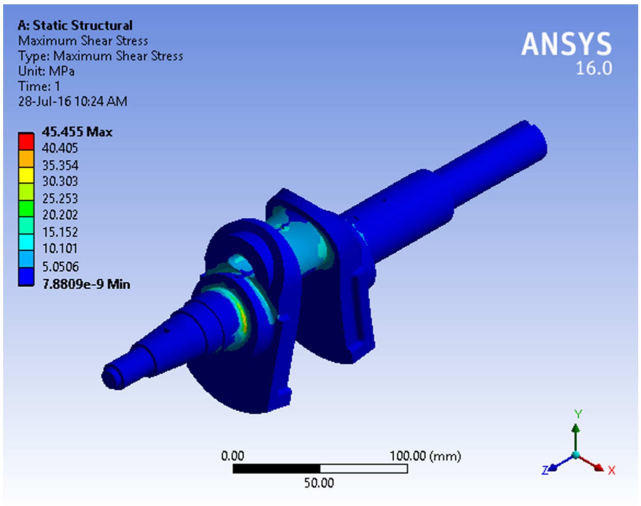

Figure 19. *Shear stress at 40 mm d_c.*

4.5. Life at Different Crankpin Diameter

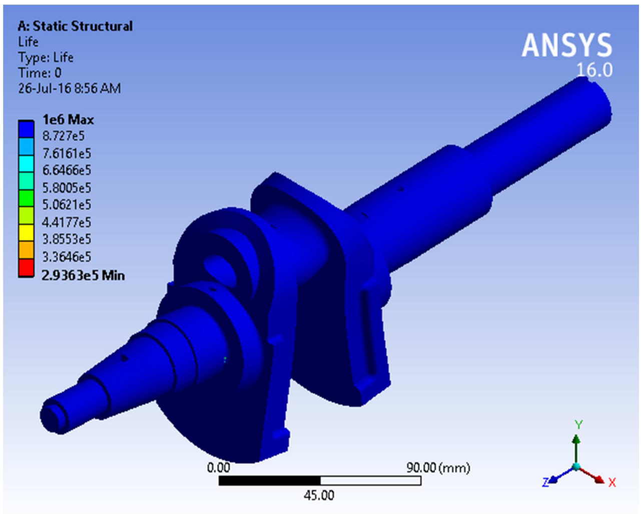

Figure 20. *Life at 37 mm d_c.*

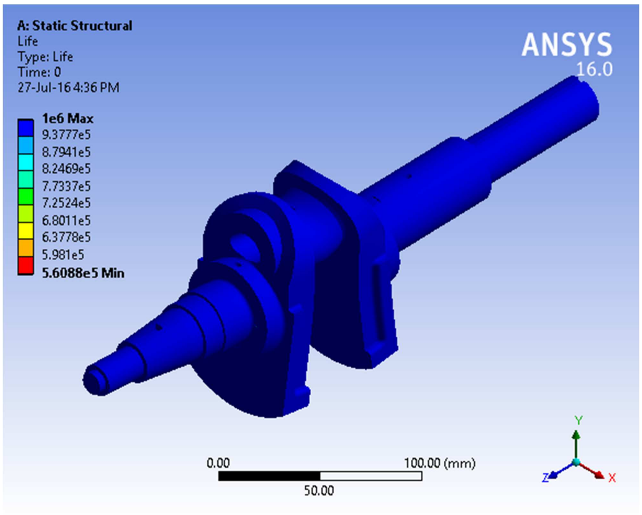

Figure 21. *Life at 38 mm d_c.*

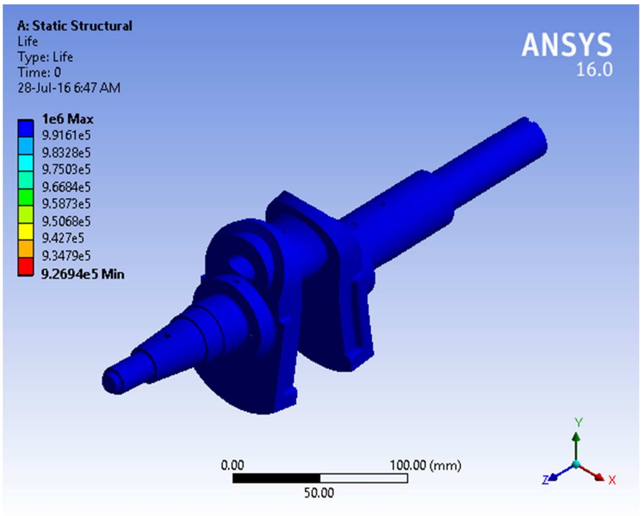

Figure 22. *Life at 39 mm d_c.*

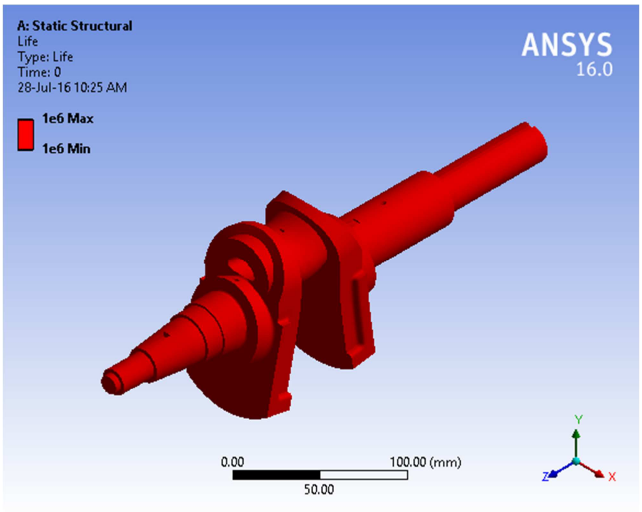

Figure 23. *Life at 40 mm d_c.*

5. Discussions

This study focuses on the single cylinder engine crankshaft for improving fatigue life; that single cylinder engine crankshaft modelling in ANSYS predicts that the maximum value of the equivalent alternating stress decreases as fatigue life increases as shown in table 4 below.

Table 4. *Alternating stress vs. cycles.*

Alternating Stress (MPa)	Cycles
3999	10
2827	20
1896	50
1413	100
1069	200
441	2000
262	10000
214	20000
138	1.e+005
114	2.e+005
86.2	1.e+006

Figure 24 shows fatigue sensitivity curve and how the fatigue results change as a function of the loading at the critical location on the model and fatigue sensitivity between 1.25 and 1.5 is dangerous region as load going to increase the material get failure at critical location.

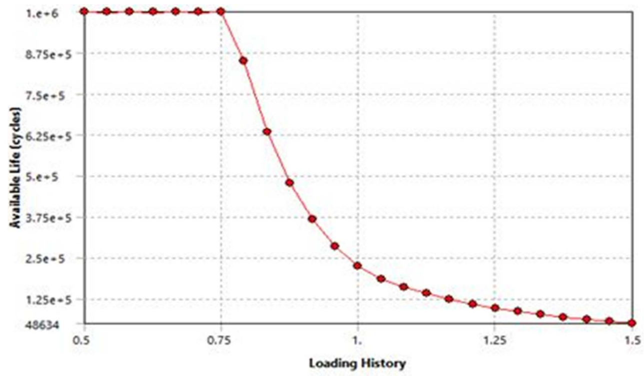

Figure 24. *Fatigue Sensitivity Curve.*

Table 5. *Number of Cycles to Failure related to crankpin fillet radius.*

Fillet radius	Von misses stresses (MPa)		Shear stresses (MPa)		Number of Cycles to Failure (N)
	Theoretical solution	Ansys solution	Theoretical solution	Ansys solution	Ansys solution
3	98.27 N/mm^2	106.65 N/mm^2	56.52 N/mm^2	60.45 N/mm^2	2.9363x10^5
3.5	95.22 N/mm^2	102.9 N/mm^2	55.29 N/mm^2	58.53 N/mm^2	3.6067x10^5
4	92.76 N/mm^2	91.42 N/mm^2	53.54 N/mm^2	51.71 N/mm^2	7.1244x10^5
4.5	89.93 N/mm^2	89.92 N/mm^2	51.93 N/mm^2	50.93N/mm^2	7.8408x10^5

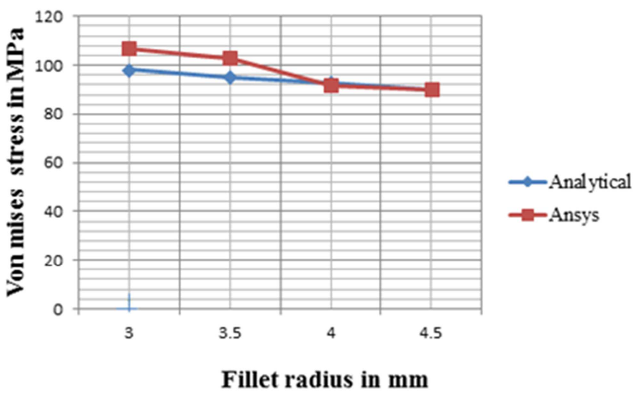

Figure 25. *Graphical Representation of Fillet radius Vs Von misses.*

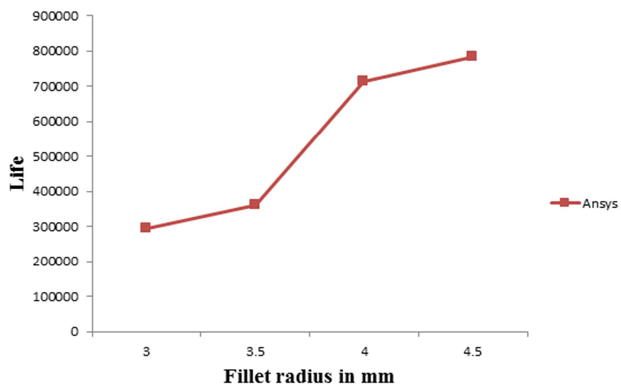

Figure 26. *Graphical representation of Fillet radius Vs Life.*

Table 6. *Number of Cycles to Failure related to crankpin fillet radius.*

Fillet radius	Von misses stresses (MPa)		Error %
	Theoretical solution	Ansys solution	
3	98.27 N/mm^2	106.65 N/mm^2	7.85
3.5	95.22 N/mm^2	102.9 N/mm^2	7.46
4	92.76 N/mm^2	91.42 N/mm^2	1.44
4.5	89.93 N/mm^2	89.92 N/mm^2	0.01

Table 7. *Number of Cycles to Failure related to crankpin fillet radius.*

Fillet Radius	Shear Stresses (MPa)		Error %
	Theoretical solution	Ansys solution	
3	56.52 N/mm^2	60.45 N/mm^2	6.50
3.5	55.29 N/mm^2	58.53 N/mm^2	5.53
4	53.45 N/mm^2	51.71 N/mm^2	3.25
4.5	51.93 N/mm^2	50.93 N/mm^2	1.96

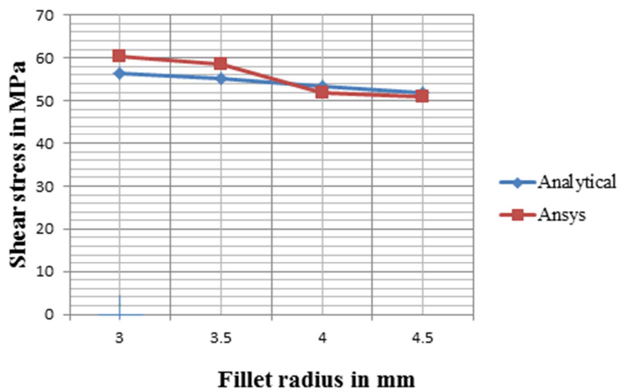

Figure 27. *Graphical Representation of Fillet radius Vs Shear stress.*

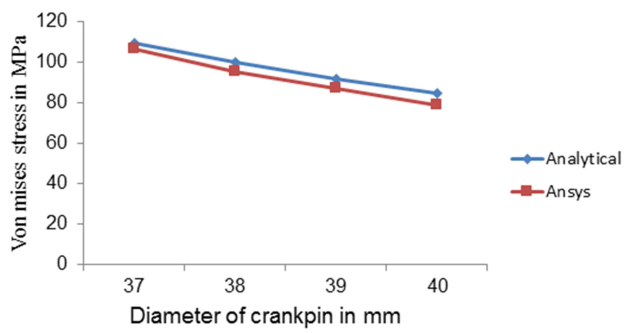

Figure 28. *Graphical representation of crankpin diameter Vs Von misses stress.*

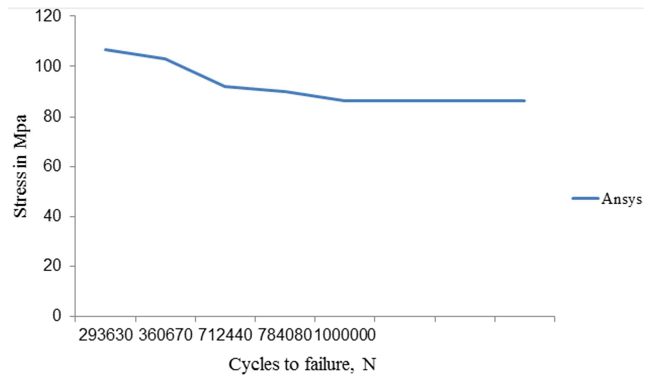

Figure 29. *Graphical representation of S-N.*

Table 8. *Number of Cycles Failure related to diameter of crankpin (d_c).*

Crankpin diameter	Von misses stresses (MPa)		Shear stresses (MPa)		Number of Cycles to Failure
	Theoretical solution	Ansys solution	Theoretical solution	Ansys solution	Ansys solution
37	109.69N/mm^2	106.65 N/mm^2	55.69N/mm^2	60.45 N/mm^2	2.9363x10^5
38	100.25N/mm^2	95.22 N/mm^2	50.91N/mm^2	54.21N/mm^2	5.6088x10^5
39	91.92N/mm^2	87.34 N/mm^2	46.47N/mm^2	49.73N/mm^2	9.2694x10^5
40	84.52N/mm^2	79.04 N/mm^2	42.92N/mm^2	45.45N/mm^2	1x10^6

Table 9. *Number of Cycles Failure related to diameter of crankpin (d_c).*

Crankpin diameter	Von misses stresses (MPa)		Error %
	Theoretical solution	Ansys solution	
37	109.69N/mm^2	106.65 N/mm^2	2.77
38	100.25N/mm^2	95.22 N/mm^2	4.93
39	91.92N/mm^2	87.34 N/mm^2	4.98
40	84.52N/mm^2	79.04 N/mm^2	6.48

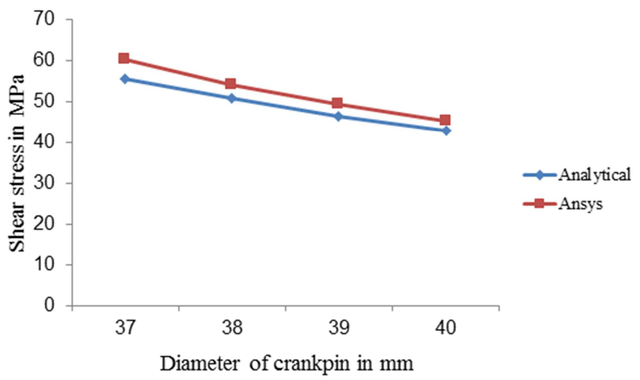

Figure 30. *Graphical representation of crankpin diameter Vs Shear stress.*

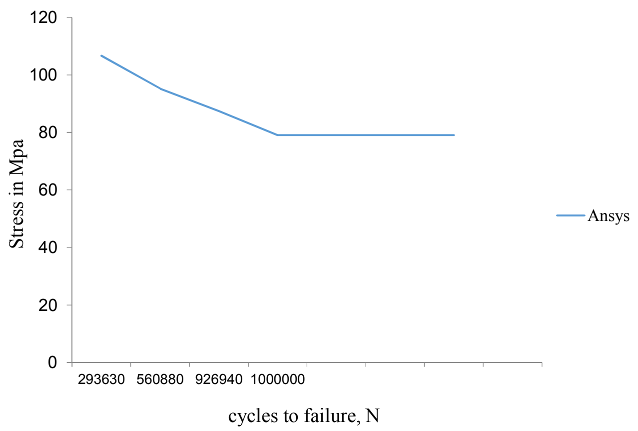

Figure 31. *Graphical representation of S-N.*

Table 10. *Number of Cycles Failure related to diameter of crankpin (d_c).*

Crankpin Diameter	Shear stresses (MPa)		Error %
	Theoretical solution	Ansys solution	
37	55.69N/mm^2	63.4 N/mm^2	7.87
38	50.91N/mm^2	54.21N/mm^2	6.08
39	46.47N/mm^2	49.73N/mm^2	6.55
40	42.92N/mm^2	45.45N/mm^2	6.51

Table 11. *Input data to obtain optimum points using Dx7 design expert.*

Constraints name	Goal	Lower limit	Upper limit
Fillet radius (mm)	is in range	3	4.5
Crankpin diameter (mm)	is in range	37	40
Von misses Stress	Minimize	79.04	106.65
Shear stress	Minimize	45.45	60.45
Cycles to failure (N)	Maximize	293630	1000000

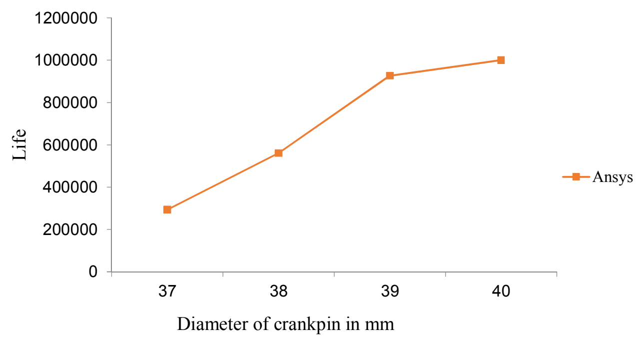

Figure 32. *Graphical representation of crankpin diameter Vs Life.*

6. Conclusion

In this study, the crankshaft model was created using SOLIDWORK modelling and then, the model created was imported to ANSYS commercial software. The analysis of the crank shaft also done using geometry optimization like cxdifferent crankpin fillet radius and crankpin diameter. The maximum stress found at the fillet areas between the crankshaft journal and crank web; that the edge of main journal is maximum stress area in the crankshaft. The FE model of the crankshaft geometry is meshed with tetrahedral elements also mesh refinement are done on the crank pin fillet and journal fillet, so that fine mesh is obtained on fillet areas, which are generally critical locations on crankshaft. The failure in the crankshaft initiated at the fillet region of the journal, and fatigue is the dominant mechanism of failure. The comparison results of all different crankpin fillet radius and crankpin diameter will show the effect of stresses on crankshaft and this will help to select optimized one. Geometry optimization resulted in 15% stress reduction of and life is optimized 62.55% crankshaft which was achieved by changing crankpin fillet radius and 25.88% stress reduction of and life is optimized 70.63% of crankpin diameter change. As the stress of the crankshaft is decreased this will increase fatigue life of the crankshaft. The crankpin fillet radius and crankpin diameter increases then von misses stress and shear stresses are decreases as well as number of cycles to failure increases.

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