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Numerical investigation on effect of thermally isolated piston on improving SI engine performance during the cold start conditions

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ABSTRACT

In low ambient temperatures, the engine will have poor output torque and power generation at the first cycles of its performance, long start time and huge noxious emissions because of insufficient temperature and quality of air-fuel mixture and failure of flame to burn the charge. It is noted that any attempt to increase the charge temperature leads to better combustion conditions. By applying thin ceramic coating over the piston top surface and increasing the temperature of engine inner walls, more efficient combustion in shorter time can be provided for the engine. In current research, a transient thermal study is developed for simulating the engine performance under cold starting conditions and predicting the effect of insulating layer thickness on the charge temperature before ignition timing. The results demonstrate that the ceramic coating can provide higher temperature for piston crown and effectively improve the temperature of charge during the first cranking cycles of engine cold start stage.

1) Introduction

At the beginning of cold start, the temperatures of intake port wall and intake valve are low. This leads to diminished kinetic reaction rates and greater clearances between piston-ring assemblies and cylinder surface during the first cycles. Partial or complete failure of the flame to burn the charge results in poor power generation and huge noxious emissions [1]. With increasing consciousness concerning environmental pollution resulting from internal combustion engines and stringent regulations such as EURO-IV and ULEV (Ultra low emission vehicle), the automobiles emissions becomes an important problem for both governments and car producers [2].

As a main solution, by adoption of catalytic converter, most of engine emission can be effectively controlled [3]. However, the conversion efficiency depends strongly on the working temperature. As a result, at cold start and warming-up periods the converter efficiency is very close to zero [4]. Preparing successful combustion for first few cycles not only improves the environment of mixture formation but also robust the output torque of first few cycles and shorten the start time. Quality of the air-fuel mixture, the extent of dilution with combustion products, charge temperature, engine speed and spark timing are recognized to be effective on combustion instability [5].

Among all mentioned factors, the charge temperature plays a decisive role for first few cycles of engine performance [6]. In some studies, heat storage systems are proposed to heat the entire engine before the cranking start [2, 7]. However, Gumus [2] showed that the efficiency of this method may be low because the most of the energy is wasted on non-essential places. In order to increase the efficiency, the effect of heating the intake air and injectors on better fuel evaporation are investigated [8, 9]. Li et al. [10] proposed also a partial heating device to achieve more effective pre-cranking system. These measures are more efficient than heating the engine entirely. When the injected fuel spray impinges on the cold intake port walls, the fuel spray loses heat to the cold engine wall quickly and condensate on the surface of the intake wall [11]. On the other hand, increase in lubricant viscosity and resistance to the motion will increase the load onto accumulator and starter [2].

It is well known that by insulating the engine and suppressing the heat rejection to the cylinder walls, the combustion temperature will increase. Thus, a thin ceramic coating layer is applied to the combustion chamber components, especially over the top of piston crown surface. In the past two decades, much attention has been paid on the study of "adiabatic or low heat rejection engines". These studies have been commonly performed on diesel engines [12-18]. As an important contributing method for improving the fuel economy and exhaust emissions, less attention has been paid to ceramic coated gasoline engines [19-22]. However, the degree of insulation is an important factor that should be ascertained from the view point of knock free performance [23].

Most of the works for the purpose of engine modification and fuel economy, have concentrated on steady state models. However, for limiting the engine emissions during the engine cold start, the steady state model can no longer contribute. The current research aims at developing a transient model for simulating the engine performance under cold starting conditions. This model has the capability in predicting the effect of insulating layer thickness on increasing the temperature of the charge before ignition.

2) Material and method

The numerical simulation of engine heat transfer is very important for evaluating the temperature variation of the engine block. So, correct transient heat transfer analysis is a pre-condition for the accurate prediction of engine performance during the cold start. The unsteady equation of heat conduction that can be conveniently expressed as [24]:

$$\rho\frac{Dh}{Dt}=\frac{Dp}{Dt}+\nabla .k\nabla T+\dot{Q} \tag{1}$$

Where in this equation, T is temperature, t is time, k is the conductivity factor of constructing material, ρ is the mass density, c is the specific heat and $\dot{Q}$ is the internal heat generation rate per unit volume D/Dt is the total derivation composed of the local time derivative and the convective derivative [25]. The pressure is assumed to be uniform throughout the boundary layer and the viscous dissipation is neglected. The unsteady energy equation can therefore be written as:

$$\rho\frac{\partial h}{\partial t}+\rho u\frac{\partial h}{\partial x}+\rho v\frac{\partial h}{\partial y}+\rho w\frac{\partial h}{\partial z}= \frac{\partial p}{\partial t}+\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+ \frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{Q} \tag{2}$$

All term of the right hand side of equation 2 is a source term and refers to the rate at which energy is delivered into the volume of fluid by means other than conduction (i.e. convection). It is noted that closed form solution of the coupled set of differential equations is commonly difficult to obtain. So, a finite element analysis as a powerful numerical approach is conducted for the problem. But before it, the initial and boundary conditions of the problem should be well specified.

During the engine cranking and warming up, the main region of the piston which is exposed to the thermal effects is top of crown surface and other regions of the piston such the lateral surfaces, rings and skirt have not any considerable heat exchange with their surrounding surfaces. The values of instant gas temperature and instant coefficient of heat transfer above the piston top surface can be found according to the following equation [26]:

(3) $$h_g = 453.6 D^{-0.214} (C_m P_g)^{0.786} T_g^{-0.525}$$

Where h_g was defined as instant coefficient of heat transfer of gas at piston top surface, D diameter of cylinder, Cm average velocity of piston, P_g instant gas pressure and T_g instant gas temperature which could be obtained from following equations.

(4) $$T_g = \frac{P_g V_x}{mR}$$

(5) $$V_x = V_c + \pi D^2 S_x / 4$$

(6) $$S_x = S\left[1 - \cos\phi + \frac{1}{\lambda}\left(1 - \sqrt{1 - \lambda^2 \sin^2\varphi}\right)\right] / 2$$

Where V_x was defined as instant cylinder volume, V_c chamber volume, m gas mass in the cylinder, S_x piston displacement, and λ stands for ratio of crank radius to connecting rod length, S stroke of piston.

It is reasonable to estimate the gas temperature as the average temperature values during an engine cycle [27]. For the considered case study, the average temperature and convection coefficient on the top of piston crown were estimated to be 650 °C and 800 W/m^{2}K, respectively. These values are very closed to thermal condition of the analysis performed for a similar engine [28].

For other positions inside the chamber, the heat flux through the walls can be experimentally evaluated using quick-response thermocouples on certain positions located in close proximity to the surfaces [29]. However, the occurred heat transfer in all four strokes of a cycle varies from very high fluxes to low or even heat flow from the walls to the gas mixture in cylinder. In order to facilitate the analysis, it is assumed that the same boundary condition is considered for piston top surface and the combustion chamber walls.

As a widespread gasoline engine used in Iran transport industry, the investigated spark ignition engine is a fuel injection, 4 stroke, 4 cylinder, water-cooled engine and its details are given in Table 1.

The computational domain of the problem includes the intake and exhaust pipes and valves, cylinder head and piston (Fig 1). Each zone has been meshed separately. Hexahedral cells have been used for the grid generation because they yield better accuracy and stability than tetrahedral cells. However, due to the complexity of the pipe geometry near the valve, the tetrahedral cells have been used to mesh the intake and exhaust pipes. A commercial finite element package COSMOS, which has the capability of 3D heat transfer equation, was used for the thermal analysis.

As can be seen, great attention was paid for modeling the geometric details of engine components. The finite element mesh of the piston model is shown in Fig 1. 111202 nodes and 66717 elements to guaranty the accuracy and acceptability of the obtained results.

Table 1: Specification of the engine under study

Bore	9.2 cm
Stroke	8.9 cm
Compression ratio	8.7
Connecting rod	13.61 cm
IVO	34° bTDC
IVC	74° aBDC
EVO	36° bBDC
IP	0.83 bar
IT	297 K
λ	1.1
RPM	3000
Ignition time	55 bTDC

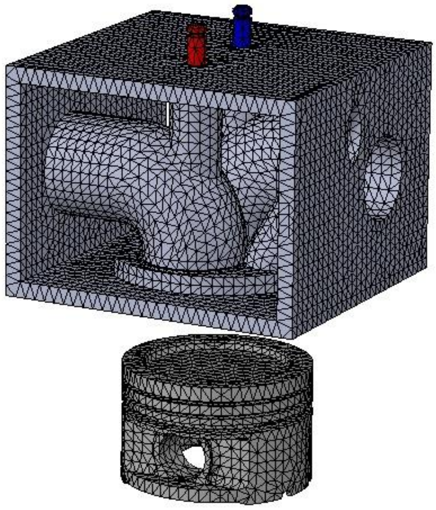

Figure 1: Finite element analysis of engine model prepared for 3D thermal analysis.

3) Results and discussion

As the air passes through the hot intake manifold, the temperature of air during the inlet stroke will generally be on the order of 25 °C to 35 °C hotter than the surrounding air temperature [30]. During the isentropic compression, not only is there an increase in pressure but the temperature within the cylinder is increased substantially due to compressive heating. Using the ideal gas relationship for the air- fuel mixture, the charge temperature immediately before the ignition can be readily found through the following relation:

(7) $$T_2 = T_1 \left(v_1 / v_2 \right)^{k-1} = T_1 \left(V_1 / V_2 \right)^{k-1} = T_1 \left(r_c \right)^{k-1}$$

(8) $$P_2 = P_1 \left(v_1 / v_2 \right)^{k} = P_1 \left(V_1 / V_2 \right)^{k} = P_1 \left(r_c \right)^{k}$$

Where k is the ratio of specific heats C_p/C_v and r_c is compression ratio. For modeling the performance of the engine during the cold start the ratio is set to be *k= 1.2* for normal condition of engine this value is *k= 1.35* [31].

On low temperature conditions, the temperature of the charge exactly before the ignition would be the most important to the cold start. So, the instantaneous heat exchange between the charge and chamber walls immediately before the ignition is investigated as transient thermal analysis in this study.

In current study, the ambient temperature during low temperature cold start is considered to be -5 °C. The idle speed of engine during cold start is 1000 *rpm*. Oil and water temperatures are fixed at -5 °C and set as boundary conditions for the calculations.

The calculations are performed for the state when both inlet and outlet valves are closed, the piston is near TDC and all the charge is enclosed with the holding chamber at the end of compression stroke. According to the timing of the inlet and outlet valves given in Table 1, it can be noted that at engine speed of 1000 *rpm* the compressed air-fuel mixture is in contact with piston crown and combustion chamber walls just for 0.005 sec before spark ignition.

For considered ambient temperature and for the engine that has reached to its normal condition, the gradients of temperature for piston valves and cylinder head are shown in Fig 2. As mentioned earlier, the instantaneous heat exchange with the engine walls determines the charge temperature before power stroke. For this purpose, another model which includes the compressed air is provided for a transient analysis (Fig 3).

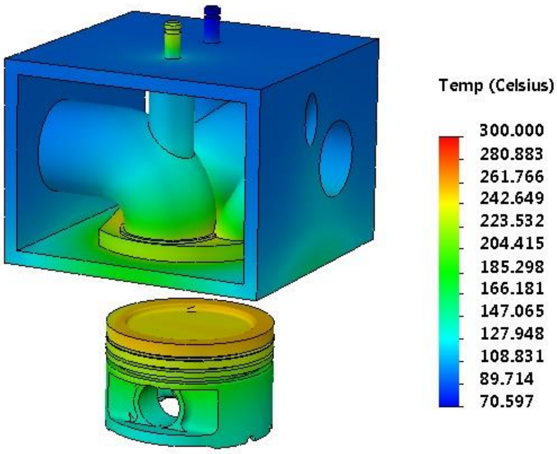

Figure 2: The temperature gradient of engine part for steady state condition.

During the isentropic compression, the temperature of air-fuel mixture increases from *268 K* to *418.63 K* according to equations 7 and 8. Then, after instantaneous heat exchange with surrounding walls, the compressed charge (before the ignition) has average temperature of 245 °C. This average temperature is obtained by importing the results of steady state analysis of engine parts into the transient model shown in Fig 3. So, it can be concluded that for optimum engine performance in ambient temperature -5 °C, the charge temperature should reach as fast as possible to this average value.

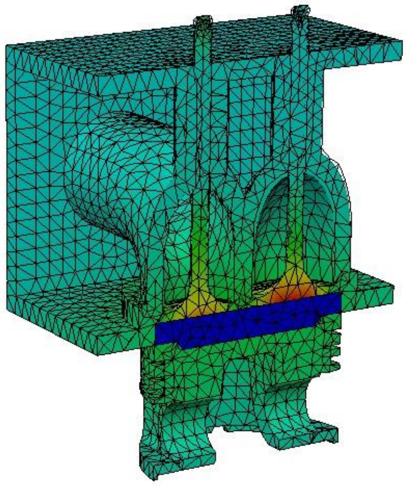

Figure 3: The model for evaluating the instantaneous transient heat exchange between engine parts and compressed air.

It is noted that for the first cranking cycles and before the steady state condition, the temperature of engine parts have a great variation with time. So, the compressed air-fuel mixture at each cycle of engine performance comes into contact with the walls with variable temperature. As the temperature of combustion chamber increases, the average temperature of charge will also increase which leads to better power generation and reduction in engine emissions [10].

For the engine equipped with conventional piston, the time rate of increase in charge temperature at first *2 min* performance of engine working on ambient temperature of -5 °C is shown in Fig 4. It can be observed that the charge temperature before the ignition is far below the needed value and engine will provide poor performance.

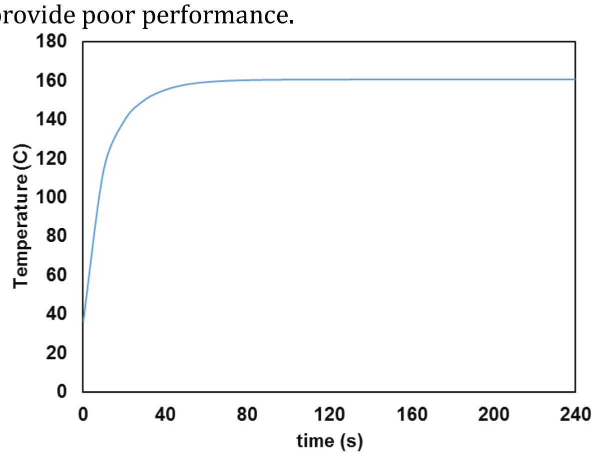

Figure 4: the time variation of air-fuel mixture temperature before the ignition for first 2 *min* performance of engine working on ambient temperature of -5 °C.

As moveable part of engine combustion chamber, the piston crown surface has a considerable effect on increasing the temperature of the charge before ignition. Therefore, increasing the temperature gradient of piston top surface can improve the engine cold start performance in low ambient temperature. It is well known that insulating the combustion chamber components of low heat rejection (LHR) engines can reduce the heat transfer between the gases in the cylinder and the cylinder wall and, consequently, increase the temperature of compressed intake air and leads to improve the combustion efficiency [32].
In the present study, the ceramic layer is considered to be deposited over the bowl lip areas of the crown surface. Choosing this area enables to decrease the risk of knock free performance for the spark ignition engine. As shown in Fig 5, the piston is coated with a zirconia-based ceramic layer over an inter-metallic alloy bond coat layer for providing better adhesion [33].

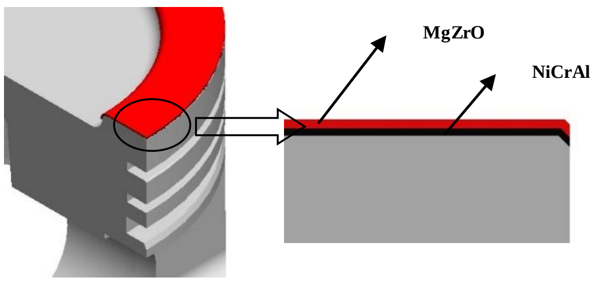

Figure 5: Thermal barrier coating thicknesses over the crown surface of a locally coated piston.

The bond and ceramic layers would be coated on the piston crown surface by using plasma spray coating technique. Thermally sprayed ceramic material has layered structures with a defect density resulting from successive impact of a multitude of fully or semi-molten particles. Material properties of the $MgZrO_3$, NiCrAl and piston material made of AlSi alloy are listed in Table 2.
Under the same boundary conditions, the temperature distributions of conventional and coated pistons are shown in Figs 6.
The thicknesses of ceramic coating layer are *0.2 mm*, *0.4 mm*, *0.6 mm* and *0.8 mm*, respectively. The bond layer has unique thickness of *0.15 mm* for all the cases. The temperatures at top surface of coated piston are found to be much higher than the temperatures obtained for the traditional piston.
For the coated pistons, the obtained temperature gradients are relatively similar to each other, and the bowl lip area of the coated pistons are found hottest region of the crown surface. However, because of low thermal conductivity of the ceramic layer, different temperatures were observed for each case.

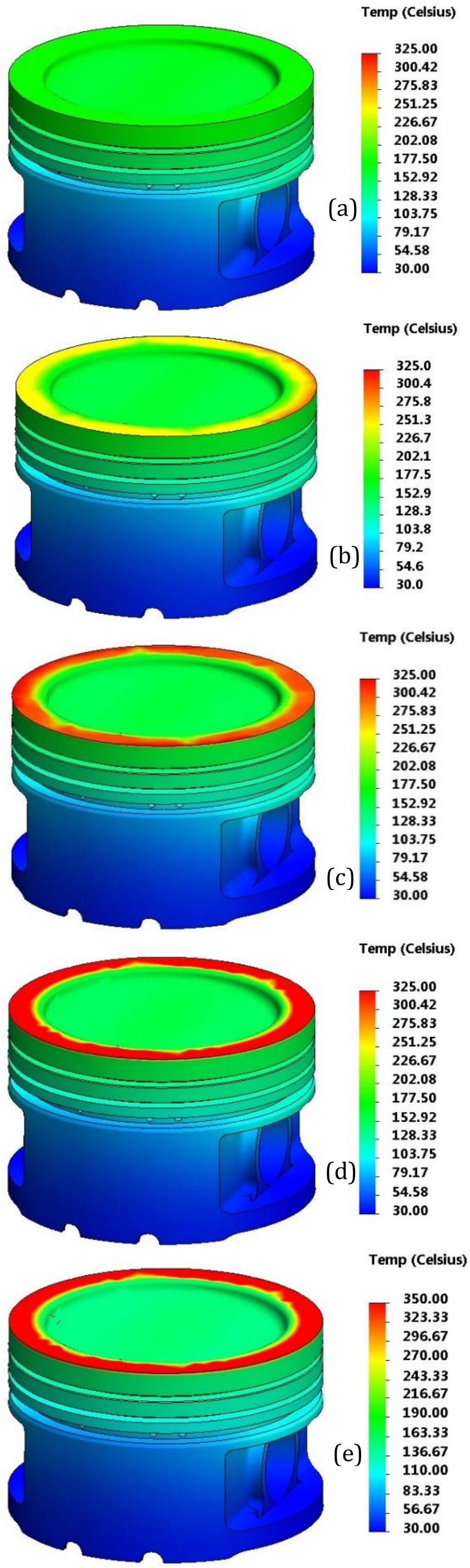

Figure 6: The effect of ceramic layer thickness on temperature distribution for conventional piston and pistons with coated top surface; the considered thicknesses are: a) 0 mm, b) 0.2 mm, c) 0.4 mm d) 0.6 mm and e) 0.8 mm, respectively

Table 2: Material properties of piston and ceramic [34].

	AlSi	NiCrAl	$MgZrO_3$
Thermal conductivity, [W/m °C]	155	16.1	0.8
Thermal expansion, 10^{-6} [1/°C]	21	12	8
Density, [kg/m^3]	2700	7870	5600
Specific heat, [J/kg°C]	960	764	650
Poisson's ratio	0.3	0.27	0.2
Young's Modulus , [GPa]	90	90	46

Fig. 7 shows the time variation of temperature distribution on top surface of conventional piston exposed to the compressed air-fuel mixture. For this purpose, the transient values of temperatures for some specified points which are diametrically distributed over top crown surface, are used for obtaining the plots. It can be observed that after 2 *min* performance of engine the maximum temperature of piston is slightly more than180 °C.
The same consideration is held for the coated pistons and results are given in Figs. 8. According to the diagrams, the temperature on bowl edge region has a very fast increase rate and in a short time reaches to its normal temperature (see Fig. 2). On the other hand, thicker ceramic layer can provide higher temperatures for the piston. So, it may have positive effect on engine cold start performance.

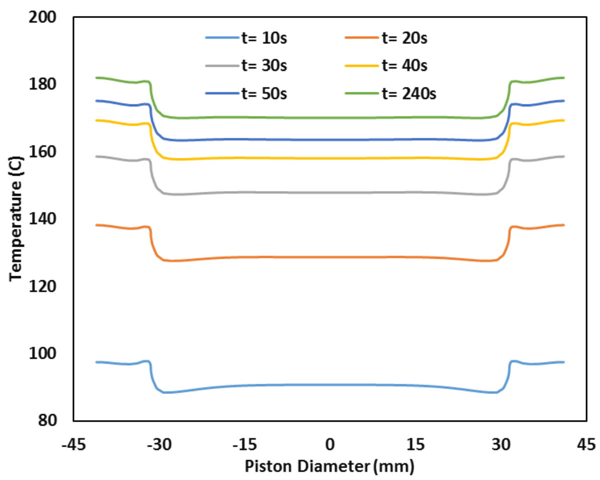

Figure 7: The time variation of temperature over the piston crown surface.

The effect of ceramic layer thickness on cycle to cycle increase in temperature of charge at the end of compression stroke is shown in Fig 9.

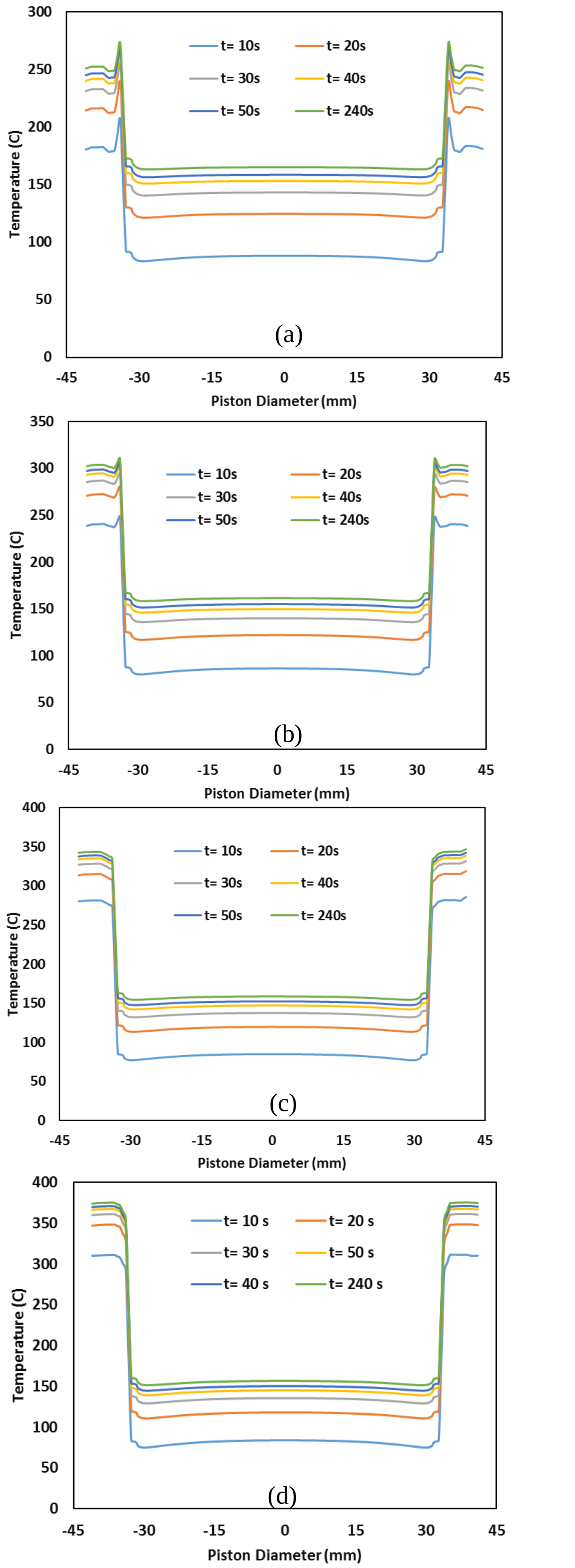

Figure 8: The time variation of temperature over the piston crown surface; the considered thicknesses are: a) 0.2 mm, b) 0.4 mm c) 0.6 mm and d) 0.8 mm, respectively.

It is noted that by using 0.2 *mm* ceramic coating layer over the crown surface of the engine piston, the charge temperature reaches to 66.67% of desired temperature for normal performance of the engine. This is 10.88% more than the value obtained for conventional engine.The use of ceramic layers with thickness of 0.4 *mm*, 0.6 *mm* and 0.8 *mm* increases the charge temperature by 68.75%, 71.67% and 72.5%, respectively. For the thicknesses more than *0.4 mm* the charge temperature have a fast approach to its normal value. So by depositing a coating layer with sufficient thickness, the engine can provide more output power and torque in a shorter period. However, no major contribution can be observed for the layers thicker than 0.6 *mm*. so, instead of vulnerable thick layers of ceramic layers, this thickness can be used as optimum choice for rapid increase in charge temperature and great improvement of engine cold start. start performance. This allows also less time for decreasing the ring-cylinder clearances and more effective compression in next cycles.

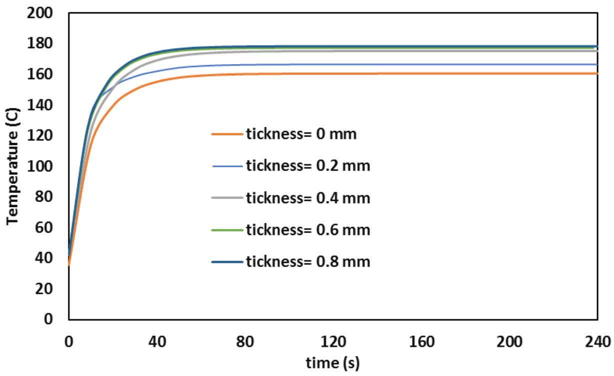

Figure 9: The time rate of increase in cycle to cycle temperature of charge before ignition.

4) Conclusion

In this work, an elaborated 3D thermal study was conducted for improving the performance of a specified gasoline engine at the beginning of cold start stage. By increasing the temperature of combustion chamber walls, the charge has higher temperature before ignition and more efficient combustion in shorter time is provided for the engine. For first 2 min operation of the engine in cold weather, the effect of ceramic layer thickness is evaluated on increasing the engine wall temperature. Cycle to cycle increase in temperature of charge at the end of compression is also investigated in this work. For the considered SI engine, a detailed model including the piston, cylinder-head and valves are constructed with high geometrical accuracy. The calculations are performed for the state when the charge is enclosed with the holding chamber at the end of compression stroke. Compared with the conventional uncoated engine, the pre-ignition temperature of the charge mixture can be increased 10.88%-16.94% depending on thickness of coating layer. According to the results, no major contribution can be observed for the layers thicker than 0.6 *mm*. so, this value is optimum choice for improvement of the engine cold start performance.

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**نویسندهٔ مسئول*

چکیده

در دماهای محیطی پایین به علت شرایط حرارتی و کیفیت نامناسب مخلوط سوخت و هوا، احتراق قادر به سوزاندن کامل مخلوط نمی باشد. به همین جهت گشتاور و توان خروجی کافی در موتور تولید نمی شود و موتور زمان راه اندازی طولانی تری لازم خواهد داشت ضمن اینکه آلودگی زیست محیطی قابل توجهی نیز تولید خواهد شد. با ایجاد پوشش سرامیکی روی دیواره تاج سنبه و افزایش دمای مخلوط قبل از جرقه زنی شمع، احتراق در مدت زمان کوتاه تری موتور را به شرایط معمول می رساند. در تحقیق حاضر، تحلیل حرارتی گذرا برای شبیه سازی عملکرد موتور در شرایط راه اندازی سرد ارائه شده است تا اثر ضخامت لایه عایق روی دمای مخلوط مورد بررسی قرار گیرد. نتایج به دست آمده به خوبی موید این واقعیت است که پوشش حرارتی تأثیر بسزایی در بهبود شرایط موتور در حالت راه اندازی سرد دارد.

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