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Analysis of Piston Head Using Different Materials

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I. INTRODUCTION

Engine pistons are one of the most complex components among all automotive and other industry field components. The engine can be called the heart of a vehicle and the piston may be considered the most important part of an engine. There are lots of research works proposing, for engine pistons, new geometries, materials and manufacturing techniques, and this evolution has undergone with a continuous improvement over the last decades and required thorough examination of the smallest details. Notwithstanding all these studies, there are a huge number of damaged pistons. Damage mechanisms have different origins and are mainly ***wear, temperature, and fatigue related***. But more than wear and fatigue, damage of the piston is mainly due to stress development, namely- ***Thermal stress, Mechanical stress.*** The main objectives are to investigate and analyze the thermal stress and mechanical stress distribution of piston at the real engine condition during combustion process.

A piston is a component of reciprocating IC-engines. It is the moving component that is contained by a cylinder and is made gas-tight by piston rings. In an engine, its purpose is to transfer force from expanding gas in the cylinder to the crankshaft via a piston rod. In engine, transfer of heat takes place due to difference in temperature and from higher temperature to lower temperature. Thus, there is heat transfer to the gases during intakes stroke and the first part of the compression stroke, but the during combustion and expansion processes the heat transfer take place from the gases to the walls.

So the piston crown, piston ring and the piston skirt should have enough stiffness which can endure the pressure and the friction between contacting surfaces. In addition, as an important part in engine, the working condition of piston is directly.

II. LITERATUREREVIEW

An optimized piston which is lighter and stronger is coated with zirconium for bio-fuel. In this regards some of the papers have been studied and details given below as The paper[1], the coated piston undergone a Von misses test by using ANSYS for load applied on the top. Analysis of the stress distribution was done on various parts of the coated piston for finding the stresses due to the gas pressure and thermal variations. Von-misses stress is increased by 16% and deflection is increased after optimization. But all the parameters are well with in design consideration.

Design, Analysis and optimization of piston [2] which is stronger, lighter with minimum cost and with less time. Since the design and weight of the piston influence the engine performance. Analysis of the stress distribution in the various parts of the piston to know the stresses due to the gas pressure and thermal variations using with Ansys.

With the definite-element analysis software, a three-dimensional definite-element analysis [3] has been carried out to the gasoline engine piston. Considering the thermal boundary condition, the stress and the deformation distribution conditions of the piston under the coupling effect of the thermal load and explosion pressure have been calculated, thus providing reference for design improvement. Results show that, the main cause of the piston safety, the piston deformation and the great stress is the temperature, so itis feasible to further decrease the piston temperature with structure optimization.

The paper [4] involves simulation of a 2-stroke 6S35ME marine diesel engine piston to determine its temperature field, thermal, mechanical and coupled thermal-mechanical stress. The distribution and magnitudes of the afore-mentioned strength parameters are useful in design, failure analysis and optimization of the engine piston.

This work [5] describes the stress distribution of the piston by using finite element method (FEM). FEM is performed by using computer aided engineering (CAE) software.

III. THEORITICALBACKGROUND

The piston are made of different materials such as Carbon steel, Cast Iron, Aluminum alloys etc. For this project material selected as Aluminum Alloy and Carbon steel.

Pistons are commonly made of aluminum alloy .Generally in this material aluminum is the main component. In addition to main component aluminum it consists of copper 4-5%, ferrous -1.3%, silicon- 16 to18%, magnesium- 0.45-65%, zinc- 1.5% and nickel- 0.1% aluminum alloy is used for excellent and lightweight thermal conductivity. Thermal conductivity is the ability of a material to conduct and transfer heat. Aluminum expands when heated and proper clearance must be provided to maintain free piston movement in the cylinder bore. Piston material was assumed to be aluminum alloy which is homogenous, isotropic and linear elastic with a Poisson's ratio of 0.33.

Carbon steel has been used to compare the above said characteristic with Aluminum alloy.

Abbreviations

m_p= mass of the piston (Kg)
V = volume of the piston (mm^3)
t_H= thickness of piston head (mm)
D = cylinder bore (mm)
P_{max}= maximum gas pressure or explosion pressure (MPa)
σ_t= allowable tensile strength (MPa)
σ_{ut}= ultimate tensile strength (MPa)
F.O.S = Factor of Safety
k = thermal conductivity
T_c = temperature at the centre of the piston head (^{0}C)
T_e= temperature at the edge of the piston head (^{0}C)
HCV = Higher Calorific Value of fuel (KJ/Kg)
BP = brake power of the engine per cylinder (KW)
N= rpm of engine
r_c =Compression Ratio
m = mass of fuel used per brake power per second
C = ratio of heat absorbed by the piston to the total heat developed in the cylinder = 5% or 0.05
b = radial thickness of ring (mm)

P_w = allowable radial pressure on cylinder wall (N/mm^2)

h_2 = axial thickness of piston ring (mm)

t_1 = thickness of piston barrel at the top end (mm)

t_2 = thickness of piston barrel at the open end (mm)

d_o = outer diameter of piston pin (mm)

Engine Specifications:

Engine make: Kirloskar

Bore Diameter: 80mm
Stroke Length: 110mm.

The Physical and material properties of Aluminum Alloy are given below:
Density – 2770 (Kg/m^3)

Poisson Ratio – 0.33

Young Modulus – $7.1x10^{10}$ (Pa)

Tensile Ultimate Strength – $3.1x10^8$(Pa)

Tensile Yield Strength – $2.8x10^8$(Pa)

Compressive Yield strength – 2.8 $x10^8$(Pa)

F.O.S = Factor of Safety = 2.25
k = thermal conductivity =174.15(W/m C)
HCV = Higher Calorific Value of fuel (KJ/Kg) = 47000 KJ/Kg (petrol)
BP = brake power of the engine per cylinder (KW) =4KW. Value obtained experimentally considering the following conditions. N=1500rpm [4]
r_c = Compression ratio 16.5, fully loaded condition.[4]
m = mass of fuel used per brake power per second (Kg/KW s) =0.25/3600 (Kg/KW s).Value obtained experimentally considering the following conditions: B.P=4KW, CV=47000Kj/kg (petrol), N=1500, fully loaded condition.
C = ratio of heat absorbed by the piston to the total heat developed in the cylinder = 5% or 0.05
P_w = allowable radial pressure on cylinder wall (N/mm^2) = 0.042 MPa

Calculation of Dimension of piston head

Thickness of Piston Head (t_H) :The piston thickness of piston head calculated using the following Grashoff's formula,

$$\mathbf{t_H} = D \sqrt{(3P)/(16\sigma_t)} \text{ in mm}$$

P= Maximum Pressure in N/mm^2 = 8 N/mm^2

D= cylinder bore/outside diameter of the piston in mm= 80mm. σ_t=permissible tensile stress for the material of the piston.
= σ_t =280/2.25=124.4 MPa.

$$\mathbf{t_H = 8.9mm}.$$

Heat Flow through the Piston Head (H)
The heat flow through the piston head is calculated using the formula

$$H = 12.56 * t_H * k * (Tc-Te) \text{ KJ/sec}$$

Where

k=thermal conductivity of material which is 174.15W/mC
Tc = temperature at center of piston head in °C.
Te = temperature at edges of piston head in °C. (Tc-Te)=75°C for Aluminium alloy.

On the basis of the heat dissipation, the thickness of the piston head is given by:

$$H = [C \times HCV \times m \times BP]$$
$$= 0.05 \times 47000 \times 0.25/3600 \times 4$$
$$= 0.6527 \text{ KJ/s}$$

$$t_H = 3.98 \text{ mm}$$

Comparing both the dimensions, for design purpose we will be considering the maximum thickness, hence required thickness of piston head is 8.9 mm.

Radial Thickness of Ring (t_1):

$$t_1 = D\sqrt{3p_w/\sigma_t}$$

Where,

D = cylinder bore in mm=80mm.

P_w= pressure of fuel on cylinder wall in N/mm². Its value is limited from 0.025N/mm² to 0.042N/mm². Here P_w value is taken as 0.042N/mm² while σ_t= 124.4Mpa for aluminum alloy.

t_1 =3mm.

Axial Thickness of Ring (t_2)

The thickness of the rings may be taken as $t_2 = 0.7t_1$ to t_1
=0.7 t_1=2.1mm.

t_2 = 2.1 mm

Number of rings (n_r) Minimum axial thickness (t_2) t_2= D/(10*n_r)

n_r = 3.86 or 4 rings.

n_r=4 rings

Width of the top land (b_1) :

The width of the top land varies from $b_1 = t_H$ to 1.2 t_H
=1.2 t_H = 1.2 x8.9 =10.68mm.

b_1=10.68 mm

Width of other lands (b2):

Width of other ring lands varies from $b_2 = 0.75t_2$ to t_2

b_2=0.75 t_2= 0.75x2.1=1.575mm.

b_2=1.575 mm

Maximum Thickness of Barrel at the top end (t_3):

Radial depth of the piston ring grooves (b) is about 0.4 mm more than radial thickness of the piston rings(t_1),therefore

b = 0.4 +t_1 =0.4+3 =3.4 mm
t_3 = 0.03D + b + 4.5 mm

t_3=10.7 mm

Thickness of piston barrel at the open end (t_4):
t_4= 0.25 t_1 to 0.35 t_1) t_4=0.25x10.7=2.675mm

t_4=2.675 mm

Piston pin diameter (d_o):
d_o=0.03D=24mm.

Theoretical Stress Calculation:
The piston crown is designed for bending by maximum gas forces Pz_{max} as uniformly loaded round plate freely supported by a cylinder. The stress acting in MPa on piston crown:

$\sigma_b=M_b/W = Pz_{max}(ri/\delta)2$ Where
Where, $M_b= (1/3)\, Pz_{max}\, r_i^3$ is the bending moment, MN m;

$W_b= (1/3)\, ri\delta^2$ is the moment of resistance to bending of a flat crown, m^3;
$P_{zmax}= Pz$, is the maximum combustion pressure, MPa =5Mpa

This value varies from 2Mpa-5Mpa in case of aluminum alloy.

$ri= [D / 2 - (s + t1 + dt)]$ is the crown inner radius, m.;

Where, Thickness of the sealing part s = 0.05D= 0.05X80=4mm. Radial clearance between piston ring and channel dt= 0.0008m Radial thickness of ring (t_1) =3mm.

Therefore, *ri=[0.08/2-(0.004+0.003+0.0008)]=0.0322m*

Thickness of piston crown δ=(0.08 to 0.1)XD= 0.085X80=7mm. σb= 5X[(0.0322/0.007)^2] Mpa= 105.8Mpa. Hence required theoretical stress 105.8 Mpa

Hence the ***design is safe. i,e*** obtained stress (105.8Mpa) < Allowable stress (124.4Mpa)

For carbon steel material
The Physical and material properties of Carbon steel is given below: [3]
Density – 2250 (Kg/m^3)
Poisson Ratio – 0.29
Young Modulus – $20.7x10^{10}$ (Pa)
Tensile Ultimate Strength – $5.4x10^8$(Pa)
Tensile Yield Strength – $13.2x10^8$(Pa)
Compressive Yield strength – 12.4 $x10^8$(Pa)

Thickness of Piston Head (t_H) : The piston thickness of piston head calculated
Using the following Grashoff's formula,

$$t_H = D \sqrt{(3P)/(16\sigma_t)} \text{ in mm}$$

P= maximum pressure in N/mm²=8 N/mm².
This is the maximum pressure that carbon steel can withstand.
D= cylinder bore/outside diameter of the piston in mm= 80mm. σ_t=permissible tensile stress for the material of the piston.
= σ_t =540/2.25=240 MPa.

$$\mathbf{t_H = 6.32mm}.$$

Heat Flow through the Piston Head (H)
The heat flow through the piston head is calculated using the formula

$$H = 12.56 t_H\, k\, (Tc-Te) \text{ KJ/sec}$$

Where
k=thermal conductivity of material which is 45W/mC , Tc = temperature at center of piston head in °C.
Te = temperature at edges of piston head in °C. (Tc-Te)=75°C for carbon steel alloy. On the basis of the heat dissipation, the thickness of the piston head is given by:

$$H = [C \times HCV \times m \times BP]$$
$$= 0.05 \times 47000 \times 0.25/3600 \times 4$$
$$=0.6527 \text{ KJ/s}$$
$$t_H = H/(12.56 \times k\, (Tc – Te))$$
$$= Hx1000/12.56 \times 45 \times 75$$
$$=15.39mm.$$
$$\mathbf{t_H=15.39\ mm}$$

Comparing both the dimensions, for design purpose we will be considering the maximum thickness, hence required thickness of piston head is 15.39mm.

Radial Thickness of Ring (t_1):

$$t_1 = D\sqrt{3P_w/\sigma_t}$$

Where,
D = cylinder bore in mm=80mm.
P_w= pressure of fuel on cylinder wall in N/mm². Its value is limited from 0.025N/mm² to 0.042N/mm². Here P_w value is taken as 0.042N/mm² while σ_t= 240Mpa for carbon steel.

$$\mathbf{t_1=2mm.}$$

Axial Thickness of Ring (t_2)
The thickness of the rings may be taken as $t_2 = 0.7t_1$ to t_1

$$\mathbf{t_2=0.7\ t_1=1.4mm.}$$

Number of rings (n_r) Minimum axial thickness (t_2) t_2= D/($10*n_r$)

n_r = 5.71 or 6 rings.

$\mathbf{n_r = 6}$

Width of the top land (b_1)

The width of the top land varies from $b_1 = t_H$ to 1.2 t_H

b_1=1.2 t_H = 1.2 x15.39=18.47mm.

$\mathbf{b_1=18.47\ mm}$

Width of other lands (b_2):

Width of other ring lands varies from $b_2 = 0.75t_2$ to t_2

=0.75 t_2= 0.75x1.4=1.05mm.

$\mathbf{b_2=1.05\ mm}$

Maximum Thickness of Barrel at the top end (t_3):

Radial depth of the piston ring grooves (b) is about 0.4 mm more than radial thickness of the piston rings(t_1),therefore

b = 0.4 +t_1 =0.4+2 =2.4 mm

t_3 = 0.03D + b + 4.9 mm

t_3=0.03x80+2.4+4.9=9.7mm.

$\mathbf{t_3=9.7\ mm}$

Thickness of piston barrel at the open end (t_4):

t_4= 0.25 t_1 to 0.35 t_1)

t_4=0.25*9.7=2.425mm

$\mathbf{t_4= 2.425\ mm}$

Piston pin diameter (d_o):

d_o=0.03D=2.4 mm.

$\mathbf{d_0= 2.4\ mm}$

Theoretical Stress Calculation:

The piston crown is designed for bending by maximum gas forces Pz_{max}as uniformly loaded round plate freely supported by a cylinder. The stress acting in MPa on piston crown:

$$\sigma_b=M_b/W\!\!\!-b=Pz_{max}(ri/\delta)^2$$

Where, M_b= (1/3) $Pz_{max}\ r_i^3$is the bending moment, MN m;

W_b= (1/3) $ri\delta^2$is the moment of resistance to bending of a flat crown, m^3;

Pz_{max}= P_z, is the maximum combustion pressure, MPa;=5 Mpa

This value varies from 2Mpa-5Mpa in case of carbon steel.

The crown inner radius,

$$ri= [D / 2 - (s + t1 + dt)]\ \ (m)$$

Where, Thickness of the sealing part s = 0.05D= 0.05X80=4mm. Radial clearance between piston ring and channel: dt= 0.0008m Radial thickness of ring (t_1) =3mm.

Therefore, *ri=[0.08/2-(0.004+0.002+0.0008)]=0.0332m*

Thickness of piston crown δ=(0.08 to 0.1)XD= 0.085X80=7mm.

σ_b= 5X[(0.0332/0.007)^2] Mpa= 112.45Mpa.

Hence theoretical stress obtained 112.45 Mpa

Hence the ***design is safe. i,e*** obtained stress (112.48Mpa) < Allowable stress (240Mpa)

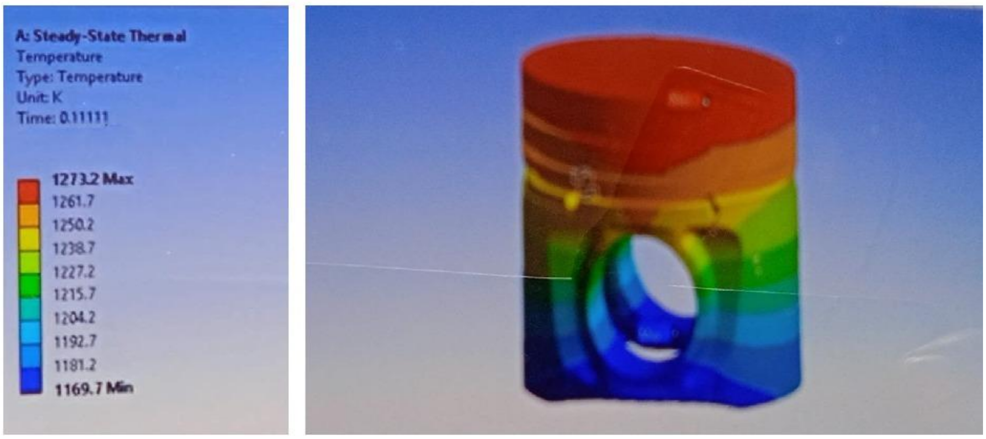
Table -1: Comparison of dimension of piston head using Aluminium alloy and carbon steel

Comparison of dimension of piston head using Aluminium alloy and carbon steel		
Dimension of piston Head	**Aluminium alloy**	**Carbon steel**
Thickness of piston head on basis of stress (tH)	8.9 mm	6.32 mm
Thickness of piston head on basis of heat flow (tH)	3.98 mm	15.39 mm
Max. thickness of piston head for design (tH)	8.9 mm	15.39 mm
Radial thickness (t1)	3.0 mm	2.0 mm
Axial thickness (t2)	2.1 mm	1.4 mm
No of rings (nr)	4	6
Width of Top Land (b1)	10.68 mm	18.47 mm
Width of other Land (b2)	1.575 mm	1.05 mm
Max. thickness of barel at the top end (t3)	10.7 mm	9.7 mm
Max. thickness of barel at the other end (t4)	2.675 mm	2.425 mm
Piston Pin Dia (d0)	2.4 mm	2.4 mm
Max. bending stress developed	105.8 MPA	112.48 MPA

IV. RESULT AND DISSCUSSION

THERMAL ANALYSIS USING CATIA

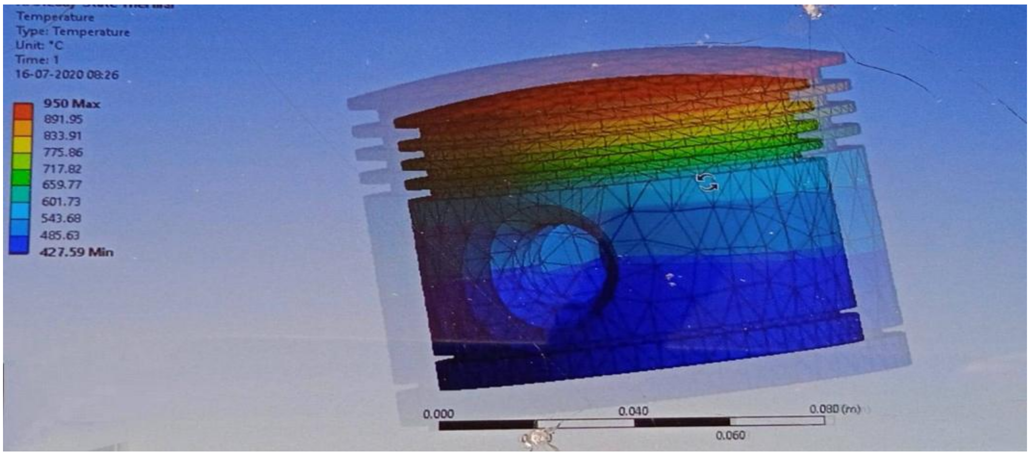
ALUMINIUM ALLOY

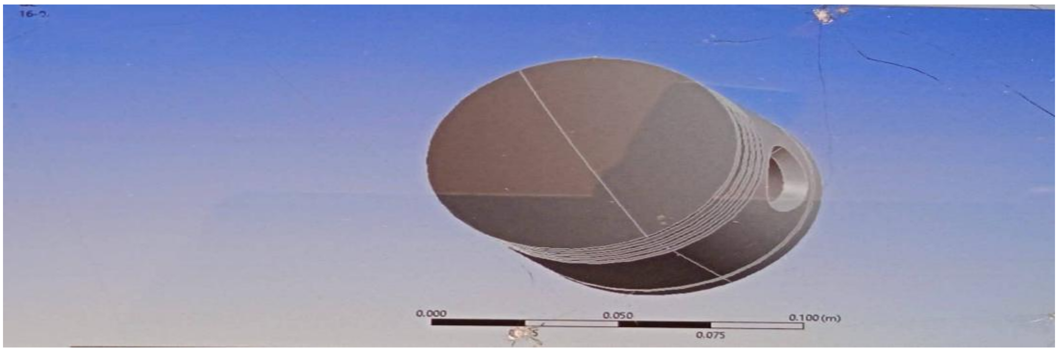


Temperature	MAX	1273k
	MIN	1169k

Aluminium Alloy

THERMAL ANALYSIS BY USING CATIA Using CARBON STEEL





Temperature	MAX	950k
	MIN	427k

Carbon Steel

The carbon steel is now widely used in piston head compared to aluminum alloy. It is found that average temperature of carbon steel is less than that of aluminum alloy .

V. Conclusion :

The first main conclusion that could be drawn from this work is that although thermal stress is not the responsible for biggest slice of damaged pistons, it remains a problem on engine pistons and its solution remains a goal for piston manufacturers. From the analysis, it is evident that thermal stress was higher than mechanically induced stress hence it could be concluded that the piston would fail due to the thermal load rather than the mechanical load and hence during optimization design, this could be put into consideration to ensure that thermal load is reduced.

The thermal and mechanical stress proportions have a direct influence on the coupled thermal-mechanical stress hence during design each load can be considered and reduced independently. It can be concluded that the piston can safely withstand the induced stresses

To satisfy all the requirements with regard to successful application of pistons, in particular mechanical and high temperature mechanical fatigue and thermal/thermal–mechanical fatigue there are several concepts available that can be used to improve its use, such as design, materials, processing technologies, etc.

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