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Analysis of Piston, Connecting Rod and Crankshaft Assembly by Applying Different Materials

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Abstract* — The purpose of this project is to improve the engine efficiency and structural as well as thermal behaviour of the piston, connecting rod and crankshaft assembly. Theoretical calculation of the piston, connecting rod and crankshaft is developed from Hero Splendor-Pro engine specifications. Modelling of parts and its assembly were done in PTC Creo Parametric 2.0 design software and Finite Element Analysis performed using analysis software ANSYS Workbench 18.1. In this research, structural and thermal analysis; and weight optimization of assembly are done by two different materials such as Aluminium alloy and Aluminium silicon carbide (AlSiC) for piston, Grey Cast iron and Aluminium silicon carbide (AlSiC) for connecting rod & High carbon steel and Forged steel for crankshaft. Structural analysis is to evaluate the total deformation, equivalent stress and equivalent elastic strain; and thermal analysis to evaluate the temperature distribution and total heat flux. By comparison of the result of analysis, we would be able to find which material is better for the assembly of IC engine.

***Keywords*— IC Engine, Piston, Connecting Rod, Crankshaft, Modelling, Creo, FEA, ANSYS.**

I. INTRODUCTION

Automobiles are being a basic necessity in this modern era and we all know very well, because of the increased use of vehicles, there is a requirement to improve performance and reduce cost of automobile components. For the betterment of these components we need to understand the new technologies which are helpful in development of components with improved build quality.

A Piston is a reciprocating component, contained within the cylinder in IC engines and piston connected with crankshaft by the connecting rod. Due to combustion of fuel, piston reciprocates and this reciprocating motion of piston, passes to crankshaft by the connecting rod which is converted into rotary motion. In this process, high thermal stress acts on the all components of assembly.

Therefore, to reduce the thermal stress acting on the all assembly components, we strive to improve the design or choose the best material for the piston, connecting rod and crankshaft.

II. ANALYTICAL DESIGN

According to function of the piston, connecting rod and crankshaft; suitable design and selection of material is required.

A. Material Selection

Piston is generally made using aluminium alloy materials but at high temperature it has poor strength and high coefficient of thermal expansions. To overcome this problem, we have selected AlSiC composite material for the piston.

Property	Al Alloy	AlSiC
Density (kg/m^3)	2770	2950
Poisson's ratio	0.32	0.30
Young modulus (GPa)	78	210
Co-efficient of expansion (1/℃)	21×10^{-6}	9.5×10^{-6}
Tensile ultimate stress (MPa)	317	610
Tensile yield stress (MPa)	170	400
Thermal conductivity (W/m℃)	113	180
Specific heat capacity (J/kg℃)	870	990

TABLE I: COMPARISON OF PROPERTIES OF SELECTED MATERIALS FOR PISTON

Mostly material of connecting rod is steel, grey cast iron, titanium alloy, manganese alloy, etc. Here, connecting rod material has better strength and durability required because of connecting rod is a member in IC engine that appear maximum forces on it.

Property	Grey Cast iron	AlSiC
Density (kg/m^3)	7250	2950
Poisson's ratio	0.211	0.30
Young modulus (GPa)	75	210
Co-efficient of expansion (1/℃)	6×10^{-6}	9.5×10^{-6}
Tensile ultimate stress (MPa)	415	610
Tensile yield stress (MPa)	274	400
Thermal conductivity (W/m℃)	53	180
Specific heat capacity (J/kg℃)	460.5	990

TABLE II: COMPARISON OF PROPERTIES OF SELECTED MATERIALS FOR CONNECTING ROD

Crankshaft provides the turning motion to the wheels, which converts the reciprocating motion of the pistons into a rotary motion. So, forged steel material, which has high strength is more feasible for crankshaft.

Property	**High carbon steel**	**Forged steel**
Density (kg/m^3)	7800	7700
Poisson's ratio	0.295	0.29
Young modulus (GPa)	200	221
Co-efficient of expansion (1/℃)	10.8×10^{-6}	11.9×10^{-6}
Tensile ultimate stress (MPa)	635	827
Tensile yield stress (MPa)	490	625
Thermal conductivity (W/m℃)	52	42.2
Specific heat capacity (J/kg℃)	495	540

TABLE III: COMPARISON OF PROPERTIES OF SELECTED MATERIALS FOR CRANKSHAFT

B. Theoretical Calculations

Design calculations of piston, connecting rod, crankshaft are done by taking the reference as a 4 stroke single cylinder air cooled Hero Splendor – Pro 100cc engine specifications.

Engine Type	Air Cooled 4-Stroke Single Cylinder
Max Power	6.15kw @8000rpm
Max Torque	8.05N.m @5000rpm
Bore Diameter	50mm
Stroke Length	49.5mm
Engine Capacity	97.2cc
Starting	Kick Start/Self Start
Compression Ratio	9.9:1

TABLE IV: ENGINE SPECIFICATIONS

1) Design of piston:

a. Design of piston head or crown

Thickness of the piston head, according to Grashoff's formula

$$t_H = D \times \sqrt{\frac{3 \times p_{max}}{16 \times \sigma_t}}$$

Where, p = Maximum gas pressure or explosion pressure in N/mm2,

D = Cylinder bore or outside diameter of the piston in mm, and

σ_t = Permissible bending (tensile) stress for the material of the piston in MPa or N/mm^2 (For Al alloy $\sigma_t = 152.2\ MPa$)

$$t_H = 50 \times \sqrt{\frac{3 \times 3.645}{16 \times 152.2}} = 3.35\text{mm}$$

b. Design of piston rings

Radial thickness of the ring

$$t_1 = D \times \sqrt{\frac{3 \times p_w}{\sigma_t}}$$

Where, p_w = Pressure of gas on the cylinder wall in N/mm^2 = 0.025 N/mm^2 to 0.042 N/mm^2

$$t_1 = 50 \times \sqrt{\frac{3 \times 0.042}{152.2}} = 1.438\text{mm}$$

Axial thickness of the ring

$$t_2 = 0.7t_1 \text{ to } t_1 = 0.92t_1 = 0.92 \times 1.438 = 1.322\text{mm}$$

Height of the top land (the distance from the top of the piston to the first ring groove)

$$h_1 = t_H \text{ to } 1.2t = 1.2 \times 3.35 = 4.02\text{mm}$$

Height of other ring land (the distance between the ring grooves)

$$h_2 = 0.75t_2 \text{ to } t_2 = 0.75 \times 1.322 = 0.991\text{mm}$$

c. Design of piston barrel

Thickness of the piston barrel at top land

$$t_3 = 0.03D + b + 4.5$$

Where, b = Radial depth of piston ring groove which is taken as 0.4 mm larger than the radial thickness of the piston ring (t_1)

$$t_3 = t_1 + 0.4\text{ mm} = 0.03 \times 50 + 1.438 + 4.5 = 7.438\text{mm}$$

Thickness of the piston barrel at bottom land

$$t_4 = 0.25t_3 \text{ to } 0.35t_3 = 0.25 \times 7.438 = 1.859\text{mm}$$

d. Design of piston skirt

Length of the piston skirt

$l_{ps} = 0.65D$ to $0.8D = 0.5\times50 = 25$mm

Total length of the piston

L = Length of skirt+ Length of ring section + Top land $= 25 + [(3\times0.911) + (2\times1.322)] + 4.02$

$= 34.397$mm

e. Design of piston pin

Outside diameter of the piston pin

$d_o = 0.3D$ to $0.45D = 16\text{mm} \approx 12\text{mm}$

Inside diameters of the piston pin

$d_i = 0.6\times d_o = 12\text{mm} \approx 8\text{mm}$

2) *Design of Connecting rod:*

a. Dimensions of cross-section of the connecting rod

Let thickness of the flange and web of the section $= t$

Width of the section, $B = 4$t

Depth or height of the section, $H = 5$t

$I_{xx} = 4I_{yy}$

Where, Ixx = Moment of inertia of the section about X-axis, and

Iyy = Moment of inertia of the section about Y-axis.

But, $I_{xx} \leq 4\, I_{yy}$

Area of the connecting rod section,

$A = 2(4\text{t}\times\text{t}) + (3\text{t}\times\text{t}) = 11\text{t}$

$$I_{xx} = \frac{1}{12}[4t(5t)^3 - 3t(3t)^3] = \frac{419}{12}t^4$$

$$I_{yy} = 2 \times \frac{1}{12}[t(4t)^3] + \frac{1}{12}[3t(t)^3] = \frac{131}{12}t^4$$

$$\frac{Ixx}{Iyy} = \frac{419}{12} \times \frac{12}{131} = 3.2$$

The force on the connecting rod (F_C) equal to the maximum force on the piston (F_P) due to gas pressure

$$F_c = F_P = \frac{\pi}{4} \text{ x } D^2 \text{ x } P_{\text{ max}} = \frac{\pi}{4} \text{ x } (50)^2 \text{ x } (3.645) = 7.153 \text{ x } 10^3 \text{ N}$$

The Critical buckling load

$W_B = F_c$ x $FOS = 7.153 \text{ x } 10^3 \text{ x } 6 = 4.291 \text{ x } 10^4$ N (Factor of safety = 6)

Radius of gyration of the section about X-axis,

$I = Ak^2$

$$k_{xx} = \sqrt{\frac{I_{xx}}{A}} = \sqrt{\frac{419}{12}t^4 \times \frac{1}{11t^2}} = 1.78t$$

Length of crank,

$$r = \frac{\text{Stroke of piston}}{2} = \frac{50}{2} = 25\text{mm}$$

Length of the connecting rod,

$L = 2\times$Stroke of piston $= 2\times50 = 100$mm

Now according to Rankine's formula,

$$W_B = [\sigma_c \text{ x } A] / \left[1 + \alpha\left(\frac{L}{k_{xx}}\right)^2\right]$$

We'll find out the value of (***t***) using

$\sigma_c = 415$ MPa $\qquad A = 11t^2$

$\alpha = \frac{1}{7500} \qquad k_{xx} = 1.78t$

$$4.291\text{x } 10^4 = [415\text{x } (11t^2)] / \left[1 + \frac{1}{7500}\left(\frac{2 \text{ x } 49.5}{1.78t}\right)^2\right]$$

$$\frac{4.291\text{x } 10^4}{415\text{x } 11} = \frac{t^2}{1 + \frac{0.404}{t^2}}$$

$$103.42 = \frac{t^4}{t^2 + 0.404}$$

$$103.42t^2 + 43.52 = t^4$$

$$t^4 - 103.42t^2 - 43.52 = 0$$

$$t^2 = \frac{103.42 \pm \sqrt{103.42 + 4(43.52)}}{2}$$

$t = 3.13$ mm ≈ 3 mm

Width of the section, $B = 4t = 12$ mm

Depth or height of the section, $H = 5t = 15$ mm

Depth near the big end, $H_1 = 1.1H$ to $1.25H = 18$ mm

Depth near the small end, $H_2 = 0.9H = 12.75$mm

Diameter near the big end = $H_1 \times B = 18 \times 12 = 216$ mm

Diameter near the small end = $H_2 \times B = 12.75 \times 12 = 153$ mm

b. Dimensions of the crankpin at the big end

Load on the crank pin

$$F_C = l_c \times d_c \times p_{bc}$$

Where, $p_{bc} = 10$ MPa

$l_c = 1.25d_c$ to $1.5d_c$

$$7.153 \text{ x } 10^3 = 1.3\, d_c^2 \text{ x } 10$$

$d_c = 24$mm

$l_c = 1.3\, d_c = 31$mm

c. Dimensions of the piston pin at the small end

Load on the piston pin

$$F_P = l_p \times d_p \times P_{bp}$$

Where, $l_p = 1.5d_p$ to $2\, d_p$

$P_{bp} = 15$ MPa

$$F_L = 2\, d_p^2 \times P_{bp}$$

$$7.153 \text{ x } 10^3 = 2\, d_p^2 \text{ x } 15$$

$d_p = 15.44 \approx 16$ mm

$l_p = 2\, d_p = 32$ mm

3) *Design of Crankshaft:*

F_P = The force on the connecting rod equal to the maximum force on the piston (F_P) due to gas pressure

Due to the piston gas pressure (F_p) acting horizontally,

There will be two horizontal reactions H_1 & H_2 at bearings 1 & 2 respectively.

Considering the crankpin as simply supported beam i.e., $H_1 = H_2$

$F_p = H_1 + H_2 = 28574$ N

b = distance between the bearings 1 & 2 is equal to twice the bore diameter $= 2D = 2\times100 = 200$ mm

$b_1 = b_2 = b/2 = 50$ mm

a. Design of crank web

Thickness of the crank web

$t = 0.4ds$ to $0.6ds = 0.22D$ to $0.32D = 0.65\, d_c + 6.35$

Where, ds = Shaft diameter in mm,

D = Bore diameter in mm, and

dc = Crankpin diameter in mm

$t = 0.32D = 0.32\times100 = 32$mm

Width of the crank web

$w = 1.125dc + 12.7 = 1.125\times24+12.7 = 39.5$ mm

Maximum bending moment on the crank web

$$M = H_1\left(b_2 - \frac{l_c}{2} - \frac{t}{2}\right) = 14287\left(50 - \frac{31}{2} - \frac{32}{2}\right) = 264309.5 \text{ N-mm}$$

Section modulus

$$Z = \frac{1}{6} \times w \times t^2 = \frac{1}{6} \times 39.5 \times 32^2 = 6741.33 \text{ mm}^3$$

Bending stress on the crank web

$$\sigma_b = \frac{M}{Z} = \frac{264309.5}{6741.33} = 39.2073 \text{N/mm}^2$$

Compressive stress on the crank web

$$\sigma_c = \frac{H_1}{wt} = \frac{14287}{39.5\times32} = 11.3030 \text{N/mm}^2$$

Total stress on the crank web

$$\sigma = \sigma_b + \sigma_c = 39.2073+11.3030 = 50.5103 \text{N/mm}^2$$

Here total stress is less than the yield strength of carbon steel (560N/mm^2)

b. Design of shaft

Diameter of the shaft

$$ds = \frac{t}{0.6} = \frac{32}{0.6} = 53.33 \text{mm}$$

III. ANALYSIS

We did two types of analysis: structural and thermal, using the ANSYS software. ANSYS is analysis software that gives the results according to the input parameters such as meshing and boundary conditions.

A. *Geometrical Modelling*

Initial 3D models of the pistons, connecting rod and crankshaft prepared from the calculated dimensions in Creo Parametric 2.0 design software, which is shown in below figures.

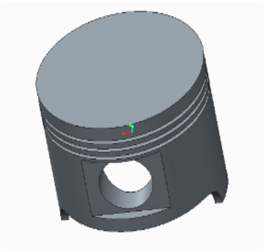

Fig 1: Piston Geometry

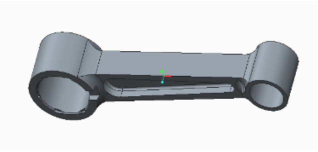

Fig 2: Connecting Rod Geometry

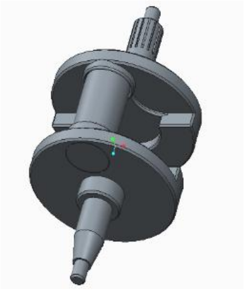

Fig 3: Crankshaft Geometry

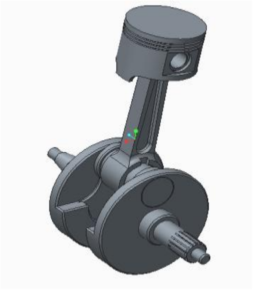

Fig 4: Assembly Geometry

B. *Meshing of 3D Model*

After modelling, mesh is generated and it contains 36920 nodes, 15844 elements and 5.219×10^{-5}m of minimum edge length.

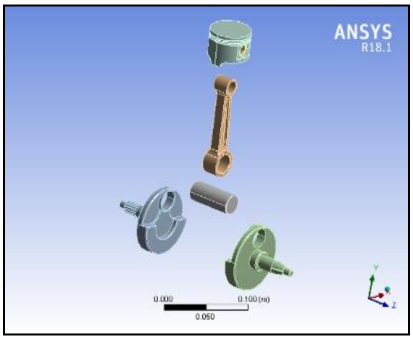

Fig 5: Exploded View of Assembly

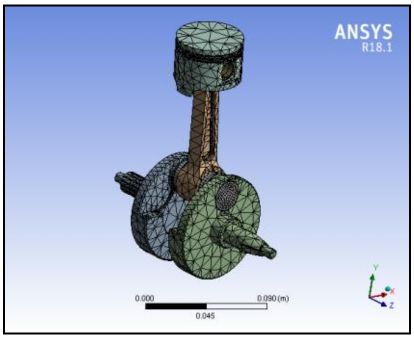

Fig 6: Meshing Model of Assembly

C. *Boundary Conditions*

1) *For The Structural Analysis:* Combustion of gases exerts pressure on the piston head during power stroke and the piston will move from Top Dead Center (TDC) to Bottom Dead Center (BDC) because of the fixed support with the connecting rod at pin hole. The gas pressure 5MPa on top surface of piston and fixed support at piston pin hole are given as boundary condition for structural analysis. Same boundary condition is given to connecting rod and crankshaft.

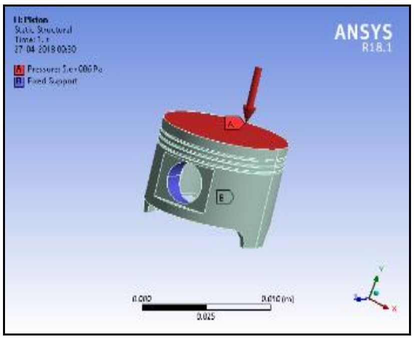

Fig 7: Structural Boundary Conditions for the Piston

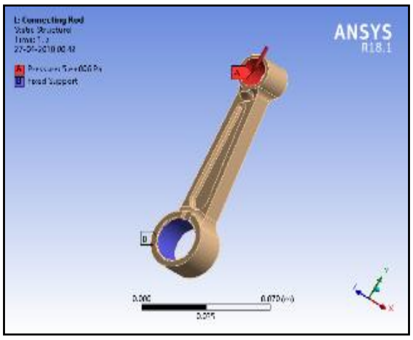

Fig 8: Structural Boundary Conditions for the Connecting Rod

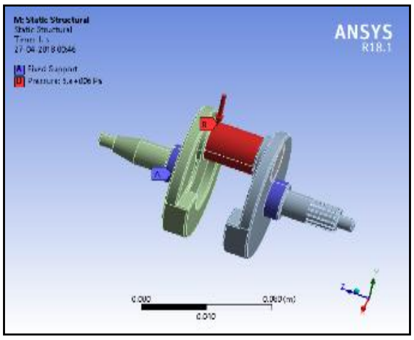

Fig 9: Structural Boundary Conditions for the Crankshaft

2) *For The Thermal Analysis*: Boundary condition of the thermal analysis is 500°C at head of both pistons and co-efficient of convection on surface of the piston is given according to the material.

3)

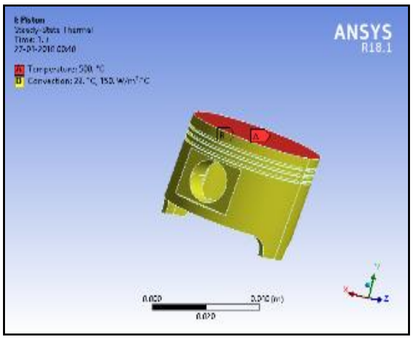

Fig 10: Thermal Boundary Conditions for the Piston

D. Analysis Solution

Finite Element Analysis of piston, connecting rod and crankshaft assembly performed for materials such as aluminium alloy and Aluminium silicon carbide (AlSiC) for piston, Grey Cast iron and Aluminium silicon carbide (AlSiC) for connecting rod & High carbon steel and Forged steel for crankshaft. The pictures of structural analysis and thermal analysis for piston performed by ANSYS is shown in figures below.

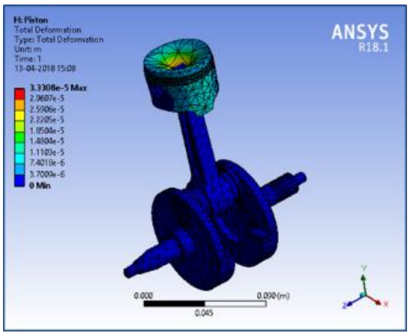

Fig 11: Total Deformation of Al Alloy Piston

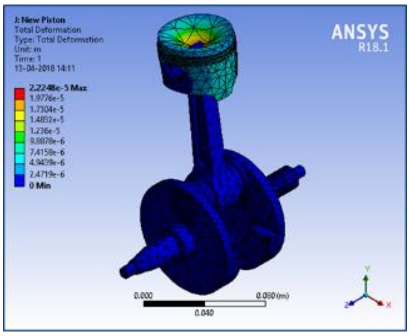

Fig 12: Total Deformation of AlSiC Piston

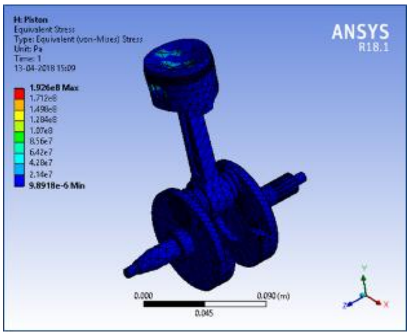

Fig 13: Stress Distribution of Al Alloy Piston

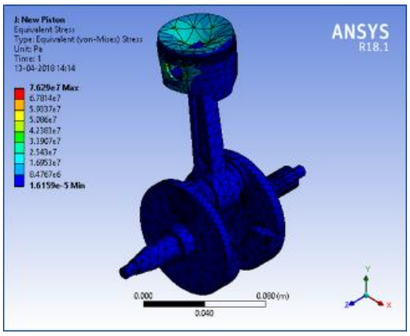

Fig 14: Stress Distribution of AlSiC Piston

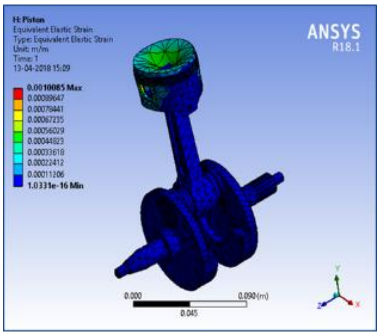

Fig 15: Strain Distribution of Al Alloy Piston

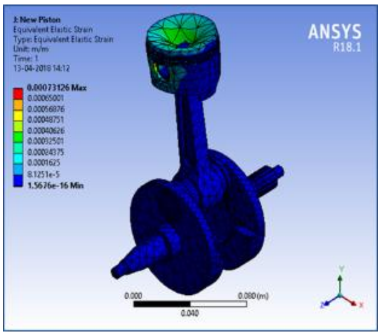

Fig 16: Strain Distribution of AlSiC Piston

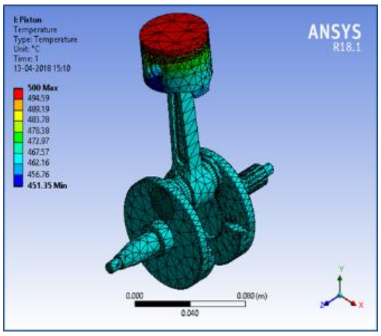

Fig 17: Temperature Distribution of Al Alloy Piston

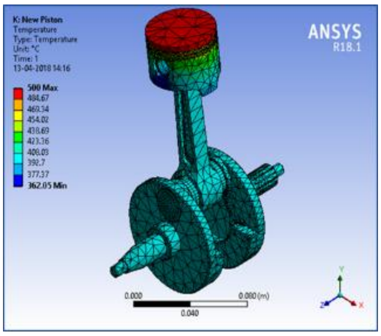

Fig 18: Temperature Distribution of AlSiC Piston

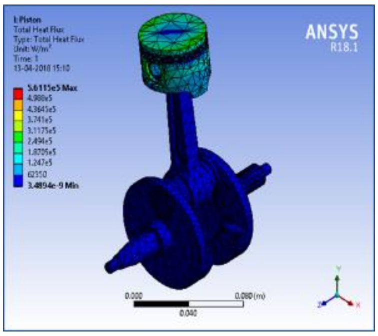

Fig 19: Total Heat Flux of Al Alloy Piston

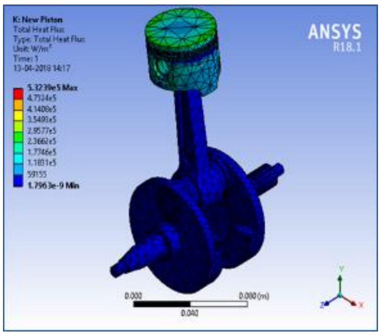

Fig 20: Total Heat Flux of AlSiC Piston

E. Analysis Results

The comparison of piston, connecting rod and crankshaft analysis result in terms of structural analysis is done under the parameters: total deformation, equivalent stress, equivalent elastic strain and thermal analysis is done under two parameters temperature distribution, total heat flux is shown in table form.

1) Analysis Results of Piston:

Parameters	Al Alloy		AlSiC	
	Min	Max	Min	Max
Total deformation (m)	0	3.3308×10^{-5}	00	2.224×10^{-5}
Stress (Pa)	9.8918×10^{-6}	1.926×10^{8}	1.615×10^{-5}	7.629×10^{7}
Strain (Pa)	1.0331×10^{-16}	0.0010085	1.567×10^{-16}	0.00073126
Temperature (°C)	451.35	500	362.05	500
Total heat flux (W/m^2)	3.4894×10^{-9}	5.6115×10^{5}	1.796×10^{-9}	5.323×10^{5}
Weight (kg)	0.24806		0.2704	

TABLE V: COMPARISON OF Al ALLOY AND AlSiC PISTON

2) *Analysis Results of Connecting Rod*:

Parameters	Grey Cast Iron		AlSiC	
	Min	Max	Min	Max
Total deformation(m)	0	0.00083158	0	0.00053993
Stress(Pa)	3.6647×10^{-11}	1.4833×10^{8}	0.00018928	1.4793×10^{8}
Strain(Pa)	7.1113×10^{-22}	0.0013713	6.2513×10^{-15}	0.0021157
Weight(kg)	0.14911		0.13676	

TABLE VI: COMPARISON OF GREY CAST IRON AND AlSiC CONNECTING ROD

3) *Analysis Results of Crankshaft*:

Parameters	High Carbon Steel		Forged Steel	
	Min	Max	Min	Max
Total deformation(m)	0	4.974×10^{-6}	0	1.7593×10^{-6}
Stress(Pa)	0.091942	3.1303×10^{7}	0.24086	6.4873×10^{7}
Strain(Pa)	1.1039×10^{-12}	0.0001846	2.6043×10^{-12}	0.00040732
Weight(kg)	1.02043		0.9506	

TABLE VII: COMPARISON OF HIGH CARBON STEEL AND FORGED STEEL CRANKSHAFT

IV. CONCLUSIONS

By observing the above analysis results of two assemblies we can conclude that using AlSiC for both piston and connecting rod; Forged steel for crankshaft is more beneficial than using Al alloys for piston, gray cast iron for connecting rod and high carbon steel for crankshaft.

- From the numerical analysis it is clear that modified components has better performance in temperature distribution and heat flux in comparison with actual components under the joint action of the thermal and mechanical loads
- While comparing stresses of the modified assembly is having less stresses than actual assembly. Hence it has more strength as compared to actual assembly.
- By changing the components material we can reduce the deformation 32.38% in piston, 35.07% in connecting rod, 35.36% in crankshaft as compared to conventional material components. So this will increase the life span of assembly.
- Weight reduction of 8.26 % in Al Alloy piston, 8.28% in AlSiC connecting rod and 6.15% in forged steel crankshaft have been observed as compared to actual material components so that we can conclude that modified assembly is having more mechanical efficiency thereby decreasing balancing and inertia problems.

REFERENCES

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