Fatigue analysis of four cylinder engine crank shaft

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Fatigue Analysis of Four Cylinder Engine Crank Shaft

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Abstract: The engine crankshaft plays a major role in the internal combustion engines. The combustion pressure is transmitted on to the crankshaft through connecting rod. The inertia forces buckle the connecting rod during transmission of motion. The crankshaft is mounted with a flywheel and the torque is transmitted to the wheels through the gear box. During combustion process huge amount of gas load acts on the piston which is transmitted to the crankshaft. This load is a variable load, due to which fatigue stresses are induced on the shaft. Variations in the strength of the shaft may lead to failure of the engine. Fatigue analysis has been carried on the multi cylinder engine crankshaft by considering two different materials.

Keywords: Crank shaft; Fatigue Stress; Variable load; Reliability;

INTRODUCTION

Crank shaft is main functional element in the internal combustion engine. It can convert piston motion into linear to rotary. It is an important part in engine it is used in all engines of applications like aircraft, ships, diesel Locomotives, compressors.

Crank shaft comprises of web, main journal, oil bore, cornod journal. Section of main and conrod journal is based on engine type and number of cylinders. Main journal helps to assemble the crank shaft to engine block and oil bore is used to supply the oil to the engine when it is in running condition. In the internal combustion engine piston imparting the forces to crank shaft, so crank shaft should withstand bending and twisting forces. Because of this reason ductile materials are used as crank shaft material and production method are casting and forging. Crank shafts are mainly fails due to fatigue because of cyclic loading so for analysing the failure behaviour of crank shaft 3D model is created with CATIA V5 and analysis has been done with HYPER MESH.

LITERATURE REVIEW

Crank shaft is basic element in the engine. It can convert linear motion into rotary motion. Piston helps to change the position of crank shaft which leads to run the vehicle engine. Crank shaft is the complex part in the engine assembly so it will get fail easily for analysing the failure behaviour of crankshaft different materials are used as crankshaft materials and failure analysis has been with different modelling techniques [1]. Fatigue analysis has been done for v8 engine crank shaft, with the help of FEM analysis stress variations are observed at critical points and load calculations also done and these values are compared with ANYSIS so by this fatigue analysis life of crank shaft can be estimated [2]. Nodular cast iron crank shafts are repeatedly failed at same position i.e. nearer to flywheel crank pin. Stresses are estimated with the help of FEM analysis, which will give first crank pin is most effected point for fracture after analysing the failure it is suggested that change the process parameters of crank shaft [3]. Crank shafts are subjected to repeated loads with different directions so fatigue failure plays an important role in consideration for design of crank shaft. By using different failure analysis techniques failure behaviour of crank shaft has been analysed

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[4]. Crank shaft is the complex part in IC engine, single cylinder dynamic analysis has been done, Crank shaft 3D model is created with solid works. State of stresses estimated by finite element analysis and fatigue life of crank shaft is estimated [5]. Fatigue failure is most common failure for crank shaft because crank shafts are subjected to cyclic and bending loads. For estimating the fatigue life of the crank shaft different analytical approaches are used i.e. total life approach and crack growth [6]. Static Failure analysis of crankshaft has been done. Crack initiation is observed at first crack pin, for understanding the failure behaviour FEM analysis has been used. Main reason for fatigue failure is crank pin external zone is affected with high cycle fatigue [9]. For design of marine crank shaft fatigue strength is the key parameter because crank shafts are subjected to bending stress and shearing stress. Fracture is always occurred at crank pin due to high cycle fatigue. Fatigue life of crank shaft estimated by FEM analysis [10]. KL Crank shaft is important part in IC engine, crank shafts are subjected to variable loads which leads to reduce the life of crank shaft, failure analysis has been observed with different approaches and out of all failures fatigue failure is the most common failure for crank shaft [13]. Failure behaviour of damaged diesel motor has been analysed and it is observed that crank pin and bearing cap got damaged due to fatigue failure which leads to failure of crank shaft and failure analysis has been observed with different modelling techniques [14].

NUMERICAL CALCULATION

Static load induceses stress in the members due to the magnitude and direction of load. The stresses do not vary for any period of time. When there is an application of load with respective to time it induceses fluctuating stresses in the member. For the design of member subjected to variable loading the material plays a major role. During the application of variable loading factor of safety is considered to improve the reliability of member. For a crank shaft maximum load is applied during the power stroke and minimum load is applied during compressive stroke. Due to the variable load during rotation of crankshaft variable stresses are induced in it. Endurance strength is considered during its design. The materials considered for the design of the crank shaft are alloy steel and forged steel. Since the material properties of member considered in the design of crank shaft are ductile Soderbergh line method is used deriving the factor of safety.

a) Fatigue Life Estimation:

When member is subjected to repeated loading it will fail before the yield point the phenomena is called fatigue to analyze the failure behavior of repeated loading member Endurance strength will be used. Endurance strength reveals that safe zone of the component under repeated loading condition.

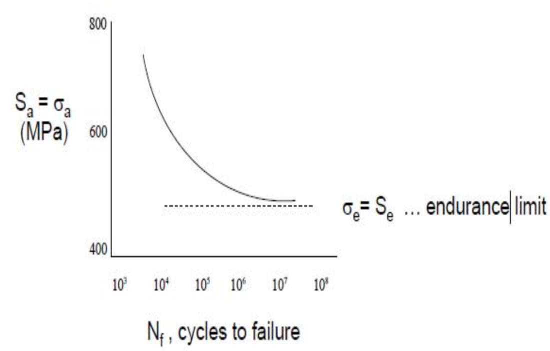


FIGURE 1. Fatigue strength (Sa) Vs Number of Cycles (Nf)

For estimating the life of the component S.N curves are used. In the S-N curve graph is drawn between fatigue strength and number of stress cycles. Endurance strength for steel is it can with stand the load without fail for infinite number of cycles is 107 so for steel material below the curve the fatigue life is infinite and for non-ferrous materials it will take more stress cycles for finding the safe zone of the component.

b) Mean Stress Effects

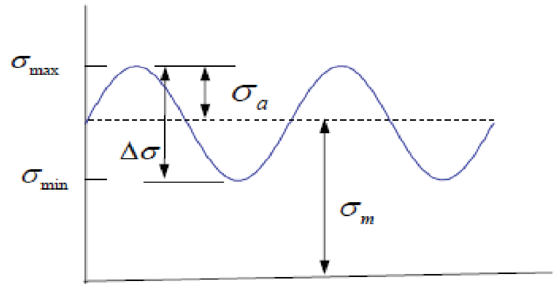


FIGURE 2. Mean Stress Effects

$\Delta\sigma = \sigma_{max}-\sigma_{min}$ Stress Range
$\sigma_a = \Delta\sigma/2 = (\sigma_{max}-\sigma_{min})/2$ Alternating Stress
$\sigma_m = [\sigma_{max}+\sigma_{min}]/2$ Mean Stress
$\sigma_{eq} = \sigma_a / (1-\sigma_m/\sigma_x)^n$
σ_{eq} is equivalent stress
σ_a is alternative stress $[\sigma_{max}-\sigma_{min}]/2$
σ_m is alternative stress $[\sigma_{max}+\sigma_{min}]/2$
σ_x is yield stress
soderberg and good man criteria if n=1
Gerber criteria if n=2
Sample calculation for r/d=0.06
Thus σ_{max} =1250 Mpa
σ_{min}=625 Mpa
$\Delta\sigma = \sigma_{max}-\sigma_{min}$
$\Delta\sigma$=1250-625=625Mpa
σ_a=625/2=312.50 Mpa.
$\sigma_m = [\sigma_{max}+\sigma_{min}]/2 = [1250+625]/2 = 1875/2 = 937.5$ Mpa
$\sigma_{eq} = \sigma_a / (1-\sigma_m/\sigma_x)^n$
$\sigma_{eq} = 312.50/(1-(937.5/625))^2 = 311.25$ Mpa

TABLE 1. Material Properties

S. No.	Properties	Forged steel	Micro Alloy Steel
1	Young's Modulus	221000 Mpa	206000
2	Poisson Ratio	0.33	0.33
3	Density	1.84E-9 ton/mm3	1.72E-09
4	Yield Strength	625 Mpa	980 Mpa
5	Ultimate Strength	827 Mpa	1100 Mpa
6	Strength Coefficient	112000	1150000000
7	Strength Exponent	-0.079	-0.061
8	Ductility Coefficient	0.671	0.18
9	Ductility Exponent	-0.597	-0.53
10	Cyclic Strength Coefficient	116000	1.42e-9 ton/mm3
11	Cyclic Strain Hardening Exponent	0.128	0.12

The four-cylinder diesel engine crank shaft has been modeled and keeping in vie the design considerations in CATIA. More care was taken during the modelling in regards with fillets, oil holes and shoulder near the web. The profile of the web is maintained during the modelling 3D model is imported in to hyper mesh and the crank shaft was meshed using hexahedron elements. The aspect ratio has been maintained during the meshing of the crank shaft. The number of elements obtained after optimized mesh are 8. The elements representing the big end journal have been constrained in U_x U_y U_z such that during rotation it can rotate about its axis. Due to this constrained reaction forceses may be obtained on these elements. Static analysis has been carried out to determine the behavior of crank shaft due to the wait of the components acting on it. In this analysis the displacement and static stresses are analyses by considering forged steel and alloy steel. Fatigue analysis is carried out to determine the reliability of the crank shaft. 400 Mpa Pressure is applied exactly where connecting holds the crank shaft.

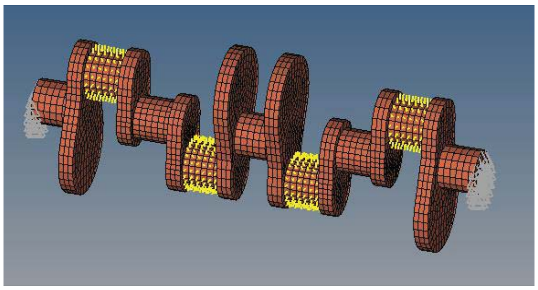


FIGURE 3. Bearing load on crank pin

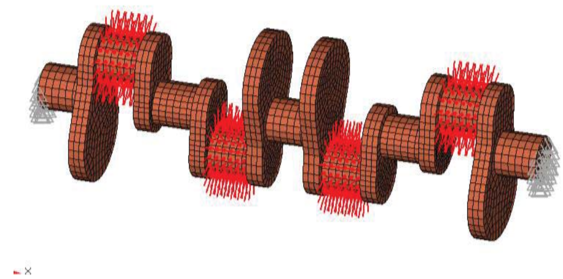


FIGURE 4. Forces on crank pin

c) Fatigue Analysis of Crankshaft

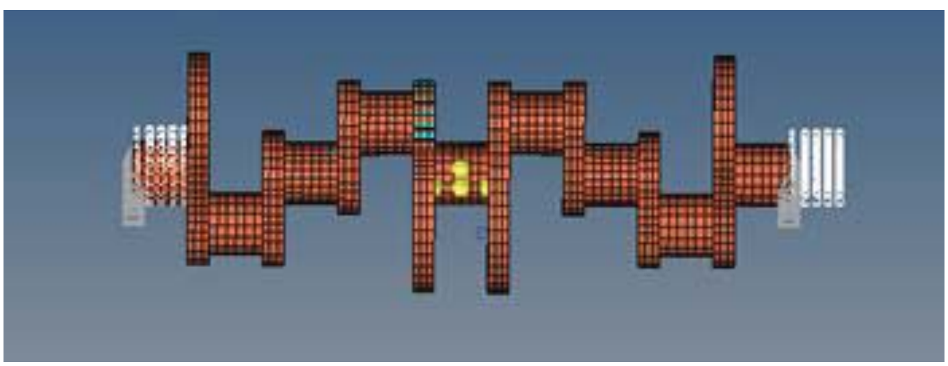


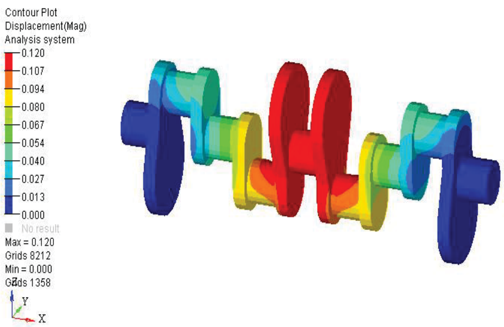
FIGURE 5. Application of fatigue load on the crank journal

RESULTS AND DISCUSSION

From the Figure.6 the max defection in the crank shaft is observed on the main journal at max pressure 40 Mpa for forged steel and from the analysis the displacement is 0. 12mm.from the stress analysis the crank pin induced with maximum stress of 486.18Mpa as the load was acting on the crank pin two and three. During the load stress the bearing stress on the main journal is minimum.

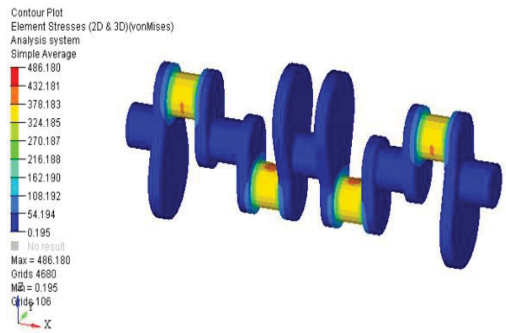
Static Analysis Results for Crank Shaft for two materials:

Forged Steel Results for 400 Mpa pressure:



Displacement is 0.120 mm

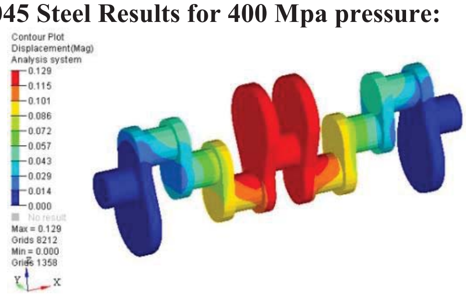
FIGURE 6. Displacement Analysis for Forged Steel



Stress is 486.180 Mpa

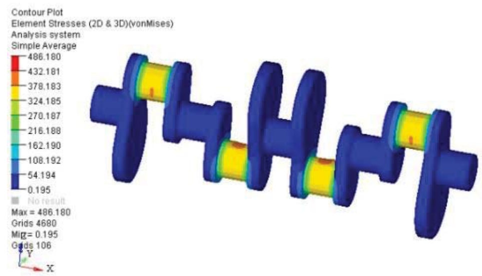
FIGURE 7. Stress Analysis for Forged Steel

AISI 1045 Steel Results for 400 Mpa pressure:



Displacement is 0.129mm

FIGURE 8. Displacement Analysis for AISI1045 Steel



Stress is 486.180 Mpa

FIGURE 9. Stress Analysis for AISI1045 Steel

From the Figure.6 the max defection in the crank shaft is observed on the main journal at max pressure 40 Mpa for forged steel and from the analysis the displacement is 0. 129mm.from the stress analysis the crank pin induced with maximum stress of 486.18Mpa as the load was acting on the crank pin two and three. During the load stress the bearing stress on the main journal is minimum.

By seeing the above two material static results for 400Mpa pressure on crank shaft. All are under the yield limit of particular material. But displacement and stresses are different for two materials by this result forged steel is having better results. Displacement and stress is less compare to two materials. Fatigue analysis is very important for crank shaft, as per OEM original equipment manufacturer the fatigue is main criteria for crank shaft. Based on that the fatigue analysis is selected for crank shaft and using Opti struct solver, fatigue process manager is used which is explained clearly as above chapter. The steps have shown clearly for fatigue test. Main requirement is load time history curve; this is given by automotive industry. Based on that for forged steel material is used for fatigue analysis and solved.

Results for Fatigue Analysis of Crank Shaft:

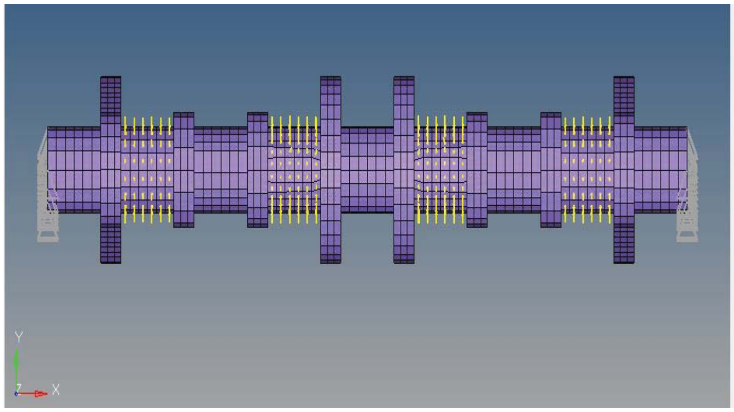


FIGURE 10. Results for forged Steel material

Average Life Results for fatigue forged Steel material. Check the top 0.1%, 1.0%, 5.0% average life, and Top 1, 2, 3 most damage elements lives.

Percentage	Stress
Top 0.1%	307.513 Mpa
Top 1.0%	378.8217Mpa
Top 5.0%	745.7221Mpa

Top Element Number	Stress
10066	300.0147
20450	300.0157
10118	300.5776

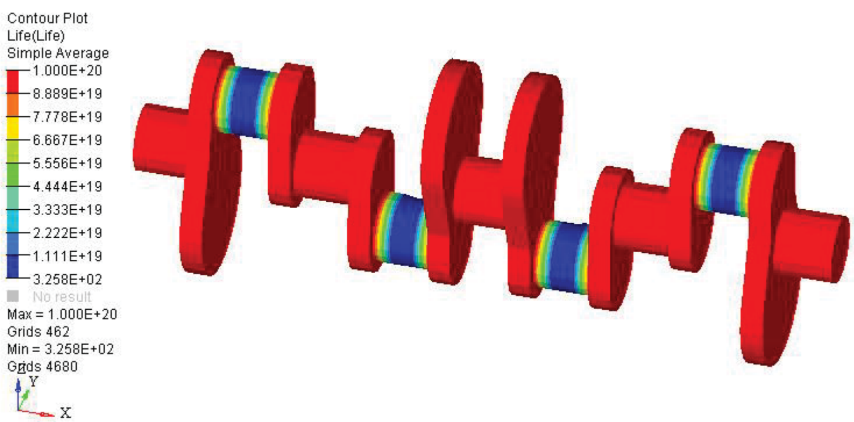


FIGURE 11. Life Cycles of a Crank Shaft

Life of crank shaft is 1E20 Cycles and Blue and Green region having less life compare to red region location. Red and green region has 1E19 cycles and blue region has 3.25E2 cycles. This is having less life for blue region. Damages happen at mentioned elements. To increase the blue region life the design of crank shaft has been changed and solved using same method as explained above. Design changes have been done using Hypermorph which is finite element model modifying tool.

CONCLUSION

Fatigue analysis results of crank shaft; the results shows base crank shaft design has high life that is 1E20 Cycles. But in base design the blue region area having less life, because were force is applied at that particular location is having less life of 3.2E+06 cycles.

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