Design And Static Stress Analysis Of Trunk Piston Of Detroit Diesel Engine

Sundaresh S[1], Sriganesh T G[2], K Chandra Sekhara Reddy[3]
[1]*BMS Institute of Technology and Management, Yelahanka, Bengaluru, India -560064*
[2]*BMS Institute of Technology and Management, Yelahanka, Bengaluru, India -560064*
[3]*BMS Institute of Technology and Management, Yelahanka, Bengaluru, India -560064*

Abstract: *Heavy-duty diesel engines are built to endure longer than commercial engines. Heavy-duty diesel engines employ crosshead pistons, whereas medium-duty diesel engines use trunk pistons. In a medium-duty diesel engine, the trunk piston is a mono piston. In this work, Finite Element Analysis (FEA) is used to create and compare a trunk piston for a Detroit diesel engine 8V92T a heavy-duty engine to a standard crosshead piston. The 3D model was made using Solid works 2021 using a Crosshead piston from a Detroit diesel engine 8V92Ta (series 92 engine). The parameters such as gas pressure, piston temperature, mass characteristics, and material properties that are evaluated while conducting the simulation for both pistons.*

Keywords: Trunk Piston, Cross head piston, static stress, simulation,

1. Introduction

Diesel engines are classified as heavy, medium, or low duty diesel engines, depending on the vehicle's purpose. If a diesel engine is used in a heavy-duty vehicle, it is categorised as a heavy-duty diesel engine, and if it is used in a light-duty vehicle, it is classified as a light-duty diesel engine. Commercial vehicles, cranes, and consumer goods use heavy-duty diesel engines all over the world. Heavy duty diesel vehicles are defined as those with a curb weight higher than 2720 kg or a gross vehicle rating greater than 3850 kg.

A diesel engine is known as a compression-ignition engine. Compression heat is used to ignite the fuel delivered into the combustion chamber in this type of internal combustion engine. Because the compression ratio is between 17:1 and 19:1, these engines have the maximum thermal efficiency when compared to other combustion engines. The four-stroke and two-stroke heavy duty diesel engines are available.

1.1 Piston

Piston is one of the most significant components of a reciprocating engine because it aids in the conversion of chemical energy generated from fuel combustion into useable (work) mechanical power. A piston is a cylindrical plug that travels up and down within a cylinder. It has piston rings to ensure a strong seal between the cylinder wall and the piston.

Types of pistons:

i. Trunk piston
ii. Crosshead piston

i. **Trunk piston**: In a medium ranger heavy duty diesel engine, trunk pistons are employed. The length of these pistons is proportional to their diameter. The primary components of this piston are the head, which must endure gas pressure, the skirt, which acts as a bearing for side force, and the wrist pin, which connects the piston to the connecting rod. Piston rings seal the combustion chamber's gases. They serve the same purpose as a piston and a crosshead (cylindrical). The most common materials for pistons are cast iron and aluminium. It has a groove for an oil ring below the gudgeon pin, in addition to the rings between the gudgeon pin and the crown, which is very useful for diesel engines. The trunk pistons are mono pistons as compared to crosshead pistons.

Fig 1: Mono piston

ii. **Crosshead piston**: Crosshead pistons are mostly utilised in heavy-duty diesel engines that run at low speeds. These pistons are divided into two parts: the crown and the floating skirt. The piston pin holds the two pieces together. Through bolts, the gudgen pin is attached to the connecting rod. A Crosshead piston's crown is made of cast iron with an aluminium floating skirt. The piston head and skirt can both be made of malleable iron; in this case, the skirt is tin-plated and must be suitable for the liner. The crosshead piston's design directs all of the gas pressure from the combustion chamber to the piston pin. The piston is guided by the floating skirt.

Fig 2: Crosshead piston

1.2 Detroit Diesel engine 8V92TA

This engine is a V-Block with a single power stroke that lasts two cycles. The cylinders in this series range in size from 6 to 16. The 6V92 and 8V92 are the most popular in this category. These engines first appeared in 1974. These engines are still used in generators, vehicles, and cranes. These engines are simple to service and maintain since individual components such as the cylinder head, piston, and bore may be replaced without removing the complete engine.

Table 1: 8V92TA engine specification

Detroit Diesel engine specification	
Bore	122.94 mm
Stroke	127 mm
Compresstion Ratio	17:1
Displacement	552 to 1472 cu
8V	12.07 l = 1511.7cc/cylinder
Max power for 8V	324 kW@2100 rpm

2. Design of Trunk piston

2.1Material selection

When comparing the new trunk piston to an existing crosshead piston from an 8V92TA Detroit diesel engine, which is composed of malleable iron or 34crnimo6, we can see that the new trunk piston is made of malleable iron or 34crnimo6. The qualities of the 34CrNiMo6 material are as follows:

Table 2: Material

Density	7800 kg/m³
Modulus of Elasticity	210 GPa
Tensile Strength	800 MPa
Yield Strength	700 MPa
Poisson's Ratio	0.3
Thermal Conductivity	42.5 W/m-K

2.2 Engine Specification

Table 3: Specification

Displacement	12093.7cc for 8 cylinders = 1511.7cc/cylinder
Max power	324 kW@2100 rpm = 40.5 kw
Compression ratio	17:1
Bore	123 mm
Stroke	127

2.3. Calculation of maximum pressure on the piston

The gas pressure of the combustion pressure acting on the piston crown or the bowel area will be calculated here. Each piston's pressure will be computed.

1. Torque

$$P = \frac{2\,\pi\,N\,T}{60}$$

w.k.t P = maximum power acting on each piston is 40.5 kw

$$40.5 \times 10^3 = \frac{2\,\pi \times 2100 \times T}{60} \qquad \mathbf{T = 184.16\ N\text{-}m.}$$

2, Gas pressure

w.k.t Velocity = $V = \frac{2\,L\,N}{60}$, $V = \frac{2 \times 0.127 \times 2100}{60}$, **V = 8.89 m/s.**

Force = $F = \frac{P}{v}$, $F = \frac{40.5 \times 10^3}{8.89}$, **F = 4555.6 N**

Pressure = $P = \frac{F}{A}$, $P = \frac{4555.6 \times 4}{\pi \times 123^2}$, **P = 0.3833 MPa**

∴ Maximum pressure = P_{max} = 15×P=15×0.3833, **P_{max} = 6 MPa per cylinder.**

a.Design calculation for trunk piston

1. Piston head thickness.

   $t_h = D \times \frac{\sqrt{3 \times P_{max}}}{16 \times \sigma_t}$

   t_h → Thickness of the piston head

   P_{max} → Gas pressure acting on the piston head N/mm^2

   σ_t → Allowable stress of the material N/mm2 (σ_t = 311.1 MPa)

   $t_h = 123 \times \frac{\sqrt{3 \times 6}}{16 \times 311.11}$ = **7mm**

2. Ring radial thickness

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

   P_w → Fuel pressure on the cylinder wall in N/mm^2. The value is limited form 0.042 to 0.0667 N/mm^2

   $$t_1 = \sqrt{\frac{3 \times 0.042}{311.11}}$$

   t_1 = 2.475mm.

3. Axial thickness of the ring t_2

   $t_2 = 0.7 \times t_1$

   $t_2 = 0.7 \times 2.475$ = **1.75mm**

4. Number of ring's

$n_r = \frac{D}{(10 \times t_2)}$

$n_r = \frac{123}{(10 \times 2)}$ =6.15 →**6 rings**

5. Width of the top ring land

   $b_1 = t_h \times 1.2$

   $b_1 = 7 \times 1.2$ = **9mm.**

6. Width of other ring's land

   $b_2 = 0.75 \times t_2$

   $b_2 = 0.75 \times 2$ = **1.5mm**.

7. Barrel's maximum thickness

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

   where,

   $b = 0.4 + t_1$

   $b = 0.4 + 2.475 = 2.875$mm

   $\therefore t_3 = 0.03 \times 123 + 2.9 + 4.5$

   t_3 = 11.09mm.

8. The open end of the pistons barrel thickness t_4

   $t_4 = 0.35 \times t_3$

   $t_4 = 0.35 \times 11.09$ = **3.9mm**

9. Maximum thrust acting on side of the cylinder

   $R = 0.1 \times P_{max} \times \frac{\pi \times D^2}{4}$

   $R = 0.1 \times 6 \times \frac{\pi \times 123^2}{4}$

   R = 7129.3 N

   The side thrust (R) also given by

   R = Bearing pressure × Projected bearing area of the piston skirt

   $R = P_b \times D \times l$

   Where P_b for the present engine is $P_b = 0.48$

   $R = 0.48 \times 123 \times l$

   R = 59.04L

   $l = \frac{7129.3}{59.04}$

   l = 120.7 mm.

   Length of the skirt = 120.7mm.

   L = Length of skirt + Length of ring section + Top land

   L = 120.7 + 22 + 9 = **151.75 mm.**

10. Gudgeon pin

   Let, d_o = Gudgeon pin outer diameter in mm.

   l = Gudgeon pin length at the samll end in the bush of the connecting rod in mm.

   P_{b1} = bearing pressure at the small end of the connecting rod bushing in N/mm^2, its value for the bronze bushing may be taken as 25 N/mm^2.

   $l_1 = 0.45 \times D$

   $l_1 = 0.45 \times 123 = 55.35$mm.

   → w.k.t load on the piston is due to gas pressure or gas load

   $\frac{\pi * D^2}{4} \times P_{max}$

$\frac{\pi \times 123^2}{4} \times 6 =$ **71293.73 N**

Load on gudgeon pin is due to combustion pressure = Bearing Pressure × bearing area

$= P_{b1} \times d_o \times l_1$

$= 25 \times d_0 \times 55.35$

= **1383.75 d_o N**

From above, we find that

$1383.75 \times d_o = 71293.73$

d_0 = 51.5 mm.

The inner diameter of the gudgeon pin (d_i) is 0.6 d_o.

0.6 × 51.5 = **30.9mm.**

11. Maximum bending moment at the middle of the gudgeon pin

$M = \frac{P \times D}{8} \rightarrow \frac{71293.7 \times 123}{8}$

$M = 1096.14 \times 10^3$ N/mm

We also know that maximum bending moment (M),

$$M = \frac{\pi}{32} \times \left(\frac{(do)^4 - (di)^4}{do}\right) \times \sigma_b$$

$$1096.14 \times 10^3 = \frac{\pi}{32} \times \left(\frac{(51.5)^4 - (30.9)^4}{51.5}\right) \times \sigma b$$

$$1096.14 \times 10^3 = 11649.26 \times \sigma_b$$

$$\sigma_b = \frac{1096.14 \times 10^3}{11649.26}$$

σ_b = 94.09 N/mm²

∴The induced bending stress in the pin is less than the permissible value of 140 MPa. Therefore, the dimensions for the pin as calculated above (i.e. d_0 = 51.5 mm and d_i = 30.9 mm) are satisfactory.

12. Piston boss

i.e 1.4 * d_o = 1.4 * 51.5 = **72.1mm**

13. Piston Rib

$$tr = \frac{th}{3} \text{ or } \frac{th}{2} = \frac{7.3}{2} = \mathbf{3.65\ mm}$$

Note: As the piston wall thickness (t4) approaches the opening end, it should be taken into consideration as 0.25 × t_3 to 0.35 × t_3

∴ 0.35 × 11.09 = **3.88 mm**.

b. Specifications of the trunk piston

Table 4: Trunk piston parameter

1.	The piston head's thickness (T_h)	7.3 mm
2.	Ring's radial thickness (t_1)	2.475 mm
3.	Ring's Axial thickness (t_2)	1.75 mm
4.	Number of ring's (n_r)	6
5.	Width of the top ring land (b_1)	9 mm
6.	Width of other ring's land (b_2)	1.5 mm
7.	Barrel's maximum thickness (t_3)	11.09 mm

8.	The open end of the pistons barrel thickness (t_4)	3.9 mm
9.	Length of the piston (L)	151.75 mm
10.	Outside diameter of the gudgeon pin (d_o)	51.5 mm
11.	Inside diameters of the gudgeon pin (d_i)	30.9 mm
12.	Diameter of the piston boss	72.1 mm
13.	Thickness of the piston ribs	3.65 mm
14.	Piston wall thickness at open end	3.88 mm

3. Static stress analysis for both crosshead and trunk piston

Ansys 2021 is used for the analysis. The static stress analysis and the steady state thermal analysis are the two basic forms of piston analysis. The static stress analysis on the crosshead piston is shown in the stages below.

Step 1: Solidworks is used to import the crosshead piston. The FEA model may be produced using FEA software, however just the most relevant parts are created in order to do the analysis. We can save time and receive better outcomes by doing so. The model may be exported as a.Step file and imported later. The model may be changed and tiny components removed, reducing the processing time.

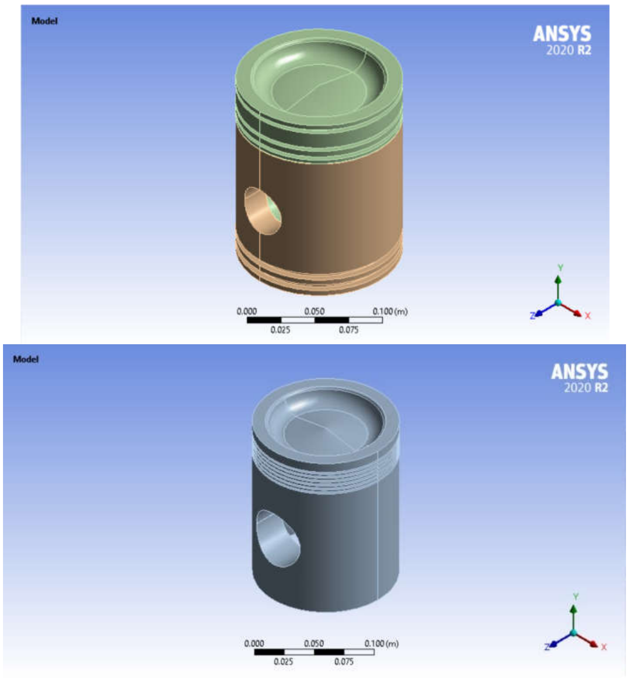

Fig 3: FEA models

Step 2: The material attributes for the FEA model are added in this stage. The attributes might be found in the engineering data base or they can be created. We're going to utilise malleable cast iron (36CrNiMo6) for this study because it's what the crosshead piston is made of (Detroit diesel engine 8V92TA). The material attributes employed in the analysis are depicted in table 2. In the pre-processing stage, the material is allocated to the model.

Step 3: The model is now broken into finite elements, which is known as meshing. The meshing method is tetrahedral, and the element size is 2.5 mm. These are added to the meshing procedure. The element size in the region where the load is applied might be considerably finer than the rest of the area for more precise results. In comparison to the remainder of the region, the top of the piston surface has a mesh size of 1 mm.

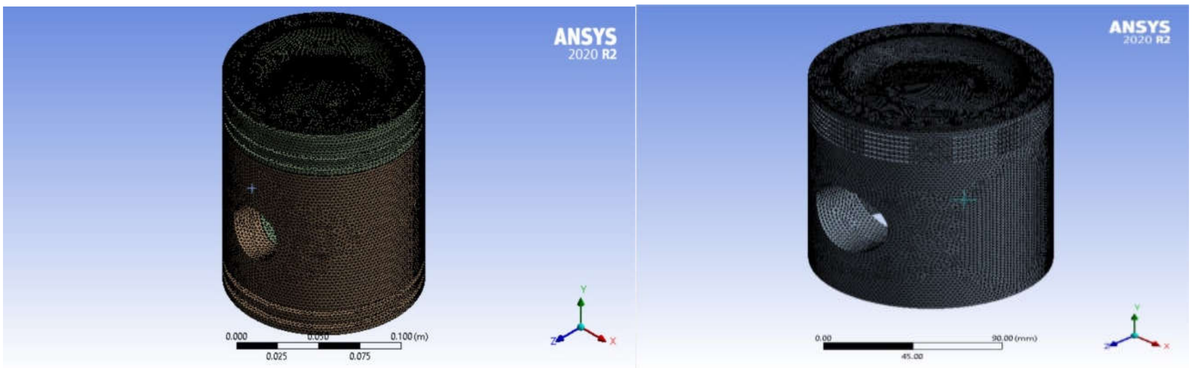

Fig 4: Mesh

Step 4: The model is given restrictions, loads, and supports after the meshing phase. The total gas pressure here is 6 mpa, as determined in the previous section. The gas pressure can be applied to the piston in the form of pressure and pressure direction. The displacement feature, which permits the piston to move in the y direction (x = 0, z = 0, y = free), restricts the piston's movement in the y direction. The piston boss provides support for the piston. That interaction may be represented by a cylinder support, which prevents the cylinder from moving in any of the three directions.

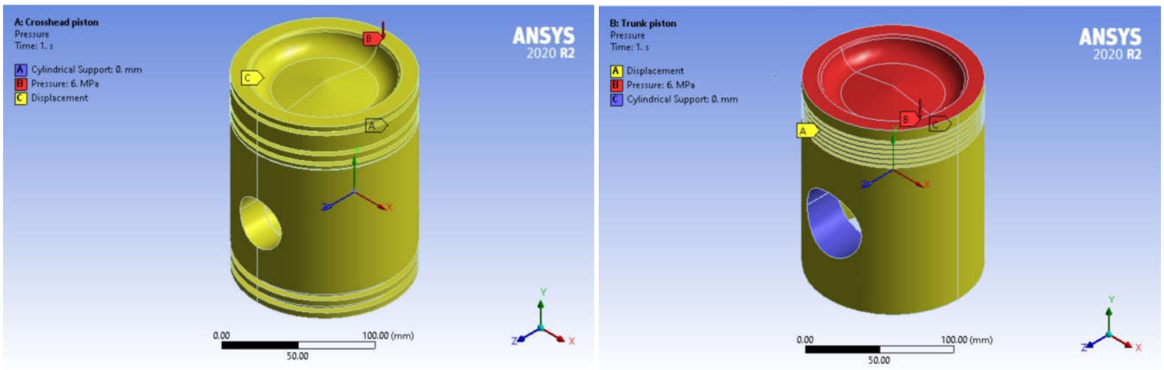

Fig 5: Boundary condition

Step 5: The required data are then entered into the solution after the constraints have been introduced. The answer contains the equivalent stress, maximum primary stress, total deformation, volume, and maximum shear elastic strain. A green tick shows on all of the processes if all of the parameters are valid and typed correctly. After you've ticked everything, you'll need to run the solver. The issue is solved by the solver, and the time it takes is determined by the mesh size and number of elements. If there are more than 1 billion items, the solver will be unable to solve the issue, and the result will not converge or will proceed to infinity.

4. Result and discussions

- The von mises stress and deformation is are obtained through ANSYS 2020 and are tabulated below

	Crosshead piston	Trunk piston
Von mises stress	88.55 MPa	58.60 MPa
Total deformation	0.007 mm	0.007 mm

The result will complete only when the inputs are correct and all the properties to solve the problem. The following are the results obtained for the crosshead piston.

1. Equivalent stress

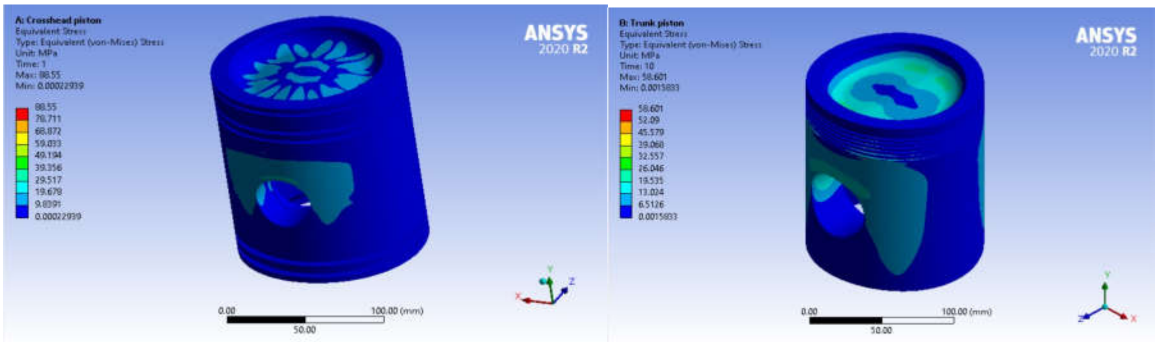

Fig 6: Von Mises stress of both the piston

2. Total deflection

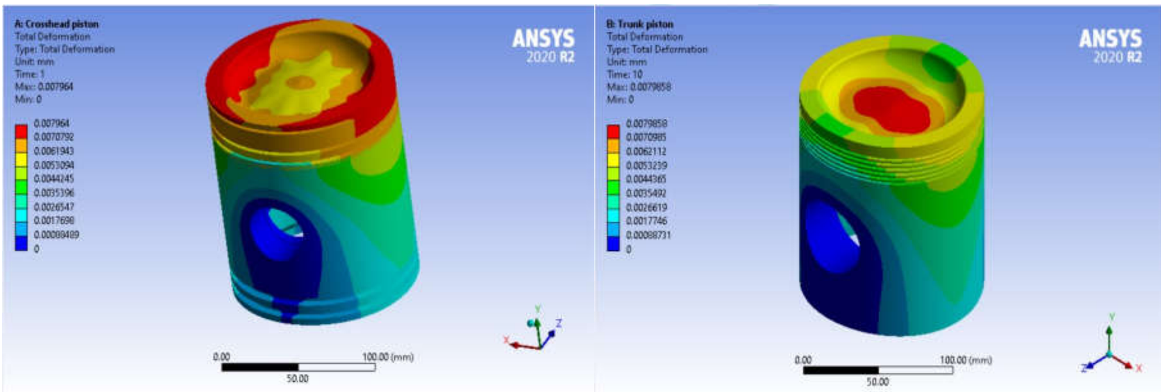

Fig 7: Deflection

By the result, we can see that the stress acing on the piston is 88.55 mpa and the deflection is 0.0079 mm which are near the sides of the piston as shown in the figure 6.

5. Conclusions

- We can see from the preceding analysis that for a load of 6 MPa, both the piston and all of the boundary conditions are the same. We can observe that the trunk piston has 58.6 MPa of stress and the crosshead piston has 88.5 MPa of stress, and the crosshead piston deformation is 0.007 mm and the trunk piston deformation is 0.007 mm under the same load.
- As a result, trunk pistons impose less stress on the frame and can handle the strains generated by a heavy-duty diesel engine. In the analysis, the stress should be minimum and is restricted within the given incremental load. The weight of the new trunk piston is less than the crosshead piston, which improves the inertia force.
- From the analysis the von mises stress of both the piston is analyzed, by correlating both the piston, we can see that the trunk piston has less stress acting on them, with the same load and the deformation of the piston is also the same as the OEM piston or crosshead piston.
- The weight of the new trunk piston is less than the crosshead piston, which improves the inertia force.

6. References

1. N. H. Hamparian,C. S. Hoffma "The Detroit Diesel Crosshead Piston" Automotive Engineering Congress,Detroit, Mich,January 11-15, 1971 .
2. M. Narender, d. Bahar, "Modelling and analysis of flat shape and trunk shape piston", International Journal of Engineering & Science Research, ISSN 2277-2685, IJESR/July 2018/ Vol-8/Issue-7/14-20.
3. D. B. Field and S. J. Hinkle, "Detroit Diesel Allison's Series 92 Engines",Automotive Engineering Congress, Detroit, Mich, February 25-March 1,1974.
4. P. A. Lakshminarayanan, Avinash Kumar Agarwal, "Design and Development of Heavy-Duty Diesel Engines",A Handbook.
5. MAHLE, "Pistons and engine testing", ISBN 978-3-8348-1590-3.
6. Sharon R. Frisch, "Analysis of a Heavy-Duty Diesel Piston Including Material, Air Gap, and Thermal Barrier Coating Effects",International Congress and Exposition, Detroit. Michigan, February 29•March 4. 1988.
7. Yanxia Wang, "Simulation Investigation on the Thermo-mechanical Coupling of the QT 300 Piston", 2009 Second International Conference on Information and Computing Science.
8. Jinfeng Pan, Roberto Nigro and Eduardo Matsuo, "3-D Modeling of Heat Transfer in Diesel Engine Piston Cooling Galleries", 2005 SAE World Congress, Detroit, Michigan, April 11-14, 2005.
9. Islam Ismail, "Thermo mechanical analysis of a direct injection heavy duty diesel engine piston using FEA", Proceedings of the ASME 2016 International Mechanical Engineering Congress & Expositionn, IMECE2016, November 11-17, 2016, Phoenix, Arizona, USA.
10. Manisha B Shinde, Sakore T. V, Katkam V. D., (2016), Design Analysis of Piston for Four Stroke Single Cylinder Engine Using ANSYS, International Journal of Current Engineering and Technology, e-ISSN 22774106.
11. Sudhanshu Maurya, Er. Akhil Vikram Yadav, Er. Anurag Singh, (2019), Modeling & Analysis of a 100cc I.C. Engine Piston, International Research Journal of Engineering and Technology (IRJET), e-ISSN: 23950056. p-ISSN: 2395-0072
12. D. Mani Sai Kamal, (2017), Structural and Thermal Analysis of IC Engine Piston Head with Concave and Convex Shapes, International Journal of Mechanical And Production Engineering, ISSN: 2320-2092.
13. Rahul D. Raut, Satish Mishra, (2015), Stress Optimization of S.I. Engine Piston, International Journal of Science and Research (IJSR), ISSN: 23197064.
14. S. Sathishkumar, (2015), Design and Analysis of IC Engine Piston using Catia and Ansis Software, Research Gate, 303810676.