Mechanical And Thermal Stresses Anylsis In Diesel Engine Poiston With And Without Different Thermal Coating Layer On Piston Head

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Abstract. The main goal of this paper are equate behavior of the piston with and without different thermal coating layer on the piston head under thermal and mechanical loads that arise in the piston due to its operating. Three dimensional model of a piston heavy diesel engine has been presented. The governing equations have been discretized using a finite-volume method (FVM) and solved using multi-physics COMSOL package version 5. The results of the numerical model are showing the distribution of temperature, temperature gradients, Von-Mises stresses, and displacement in the diesel engine piston with and without 200 μ m of thermal coating layer as (La2Zr 2O7) which have low thermal conductivity. The results show great improving in the performance of the piston with thermal coating layer.

Keywords: Diesel engine piston, Thermal analysis, Thermal barrier coating, 3D finite volume method.

1. Introduction

In the combustion chamber engines, some of the parts like cylinder head, cylinder liner, piston and valve are most thermal loaded parts because they are in direct contact with the flame due to this they losses their strength and slightly deform from its original state. So it becomes important to calculate the piston stress distribution in order to control the deformations within acceptable levels. The stress distribution enables the designer to optimize the thermal aspects of the piston design at lower cost. The mechanical stress and

thermal stress level depends on the distribution of temperature in the parts, thermal expansion coefficient, young modulus of elasticity, thermal load, design of the parts and cooling conditions [1,2]. The diesel engine is undergoing continuous technological progression to become an adiabatic engine improvement in performance and durability of IC engine. A major breakthrough in diesel engine technology to make it adiabatic can be achieved by coating the various parts like the cylinder wall, combustion chamber, cylinder head, piston body, valves etc., with ceramic materials having very low thermal conductivity [1]. The coating material of the adiabatic engine must have a high temperature resistance, low thermal expansion coefficient, low friction characteristics, good thermal shock resistance, high strain tolerance, low sintering rate of the porous microstructure, lightweight and durability [3, 4]. The ceramic-coated piston has been widely used in the internal combustion engine, and the ceramic coating can be used as an insulating layer that reduces much of the heat loss of the internal combustion engine and obtains higher efficiency [5].

The application of Thermal barrier coatings on the diesel engine piston head reduces the heat loss to the engine cooling-jacket through the surfaces exposed to the heat transfer such as cylinder head, liner, piston crown and piston rings. It is important to calculate the piston temperature distribution in order to control the thermal stresses and deformations within acceptable levels [6].

In the past two decades, much attention has been paid to the study on 'adiabatic' or 'low heat rejection' engines. These studies have been commonly performed on Diesel engines [7]. Uzun et al. [8] studied an experimental investigation into the effects of ceramic coatings on the performance of a diesel engine and exhaust emissions. Ceramic coatings can eliminate visible smoke, inhibit the formation of NOx reduce CO and particulate emissions, and improve combustion efficiency. The performance of the diesel ceramic coating was tested on a hydraulic engine dynamometer. The coatings were being evaluated for their ability to control particulate emissions, for emissions in exhaust gases for smoke, horsepower, speed and fuel rate. CO and hydrocarbon levels were lower than baseline levels.

Buyukkaya and Cerit [9] investigated thermal analyses on a conventional (uncoated) diesel piston, made of aluminum silicon alloy and steel, thermal analyses were performed on pistons, coated with $MgO - ZrO2$ material by means of using a commercial code, namely ANSYS. Those results of four different pistons were compared with each other. The effects of coatings on the thermal behaviors of the pistons were investigated.It has been shown that the maximum surface temperature of the coated piston with material which has low thermal conductivity was improved approximately 48% for the AlSi alloy and 35% for the steel.

Bhagat and Jibhakate [10] described the stress distribution of the seizure on piston four stroke engine by using FEA. The finite element analysis was performed by using computer aided design (CAD) software. The main objective was to investigate and analyze the thermal stress distribution of piston at the real engine condition during combustion process. That paper describes the mesh optimization with using finite element analysis technique to predict the higher stress and critical region on the component. The optimization is carried out to reduce the stress concentration on the upper end of the piston i.e (piston head/crown and piston skirt and sleeve). With using computer aided design (CAD), Pro/ENGINEER software the structural model of a piston will be developed. Furthermore, the finite element analysis performed with using software ANSYS.

Rakopoulos and Mavropoulos [11] used a piston model for the calculation of the temperature field and heat flow field under steady and transient engine operating conditions. Three-dimensional finite-element analyses were implemented for the representation of the complex geometry metal components and found a satisfactory degree of agreement between theoretical predictions and experimental measurements.

Muhammet Cerit [12] determined the temperature and the stress distributions in a partial ceramic coated

spark ignition (SI) engine piston. Effects of coating thickness and width on temperature and stress distributions were investigated including comparisons with results from an uncoated piston. It was observed that the coating surface temperature increase with increasing the thickness in a decreasing rate. Surface temperature of the piston with 0.4 mm coating thickness was increased up to 82 ° C. The normal stress on the coated surface decreases with coating thickness, up to approximately 1 mm for which the value of stress was the minimum. However, it rises when coating thickness exceeds 1 mm. As for bond coat surface, increasing coating thickness, the normal stress decreases steadily and the maximum shear stress rises in a decreasing rate. The optimum coating thickness was found to be near 1 mm under the given conditions.

Li [13] used a three-dimensional finite element model of an aluminum diesel engine piston to calculate operating temperatures. He showed that skirt contours played an important part in the reduction of scuffing and friction.

Prasad et al. [14] used thermally insulating material, namely partially stabilized zirconia (PSZ), on the piston crown face and reported a 19% reduction in heat loss through the piston.

Pierz [15] investigated the thermal barrier coating development for diesel engine aluminum piston he found that the resulting predicted temperatures and stresses on the piston, together with material strength information, the primary cause of coating failure is proposed to be low cycle fatigue resulting from localized yielding when the coating is hot and in compression.

The main scope of this work is to reduce the mechanical and thermal stresses level depends on the distribution of temperature in the piston and to increase the life span of the coated pistons of the engine using a thermal barrier coating (TBC) technology. The best performance can be obtained by combining the chosen materials and its own properties combines to create better performance.

2. Numerical Model

Three dimensional finite volume method (FVM) model of a diesel engine piston has been presented. The model accounts for mechanical and thermal loads that arise in the engine piston due to its operating. The geometry of the piston is shown in Figure 1. The piston sits on the connecting road with crankshaft.
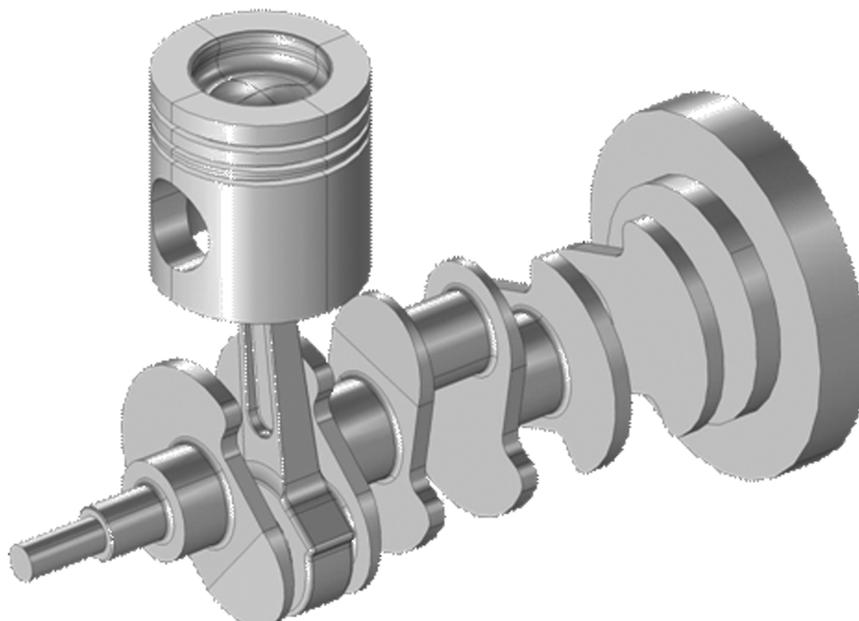

Figure. 1 Geometry of the engine diesel piston.

2.1 Computational domain

A computational model of an entire diesel engine piston would require very large computing resources and excessively long simulation times. The computational domain in this study is therefore limited to a sector from the cylindrical geometry of the piston. The three dimensional computational domain of the diesel engine piston used in the model is shown in Figure 2.
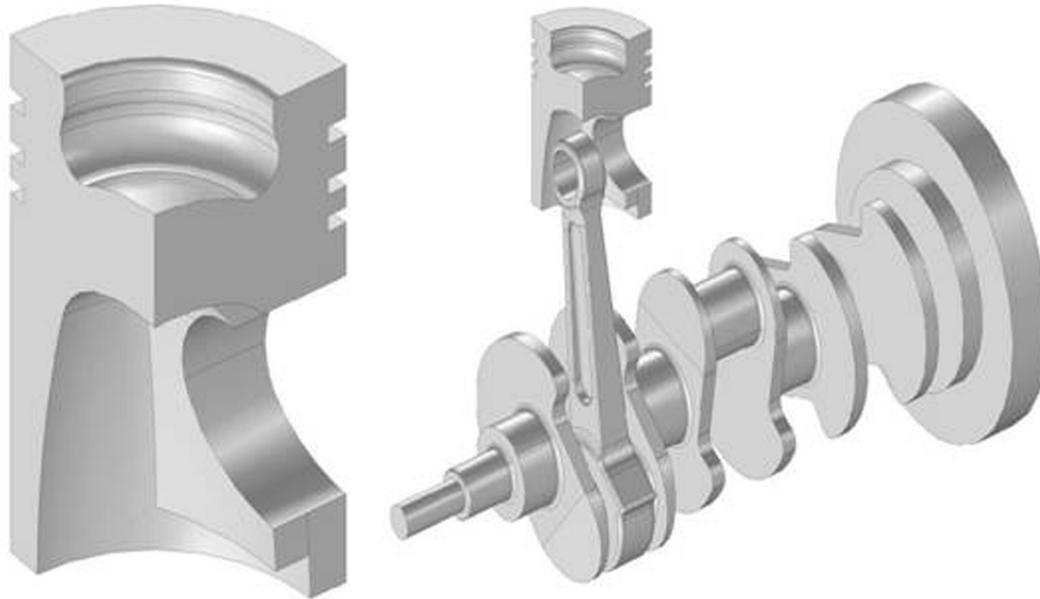

Figure. 2. Three dimensional computational domain of the diesel engine piston.

2.2 Modelling equations

The prediction of the temperature distribution in the diesel engine piston, involves the solution of the heat transfer equation; heat conduction and heat convection with the appropriate boundary conditions.
The heat transfer in the diesel engine piston is governed by [16];

$$\rho c_p \frac{\partial T}{\partial t} + \rho c_p \mathbf{u}.\nabla T = \nabla.(k\nabla T) + Q \qquad (1)$$

Where is the density [kg/m3], cp is the heat capacity [kJ/kg.K], k is the thermal conductivity [W/m.K], u is the velocity vector [m/s], and Q is the heat source [W].
The diesel engine piston is subjected to the pressure force of the hot gases as shown in Figures 3 and 4. The heat transfer coefficient at the diesel engine piston head is calculated using the following equation [17];

$$h_g(\theta) = 3.26\big(P_g(\theta)\big)^{0.8}\big(6.18 \times V_p\big)^{0.8}\big(b^{-0.2}\big)\big(T_g(\theta)\big)^{-0.55} \qquad (2)$$

Where hg is the heat transfer coefficient [W/m2.K], Pg is the gas pressure [Pa], Tg is the gas temperature [K], Vp is the mean piston speed [m/s], b is the piston bore [mm], and is the crank shaft angle [degree].
Because of high variation of the temperature and pressure inside the combustion chamber during the engine cycle, the resulting gas temperature (Tgr) and mean heat transfer coefficient (hgm) can be

calculated as [17];

$$T_{gr} = \frac{(h_g T_g)_m}{h_{gm}}$$

(3)

$$h_{gm} = \frac{1}{720} \int_0^{720} h_g(\theta) d\theta$$

(4)

$$(h_g T_g)_m = \frac{1}{720} \int_0^{720} h_g(\theta) T_g(\theta) d\theta$$

(5)

Mechanical and thermal stresses will be analysed with the following equations [16];

$$-\nabla.\sigma = \mathbf{Fv}$$

(8)

The thermal strains resulting from a change in temperature of an unconstrained isotropic volume are given by [16];

$$\epsilon_{th} = \alpha(T - T_{ref})$$

(9)

Where is the thermal expansion [1/K], and Tref is the valve reference temperature.

The analysis was performed under the worst thermal loading condition of rated power. The engine specification and operating condition are summarized in Table 1.

Table 1 Engine specifications and operating conditions

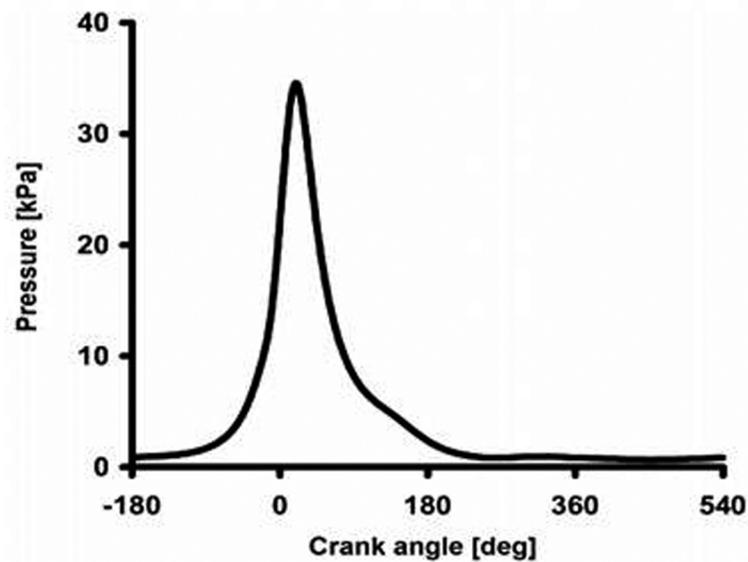| Component	Value
Piston bore [mm]	87.3
Connecting rod [mm]	130
Cylinder number	4
Int. valve number.	4
Exh. valve number	4
Compression ratio	8.6
Valve lift [mm]	9.5
Engine speed [rpm]	4700
Air fuel ratio	12
Power [hp]	66
Piston stroke [mm]	66.7
Mean piston speed [m/s]	10.4

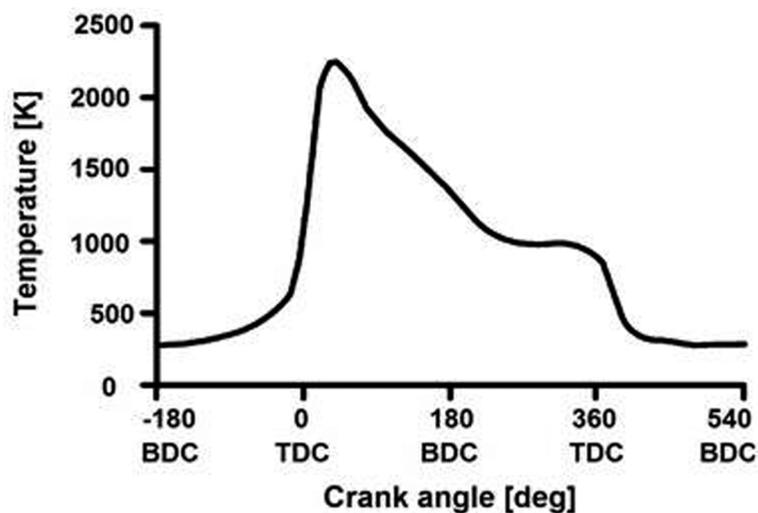Figure. 3 Diagram of measured gas pressure in the cylinder.

Figure. 4 Diagram of measured gas temperature in the cylinder.

2.3. Computational procedure

The governing equations were discretized using a finite-volume method and solved using multi-physics COMSOL package Version 5. Stringent numerical tests were performed to ensure that the solutions were independent of the grid size. A computational quadratic mesh consisting of a total of 359378 domain elements, 52230 boundary elements, and 5686 edge elements was found to provide sufficient spatial resolution (Figure 5). In addition to the heat transfer boundary conditions, the mechanical boundary conditions were applied as shown in Table 2. The coupled set of equations was solved iteratively, and the solution was considered to be convergent when the relative error was less than $1.0 \times 10\text{-}6$ in each field between two consecutive iterations.
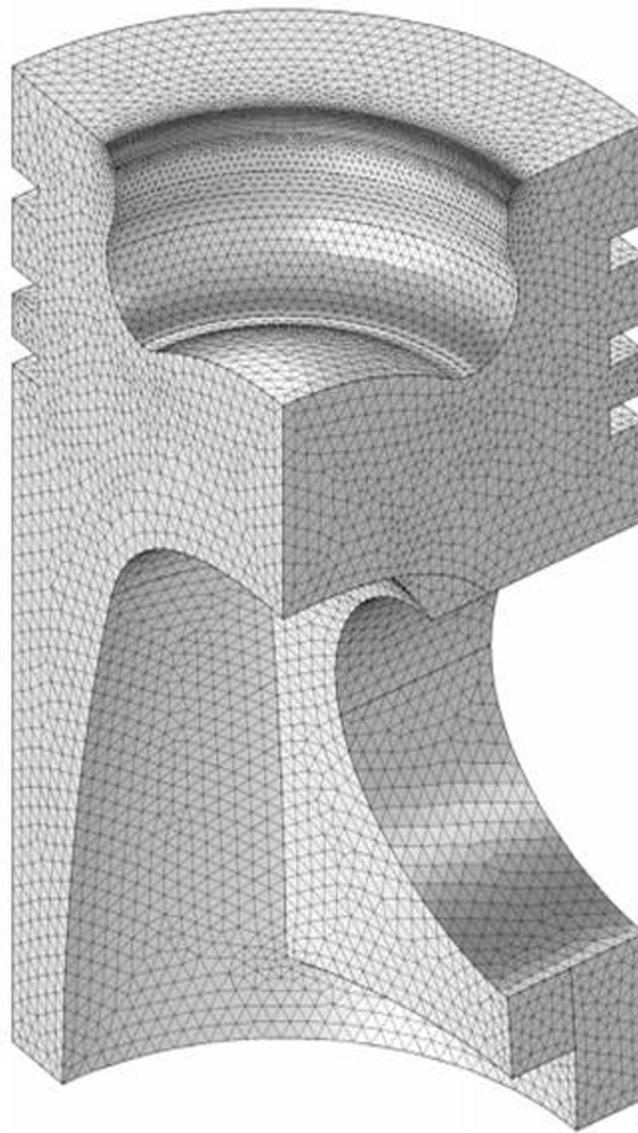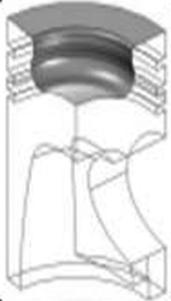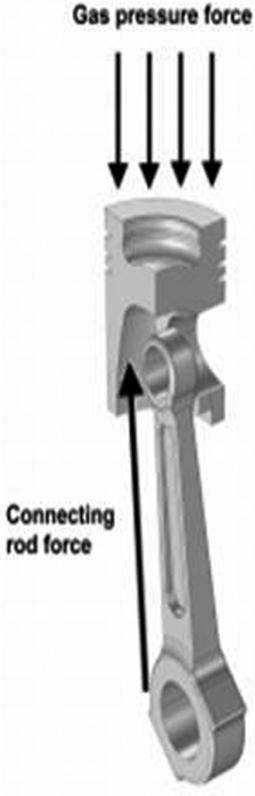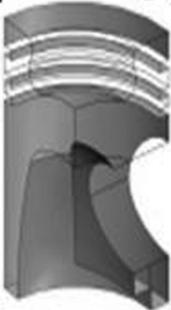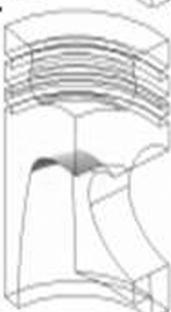
Figure. 5 Computational mesh of the computational domain.

Table 2 Boundary conditions

heat transfer boundary conditions		mechanical boundary conditions
	$T_{gf} = 1000\ ^{\circ}C$ $h_{gm} = 800\ W/m^2.^{\circ}C$	
	$T_{gf} = 80\ ^{\circ}C$ $h_{gm} = 700\ W/m^2.^{\circ}C$	**Gas pressure force**
	$T_{gf} = 80\ ^{\circ}C$ $h_{gm} = 500\ W/m^2.^{\circ}C$	
	$T_{gf} = 80\ ^{\circ}C$ $h_{gm} = 250\ W/m^2.^{\circ}C$	**Connecting rod force**
	$T_{gf} = 80\ ^{\circ}C$ $h_{gm} = 1500\ W/m^2.^{\circ}C$	

2.4. Comparing with real damage cases

The results of the numerical model are showing the temperature gradients in the diesel engine piston. The maximum values of the temperature gradients appear around the piston head and also occur in the rings grooves. This leads to develop and increase of the maximum thermal stress, and as a result, it contributes to the formation of microcracks. At the worst conditions for the cooling of the piston such as distortion of the piston head, this stress causes radial cracks at the edge of the piston. In addition, microcracks lead to loss in material of the piston and consequently to its defects. Examples of damage of the diesel engine piston shows in Figure 6 with good similarity to the numerical model.

3. Effect of The Coated Layer on The Piston Performance

In addition to revealing the detail of mechanical and thermal phenomena inside the engine piston, the comprehensive three-dimensional model can also be used to investigate the sensitivity of certain parameters on piston performance. The validated model is now ready for studying the effects of the coating of the top surface of the piston with different types of thermal barrier coatings material on the piston performance. The performance characteristics of the engine piston based on a certain parameter can be obtained by varying that parameter (material properties of the coated layer) while keeping all other parameters constant at base case conditions. Results obtained from these parametric studies will allow the identification of the critical parameters for piston performance as well as the sensitivity of the model to these parameters. The top surface of the piston head is coated with 200 μ m thickness of various thermal barrier materials (Figure 7). Material properties of each coated layer are shown in Table 3.

Table 3 Material properties of the coated layer on the top surface of the piston

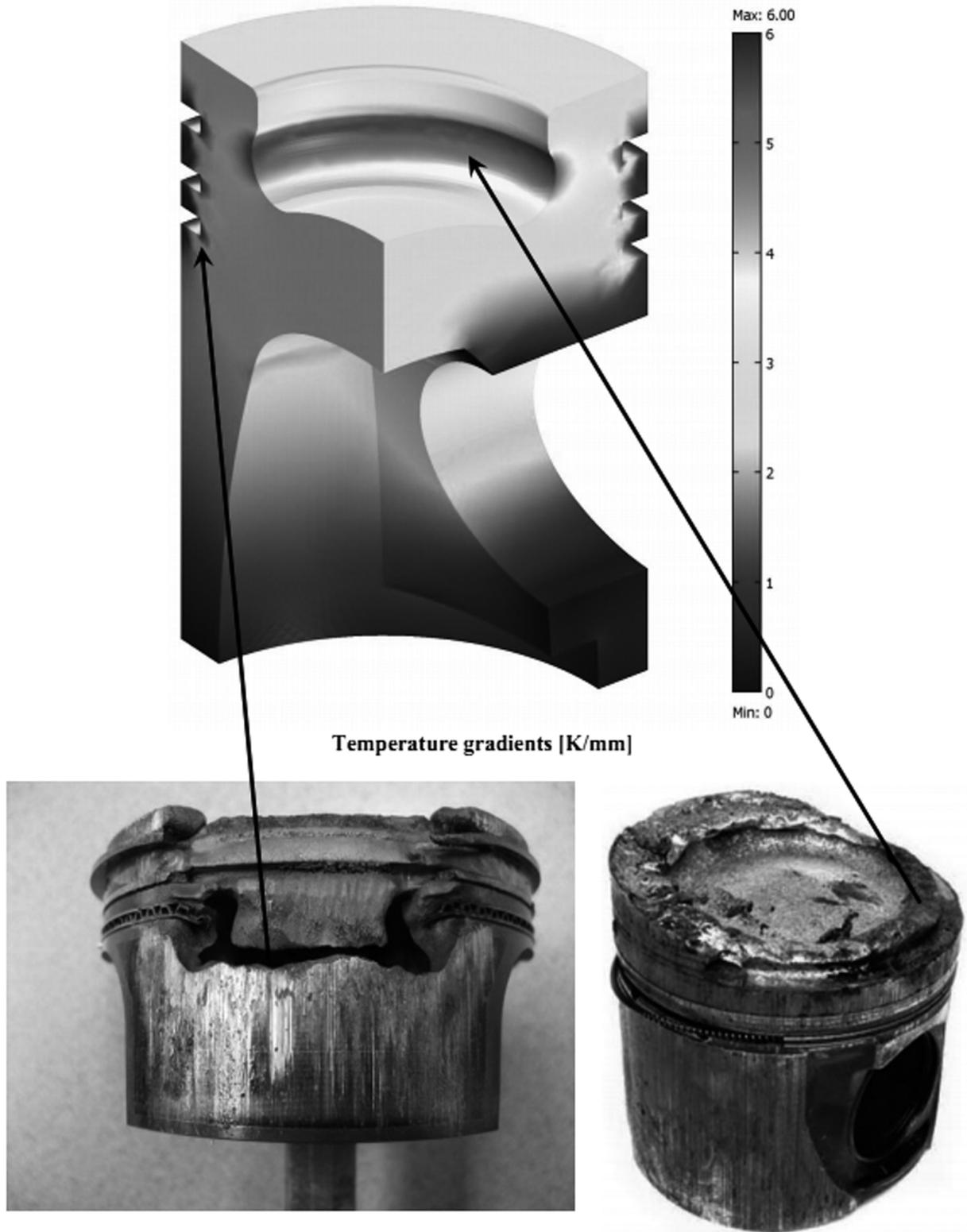| Case no.	Material	Density [g/cm^3]	Thermal conductivity [W/m.K]	Thermal expansion coeff. [1/K]
1	Gd$_2$Zr$_2$O$_7$	7.0	1.6	11.6e^{-6}
2	MgO-ZrO$_2$	3.0	1.57	09.9e^{-6}
3	La$_2$Zr$_2$O$_7$	6.0	1.5	09.7e^{-6}
4	Sm$_2$Zr$_2$O$_7$	6.7	1.5	10.8e^{-6}

Figure. 6 Visual comparison between the result of the numerical model and examples of the real damage pistons.
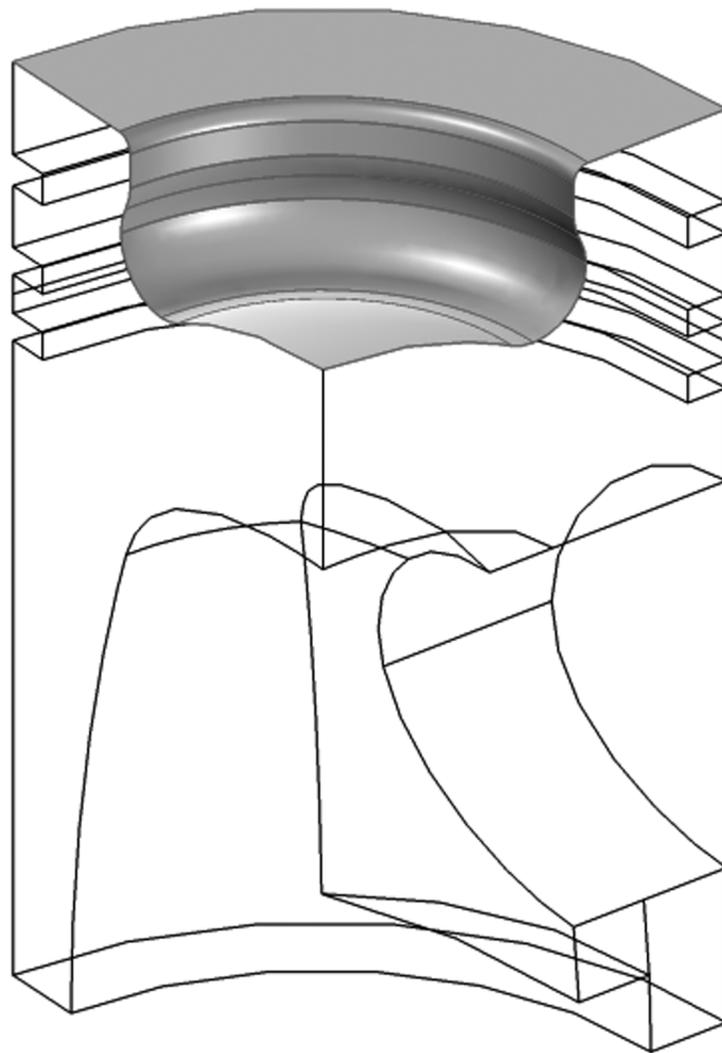
Figure. 7 Thermal barrier material coated layer on the top surface of the piston head.

4. Results

Results obtained from the mechanical and thermal phenomena inside the engine piston analyses by the finite volume technique, the temperature and stress variations for the uncoated top surface of the piston and coated top surface of the piston by the ceramic material are discussed systematically. For thermal barrier coatings in an engine, heat conduction is generally more dominant than radiation; hence, the thermal conductivity of material is very important for estimating temperature distributions and heat flows. The temperature distribution of the piston without and with coating layer is shown in Figure 8. As expected, the high temperatures are observed at the crown center and bowl lip areas, since it is subjected to the heat flux circumferentially. The maximum temperature is at the center and the minimum is at the bottom of the crown bowl on the piston top surface. In the radial direction, the temperature decreases from the crown center to the bottom of the bowl, then it increases towards the bowl lips and finally decreases again at the edge of the crown surface.

Von-Mises stress distributions versus distance for uncoated and coated top piston surfaces are shown in Figure 9. In the coating case, the Von-Mises stress distributions decreases due to decreasing in the thermal conductivity of the ceramic material.

Temperature gradients distribution for uncoated and coated pistons on the top piston surfaces are shown in Figure 10. The maximum values of the temperature gradients appear around the piston head and also occur in the rings grooves. This leads to develop and increase of the maximum thermal stress, and as a result, this stress causes radial cracks at the edge of the piston. By using coating layer on the top piston surfaces, the temperature gradients distribution has been decreased due to the low thermal conductivity of the ceramic material and this leads to reduce failure in the piston.

Figure 11 shows displacement distribution of the standard piston and coated pistons. Ceramic coating material has a low heat transfer coefficient and this reduce the displacement in the piston.
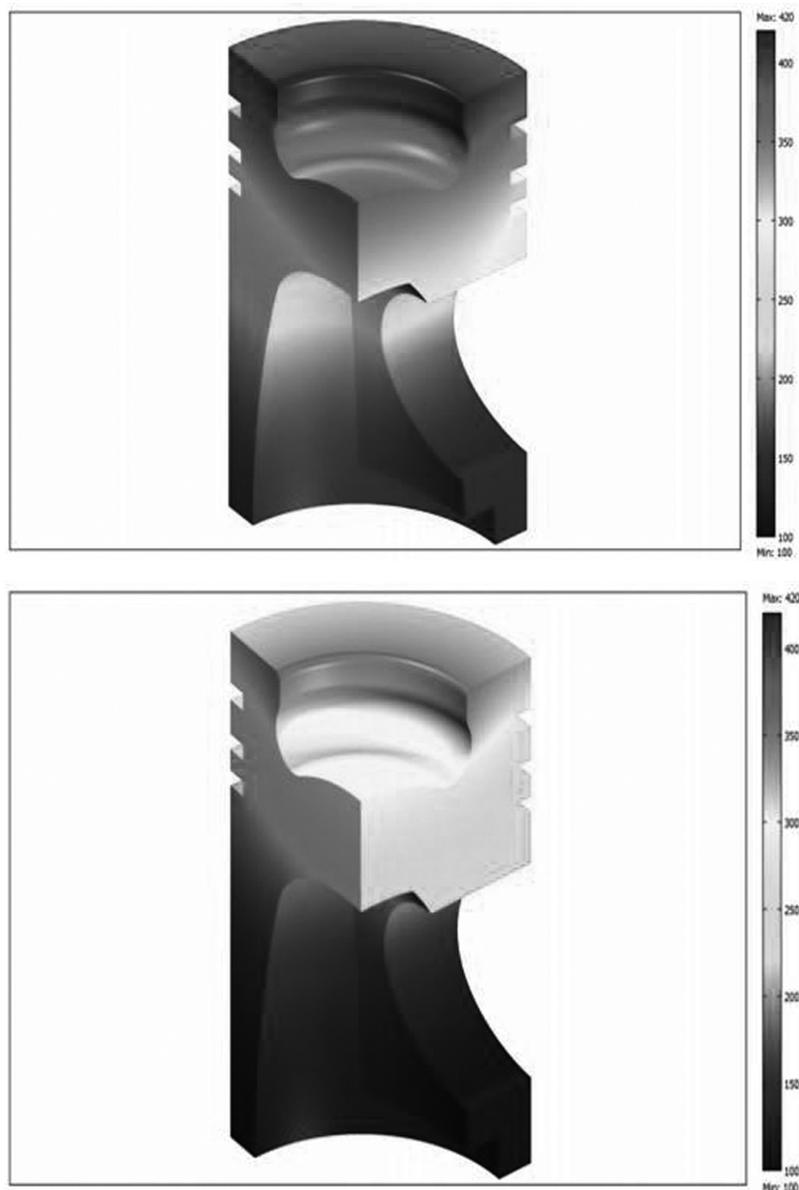

Figure. 8 Temperature distribution of the engine piston [oC] without (upper) and with (lower) coating layer on the top surface of the piston head.
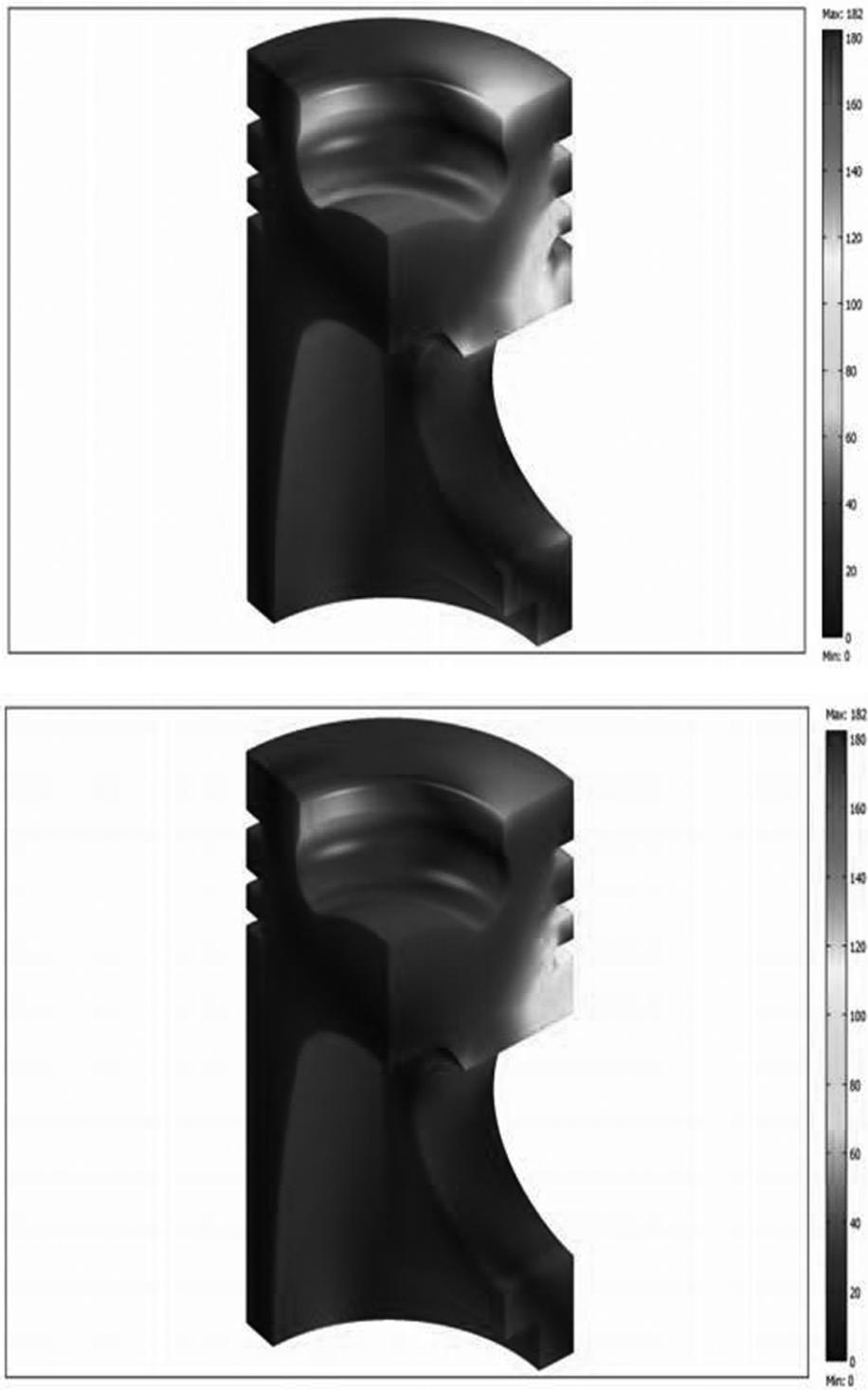
Figure. 9 Von-Mises stress distribution of the engine piston [MPa] without (upper) and with (lower) coating layer on the top surface of the piston head.
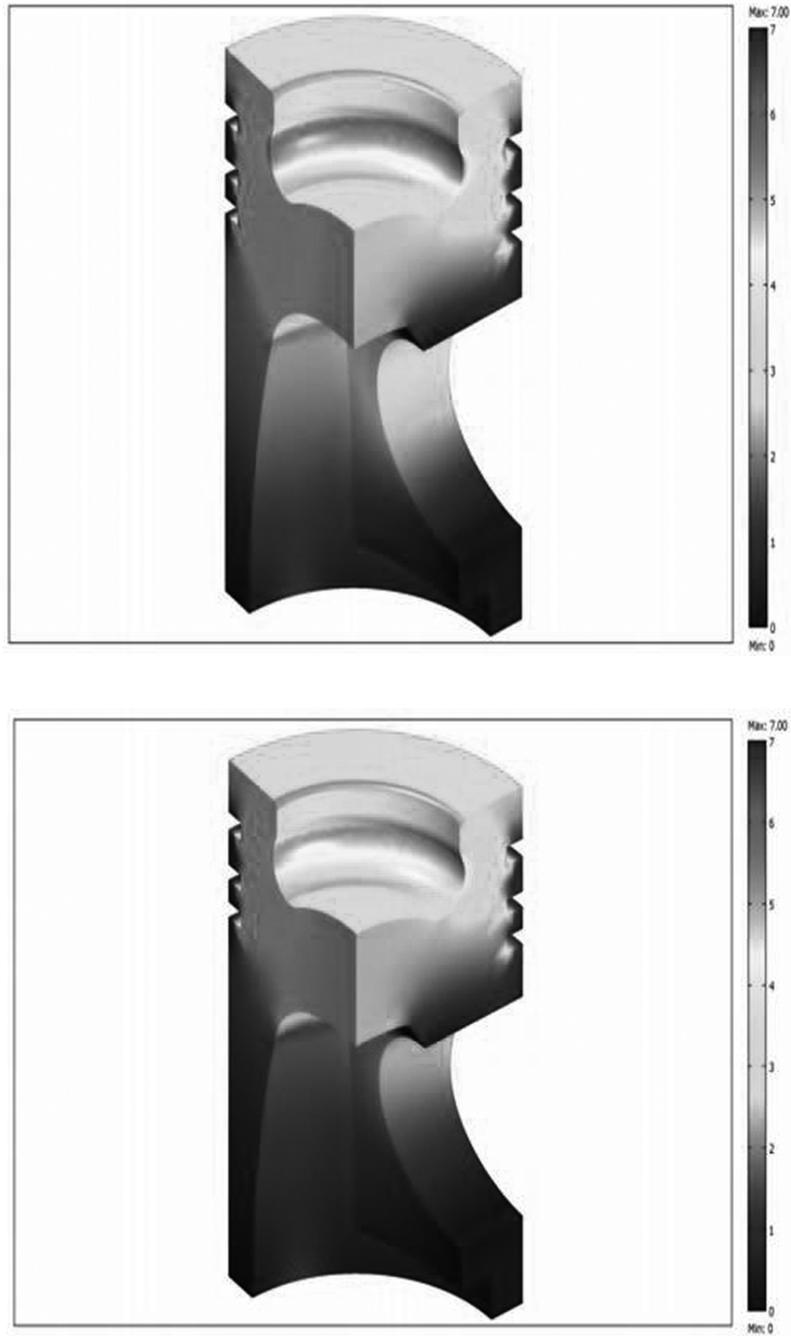
Figure. 10 Temperature gradients distribution of the engine piston [K/mm] without (upper) and with (lower) coating layer on the top surface of the piston head.
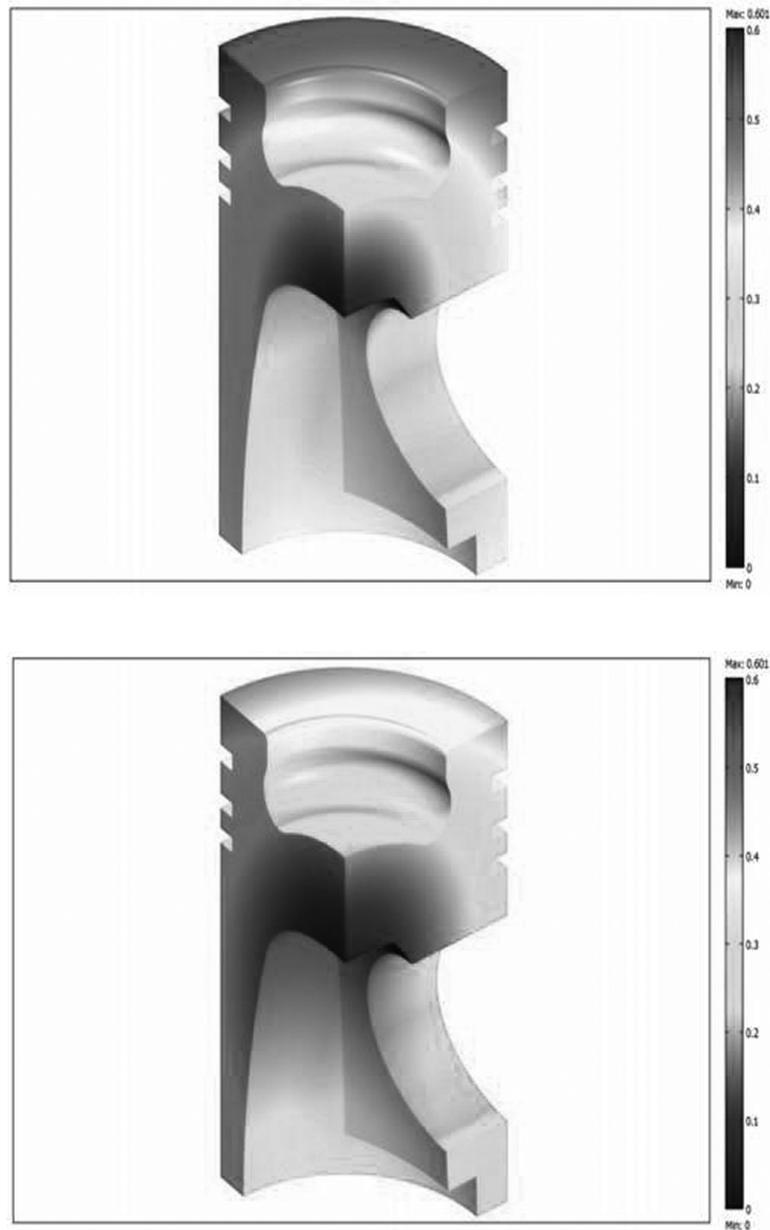
Figure. 11 Displacement distribution of the engine piston [mm] without (upper) and with (lower) coating layer on the top surface of the piston head.

5. Conclusion

The purpose of this work was to compare the behaviour of the piston without and with coating layer on the top of piston under thermal and mechanical loads. The obtained results shows that the thermal and mechanical stresses induced in piston with coating layer are less as compare to the piston without coating layer. The coating material has been applied successfully to the engine piston as the thermal barrier coating of a diesel engine prevents excessive heat loss during combustion. Through the analysis, it is concluded that the main factor influencing the piston intensity is the temperature, thus providing basis for the optimization design of the piston. The stress and the deformation of the piston are mainly determined

by the temperature, so it is necessary to decrease the piston temperature through structure improvement. The results of the numerical model are showing the temperature gradients in the diesel engine piston. The maximum values of the temperature gradients appear around the piston head and also occur in the rings grooves. This leads to develop and increase of the maximum thermal stress, and as a result, it contributes to the formation of micro cracks. It is important to calculate the piston temperature distribution in order to control the thermal stresses and deformations within acceptable levels.

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