COMPUTER AIDED ANALYSIS OF PISTON WITH THERMAL BARRIER COATING ON CROWN

Manish Kumar
M.Tech. CAD, HBTI Kanpur India

Abstract:* A piston works in variably complicated conditions, and is subjected to not only the gas pressure due to the combustion, but also due to thermal loads. The top land is known as the piston crown. Its behavior is affected by the temperature variations causing for thermal stresses and thermal fatigue in working. In the present work, the thermal boundary conditions are obtained to calculate the temperature distribution in piston. The analysis is carried out to identify the maximum and minimum temperature, stress and deformation location in piston. The temperature profile of aluminium alloy piston with and without a ceramic coating piston is calculated with apply the convective boundary conditions and ambience temperature. The modeling of piston is carried out in CATIA software whereas ANSYS workbench is used for the Finite Element Analysis. Von Mises stresses criterion is used in Finite element analysis. Finally, it is concluded that heat transfer with a coating on the piston head reduces and the thermal-mechanical stresses caused by temperature and pressure also reduce.
***Keywords:* Piston Crown, Thermal Barrier Coating, Temperature, Heat Transfer, FEM**

I. INTRODUCTION

General piston is cylindrical body, according to the working conditions and requirements of the different engine piston itself is constructed with a wide range, generally the piston is divided into three parts, head, skirt and piston pin for this little thing. The head refers to the part of the piston top ring groove. The skirt portion of the piston means of the lower portion of the piston, its role is to keep the piston in the reciprocating movement of the vertical posture, that is, the guide portion of the piston. The pistons of the automobile are mostly made of cast iron, steel and aluminium alloy through the casting and forging. The main reason for applying these methods is to produce the components integrally and to reach high productivity with the lowest cost and optimized shape. The piston design is complicated because the engine is to work in variably complicated conditions. The piston is subjected to the varying pressure and temperature by the gases and inertia forces due to the acceleration/retardation in a cycle. Piston temperature distribution is important to find out the thermal stresses and its deformations during the design of a piston. [1]. T. Morel et al. showed the temperature and heat flow distribution analytically of an S.I. engine. An advanced engine simulation code was used to solve to engine performance and thermal boundary conditions in WOT condition [2]. Douglas M. Baker et al. discussed about a methodology for a coupled thermodynamic and heat transfer analysis for diesel engine with 1-D and 2-D heat flow finite element models to find the piston and liner temperatures and heat transfer rate [3]. Hidehiko Kajiwara et al. discussed the analytical approach for prediction of piston temperature with provides a piston cooling gallery using CFD tools [4]. The coating on piston face enhances the performance of an engine. The coating provides a low heat rejection phenomenon to the coolant and oil. That enhances the power output and better efficiency. Low basic fuel consumption and high heat release rate. The temperature of exhaust gases are with coating. Winkler M.F., et al described the thermal barrier coating effects on diesel engine performance and components lifetime. Mahdi Hamzehei et al. measured the temperature of piston and cylinder head in a 4-cylinder gasoline engine at actual process [5]. Krisztina Uzuneanu et al. modeled the heat transfer in the piston head of a spark ignition engine supplied with ethanol-gasoline blend using simple thermal networks to multidimensional differential equation modeling [6]. P. Gustof et al. determined the temperature distribution in the piston in initial phase of the work of the turbocharged diesel engine by menas of the two-zone combustion model and the finite element method [7].

II. BACKGROUND

Material model and material properties plays an important role in the result of design and application of the product in servicing environment. The cyclic material properties are used to calculate the elastic-plastic stress-strain response and the rate at which fatigue damage accumulate due to each fatigue cycle. The most commonly used materials for pistons of I.C. engines are cast iron, cast aluminium, forged aluminium, cast steel and forged steel. The cast iron pistons are used for moderately rated engines with piston speeds below 6 m / s and aluminium alloy pistons are used for highly rated engines running at higher piston speeds. A coating is a covering that is applied to the surface of an object [8]. By attaching an adherent layer of a low thermal conductivity material to the surface of a diesel engine piston, a temperature drop can be induced across the thickness of the layer. This results in a reduction in the metal temperature of the component to which it is applied. Using this approach temperature drops of up to 170 degree C at the metal surface have been estimated for 50 μm thick yttria stabilized zirconia coatings.. However, the development of a “prime reliant” TBC system, for which the probability of failure is sufficiently low, would allow these coatings to be used to increase the engine operating temperature and lead to significant improvements in engine performance.

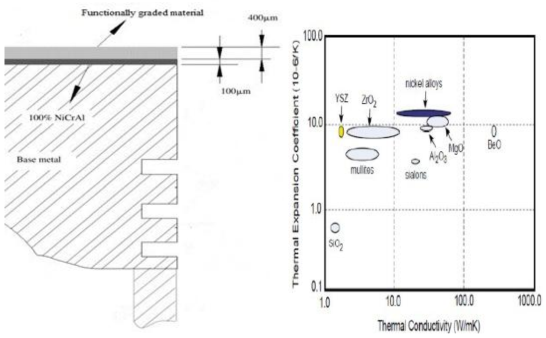


Fig 1: (A) piston ceramic coating, (B) Materials for thermal barrier coating [9]

III. STEADY STATE ANALYSIS

The analysis is carried out in steady state for the piston in this work. A piston has three types of loading during operation. Peak combustion pressure, thermal loads by combustion and inertia loads due to mass.

A. Peak Combustion Pressure

The gas pressure model is described without the effect of temperature acting on the piston crown, combustion chamber surface, field of fire, ring grooves and so on as shown in Figure 2.

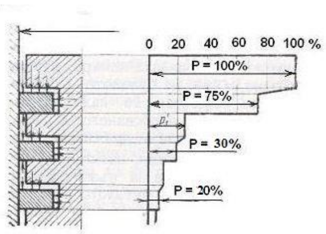


Figure 2: Variation of gas pressure along cylinder axis

The maximum explosive pressure is taken as 10 times of mean effective pressure. The maximum gas pressure is 5.5 MPa for pressure loading on piston top which is the 10 times of mean effective pressure approximately [10].

B. Inertia Loads

Because the piston does the reciprocating motion in the cylinder, according to the Dynamics of Engine, this process can produce the reciprocating inertial force P_j. Its value is proportion to the acceleration of the piston. The mass of the piston is supposed to be concentrated in its centre of gravity. The maximum reciprocating inertial force can be calculated as:

$$P_j = -m_j(1+\lambda)R\omega^2$$

In ANSYS, the inertial force is not loaded directly, but acceleration is loaded in it. Software calculates the inertial force automatically with take the mass of piston into account. The acceleration is obtained as:

$$a = R\omega^2(\cos\alpha + \lambda\cos 2\alpha)$$

Among them,

ω is the crank angle: ω = RPM π / 30

λ is the link rod ratio: λ = R/L

Three freedom degrees of the piston pin are restrained to let the piston in a static condition to eliminate the revolving of the piston around the piston pin. The selection of the displacement boundary condition is very important to the finite element analysis. If the selection is not correct, it will affect the calculation precision. On the moment of the maximum gas pressure, the pin contacted to the surface of the pin hole [10].

C. Thermal Loads

Heat transfer analysis of engine is carried out using convection heat transfer, to calculate the heat transfer coefficient and ambience temperature on piston boundaries, hence third kind boundary condition is used [11].

$$-k\frac{\partial T}{\partial n} = h(T - T_f)$$

Where, *T* is the surface temperature, *n* is the exterior normal vector for the object boundary, h is the convection heat transfer coefficient, *k* is the thermal conductivity of object and T_f is the ambience temperature at the boundary of the object.

Gas temperature is a function of crank angle in an engine and varies with crank angle. So, for steady state thermal analysis we can use the following relationship to obtain mean gas temperature at mean gas pressure [12],

$$T_{gm} = \frac{\mathbf{p_m\ V\ M}}{\mathbf{m\ R_u}}$$

The average heat transfer coefficient at the gas side is calculated with assumption Nusselt, Reynolds and Prandtl number relationship for turbulent flow in pipes and over flat plates [12]:

$$Nu = a\,Re^m\,Pr^n$$

In present work, the mean heat transfer coefficient is calculated from empirical relationship of Nusselt's equation [13] at mean pressure and bulk gas temperature.

$$h_m = 5.41\times10^{-4}P^{\frac{2}{3}}Tg^{\frac{1}{3}}(1+1.24C_m)$$

Table 1: Gas Parameters

Parameter	Value
Bulk gas temperature, T_{gm}	1162 K
Gas heat transfer coefficient, h_m	293 W/m^2K
Cylinder wall temperature, T_{wg}	245 ^0C
Wall thermal conductivity, k	46 W/mK
Wall thickness, t_w	6 mm
Water temperature, T_{cm}	80 ^0C
Water heat transfer coefficient, hc	1400 W/m^2K
Molar mass of combustion gas, M	29 kg/KMole
Combustion gas density	1.127 kg/m^3

Steady State Convective Boundary Conditions

The piston gets heat at crown from combustion with convection mode and transfers this heat to the whole piston in conduction mode and from warm piston to coolant; heat is transferred in convection mode of heat transfer. Hence, the convection heat transfer takes place at the boundary of the piston. The piston is distinguished in many surfaces as shown in Figure 3 with different ambience temperature and heat transfer coefficient.

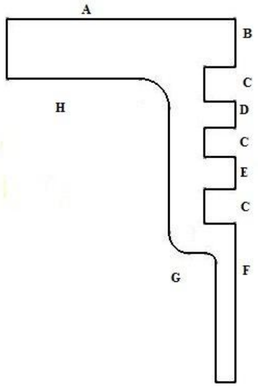


Figure 3: Heat exchange surfaces of the piston

The boundary conditions are calculated of piston surfaces from literature with numerical means of surfaces from A to H, At piston crown face, A: Ambience Temperature is bulk gas temperature and gas side heat transfer coefficient. Ring land surface, B: Since there is a small gap between piston crown and the liner, the captured gas temperature is the mean temperature of the crevice surfaces [14]. Therefore, the heat is conducted through the gas with the convective heat transfer coefficient:

$$h = \frac{k}{\delta}$$

Where δ is crevice clearance, 0.5 mm for the present work, for k = 0.047 W/ m K for gas

Ring groove face, C: the rings are placed to make a tight seal for gases. The heat transfer modeling in this region is obtained through ring resistance approach. The oil film resistance R2 is neglected in this face. The heat transfer in rings is shown in Figure 4,

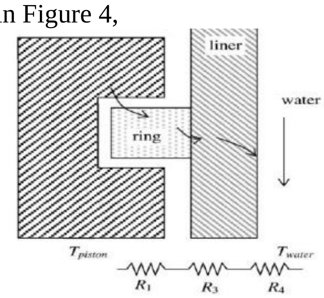


Figure 4: Thermal circuit of heat transfer resistances in the region of the rings

The resistances are:

$$R_1 = \frac{\ln(r_2 / r_1)}{2\pi L_{1kring}} \quad R_3 = \frac{\ln(r_4 / r_3)}{2\pi L_3 k_{liner}}, \quad R_4 = \frac{1}{h_{water} A_s},$$

A heat transfer path is considered (Figure 4) and the values of R1, R3 and R4 are calculated. The effective heat transfer coefficient is obtained from,

$$h_{eff} = \frac{1}{R_{tot} A_{eff}}$$

Where, A_{eff} is the piston surface in contact with the ring and R_{tot} is the total resistance. Other ring land D and E: the heat transfer coefficient at this face is taken as the one third of the gas heat transfer coefficient and the ambience temperature is the 69.5 % and 66.5% of the first ring land temperature for D and E respectively [15]. Skirt F: The heat transfer at this face through convection with low coefficient [16]. Piston underside heat transfer G and H: the piston underside is divided into two regions: Region H, the crown underside is cooled by splash cooling type. The ambience temperature is taken in this region the oil temperature [15]. The value of convective heat transfer coefficient is calculated from the following equation

$$h_H = 900\left(\frac{RPM}{4600}\right)^{0.35} \qquad h_G = 240\left(\frac{RPM}{4600}\right)^{0.35}$$

The surface boundary temperature and convective heat transfer coefficient for given Figure 3 is given in Table 2 below:

Table 2: Piston Surface Boundary Conditions

Thermal boundary	Ambience temperature [^{0}C]	Heat transfer coefficient [W/m^{2}K]
Piston crown, A	890	293
Crevice surface, B	245	94
Rings, C	80	485
Ring land, D	170	98
Ring land, E	162	98
Skirt, F	80	65
Underskirt, G	120	162
Under crown, H	120	608

The specifications of the engine under study to establish the thermal analysis and to design the piston for to determine the temperature distribution and stress analysis is given below. The dimensions of piston are calculated from the design fundamentals [17].

Table 3: Engine Specifications

Bore	80 mm
Stroke	110 mm
Connecting Rod	234 mm
Compression Ratio	16.5
Rated Power	3.7 kW @ 1500 RPM
SFC	245 g / kW h

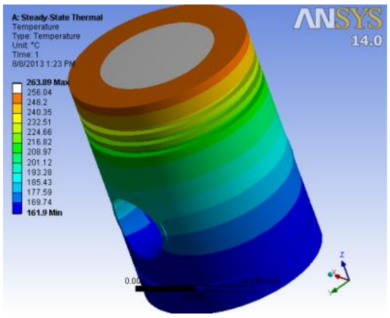
Table 4: Piston dimensions

Piston Parameter	Value [mm]
Piston diameter	79
Top ring land	11
Crown thickness	10
Other ring land	2.5
Radial thickness of groove	3
Piston pin diameter	30
Compression height	60
Total height of piston	105

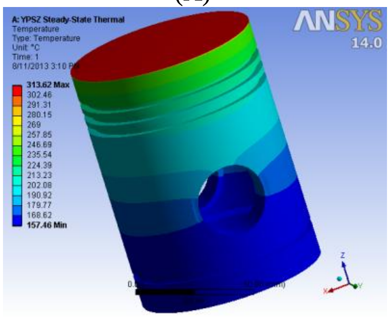
IV. RESULTS AND DISCUSSIONS

A. Temperature Distribution

Based on the established geometrical model and finite element model as well as the established boundary conditions and with the static thermal analysis module of the finite element analysis software, the piston temperature changes between 161.90C to 263.890C with the maximum temperature at the piston top and minimum temperature at the lower part of the piston skirt. The piston temperature changes uniformly from the piston top to the bottom, without any sharp change phenomenon. The piston temperature profile of piston with coating is 313.62 0C and minimum on skirt is 157.460C.



(A)



(B)

Figure 5: Temperature of piston (A) without coating (B) with coating

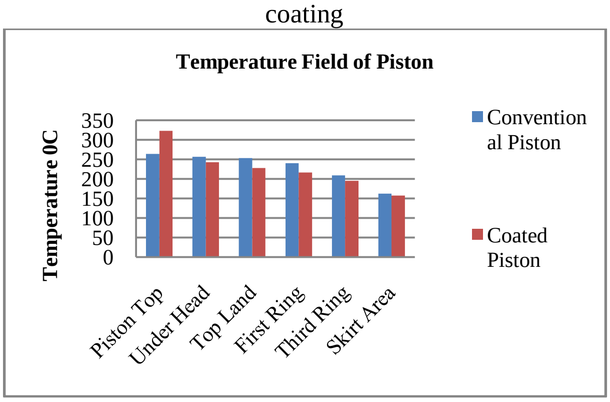
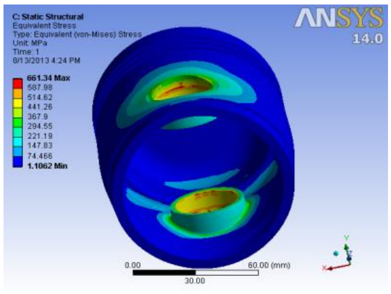


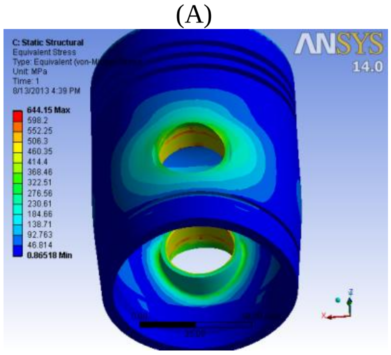
Figure 6: Comparison of temperature distribution

B. Stress Analysis

In this section, the coupling stresses without coating on aluminium alloy piston are calculated. The thermal and mechanical boundary conditions are given into ANSYS work bench. For uncoated piston The maximum value of stress near piston pin hole is found 661 MPa, and total deformation is 0.33243 mm. And for coated piston, the maximum value of stress is obtained from software is 644 MPa near the piston pin region. The stress varies from maximum to minimum near the pin area on skirt of the piston and total deformation is 0.31335 mm.

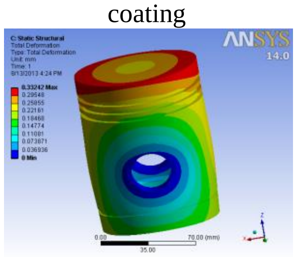


(A)

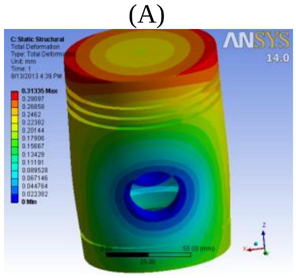


(B)

Figure 7: Stress of piston (A) without coating (B) with coating



(A)



(B)

Figure 8: Deformation of piston (A) without coating (B) with coating

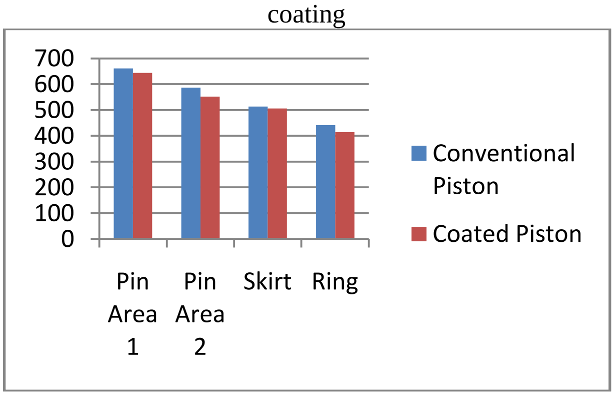


Figure 9: Comparison of stress distribution (MPa)

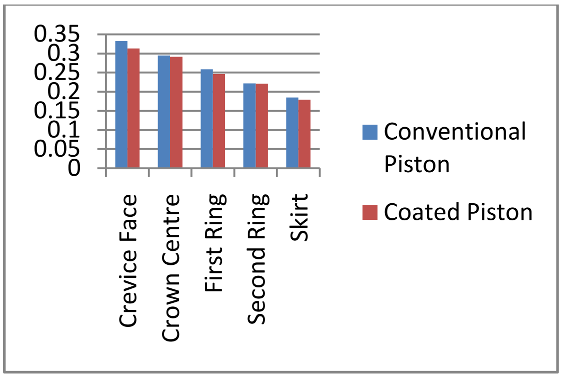


Figure 10: Comparison of Total Deformation

V. CONCLUSIONS

From the analysis for work, the following conclusions are made, For temperature profile of piston, The maximum temperature is obtained for conventional piston is 263.89 0C, and for coated piston is 313.62 0C. The heat flux on piston crown for conventional piston is 78244 W/m2 and for coated piston is 69717 W/m2. Therefore, it is conclude that the surface temperature is higher for coated piston is higher about 19 % respectively. The heat flux is reduced with ceramic coated piston by 10.89%. It is because the heat insulation of a material is proportional to the thermal diffusivity. The heat transfer to the material decreases as the thermal diffusivity coefficient decreases. In reduction the heat transfer through piston crown, the performance of engine increases. The coupled stresses in piston by temperature, pressure and inertia forces are calculated. Coupled stress for conventional piston near piston pin hole region is 661.34 MPa. The total deformations are 0.33242 mm respectively. The stresses of coated piston are 644.15 MPa and total deformation is 0.31335 mm. Therefore, it is conclude that the reduction in stress with ceramic coating on piston is 2.59 % and reduction in total deformation by 5.73%.

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