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Design Evolution of a Diesel Engine Piston

Pedireddla Srinu[#1] & Mr.V.Tirupathi Rao[#2]
[#1] M.Tech Scholar(CAD/CAM), Department Of Mechanical Engineering,
Gonna Institute Of Information Technology and Sciences, Visakhapatnam, AP, India.
[#2] Assistant Professor, Department Of Mechanical Engineering,
Gonna Institute Of Information Technology and Sciences, Visakhapatnam, AP, India.

ABSTRACT

A piston is a major component of to and fro motion engines. It transfers the power form expanding gases in the cylinder to the crank shaft through the con rod. Now a three dimensional solid model of piston along with different dimensions is designed with the help of AutoCAD software. By applying the same pressures and temperatures for different Piston head diameter with the help of AutoCAD and Ansys 16.0 Software, we can get the evaluated results of von-misses stresses, thermal flux, and thermal gradient and directional deformations for that piston design. General materials for the piston is Structural steel, cast steel, forged steel, Aluminum alloys, and Nickel alloys. In our project Cast Aluminum alloy is used as piston material. It is an excellent material in thermal conductivity and also light in weight. The stress analysis results also help us to improve component design and also have given us the knowledge on the deformation and thermal handling sections on the piston.

Key Words: Diesel Engine, Piston, Piston head, Heat Transfer, Simulation, Structural and Thermal analysis.

1. INTRODUCTION

A diesel engine is an internal combustion engine. Here, the fuel is burned inside the cylinders where power is produced. Internal combustion wastes very less energy as compared to external combustion engine because the heat doesn't have to flow out from where it's produced in the cylinder. That is the reason we use internal combustion engines (they produce more energy from the same volume of fuel). Piston is subjected to very high mechanical and thermal stresses. As there is very large temperature difference between the piston head and cooling parts induces much thermal stresses in the piston. Thus, it has become very important to discuss the thermal and mechanical stresses to improve the quality and performance of the piston. In spite of all the improvements and advancements in the manufacturing process there exists large number of defective or damaged pistons. Thermal and mechanical stresses plays a prominent role in the designing of pistons. Thus finite element analysis is done for stresses, temperature, and deformation. Structural, thermal and thermo-mechanical stresses and temperature gradient are obtained from the analysis. A detailed stress analysis of piston is done under various thermal and structural boundary conditions with different piston head dimensions which are applied to the finite element model of the piston.

2. RELATED WORK

In the recent past, the demand for diesel engines has increased rapidly. This is mainly because of their higher thermal efficiency, better performance and reliability. In the earlier days, diesel engines were considered aspollutants compared to petrol/gasoline engines. Later, with the continuous improvements in the technology, there is a considerable reduction in the emission levels in diesel engines. The efforts are continuing in the direction of improving the overall engine performance. Here we have emphasized on the design of the Piston where all the process of conversion of chemical energy to heat energy and then to mechanical energy takes place. The study of the effect of different Piston diameters on the Thermal loads and the Mechanical stresses are studied. We have chosen the piston head diameter randomly from the previous studies so that a study could be done on the specific area.

3. MATERIAL PROPERTIES

As compared with the previous materials used for the manufacturing of a Piston, we can choose cast aluminium alloy for the current material. We can see that the tensile yield strength and the compressive yield strength are good and it is a very light weight and corrosion resistant. It has a very good life as a Piston material.

Cast Aluminum Alloy	
Physical Properties	Metric
Density	2650kg/m3
Mechanical Properties	Metric
Tensile strength,Ultimate	850Mpa
Tensile Strength,Yield	530 MPa
CTE, linear	18µm/m- °C
Specific Heat Capacity	0.461kJ/kg- °C
Thermal Conductivity	228 W/m- K
Melting Point	1370- 1630°C

4. GEOMETRY

The image below shows the geometry of the piston as described in the Literature. The piston created in AutoCAD is imported to Ansys software for further analysis. The following types of boundary conditions re-applied. In this geometry, the pressures are applied on the top of the piston and the temperatures are applied on the whole body.In this section ANSYS 16.0 was used for making 2D geometry with twoextreme head diameters. Here we have chosen these limits for the purpose of domestic use of the engine so higher limit is taken as 110mm for piston head diameter and lower limit is taken as 70mm piston head diameter. Smooth mesh can be created taking mesh size of 0.005 m. But in case of smooth mesh the iterations per time step increases, hence a coarse grid mesh is

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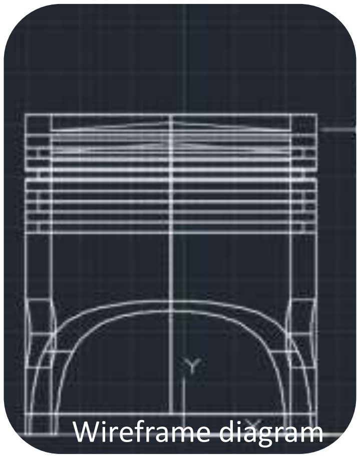
preferred. The mesh generated is shown in Fig. 2.



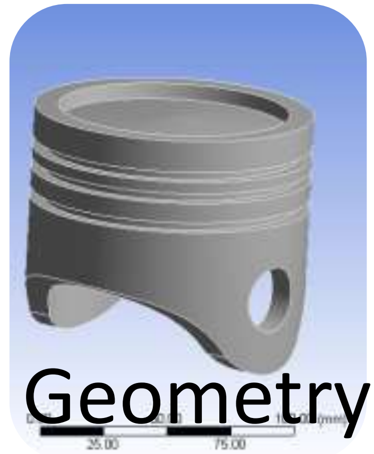


TABLE 1
Model (B4) > Geometry

Properties	
Volume	1254 mm³
Mass	0.46059 kg
Scale Factor Value	1.
Statistics	
Active Bodies	1
Nodes	130931
Elements	74604
Analysis Type	3-D

4.1 FINITE ELEMENT MODEL

This is performed by using computer aided design software. The main objective is to analyze the thermal stress distribution of piston at the combustion process. The analysis is performed to reduce the stress on the head of the piston. With the help of computer aided design software the structural model of a piston is designed. The finite element analysis is done using ANSYS software .

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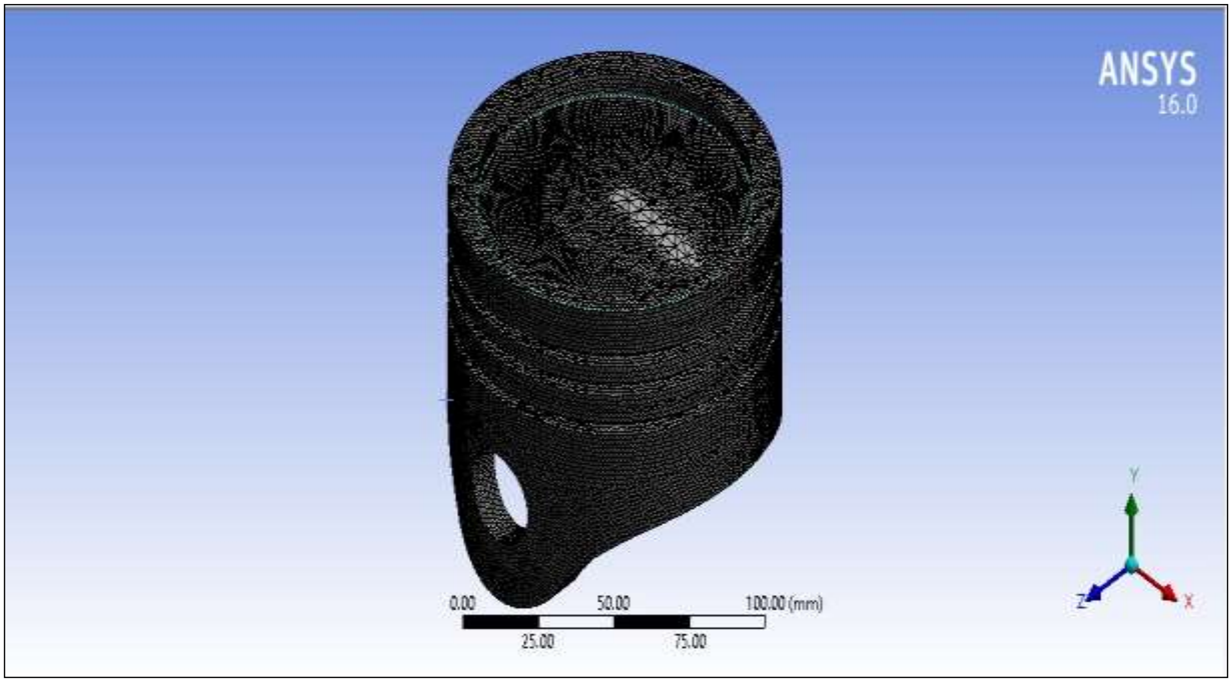

TABLE 2
Model (B4) > Mesh

Object Name	*Mesh*
State	Solved
Display	
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Minimum Edge Length	6.62690 mm
Inflation	
Maximum Layers	5
Growth Rate	1.2
Inflation Algorithm	Pre
View Advanced Options	No
Patch Conforming Options	
Triangle Surface Mesher	Program Controlled

Advanced	
Number of CPUs for Parallel Part Meshing	Program Controlled
Shape Checking	Standard Mechanical
Element Midside Nodes	Program Controlled
Straight Sided Elements	No
Number of Retries	Default (4)
Extra Retries For Assembly	Yes
Rigid Body Behavior	Dimensionally Reduced
Mesh Morphing	Disabled
Defeaturing	
Statistics	
Nodes	130931
Elements	74604
Mesh Metric	None

TABLE 3
Model (B4) > Mesh > Mesh Controls

Object Name	*Face Sizing*
State	Fully Defined
Scope	
Scoping Method	Geometry Selection
Geometry	24 Faces
Definition	
Suppressed	No
Type	Element Size
Element Size	1.5 mm
Behavior	Soft

TABLE 4
Model> Static Structural> Loads

Object Name	*Fixed Support*	*Pressure*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	2 Faces	3 Faces
Type	Fixed Support	Pressure
Suppressed	No	
Define By		Normal To
Magnitude		7. MPa (ramped)
Line Thickness	Single	
Display Type	Lines	

5. CALCULATIONS

The initial volume is 1200cc of air at about 1 atm pressure and 30°C (303.15K), and the compression ratio is 15:1 (i.e., you'll compress the air down to 80 cc, which is 1/16-th of the original volume), let's find out what the resulting temperature and pressure will be.1200cc of air is 0.0012 cubic meter. At room temperature and 1 Atm pressure it is roughly .048 moles.

5.1 CALCULATIONS FOR THE 70MM PISTON HEAD DIA

Thermal Calculation:

Piston diameter D = 70 mm

Barrel length L = 79 mm

Pressure P_1 = 1.0 bar,

Initial Temperature T_1 = 27 + 273 = 300 K

Cut-off ratio ρ = 8 / 100 V_S

In -Diesel Cycle.

Pressures and Temperatures at Salient Points

Now, Cylinder swept volume (V_s),

$V_S = \pi /4\ D^2.L$

$= (\pi / 4) \times 0.070^2 \times 0.079$

$V_S = 0.0003040276$ m3.

$V_1 = V_S + V_C = V_S + [V_S / (r-1)]$

$= [r /(r-1)] \times V_S$

$= [18 / (18-1)] \times 0.0003040276$

$V1 = 0.0003219115$ m3.

From the Ideal Gas Law,

$P_1 = mRT_1$

$m = p_1V_1 / RT_1$ [R= Gas constant $JK^{-1}mol^{-1}$]

$= [(1\times10^5 \times 0.0003219115) / (8.3144 \times 300)]$

$m = 0.0003079620$ Kg / cycle

For the Adiabatic (Isentropic) Process 1-2

$P_1V_1^{\Upsilon} = p_2V_2^{\Upsilon}$

$(P_2 / p_1) = (V_1 / V_2)^{\Upsilon} = r^{\Upsilon}$ [r = Compression ratio for diesel engine=10 - 18]

$P_2 = p_1. r^{\Upsilon} = (1 \times 18^{1.4})$

$P_2 = 57.198$ bar.

$(T_2 / T_1) = (V_1 /V_2)^{\Upsilon-1}$

$(r)^{\Upsilon-1} = (18)^{1.4-1}$

$(r)^{\Upsilon-1} = 3.1776.$

$T_2 = (T_1 \times 3.1776)$

T_2 = 953.301 K.

$V_2 = V_C$ = $[V_S / (r-1)]$

= [(0.0003040276/ (18-1)]

V_2 = 0.00001884 m3.

% Cut-off ratio = $[(\rho-1) / (r-1)]$

8/100 = $[(\rho-1) / (18-1)]$

ρ = (0.08 x 17) + 1

ρ = 2.36.

V_3 = $\rho . V_2$

= 2.36 x 0.000017884

V_3 = 0.000042206 m3.

For the Constant Pressure Process 2-3

(V_3 / T_3) = (V_2 / T_2)

T_3 = $(T_2 \times V_3) / V_2$

= (953.30145 x 0.000042206) / 0.000017884

T_3 = 2249.7914 K.

For the Isentropic Process 3-4

$P_3V_3^{\Upsilon} = p_4V_4^{\Upsilon}$

P_4 = $P_3 \times (V_3 / V_4)^{\Upsilon-1}$

= $P_3 \times 1 / (7.627)^{1.4}$

P_4 = 3.327 bar.

$T_4 / T_3 = (V_3 / V_4)^{\Upsilon-1}$

$(1 / 7.627)^{1.4-1}$ = 0.44366.

T_4 = T_3 x 0.44366

T_4 = 998.1556 K

$V_4 = V_1$ = 0.0003219115 m3.

Mean Effective Pressure

$P_m = [p_1 (r)^{\Upsilon} [\Upsilon (\rho-1) - r^{1-\Upsilon} (\rho \times \Upsilon)] / [(\Upsilon-1) (r-1)]$

$= \{57.198\ 1 \times (18)^{1.4} [1.4 (2.36-1) - (18)^{1-1.4} (2.36 \times 1.4)]\} / [(1.4-1) (18 - 1)]$

= 57.198 x 1.904 – [0.3146 x (3.327 -1)] / (0.4x 17)

= 8.411x (1.904 – 0.732)

P_m = 9.8498 bar

Power of the Engine, P

Work done per cycle = $p_m \times V_S$

= $(9.849 \times 0.0003040276 \times 10^5) / 10^3$

= 0.002994 KJ / cycle.

Work done per second = Work done per cycle x no. of cycles per second

= (0.002994 x 380) / 60

Power of the engine = 1.869 KW.

Design Calculation.

Thickness of the Piston Head According to Grashoff's formula the thickness of the piston head is given by

$t_h = D\sqrt{(3p_{max}/16\sigma_t)}$

Where, σ_t = 1200 M Pa [σ_t= Tensile Strength for Cast Aluminum]

t_h = 70 x √(3 x 57.198)/(16 x 1200)

t_h = 6.61 mm.

The maximum thickness from the above formula is **t_h is 6.61 mm**. Thickness of the piston head by heat transfer, considering piston as a circular plate.

$t_h = H/(12.56\ k \times (T_c - T_e))$ [$(T_c - T_e) = 220^o$ c]

[K = Heat conductivity Factor = 49 w/m/°C]

[T_c = Temperature at the skirt Centre]

[T_e =Temperature at the skirt Edge]

[H = Heat flow through the head]

H = C x HCV x m x B.P.

[m = fuel mass = 162.8970 x 10^6 Kg /B.P/sec]

[C = Constant value = 0.05]

[HCV = Higher calorific Value = 45 x 10^3kj/kg]

H = 0.05 x 45 x 10^3 x 162.8970 x 10^6 x 1.869

H = 0.692 KW. = 692 W.

t_h = 692/(12.56 x 49 x 220)

= 0.00511090 m.

t_h = 5.110 mm.

Taking the larger of the two values, we shall adopt t_h= 6.61 mm.

Radial thickness of the ring (t_1)

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

t_1 = 70√ (3 x 0.742) / (1200) [pw = Pressure at cylinder wall = 0.742 N/MM^2]

t_1 = 3.027 mm.

Axial thickness of the ring (t_2)

$t_2 = 0.85t_1$ to t_1

= 0.85 x 3.027

t_2 = 2.57 mm.

Width of the Top Land (b_1)

b_1 = 1.2 x 6.61

b_1 = 7.932mm ≈ 7.9 mm.

Width of the Ring (b_2)

b_2 = 0.75 t_2

b_2 = 0.75 x 2.57

b_2 = 1.925 mm ≈ 2 mm.

Piston barrel thickness (b)

t_3 = 0.03D + b + 4.5 mm

b = t_1 + 0.4

b = 3.027 + 0.4

b = 3.427 mm.

t_3 = 0.03(70) + 3.427+ 4.5 mm

t_3 = 10.027 mm ≈ 10.1mm.

Piston Wall Thickness (t_4)

t_4 = 0.33 x t3

t_4 = 0.33 x 10.1

t_4 = 3.333 mm.

Skirt Length (l)

l = (0.6 D to 0.8 D)

= 0.65 x 70

l = 49mm.

Total Length of the Piston (L)

L = Skirt Length + ring section Length + Top Land

L = 49 + 10.1 + 7.9

L = 67mm.

Centre of the skirt above the Centre of the skirt

= 0.04 x D

= 0.04 x 70

= 2.8 mm.

Gudgeon pin length in the con-rod bushing (l_1)

l_1 = 45% of the piston diameter

= 0.45 x 70

l_1 = 31.5mm.

Gudgeon pin diameter (d_o)

d_o = (0.28 D to 0.38 D)

= >0.34 x 70 =>d_o=23.8 mm.

5.2 CALCULATIONS FOR THE 110MM PISTON DIAMETER

Thermal Calculation:

Piston dia D = 110 mm

Barrel length S = 110 mm,

Pressure P_1 = 1.0 bar,

Initial Temperature T_1 = 27 + 273 = 300 K

Cut-off ratio ρ = 8 / 100 V_S

In -Diesel Cycle.

Pressures and Temperatures at Salient Points

Now, Stroke volume,

$V_S = \pi /4\, D^2.L$

$= (\pi / 4) \times 0.110^2 \times 0.110$

$V_S = 0.00104536495548$ m3.

$V_1 = V_S + V_C = V_S + [V_S / (r-1)]$

$= [r /(r-1)] \times V_S$

$= [18 / (18-1)] \times 0.00104536495548$

$V1 = 0.00110685701168$ m3.

<u>From the Ideal Gas Law</u>

$P_1 = mRT_1$

$m = p_1V_1 / RT_1$

$= [(1 \times 10^5 \times 0.00110685701168) / (287 \times 300)]$

$m = 115.699$ Kg / cycle

<u>For the Adiabatic (Isentropic) Process 1-2</u>

$P_1V_1^{\Upsilon} = p_2V_2^{\Upsilon}$

$(P_2 / p_1) = (V_1 / V_2)^{\Upsilon} = r^{\Upsilon}$

$P_2 = p_1 . r^{\Upsilon} = (1 \times 18^{1.4})$

$P_2 = 57.198$ bar.

$(T_2 / T_1) = (V_1 / V_2)^{\Upsilon-1}$

$(r)^{\Upsilon-1} = (18)^{1.4-1}$

$(r)^{\Upsilon-1} = 3.1776.$

$T_2 = (T_1 \times 3.1776)$

$T_2 = 953.301$ K.

$V_2 = V_C = [V_S / (r-1)]$

$= [(0.00104536495548 / (18-1)]$

$V_2 = 0.00002597792748$ m3.

% Cut-off ratio $= [(\rho-1) / (r-1)]$

$8/100 = [(\rho-1) / (18-1)]$

$\rho = (0.08 \times 17) + 1$

$\rho = 2.36.$

$V_3 = \rho . V_2$

$= 2.36 \times 0.00002597792748$

$V_3 = 0.00006130790885$ m3.

<u>For the Constant Pressure Process 2-3</u>

$(V_3 / T_3) = (V_2 / T_2)$

$T_3 = (T_2 \times V_3) / V_2$

$= (953.30145 \times 0.00006130790885) / 0.00002597792748$

$T_3 = 2249.7914$ K.

<u>For the Isentropic Process 3-4</u>

$P_3V_3^{\Upsilon} = p_4V_4^{\Upsilon}$

$P_4 = P_3 \times (V_3 / V_4)^{\Upsilon-1}$

$= P_3 \times 1 / (7.627)^{1.4}$

$P_4 = 3.327$ bar.

$T_4 / T_3 = (V_3 / V_4)^{\Upsilon-1}$

$(1 / 7.627)^{1.4-1} = 0.44366.$

$T_4 = T_3 \times 0.44366$

$T_4 = 998.1556$ K

$V_4 = V_1 = 0.00079492458112$ m3.

Theoretical Air Standard Efficiency

$\eta_{\text{diesel}} = 1-\{1 / ^{\Upsilon} (r)^{\Upsilon-1} [(\rho^{\Upsilon-1}) / \rho -1]\}$

$= 1- \{1 / ^{1.4} (18)^{1.4-1} [(2.36^{1.4-1}) / (2.36 - 1)]\}$

$= 1 - [(0.22478 \times 2.327) / 1.36] = 0.6153.$

$= 0.6153 \times 100$ %

$\eta_{\text{diesel}} = 61.53\%$

Mean Effective Pressure

$P_m = [p_1 (r)^{\Upsilon} [\Upsilon (\rho-1) - r^{1-\Upsilon} (\rho \times \Upsilon)] / [(\Upsilon-1) (r-1)]$

$= \{57.198\ 1 \times (18)^{1.4} [1.4 (2.36-1) - (18)^{1-1.4} (2.36 \times 1.4)]\} / [(1.4-1) (18 - 1)]$

$= 57.198 \times 1.904 - [0.3146 \times (3.327 -1)] / (0.4 \times 17)$

$= 8.411 \times (1.904 - 0.732)$

$P_m = 9.8498$ bar

Power of the Engine, P

Work done per cycle $= p_m \times V_S$

$= (9.849 \times 0.0003040276 \times 10^5) / 10^3$

$= 0.002994$ KJ / cycle.

Work done per second = Work done per cycle x no. of cycles per second

$= (0.002994 \times 380) / 60$

Power of the engine = 1.869 KW.

Design Calculation.

Thickness of the Piston Head According to Grashoff's formula the thickness of the piston head is given by

$t_h = D\sqrt{(3p_{max}/16\sigma_t)}$

Where, $\sigma_t = 1200$ M Pa [σ_t= Tensile Strength for Cast Aluminum]

$t_h = 110 \times \sqrt{(3 \times 57.198)/(16 \times 1200)}$

$t_h = 10.339$ mm

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The maximum thickness from the above formula is **t_h is 10.339mm**.

Thickness of the piston head by heat transfer, considering piston as a circular plate.

t_h = $H/(12.56\ k \times (T_c - T_e))$ [$(T_c - T_e) = 220^o$ c]

[K = Heat conductivity Factor = 49 w/m/^{o}C]

[T_c = Temperature at the Centre of the skirt]

[T_e =Temperature at the Edge of the skirt]

[H = Heat flow through the head]

H = C x HCV x m x B.P.

[m = Fuel Mass = 162.8970 x 10^6 Kg /B.P/sec]

[C = Constant = 0.05]

[HCV = Higher calorific Value =45 x 10^3kj/kg]

H = 0.05 x 45 x 10^3 x 162.8970 x 10^6 x 1.869

H = 0.692 KW. = 692 W.

t_h = 692/(12.56 x 49 x 220)

= 0.00511090 m.

t_h = 5.110 mm.

Taking the larger of the two values, we shall adopt t_h= 10.339 mm.

Radial thickness of the ring

t_1 = $D \sqrt{(3p_w / \sigma_t)}$ [pw = Pressure on the gas cylinder wall = 0.742 N/MM^2]

t_1 = $110\sqrt{(3 \times 0.742) / (1200)}$

t_1 = 4.7376 mm

Axial thickness of the ring

t_2 = $0.85t_1$ to t_1

= 0.85 x 4.7376

t_2 = 4.026 mm.

Width of the Top Land(b_1)

b_1 = 1.2 x 10.7376

b_1 = 12.88512 mm ≈ 12.9 mm.

Width of Second Ring (b_2)

b_2 = 0.74 t_2

b_2 = 0.74 x 4.026

b_2 = 3.019 mm ≈ 3.1 mm.

Piston barrel Max thickness (t_3)

t_3 = 0.03D + b + 4.5 mm

b = t_1 + 0.4

b = 4.7376+ 0.4

$b = 5.1376$ mm.

$t_3 = 0.03(110) + 3.427 + 4.5$ mm

$t_3 = 9.9676$ mm ≈ 10 mm.

Piston Wall Thickness t_4

$t_4 = 0.33 \times t3$

$t_4 = 0.33 \times 10$

$t_4 = 3.289$ mm.

Length of the skirt

$l = (0.6 D \text{ to } 0.8 D)$

$= 0.65 \times 110$

$l = 71.5$ mm

Total Length of the Piston

L = Skirt Length + Ring Section Length + Top Land

$L = 74.5 + 10 + 12.9$

$L = 97.3676$ mm.

Centre of the skirt above the Centre of the skirt

$= 0.04 \times D$

$= 0.04 \times 110$

$= 4.4$ mm.

Gudgeon pin length in the con-rod bushing

l_1 = 45% of the piston diameter

$= 0.45 \times 110$

$l_1 = 49.5$ mm

Gudgeon pin diameter (d_o)

$d_o = (0.28 D \text{ to } 0.38 D)$

$= 0.34 \times 110$

$d_o = 37.4$ mm.

Based on these calculations, the piston has been designed and the calculated temperatures has been applied. Therefore, the highest temperature reached in the combustion chamber is nearly 700°C.

We have given the reference temperature as 303.15 K as it is usual on the roads taken as an average of all the seasons. The highest temperature that is reached after compression in an

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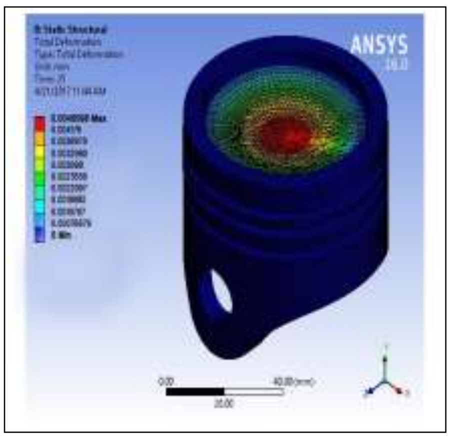
engine is about 700°C so the working boundary conditions are applied at 700°C. After the combustion, the temperatures may reach up to 2000°C. Let us check for solution using the boundary conditions.

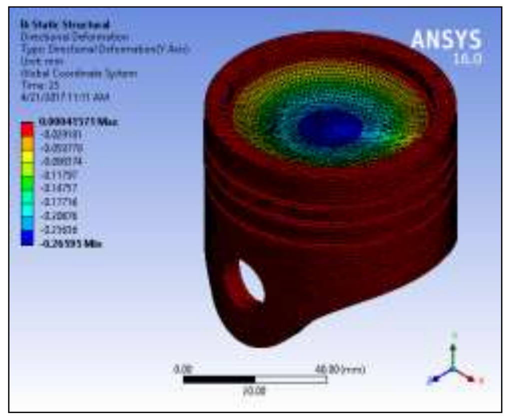
6. RESULTS

After applying the Pressures, the following effects have been resulted.

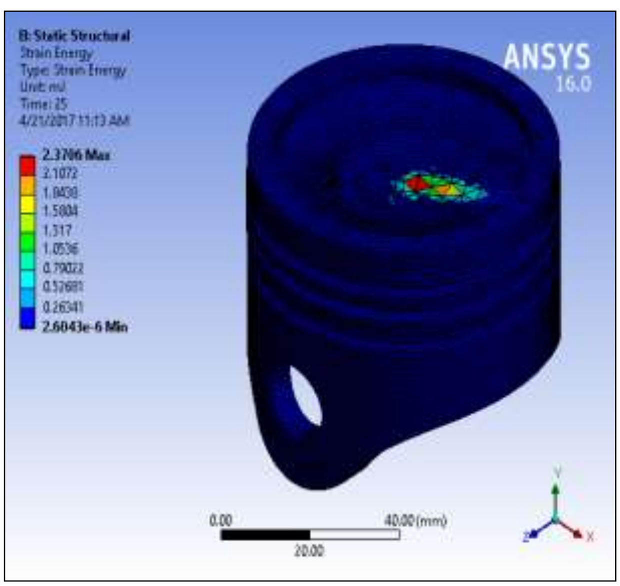
6.1 Let us look at the figures for various factorson a 70mm piston dia.



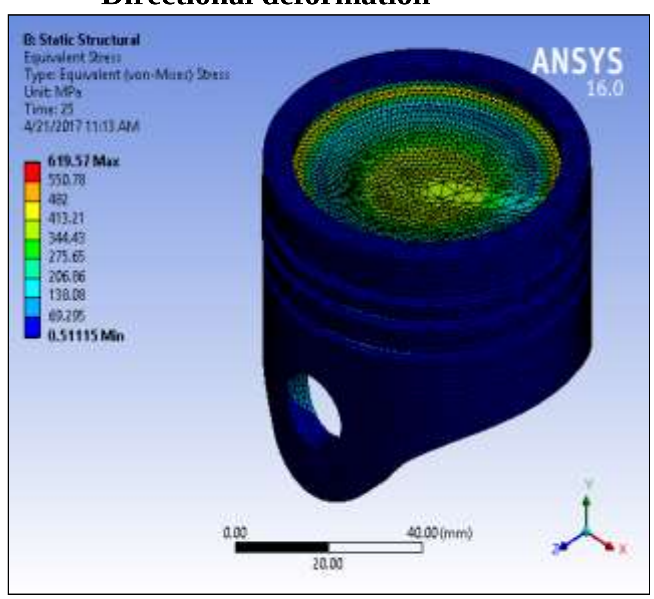
Total deformation



Directional deformation



Strain Energy



Equivalent stress

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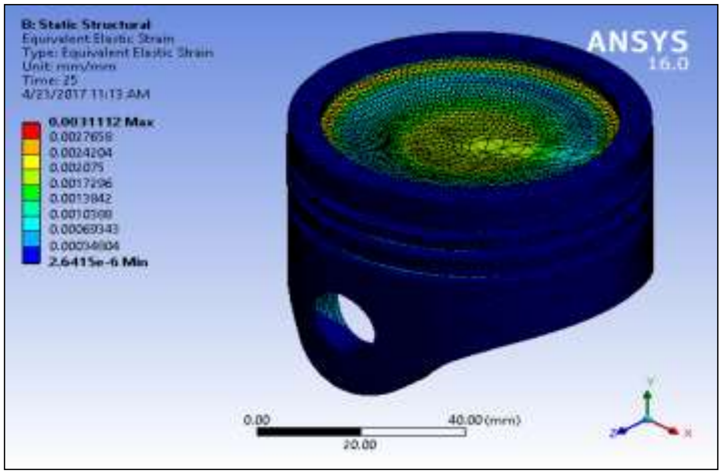

Equivalent Elastic Strain

TABLE 5

Model > Static Structural > Solution > Results

Object Name	*Total Deformation*	*Equivalent Elastic Strain*	*Equivalent Stress*	*Strain Energy*	*Directional Deformation*
Results					
Minimum	0. mm	2.6415e-006 mm/mm	0.51115 MPa	2.6043e-006 mJ	-0.26595 mm
Maximum	4.5098 e-003 mm	3.1112e-003 mm/mm	619.57 MPa	2.3706 mJ	4.1571e-004 mm
Minimum Value Over Time					
Minimum	0. mm	2.6415e-006 mm/mm	0.51115 MPa	2.6043e-006 mJ	-0.26595 mm
Maximum	0. mm	2.6415e-006 mm/mm	0.51115 MPa	2.6043e-006 mJ	-0.26595 mm
Maximum Value Over Time					
Minimum	4.5098 e-003 mm	3.1112e-003 mm/mm	619.57 MPa	2.3706 mJ	4.1571e-004 mm
Maximum	4.5098 e-003 mm	3.1112e-003 mm/mm	619.57 MPa	2.3706 mJ	4.1571e-004 mm
Information					
Time	25. s				
Load Step	25				
Sub-step	1				
Iteration Number	25				

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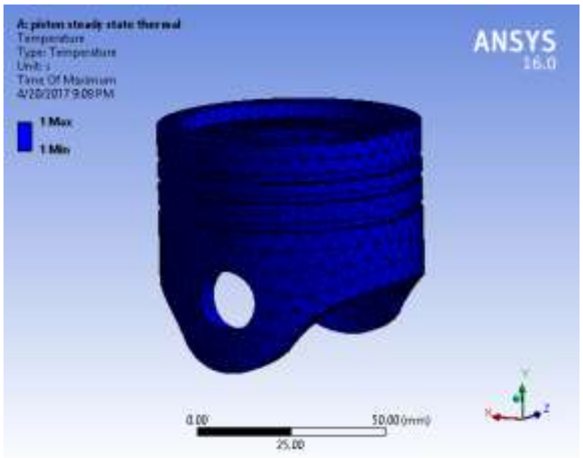
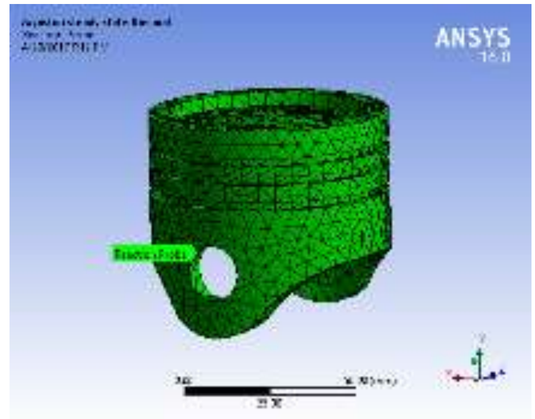
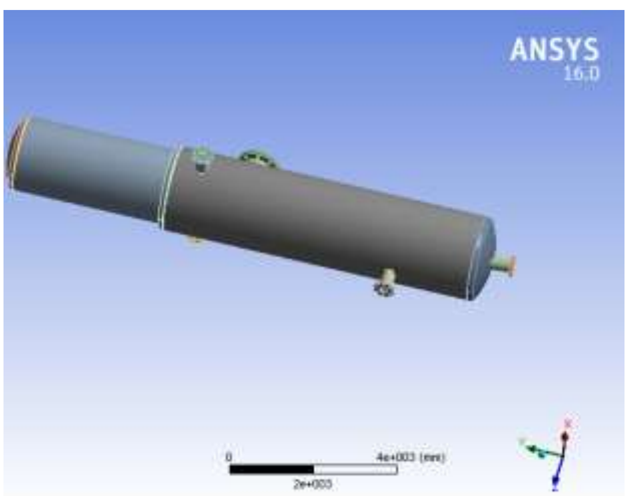
6.2 Results for the effect of temperature on the 70mm dia Piston.



Reaction Probe 2



Total heat flux



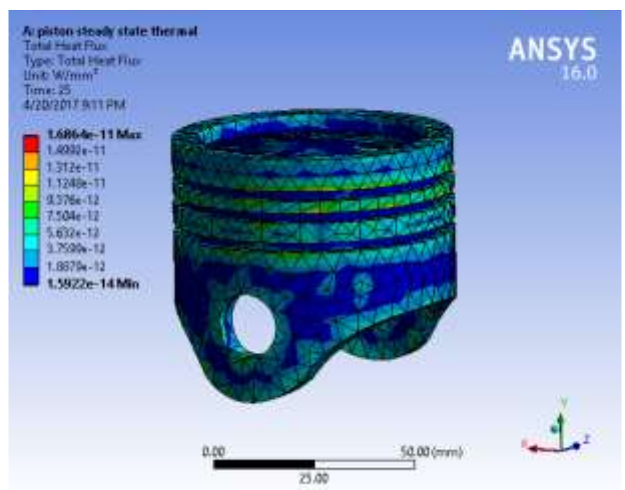


TABLE 6
Model> Steady-State Thermal > Solution> Results

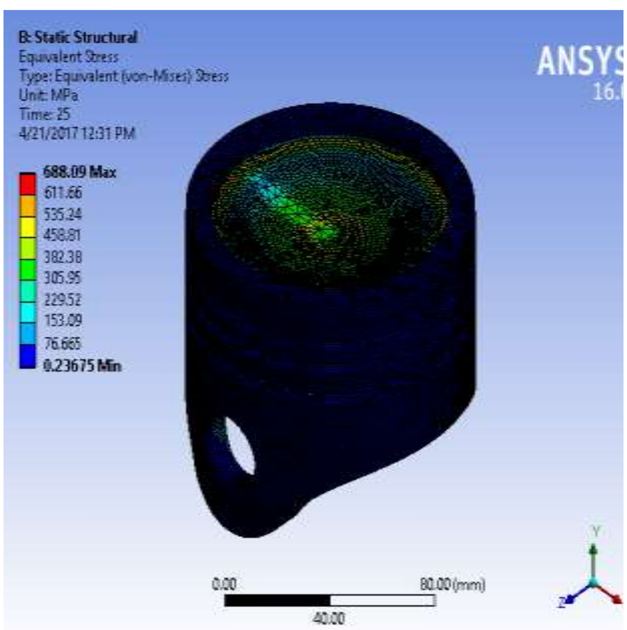
Object Name	*Temperature*	*Total Heat Flux*
State	Solved	
Scope		
Scoping Method	Geometry Selection	
Geometry	All Bodies	
Definition		
Type	Temperature	Total Heat Flux
By	Time Of Maximum	Time
Identifier		
Suppressed	No	
Display Time		Last
Calculate Time History		Yes
Results		
Minimum	1. s	1.5922e-014 W/mm²
Maximum	1. s	1.6864e-011 W/mm²
Integration Point Results		
Display Option		Averaged
Average Across Bodies		No
Minimum Value Over Time		
Minimum		9.696e-015 W/mm²
Maximum		2.3555e-014 W/mm²
Maximum Value Over Time		
Minimum		1.6864e-011 W/mm²
Maximum		1.6864e-011 W/mm²
Information		
Time		25. s
Load Step		25
Sub-step		1
Iteration Number		25

Model> Steady-State Thermal > Solution> Probes

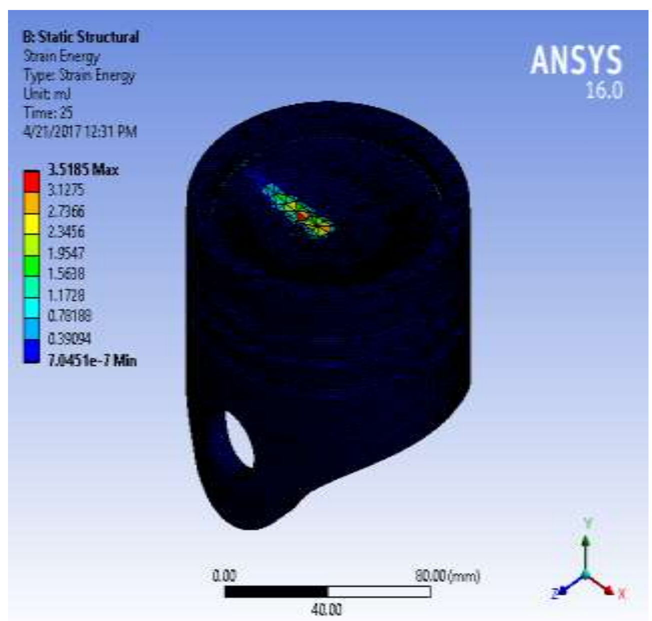
Object Name	*Reaction Probe*	*Reaction Probe 2*
State	Solved	
Definition		
Type	Reaction	
Location Method	Boundary Condition	
Boundary Condition	Temperature	Convection
Suppressed	No	

Options		
Display Time	End Time	
Results		
Heat	1.5349e-002 W	-1.5349e-002 W
Maximum Value Over Time		
Heat	1.5349e-002 W	-1.5349e-002 W
Minimum Value Over Time		
Heat	1.5349e-002 W	-1.5349e-002 W

6.3 LET US LOOK AT THE FIGURES FOR VARIOUS FACTORS LIKE TOTAL DEFORMATION, STRESSES ETC ON A 110MM PISTON DIAMETER.



Equivalent stress



Strain Energy

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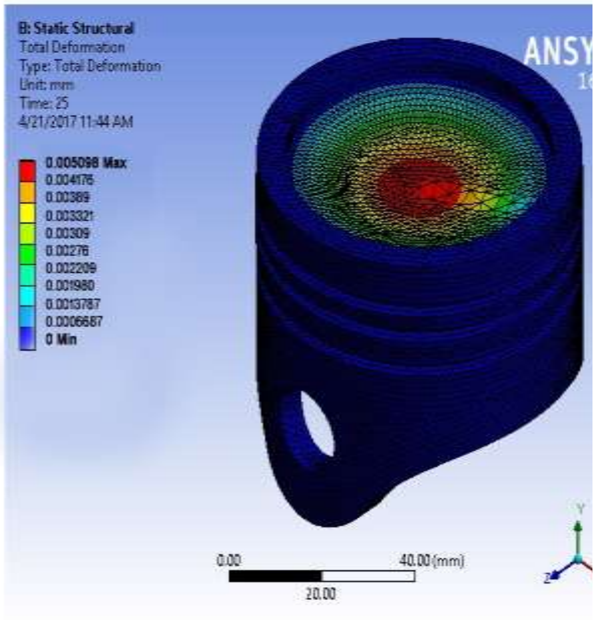
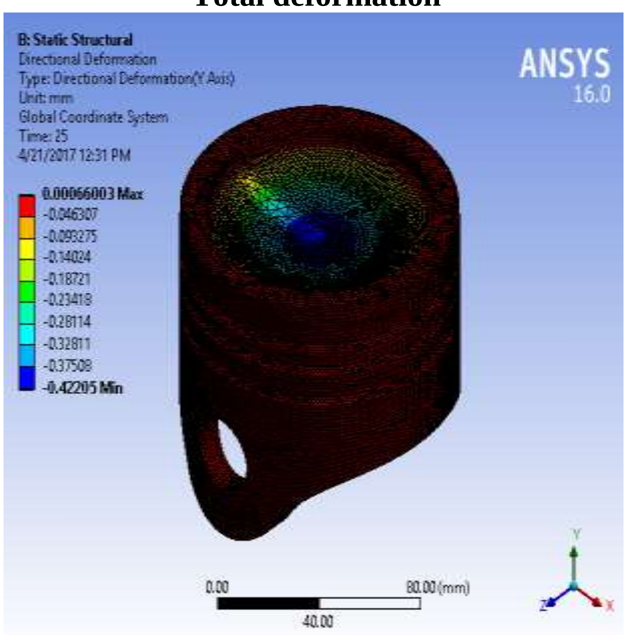


Total deformation



Directional deformation

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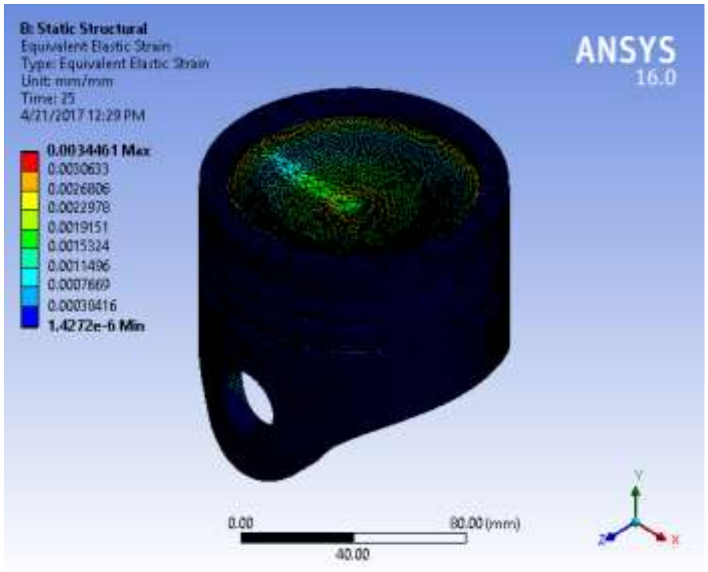


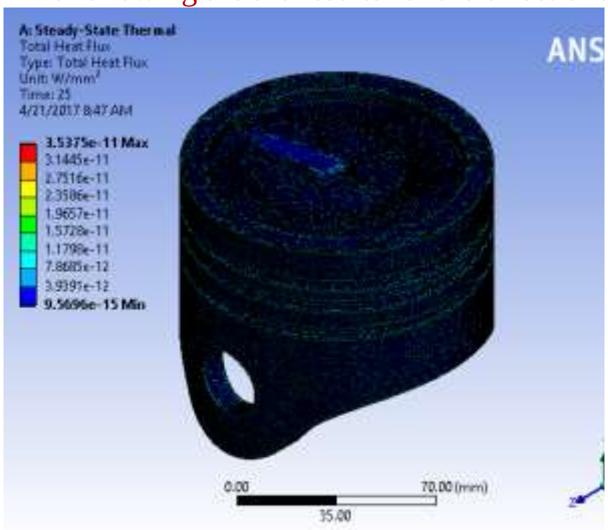
Equivalent Elastic Strain

TABLE 7
Model > Static Structural > Solution > Results

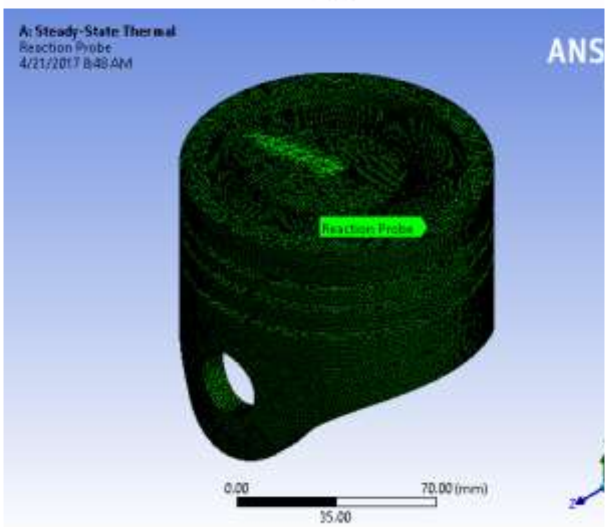
Object Name	*Total Deformation*	*Equivalent Elastic Strain*	*Equivalent Stress*	*Strain Energy*	*Directional Deformation*
Results					
Minimum	0. mm	1.4272e-006 mm/mm	0.23675 MPa	7.0451e-007 mJ	-0.42205 mm
Maximum	5.098 e-003 mm	3.4461e-003 mm/mm	688.09 MPa	3.5185 mJ	6.6003e-004 mm
Minimum Value Over Time					
Minimum	0. mm	1.4272e-006 mm/mm	0.23675 MPa	7.0451e-007 mJ	-0.42205 mm
Maximum	0. mm	1.4272e-006 mm/mm	0.23675 MPa	7.0451e-007 mJ	-0.42205 mm
Maximum Value Over Time					
Minimum	5.098 e-003 mm	3.4461e-003 mm/mm	688.09 MPa	3.5185 mJ	6.6003e-004 mm
Maximum	5.098 e-003 mm	3.4461e-003 mm/mm	688.09 MPa	3.5185 mJ	6.6003e-004 mm
Information					
Time	25. s				
Load Step	25				
Sub-step	1				
Iteration	25				

Number	

6.4 The following are the results for the effect of temperature on the 110mm dia Piston.



Total heat flux



Reaction Probe

6.5

Directional Heat flux

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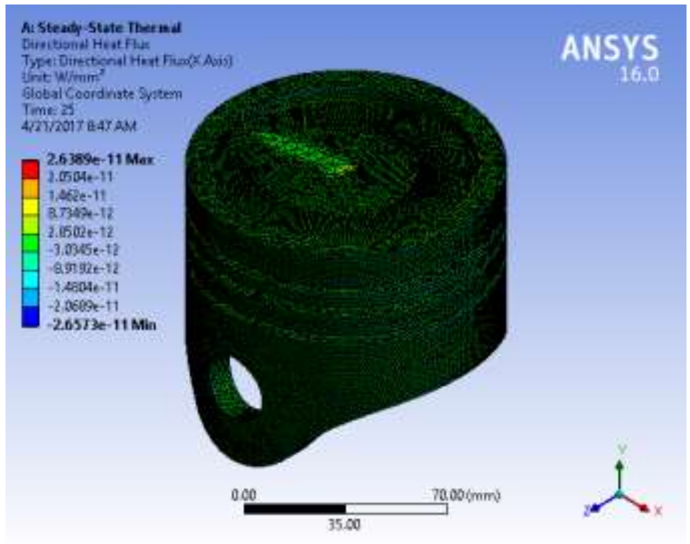

TABLE 8
Model > Steady-State Thermal > Solution > Results

Object Name	*Temperature*	*Total Heat Flux*	*Directional Heat Flux*	*Thermal Error*
Results				
Minimum	950. K	9.5696e-015 W/mm²	-2.6573e-011 W/mm²	9.6623e-023
Maximum	950. K	3.5375e-011 W/mm²	2.6389e-011 W/mm²	3.2103e-018
Minimum Value Over Time				
Minimum	950. K	9.117e-015 W/mm²	-2.6573e-011 W/mm²	6.2327e-023
Maximum	950. K	1.1645e-014 W/mm²	-2.6573e-011 W/mm²	9.7432e-023
Maximum Value Over Time				
Minimum	950. K	3.5375e-011 W/mm²	2.6381e-011 W/mm²	3.208e-018
Maximum	950. K	3.5375e-011 W/mm²	2.6389e-011 W/mm²	3.2105e-018
Information				
Time	25. s			
Load Step	25			
Sub-step	1			
Iteration Number	25			

7. CONCLUSION

This study, has been conducted using Ansys 16.0 software. The effect of temperature and pressure have been observed on a Piston material made of cast aluminum alloy. As per the simulations done above and the results obtained have given us a clear statistical data about the calculated pressure and temperatures after being applied on the 70 mm and the 110 mm diameter piston heads. The resulted deformations, stresses and fluxes have been studied and the results

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looks quite promising. Comparatively, a change in piston design has been made. Here, the excess material of the piston aiding in higher weight has been removed. The stresses and temperatures applied on both of the configurations are under allowed limits. It was found that the design parameter of the piston with these modifications gives us the improvements as expected in the existing results.

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9 .ABOUT THE AUTHORS

PEDIREDDLA SRINU is currently pursuing his 2 Years M.Tech in Department Of Mechanical Engineering, Gonna Institute Of Information Technology and Sciences, Visakhapatnam, AP, India. His area of interest includes CAD/CAM.

MR.V.TIRUPATHI RAO is currently working as Assistant Professor in Department of Department Of Mechanical Engineering, Gonna Institute Of Information Technology and Sciences, Visakhapatnam, AP, India. He has more than 3 years of teaching experience in engineering colleges. His area of interest includes CAD/CAM.