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Design and Analysis of Four Stroke Six Cylinder Diesel Engine Crankshaft with Geometric Optimization

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ABSTRACT

Crankshaft is a component in an engine which converts the reciprocating motion of the piston to the rotary motion. The aim of this project is to provide efficient model of crank shaft within minute changes in geometry for reducing weight. The less weighted crankshaft improves the mechanical efficiency and power output of engine and also reduces the production costs. Initially data collection and literature survey will be done to understand methodology and selection of material. 3D model will be prepared according to the dimensions' obtained from theoretical calculations. Analysis will be done to validate static structural and dynamic results; geometric modifications will be done according to the obtained results to improve the structural stability, material will be removed where stress is not affected to reduce the weight, The material was changed for modified model for further reducing of weight. Analysis will be done to compare modified with existing model, conclusion will be made according to the obtained results

I. Introduction

The crankshaft, sometimes casually abbreviated to crank, is the part of an engine which translates reciprocating linear piston motion into rotation. To convert the reciprocating motion into rotation, the crankshaft has "crank throws" or "crankpins", additional bearing surfaces whose axis is offset from that of the crank, to which the "big ends" of the connecting rods from each cylinder attach.

Four stroke diesel engines gives high thermal efficiency compared to petrol engines because of high compression ratios. Diesel engines are generally used in heavy duty vehicles such as buses, trucks etc. because of its higher torque and greater mileage than petrol engine. Geometric optimization is a method of reducing the weight on non effecting stress areas of crankshaft.

For finding non effecting stress areas, structural analysis was conducted for finding various stress values on various parts of crankshaft. According to that values non effecting stress areas was find out. Stress relieving holes were applied on non effecting stress areas of crankshaft. Geometric optimization is applied because to reduce weight.

The materials used in project are carbon steel and aluminium A360 alloy. Carbon steel is steel in which the main alloying constituent is carbon in the range of 0.12-2.0 percentage. As the carbon percentage content rises, steel has the ability to become harder and stronger through heat treating. Aluminium alloy 360 is having excellent pressure tightness and fluidity. The main use of aluminium is its lightweight comparing to other materials .It has high silicon content and offers high corrosion resistance as well as high strength even in elevated temperatures. This high resistance extends the life of equipment, significantly reduces maintenance costs. The automotive use of aluminium is expected to double in next ten years.

II. Procedure

In project four stroke six cylinder diesel engine (AGCO sisu power 66 CTA bus model engine) existing crankshaft was used. Initially existing crankshaft was geometrically optimised to reduce weight and production costs so that mechanical efficiency of engine can improve. The crankshaft of existing model has been analysed initially with carbon steel material. Structural analysis was conducted for finding the deformation ,stresses and strain values. For finding the frequencies of crankshaft in various directions model analysis was conducted. After that transient structural analysis was conducted for every 5000 revolutions to find the real stresses(time dependent loads)in real working conditions

Fatigue analysis was performed for calculating the life of crankshaft and according to minimum life of crankshaft, safety factor for various portions of crankshaft (crankweb,crankpin etc.,)was analyzed. After analysis of existing model crankshaft, for reducing the weight the non effecting stress areas were findout and on that portions stress relieving holes was applied. The modified model was analysed for finding the difference between existing and modified models. The difference of values are within the limits. So we can use the modified model of carbon steel crankshaft for further optimisation procedure. The modified model was again optimised by reducing the thickness of crankweb on non effecting stress areas portion. By reducing the thickness the weight of crankshaft has been considerably reduced. The modified model with reduced thickness has analysed by structural analysis,trancient analysis,fatigue analysis,model analysis for knowing the values and finding the difference between the values with existing crankshaft model,the difference of values are within the limits. So we can use the modified model with reduced thickness for further optimisation.

The thickness of modified model was reduced again by reducing the weight on non effecting stress areas of crankweb.Analysis was conducted for knowing the values and finding the difference between modified model with reduced thickness of 40 mm with existing model.The difference of values are with in the limits.We can use the modified model with reduced thickness of 40mm. Further optimization of crankshaft by reducing the thickness of crankweb was failed(difference of values are not within limits).

So we can finally use modified crankshaft with reduced thickness of 40 mm crankweb. Now for further reducing of weight of crankshaft we can use aluminium alloy 360 material was used.This material is very light weight material. So aluminium 360 alloy was used instead of carbonsteel for modified crankshaft with thickness of 40mm crankweb.After changing of material the weight of crankshaft has considerably reduced.But for finding the difference of values between aluminium alloy 360 crankshaft with existing model,analysis was performed and difference of values are with in the limits. So finally we can use the aluminium 360 alloy crankshaft in 4stroke 6cylinder diesel engine. The production costs are considerably reduced because of less weight and we can improve mechanical efficiency. After changing of material to aluminium 360 alloy,73 percent weight has reduced comparing with existing model crankshaft. Even though the life of aluminium used crankshaft was decreased 8percent comparing to carbon steel existing crankshaft,the weight of aluminium 360alloy crankshaft was very reduced so we can reduce production costs and we can improve mechanical efficiency.The production costs are very much saved by using of aluminium Alloy360 crankshaft. The existing crankshaft has 224kgs weight and modified model has 219kgs weight.The modified model with reduced thickness of 50 mm crankweb has 190kgs weight and modified model with reduced thickness of 40 mm crankweb has 165 kgs weight.After changing of material to aluminium alloy 360 the crankshaft weight has considerably reduced to 62 kgs. Lighter crankshaft improves higher mechanical efficiency and higher power output.

Analysis

Analysis software was developed by swason analysis systems and analysis was conducted for analysing the working conditions whether really meeting the requirements or not.Analysis was conducted by three stages.

Preprocessor:In this stage,initially element type was assigned and geometry definitions was given,mesh generation is carried out because any object has infinite degrees of freedom and its just not possible to solve the problem in this format.FEM reduces the degrees of freedom from infinite to finite with the help of meshing.

Solution processor:Main analysis is carried out that is applying loads and obtaining the solutions in this stage.

Post processor:In the postprocessor, already obtained solution in solution processor is analysed with help of graphs and comparing with other parameters to view results.

In the paper static structural analysis was performed for finding the stresses ,deformations and strains on various portions of crankshaft. Model analysis was performed for finding the frequency values in various directions of crankshaft. Trancient structural analysis was performed for finding the time dependent stresses or real stresses in real running working conditions. Fatigue analysis was performed for finding the crankshaft life and factor of safety.

III. Design of Multi Cylinder Engine Crank Shaft (Diesel Engine)

Specifications

ENGINE TYPE	66 CTA	74 CTA	84 CTA
Rated power (kw)	98 - 175	135 - 215	187 - 305
Nominal speed (rpm)	2200	2200	2200
Rated torque (Nm)	550 - 1000	670 - 1150	1185 - 1600
At speed (rpm)	1500	1500	1500
Number of cylinders	6	6	6
Displacement (Litres)	6,6	7,4	8,4
Cylinder bore (mm)	108	108	111
Stroke (mm)	120	134	145
Rotation direction (seen from flywheel end)	CCW	CCW	CCW
Aspiration	Turbocharged and charged air cooled		
Type of intercooler	Air to air	Air to air	Air to air
Emission certification	EU Stage 3 A / EPA Tier3		
Injection system	Common Rail	Common Rail	Common Rail

Pressure Calculations

Engine type: Air cooled 4-stroke (AGCO SISU POWER 66CTA model engine)

Number of cylinders =6

Bore diameter (D) = 108 mm

Stroke length (l) = 120mm

Maximum combustion pressure =10.496 N/mm^2

Displacement =6600CC

Compression ratio =23:1

Density of Diesel = 874.6081kg/m^3 at 15°C

= 0.0000008746081kg/mm^3

T=288.855K

Molecular weight for diesel is 200 g/mole

P=10.49N/mm^2

We know that force on the piston i,e: gas load

F_P = 0.78539 ×108^2×10.496

=95877.81N

=95.87KN

In order to find the thrust in connecting rod we should find out angle of inclination of connecting rod with line of stroke (i,e: angle f) (q lies between 30° to 40°)

Sinf = sinq /n=sin40/6=0.107

f= 6.14°

Assume that the distance (*b*) between the bearings 1 and 2 is equal to twice the piston diameter (*D*).

b = 2*D* = 2 × 108 =216mm

Due to this piston gas load (F_P) acting horizontally, there will be two horizontal reactions *H*1and *H*2 at bearings 1 and 2 respectively, such that

*b*1 = *b*2= 108mm

$H1= F_P×b1/b$

= 95.87×108/216=47.935 KN

And $H2 = F_P×b2/b$

= 95.87×108/216=47.935 KN

Assume that the length of the main bearings to be equal, *i.e.*,

*c*1 = *c*2 = *c* / 2.

We know that due to the weight of the flywheel acting downwards, there will be two vertical reactions *V*2 and *V*3 at Bearings 2 and 3 respectively, such that

v_2 = w× c1/c

= w× 0.5= 54 KN

And V_3= w× c2/c

= w× 0.5=54 KN

(*a*) *Design of crankpin:*

dc = Diameter of the crankpin in mm,

lc = Length of the crankpin in mm,

s*b* = Allowable bending stress for the crankpin in N/mm^2=75 N/mm^2

We know that bending moment at the centre of the crankpin

$M_c= H_1×b_2$=47.935 ×108= 5176.98 KN

$M_c= (\pi/32)\, d_C^3 s_b$

5176.98 ×10^3 N = $(\pi/32)\, d_C^3$ ×75

d_C=88.92 mm

The length of the crankpin is given by (l_c)

L_c= *Fp/Dc.Pb=95870/88.92*13.49*

Where p_b = Permissible bearing pressure in N/mm^2.

(p_b LIES IN BETWEEN 10.00-13.5 N/mm^2).

= = 79.87mm

(*b*) *Design of left hand crank web:*

The thickness (*t*) of the crank web is given empirically as

t = 0.65 *dc* + 6.35 mm

= (0.65×88.92) +6.35

=64.148 mm

Where,

d_c = Crankpin diameter in mm,

The width of crank web (*w*) is taken as

w = 1.125 *dc* + 12.7 mm

=1.125× (88.92) +6.35

= 112.735 mm

We know that maximum bending moment on the crank web,

$M=H_1\ (b_2\text{-}lc/2\text{-}t/2) = 47.935(108\text{-}\ 39.935\text{-}32.074) = 1725.22\text{KNmm}$

Section modulus, $Z=1/6*w.t^2=77316.78\text{mm}^3$

Bending stress,

$s_b =$

$= 0.02231\text{KN/mm}^2$

$= 22\ \text{N/mm}^2$

And direct compressive stress on the crank web,

$s_c =$

$= 6.62843\times10^{-3}\ \text{KN/mm}^2$

$=6.62\ \text{N/mm}^2$

Total stress on the crank web

$s_b + s_c =$ Bending stress + Direct stress

$= 22+6.62$

$= 28.62\ \text{N/mm}^2$

Total stress is less than the allowable stress (75 MPa). Hence the design is safe.

(c) *Design of right hand crank web:*

The dimensions of the right hand crank web (*i.e.* thickness and width) are made equal to left hand crank web from the balancing point of view.

The thickness (t) of the crank web= 64.148mm

The width of crank web (w) = 112.735 mm

Design of the crankshaft when the crank is at the angle of maximum twisting moment:

We know that piston gas load F_P = (take p`=6)

$= 0.78539\times108^2\times6$

$= 54.96\text{KN}$

$F_Q = 55.27\text{KN}$

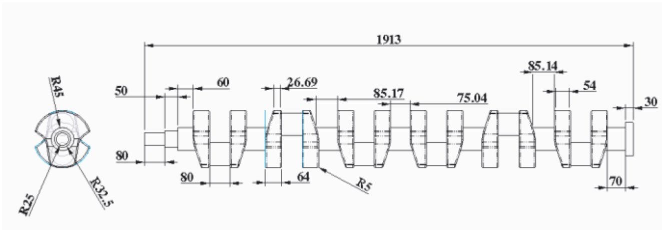
$F_T = F_Q\ \text{Sin}(q+f) = 55.27\times \text{Sin}(40+6.14)$

$= 39.85\text{KN}$

$F_R = F_Q\ \text{Cos}(q+f) = 55.27\times\text{Cos}\ (40+6.14)$

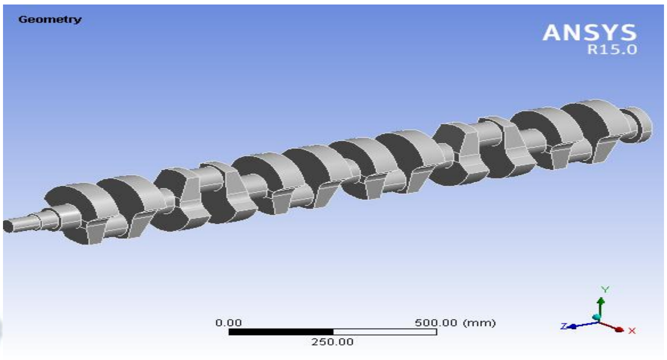
$=38.29\text{KN}$

IV. Structural Analysis of Crankshaft Existing Model

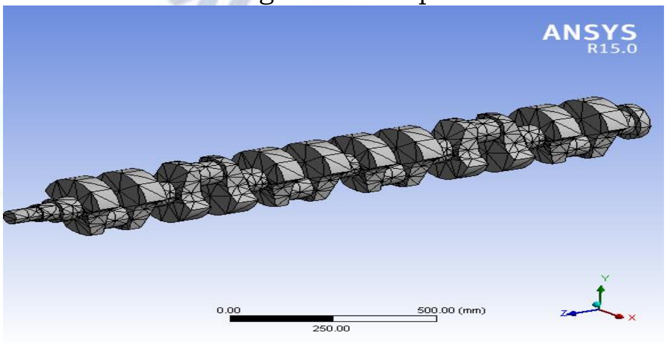


6 CYLINDER CRANK SHAFT

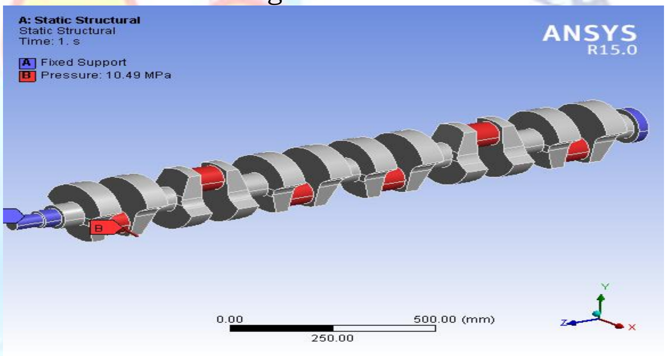
Material: Carbon Steel



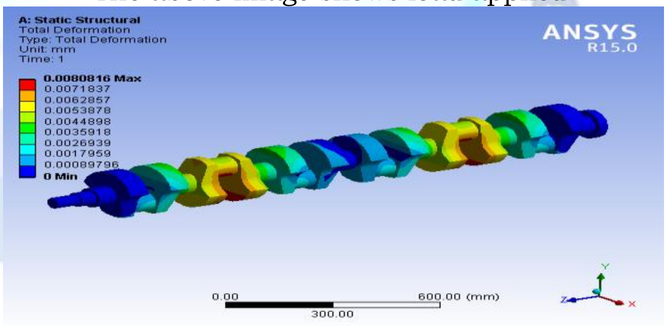
The above image shows imported model



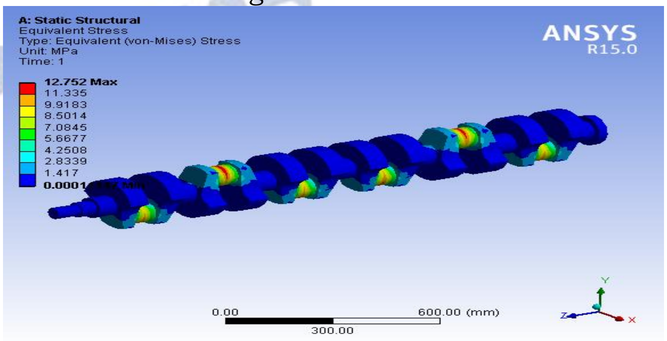
The above image shows meshed model



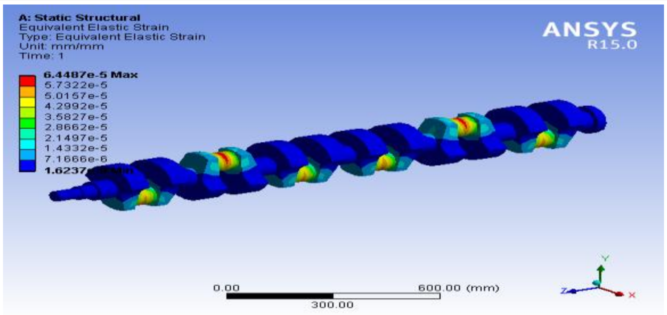
The above image shows load applied



The above image shows total deformation

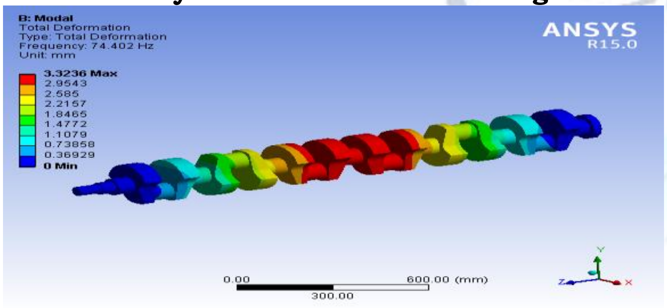


The above image shows stress

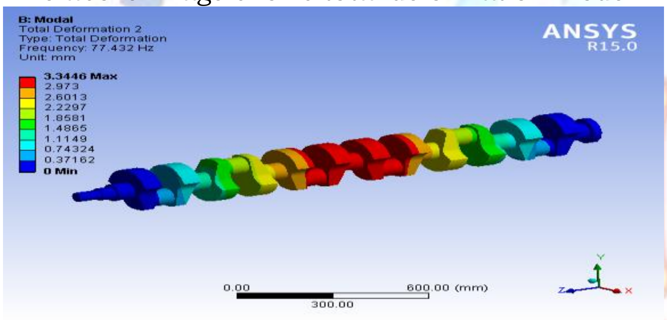


The above image shows strain

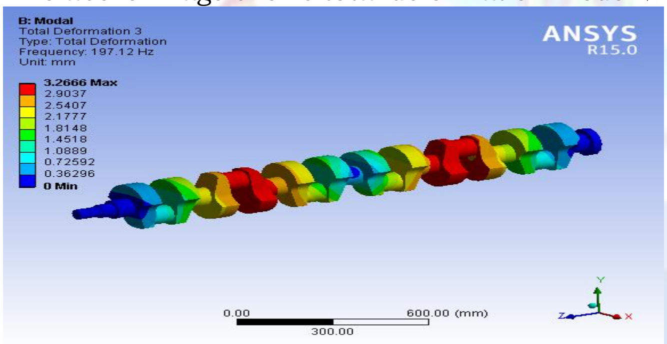
Model Analysis of Crankshaft Existing Model



The above image shows total deformation mode 1

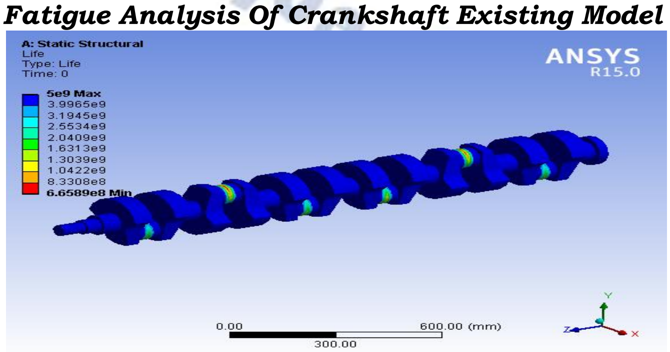


The above image shows total deformation mode 2

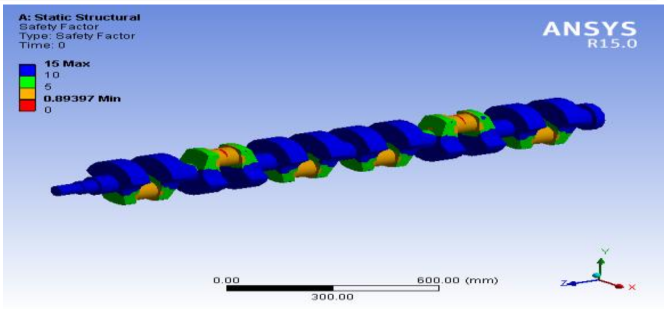


The above image shows total deformation mode 3

Fatigue Analysis Of Crankshaft Existing Model

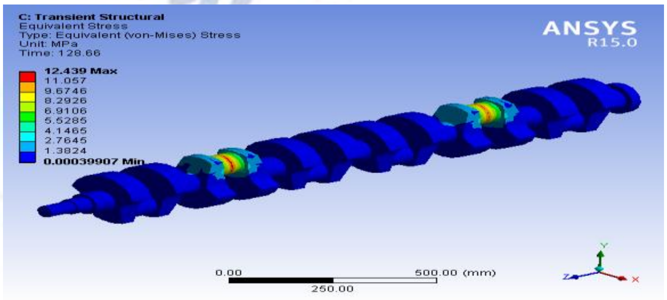


The above image shows life

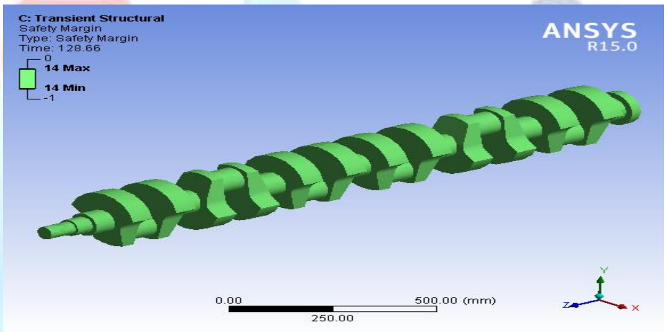


The above image shows safety factor

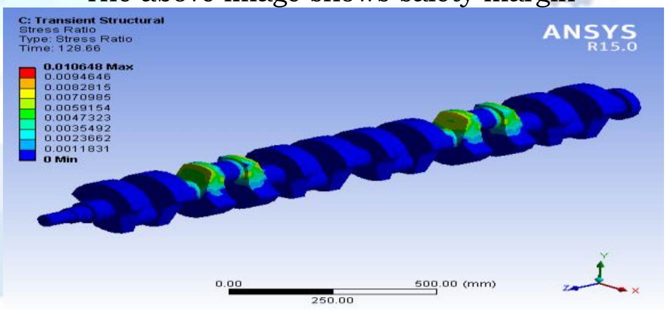
Transient structural analysis of crankshaft existing model at time 128.66 sec



The above image shows stress

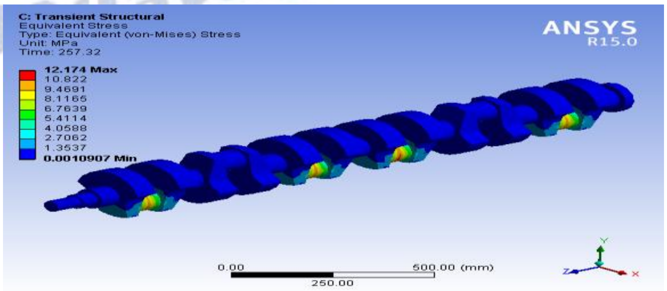


The above image shows safety margin

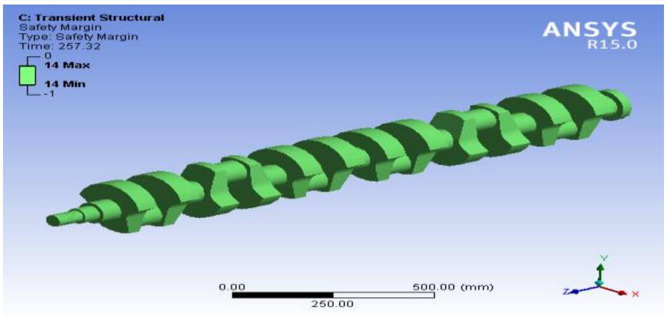


The above image shows stress ratio

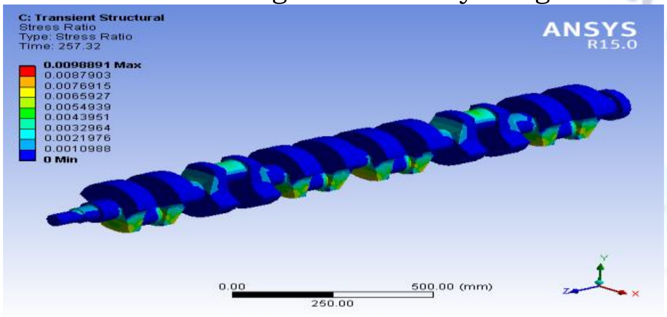
Transient Structural Analysis of Crankshaft Existing Model At Time 257.32 Sec



The above image shows stress

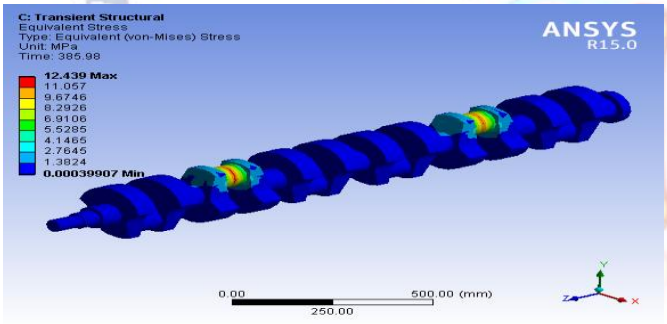


The above image shows safety margin

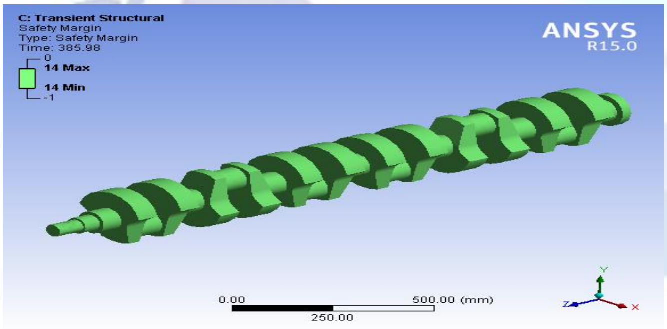


The above image shows stress ratio

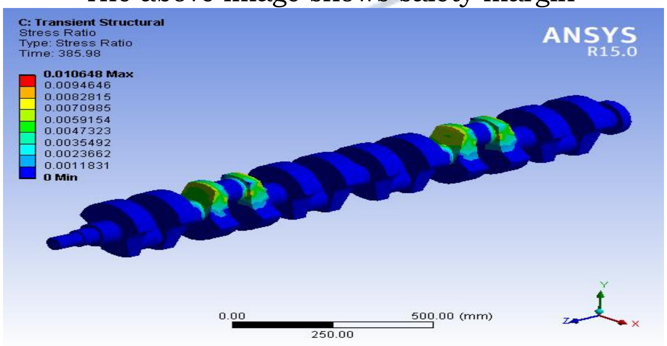
Transient Structural Analysis of Crankshaft Existing Model At Time 385.98 Sec



The above image shows stress

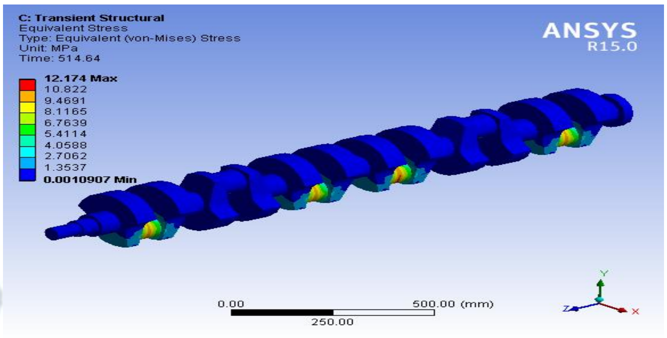


The above image shows safety margin

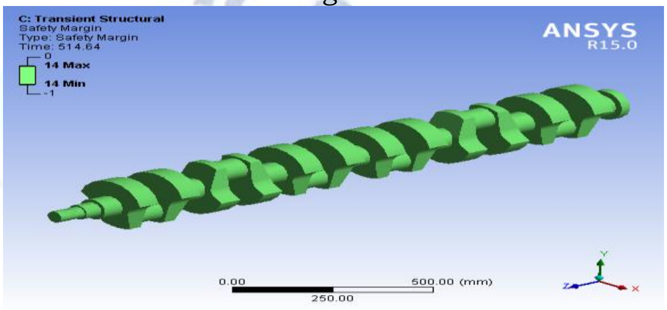


The above image shows stress ratio

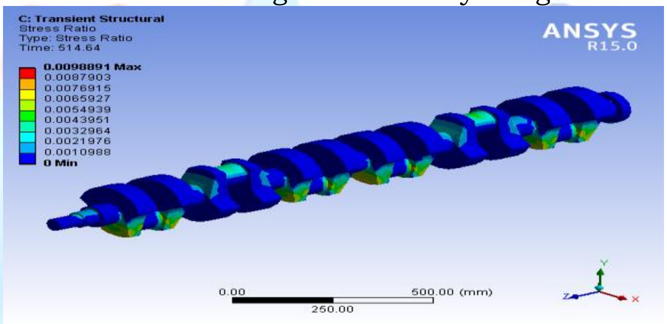
Transient Structural Analysis of Crankshaft Existing Model at Time 514.64 Sec



The above image shows stress



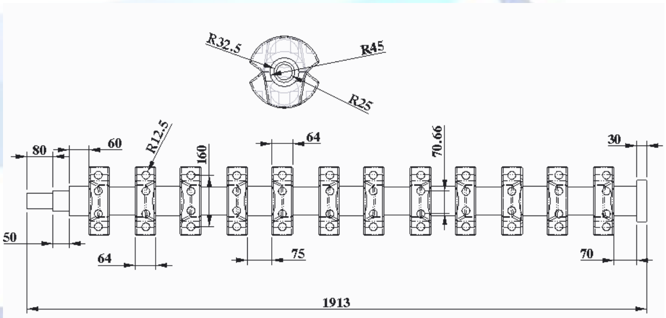
The above image shows safety margin



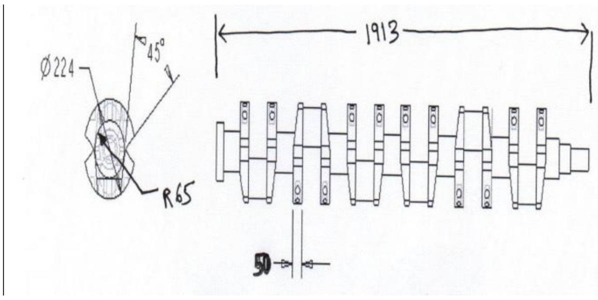
The above image shows stress ratio

V. Structural Analysis of Crankshaft Modified Model

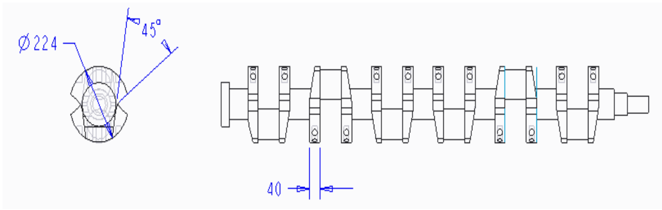
2D DRAFTING



REDUCED THICKNESS 1

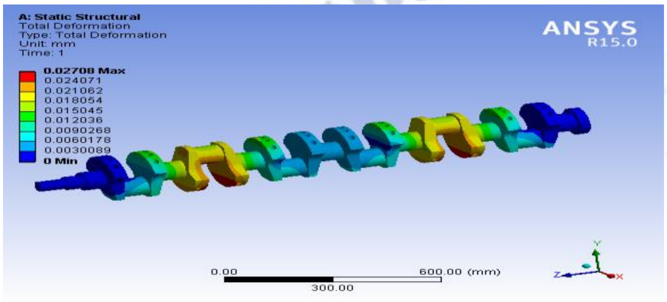


REDUCED THICKNESS 2
3.

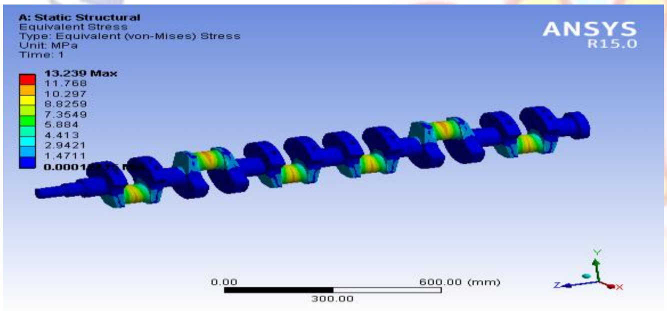


Structural Analysis of Crankshaft Modified Model With Reduced Crankweb Thickness 40 mm

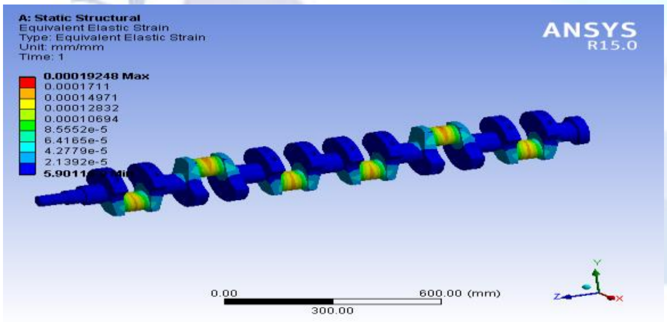
Material: 2 Aluminium Alloy 360



The above image shows total deformation

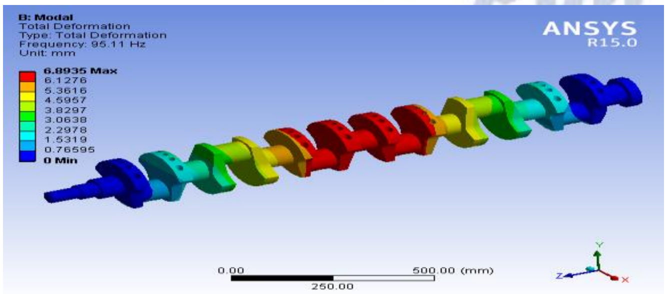


The above image shows stress

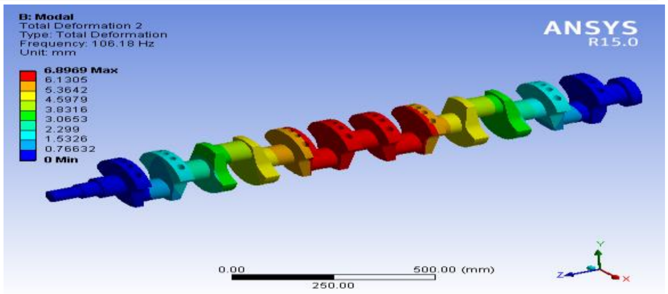


The above image shows strain

Model Analysis of Crankshaft Modified Model with Reduced Thickness 2



The above image shows total deformation mode 1



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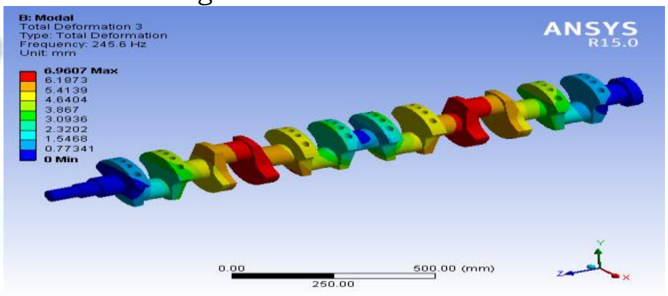
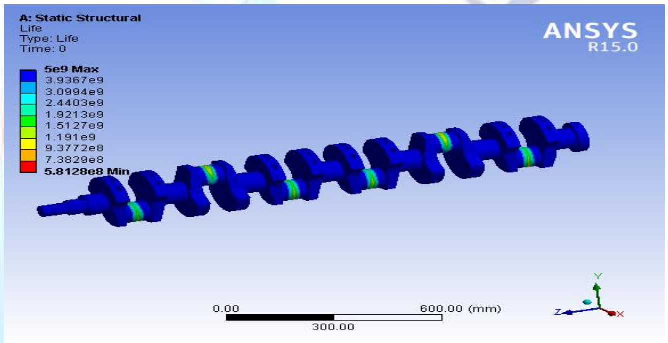
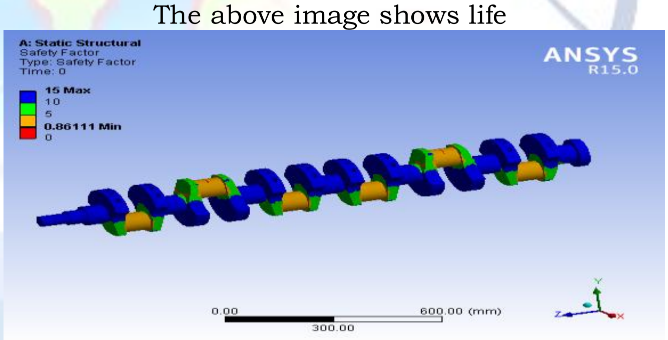


The above image shows total deformation mode 3

Fatigue Analysis of Crankshaft Modified Model With Reduced Thickness 2

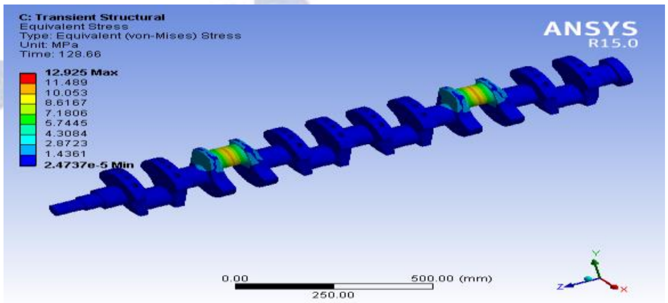


The above image shows life

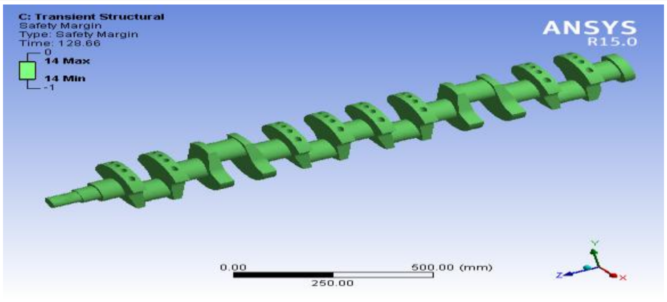


The above image shows safety factor

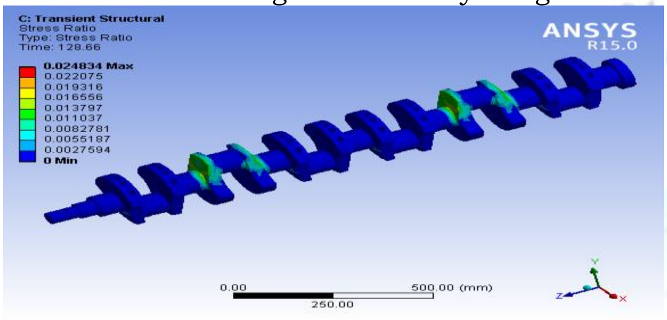
Transient Structural Analysis of Crankshaft Modified Model With Reduced Thickness 2 At Time 128.66 Sec



The above image shows stress

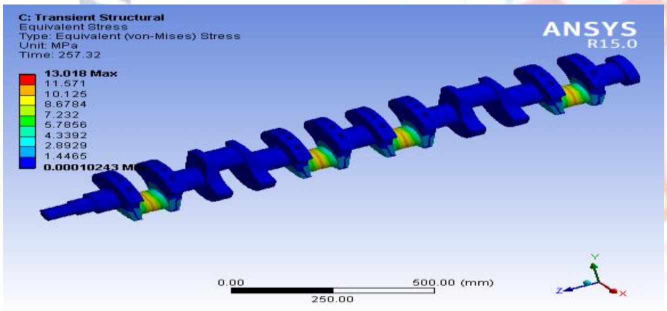


The above image shows safety margin

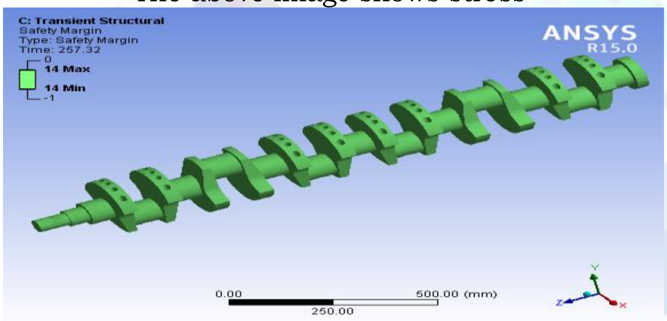


The above image shows stress ratio

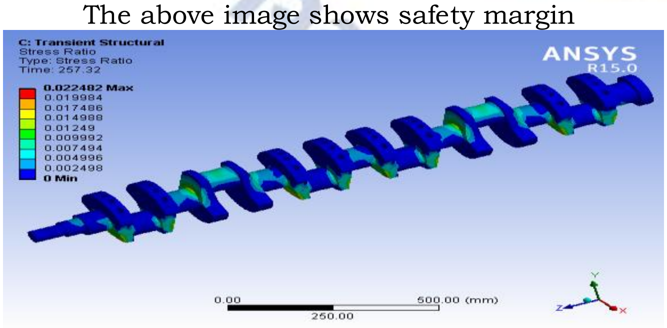
Transient Structural Analysis of Crankshaft Modified Model With Reduced Thickness 2at Time 257.32 Sec



The above image shows stress

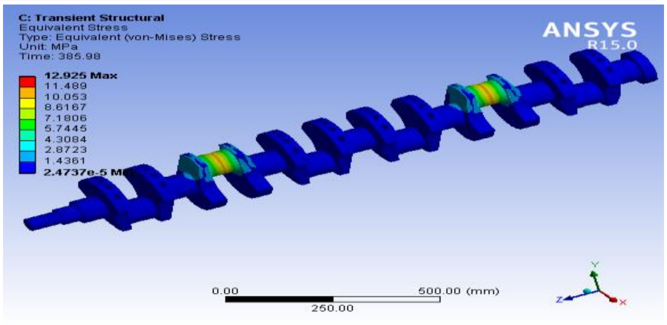


The above image shows safety margin

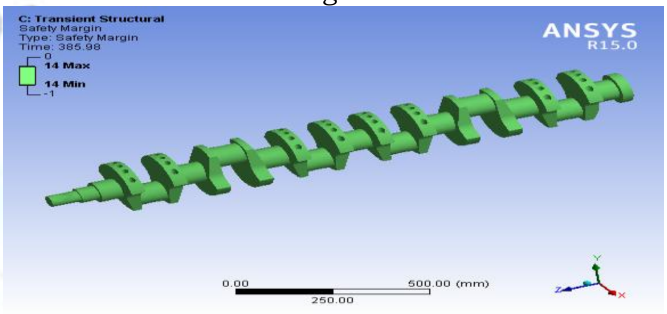


The above image shows stress ratio

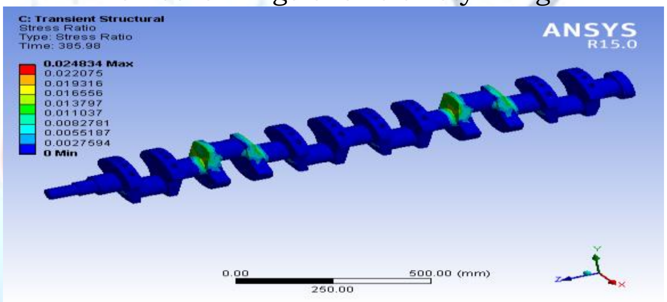
Transient Structural Analysis of Crankshaft Modified Model With Reduced Thickness 2at Time 385.98 Sec



The above image shows stress

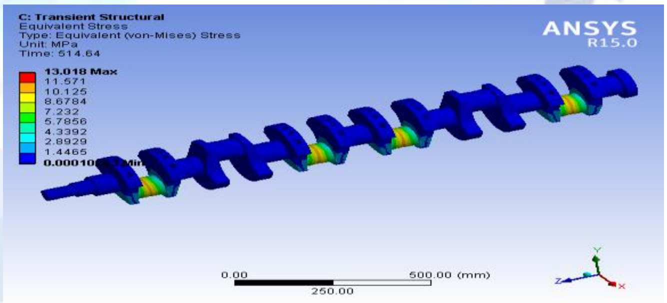


The above image shows safety margin

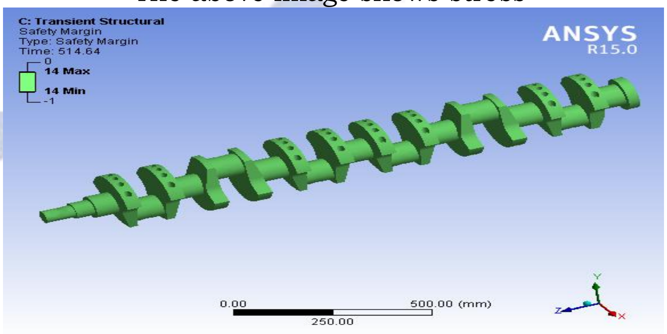


The above image shows stress ratio

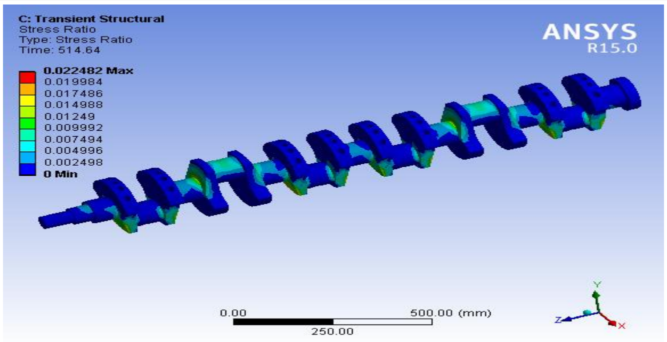
Transient Structural Analysis of Crankshaft Modified Model With Reduced Thickness 2at Time 514.64 Sec



The above image shows stress



The above image shows safety margin



The above image shows stress ratio

Results Table and Graphs

STRUCTURAL ANALYSIS					
	existing	modified	Reduced thickness1	Reduced thickness2	Material 2 Reduced thk 2
Total deformation	0.00080816	0.0073031	0.0081002	0.008708	0.02708
Stress	12.752	12.579	12.601	13.292	13.239
Strain	6.4487e-5	6.3626e-5	6.3191e-5	6.6596e-5	0.00019248

model analysis					
	existing	modified	Reduced thk1	Reduced thk 2	Material 2 Reduced thk2
Total deformation mode 1Hz	74.402	75.879	86.811	94.509	95.11
Total deformation mode 2 Hz	77.432	79.496	93.903	105.95	106.18
Total deformation mode 3 Hz	197.12	200.81	227.73	244.11	245.6

FATIGUE ANALYSIS	existing	modified	Reduced thickness1	Reduced thickness2	Material 2 Reduced thk2
LIFE	5e9	5e9	5e9	5e9	5e9
Safety factor	0.89397 to 15	0.90626 to 15	0.90466 to 15	0.85767 to 15	0.86111 to15

TRANSIENT STRUCTURAL ANALYSIS existing				
time	128.66 sec	257.32 sec	385.98 sec	514.64
stress	12.439	12.174	12.439	12.174
Safety margin	14	14	14	14
Stress ratio	0.010648	0.0098891	0.010648	0.0098891

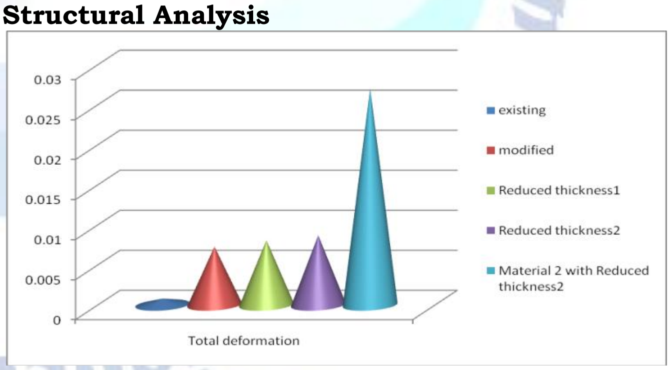
TRANSIENT STRUCTURAL ANALYSIS modified				
time	128.66 sec	257.32 sec	385.98 sec	514.64
stress	12.402	12.25	12.402	12.25
Safety margin	14	14	14	14
Stress ratio	0.011872	0.010532	0.011872	0.010532

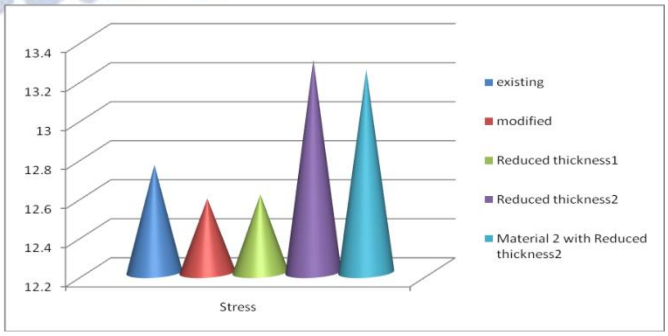
TRANSIENT STRUCTURAL ANALYSIS Reduced thickness1				
time	128.66 sec	257.32 sec	385.98 sec	514.64
stress	12.499	12.179	12.499	12.179
Safety margin	14	14	14	14
Stress ratio	0.011243	0.010063	0.011243	0.010063

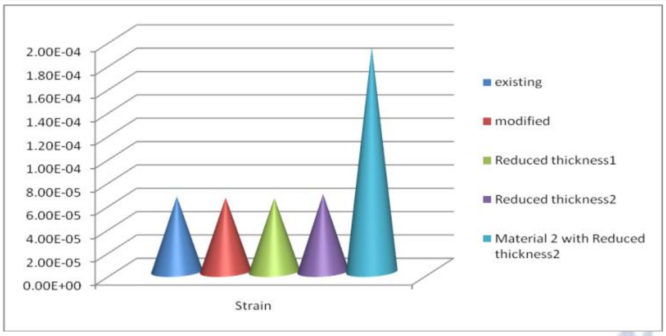
TRANSIENT STRUCTURAL ANALYSIS Reduced thickness 2				
time	128.66 sec	257.32 sec	385.98 sec	514.64
stress	12.994	13.098	12.994	13.098
Safety margin	14	14	14	14
Stress ratio	0.011863	0.010899	0.011863	0.010899

TRANSIENT STRUCTURAL ANALYSIS Material 2(al36o) with Reduced thickness2				
time	128.66 sec	257.32 sec	385.98 sec	514.64
stress	12.925	13.018	12.925	13.018
Safety margin	14	14	14	14
Stress ratio	0.024834	0.022482	0.024834	0.022482

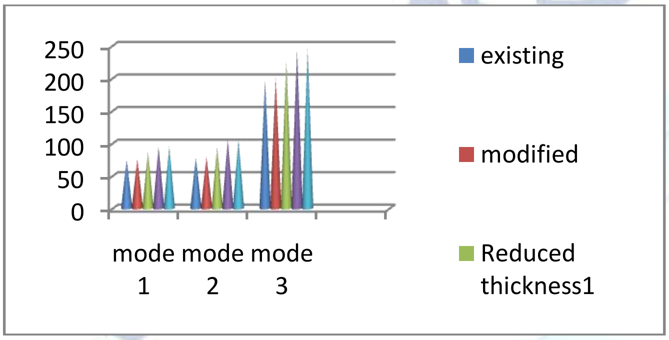
Structural Analysis



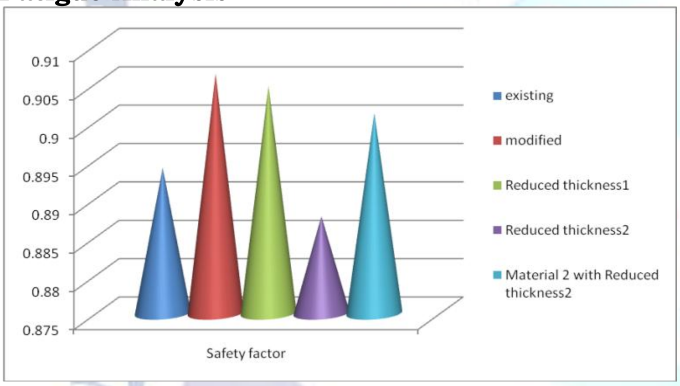




Model Analysis



Fatigue Analysis



VI. Conclusion

This project work deals with Design and analysis of four stroke six cylinder diesel engine crankshaft with geometric optimization. Initially carbonsteel used crankshaft was geometrically optimised for reducing weight and after changing material to aluminium 360 alloy,73 percent weight has reduced compared to carbonsteel used existing crankshaft.Based on the results we conclude that aluminium used crankshaft will be the better option to use because of its light weight and reduce production costs compared to carbon steel used crankshaft.

Structural, model, transient and fatigue analysis was done to obtain the result values for different operation conditions /times and also to find stress concentrated locations; then model was modified to reduce stress concentration and analysis was done; to evaluate the results according to the obtained results the modified model was showing better characteristics than existing model.

Model was again modified to reduce some more stress and stress due to self weight by reducing thickness at unwanted areas like where stress was very less.

Analysis was done on reduced thickness model then thickness was reduced again analysis was done; while observing the results for all above analysis,crankshaft with modified and reduced thickness of 40mm crankweb was the best among previous models; Analysis was carryout by using AL360 instead of carbon steel, AL360 was an alloy of aluminum, silicon, bronze Etc which is prepare for high strength and for costing/steeling process.

As per our observation companies can reduce cost and efforts by doing minute modifications to the crankshaft. Instead of carbon steel Aluminium360 alloy will be the better option for crankshaft material.

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