Computational study of effect of Ceramic coatings and piston bowl depth on heat fluxes of Re-entrant type piston

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Abstract. The current research article investigates the variation of heat flux with geometric modification on the re-entrant type piston bowl by varying bowl depth by computational domain. The contribution of ceramic coating on heat flux is also investigated in this research report. Computational study was carried out by using ANSYS 19.2 software in order to obtain the augmented results. The variation of piston bowl depth of Re-entrant type piston is accomplished as there is a novel need to study the variation in heat flux of piston at different bowl depths. The impact of the thickness of the ceramic coating is also evaluated. The focus of this article is to study the effects of combustion on the piston of a compression ignition engine. It was noticed that with the decrease of bowl depth and addition of ceramic coating there is a decrease in heat flux. Around 27.75% depreciation in the heat flux was noticed for the uncoated piston with a bowl depth of 15mm in comparison with 27 mm bowl depth and a 25% decrement with the addition of a ceramic coating of 0.4 mm and also a 55% decrement in the heat flux was observed with 1 mm coating to the piston with 15 mm bowl depth in reference to 27 mm bowl depth of respective coating thickness. Disparity of maximum heat flux and heat losses with increasing bowl depth is also analyzed in the inquiry.

Keywords: Re-entrant type piston, Thermal barrier coating, Heat flux, Piston bowl depth

1. Introduction

Enhanced torque production and high load carrying capability is the noteworthy benefit of diesel engines. The shape of the combustion chambers play a vital role in the comprehensive mixing of air and fuel in such engines. The variation of the piston bowl profile is the renowned factor in order to obtain optimal engine performance and emissions [1]. Squish, swirl and Turbulence are highly influenced by piston crown geometry and the intake system [2]. Piston bowl geometry has a greater influence on intensified squish near Top Dead Centre. Thus, the attainment of a thorough combustion process is effected by piston bowl shape [3]. To acquire complete and uniform mixing, there is always a potential need to change the shape of the piston bowl contour [4]. Donateo et.al also showed that the process of changing the shape of the piston is cost-effective [5].

The present work involves the usage of re-entrant type piston crown geometry with chamfer. Comparatively lower emission rates are the conspicuous advantage of Re-entrant type piston [6]. Narrow throat of re-entrant type piston imparts potent squish and reverse squish flow thereby augment the air-fuel mixing [7]. Foremost reason for lesser emissions is that there is a remarkable increment in squish, swirl and Turbulence Kinetic Energy during compression stroke as a result of symmetric vortices of Re-entrant piston bowl [8-10]. The
optimum re-entrant angle is prominent in having decremental HC, CO and smoke emissions [11, 12]. Mamilla et al. concluded that there is an enhanced brake thermal efficiency and peak pressure heat release rate with re-entrant piston [13]. Chamfering is basically done to split the fuel into two different portions. One of the portions goes to the depth of the bowl and works similarly as that of the re-entrant bowl and fuel in the chamfered region utilizes surface air for comprehensive combustion. Rapid air-fuel mixing is the utmost motto in the utilization of a chamfered bowl [14]. The prime limitation of re-entrant piston is that the increase in the re-entrant angle and complexity of crown geometry results in diminished thermal stability of the entire structure. The inquiry of the after-effects of combustion and thermal stability of re-entrant piston is the prime focus of the current investigation. Therefore, to inflate the thermal stability there is a potential need to study the consequence of coatings on the piston crown.

Thermal barrier coatings (TBCs) are generally coated on any material to degrade heat rejection [15]. Enhanced power density, fuel efficiency increment and better combustion characteristics leading to multi-fuel capacity is the pronounced supremacy of TBCs in diesel engines [16, 17]. Ahmanemi et al. noticed that TBCs improved the engine power by 8% and there was a notable decrement in specific fuel consumption by 15-20% [18]. The current work puts forward the study of the effect of ceramic D8 coatings on the piston crown. Jalaludin et al. in one of their research articles concluded that lower conductivity of ceramic coating helped in a 98% reduction in heat flux when compared to uncoated piston [19]. Elevated thermal efficiency and declined emissions are observed by Sethuraman et al. by coating creaming coatings on piston crown [20].

In this research article computational study was carried out by using ANSYS 19.2 commercially available software. The entire practical conditions are incarnated in the computational domain. The chief reason for opting for a computational study is to clearly study the scenario of piston bowl depth variation and to obtain the augmented results.

2. Experimentation

Fig. 1 depicts the entire workflow of the current research investigation. Initially, the model was created as the per the dimensions. Later the entire simulations were carried out using ANSYS 19.2 general purpose software.
2.1 Creating model

General purpose package software CATIA V5 has been employed for modelling chamfered re-entrant type piston. The designed piston can be utilized in a diesel engine. The dimensions of the piston inculcated in the present inquiry have been adopted as mentioned by Muhammet Cerit and Mehmet Coban [21]. The initial 3D model of the designed piston is as portrayed in Fig. 2.

![3D model of the Piston](image)

Figure 2. 3D model of the Piston

The main focus of this analysis is to illustrate the effect of variation of piston bowl depth on the thermal stability of the piston. The initial depth of the piston bowl was 27 mm. The depth variations have been done in three iterations namely 25 mm, 20 mm and 15 mm. Chamfer radius of 5 mm and the length of 160 mm were kept constant in each of these cases as represented in Fig. 3.

![Variations in piston bowl depth](image)

a) 15 mm  
b) 20 mm
2.2 Generating mesh
Meshing plays a vital role in obtaining accurate results. The entire piston has been discretized into nodes and elements. Table 1 shows the mesh details of coated and uncoated Re-entrant type piston with 27 mm bowl depth.

| Mesh details                  | Uncoated piston | Coated piston |
|-------------------------------|-----------------|---------------|
| Number of elements            | 165057          | 181262        |
| Number of nodes               | 265130          | 298376        |
| Average element size (mm)     | 1               | 1             |

Bob et al. scrutinized the intensive effect of the shape of nodes in a mesh in obtaining augmented results [22]. There are typically two kinds of meshes namely structured and unstructured meshes. Unstructured triangular meshes are currently a part of this study. Massive parallel computations can be done comprehensively by such kind of meshes [23]. Unconditionally well adaptable way of unstructured mesh, when compared with structured mesh with complex geometry, makes it more predominant in computational fluid dynamics [24]. Less computational time and during the process of adaptive mesh refinement, no moderation is required in terms of the basic algorithm, the mesh always remains as unstructured triangular mesh.

Stable and accurate results are also one of the major criteria for implementing triangular meshes. Also, second-order accuracy can be obtained more effortlessly [25, 26]. Fig. 4 renders the completely meshed uncoated chamfered piston.

The mesh details of the coated piston crown is also shown. A coating of 1mm thickness and 0.4 mm thickness has been created with ANSYS 19.2 software. As expected with the coating, the number of cells...
increased. Fig. 4b shows this trait. Table 1 clearly portrays that there is a significant increment with the number of cells.

![Coated vs Uncoated Meshes](image)

**Figure 4.** Mesh for coated and uncoated pistons

2.2.1 Mesh independent test

Initially, mesh independent test was done on uncoated re-entrant type piston with 27 mm bowl depth with the maximum temperature being 1200 K. To account for the independency of the mesh, two distinct mesh sizes namely medium mesh and fine mesh were utilized that was available in Ansys 19.2 software which is depicted in Fig. 5. From the outcomes, it can be concluded that there is no significant variation in the obtained results. The computational time was very high in solving fine mesh grids when compared to a medium-mesh. Drastic abatement in computational time in getting results of medium meshes is the prime reason for their usage. The procedure utilized by Yang et al. is followed in this study [27].

![Medium vs Fine Meshes](image)

**Figure 5.** Meshes used to conduct mesh independent test
2.2.2 Material properties

The current investigation utilizes a cast alloy as a piston material in the computational domain, having a density of 2680 kg/m$^3$ and Young’s modulus being 73 GPa. It possesses a thermal conductivity of 152 W/m$^2$ °C and has a specific heat of 963 J/kg °C. Ceramic 8D coating is used on the piston crown region with 0.4mm and 1mm thickness respectively. Basically, ceramic coatings have high Knoop hardness. The coating used in the present work is encompassed with a density of 4900 kg/m$^3$ and has an isotropic thermal conductivity of 4.5 W/m°C.

2.2.3 Boundary conditions

The boundary conditions on various parts of the piston are adapted from the work done by Kumarasekaran and Safdari [28]. Fig. 6 depicts the various boundary conditions. The maximum temperature was 1200 K and it is found on the piston crown region. quadrant of the piston design is used to portray the entire boundary conditions.

![Image of various boundary conditions on the piston]

**Figure 6.** Various boundary conditions on the piston

2.3 Transient thermal analysis

Transient thermal Analysis of piston was showcased in the present work by general-purpose ANSYS 19.2 software. The main motive for employing transient thermal analysis is to obtain the changes in temperature and heat flux profiles with reference to time. As it was noticed that temperature changes drastically with time in an IC engine, it was imperative to use transient analysis. In any practical application, the amount of heat entering the piston is not always equal to the amount of heat convected and conducted through the piston rings to air or any other coolant.

In order to realize the distribution of temperature and variation of heat flow rate across the domain, solution for fundamental equations like heat conduction, contact equation and convection equation has to be delineated. The equations for transient thermal analysis is as represented by Subodh et al. [29].

The general heat conduction equation is given by,

$$K \nabla^2 T + q_e - \rho C_{p,\text{air}} \frac{dT}{dt} = 0$$

(1)
where \( K \) stands for thermal conductivity, \( q_e \) represents internal heat generation, \( \rho \) represents the density of the material. \( C \) stands for specific heat.

Conductive boundary representation by variational formulation is shown by

\[
\frac{\partial \phi}{\partial \theta} = [k_{ij}] \{t\} = [k]\{e\} \{t\}^e \tag{2}
\]

\([k]\) = Global stiffness matrix

The global matrix is obtained as mentioned by Samria et al [30]

The governing equation for the contact boundary can be portrayed as

\[ q_c = h_c (T^e - T^p) \tag{3} \]

The variational formulation for contact boundary between two elements is noted as

\[
\chi_{bcont.} = \frac{h_s}{2} \int_0^{\pi} r_s \frac{t^p_s - t^e_s}{2\pi r} dr \tag{4}
\]

Upon differentiation of the equation (4) with respect to surface yield Temperature, a linear equation is obtained as rendered in equation (5) which is the part global set of equations

\[
\frac{\partial \phi_{bcont}}{\partial \theta} = 2h_s r \frac{t^p_s - t^e_s}{2\pi r} \frac{1}{2} \{t^e_s - t^p_s\} \tag{5}
\]

The general heat convection equation is given by

\[
-K \frac{\partial T}{\partial n} = h \left( T^e - T^p \right) \tag{6}
\]

Variation formulation is represented by,

\[
\frac{\partial \phi_{bconv}}{\partial \theta} = 2h_m r \frac{t^p_m - t^e_m}{2\pi r} \frac{1}{2} \{t^e_m - t^p_m\} \tag{7}
\]

Here, \( r \) = \( r_{ij} \), \( r_{ij} = r - r_{ij} \), \( r_{ij} = r_{ij} \), \( e = e_{ij} \), \( \epsilon = \epsilon_{ij} \), \( \frac{\partial \phi_{bconv}}{\partial \theta} = H \{t\} \{t\}^\infty \tag{8} \]

Thus, a combination of variation integrals of the heat transfer equation is

\[
([K] - [H])\{T\} = \{h_f\} \tag{9}
\]
where \([K]=\) conduction matrix, \([H]=\) convection matrix, \([T]=\) unknown temperature at each nodal point, \(h_f\) = column vector with a known quantity. The global matrices introduced above are of the order of \((N \times N)\) where \(N\) stands for the number of nodes.

Equation (10) is solved at every nodal point to attain the desired solution.

3. Results and discussion

3.1 Validation

The present work has been numerically validated. The general heat conduction equation has been solved to obtain augmented values of heat flux. The values obtained for heat flux numerically is compared with the results delivered from the computational domain. The volumetric expansion accounting for the volumetric deformation is also solved numerically. Equation (11) depicts the co-relation for obtaining volumetric expansion.

\[
K \nabla^2 T + q_e - \frac{\partial T}{\partial t} = 0
\]  

(11)  

\[
\Delta V = 3\alpha V_0 \Delta T
\]  

(12)  

where \(\alpha\) is the coefficient of thermal expansion and \(V_0\) is the initial volume.

The results obtained in the computational domain are in good agreement with those obtained mathematically.

3.2 Impact of heat flux on piston crown without coating

The variation of heat flux is more pronounced in the piston bowl region as portrayed by the Fig. 7. So the crown region is more intensely studied in this section. From these figures, it can be noticed that the red region i.e. the region with maximum heat flux occupies the majority of surface area within the piston crown. Main cause for the present characteristic is that the crown region is exposed to higher combustion temperature as the entire process of combustion is incarnated within the crown region. It can also be perceived that the maximum temperature of the entire combustion cycle is taken up by the piston crown. In the case of a 15mm depth piston bowl, the heat flux value inflated to around 13395000 W/m². There is an unpredictable variation of maximum heat flux with the increment in the bowl depth. At 27 mm there is a comprehensive enhancement of heat flux to 14880000 W/m². With the bowl depth augmentation, there is an accrual of maximum heat flux. From the images, it can be observed that as the depth increases there is a conspicuous narrowing of the region where increased heat flux is found. It can be stated that as there is a narrowing of the red region, a considerable decrement in its area can be observed. This can be the prime reason for the escalation of maximal heat flux with bowl depth. It can be discerned that both increases in maximum heat flux and narrowing of the area is very less as outlined by Fig. 6. Maximum heat flux variation with respect to depth is as illustrated in Table 2.
Table 2. Variation of maximum heat flux with respect to bowl depth

| Bowl depth (mm) | Max heat flux (W/m²) |
|----------------|----------------------|
| 15             | 13395000             |
| 20             | 14254000             |
| 25             | 14801000             |
| 27             | 14880000             |

Figure 7. Variation of heat flux in different bowl depths

3.2.1 Expansion behaviour of piston with varying depth

The presence of heat is the supreme motive for the expansion of the piston. Analysis of volumetric deformation in reference to piston bowl depth is encompassed in the Fig. 7. The outcome renders that with the increase the bowl depth, volume expansion is decreased. With the inflated bowl depth, there is an enhanced surface area. Hence, the area available for heat flow is increased. This might be the probable reason for diminished volumetric expansion with bowl depth increment. Fig. 7 clearly depicts the trend of
decrement. These data show that there is a negligible volumetric deformation. This depicts the fact that irrespective of bowl depth, geometry as a whole has a stable configuration and has the capability to withstand the fluctuating temperature.

Figure 8. Variation of volumetric deformation with bowl depth

3.3 Effect of ceramic coating on variation of heat flux

Ceramic coatings are applied on the piston crown in 2 iterations of 0.4 mm and 1mm. A significant decrement in heat flux is the major insight with coating the piston crown region. Substantial reduction of heat flux around 20-25% with 0.4 mm coating thickness and around 50-55% with 1 mm coating thickness is realized. These can be accounted for the fact that most of the heat on the crown region has been absorbed by the ceramic coating. It is a known fact that heat flux is a product of heat transfer coefficient and the difference of temperature between the coating and the modelling surface. There is a drop in the heat flux with inflating surface temperature. This mechanism of ceramic coatings play a vital role in its insulation behaviour. It is evident that the ceramic coatings have effectively acted as a thermal barrier. Fig. 8 shows the heat flux feature on 27 mm bowl depth with 0mm, 0.4mm and 1mm coating. It clearly depicts that the awful diminishing of heat flux is the function of the thickness of the ceramic coating. With no coating, a larger variation of heat flux is encompassed in the crown region. This fluctuation is chiefly due to the more complex geometry of the piston crown. With the coating, extremely high heat is resisted due to the high insulating capability of the coating. This in turn, would enhance the heat flow rate within the combustion chamber thereby raise the combustion efficiency and fuel efficiency of an engine.

3.3.1 Thickness effect on heat flux

Comprehensive heat flux decrement with inflating thickness can be accounted for from the Fig. 9. There is a clear trend of increasing average heat flux with bowl depth in the absence of thermal barrier coating. With the incremental coating thickness, there is a substantial decrement in heat flux. More the thickness of the coating, more is the heat being retained in the combustion chamber and better the entire process of combustion. 0.4 mm thickness shows the trend of increment of heat flux with increasing bowl depth. The trend remains unchanged even for 1 mm coating thickness. It can be noticed at 0.4mm and 1mm thickness with respect to uncoated piston has a decrement of heat flux. There is a percentage depreciation of 27.29%, 21.60%, 20.87% and 20.30% for 15 mm, 20 mm, 25 mm and 27 mm bowl depth respectively. The
percentage depletion of 56.22%, 56.30%, 55.84% and 55.53% can be seen for 15 mm, 20 mm, 25 mm and 27 mm bowl depth respectively in reference to the uncoated piston. Noteworthy outcome is that for 1mm coating thickness there is a substantial decrement of heat flux in comparison of 0.4 mm thickness. Enhanced Knoop hardness with incremental thickness is the major reason for its stability [31]. The entire thermal load, as well as the cooling load on the piston, is reduced by the increased TBC thickness [32].

**Figure 9.** Variation of heat flux on 27 mm bowl depth with varied thickness of ceramic coating

**Figure 10.** Behaviour of heat flux with piston bowl depth
For longer life of an engine usage of agreemental thickness of TBC is recommended. Lesser the thickness, a larger thermal gradient would be created between the coating and crown which might reduce the life of the piston [33]. Thus, a significant decrement in heat flux with coating thickness would uplift the thermal stability of an engine.

### 3.4 Surface to volume ratio effect

Surface to volume ratio effect on heat flux has been studied briefly in this work. The values of surface to volume in reference to piston bowl depth for various coating thickness are as shown in Table 3.

| Coating Thickness (mm) | Piston bowl depth (mm) | Surface to volume ratio (m⁻¹) |
|------------------------|------------------------|-----------------------------|
| 0.00                   | 15                     | 153.94                      |
| 0.00                   | 20                     | 157.61                      |
| 0.00                   | 25                     | 162.26                      |
| 0.00                   | 27                     | 164.98                      |
| 0.40                   | 15                     | 182.59                      |
| 0.40                   | 20                     | 189.08                      |
| 0.40                   | 25                     | 197.53                      |
| 0.40                   | 27                     | 201.50                      |
| 1.00                   | 15                     | 181.80                      |
| 1.00                   | 20                     | 188.14                      |
| 1.00                   | 25                     | 196.40                      |
| 1.00                   | 27                     | 200.28                      |

There is an increment in surface to volume ratio with the increasing bowl depth as expected. This trend remains unchanged with increasing coating thickness. The trait of heat flux variation with the surface to volume ratio is portrayed by the Figs. 11, 12, and 13. The plots clearly show that with the increment of the surface to volume ratio there is an inflating heat flux and the raise in heat flux occurs linearly. The average heat flux is enhanced by 5.26%, 10.56% and 12.28% in an uncoated piston crown. At 0.4mm coating the inflation is 5.04%, 11.36%, 13.90%. Increment of 5.05%, 11.51% and 14.05% is seen with 1mm coating thickness. The significant heat flux increase shows that heat losses are also increased with increased surface to volume ratio. These results are in good agreement with the outcomes of Helgi et al. in one of their research articles [34]. Amir et al. also noticed that with a drastic enhancement of the heat flux with the surface to volume ratio, there is a pronounced inflation of heat losses [35]. These heat losses are not at all permissible in an engine which in turn may reduce the engine efficiency. Thus, it can be stated that the lesser the surface to volume ratio lesser is the heat losses. Thermal stability is higher with lesser surface to volume ratio. These results also show lesser the surface to volume ratio lesser is the bowl depth. This can be correlated to the fact that lesser bowl depth has low heat flux and in turn significantly lesser heat losses. The piston bowl with lesser depth has supreme thermal stability when compared to pistons with increased bowl depth.
**Figure 11.** Average heat flux vs surface to volume ratio (without coating)

**Figure 12.** Average heat flux vs surface to volume ratio (With 0.4mm coating thickness)

**Figure 13.** Average heat flux vs surface to volume ratio (With 1mm coating thickness)
4. Conclusions

- The narrowing of the region with maximum heat flux can be accounted for the inflated maximum heat flux with increasing bowl depth.
- Principal heat flux variation is majorly seen on the piston crown region, as this is more prone to the combustion process. Further, the decrement in the piston bowl depth, the average heat flux is decreased.
- Around 27.75% decrement of heat flux is seen for 15 mm bowl depth when compared to 27 mm bowl depth.
- A comprehensive decrement of heat flux can be portrayed in the presence of coating. Further, with the enhanced coating thickness, there is a conspicuous decrement in the heat flux.
- Coating thickness of 0.4 mm showed a pronounced diminishing of heat flux by 20-25%. More than 50-55% heat flux decrement can be observed in the presence of 1mm thickness coating.
- With a lesser thickness, when subjected to longer duration of work, a larger temperature gradient may be created which would result in the depreciation of engine life.
- Surface to volume ratio increases with the increase in the bowl depth and linear increment of heat flux can be seen with increasing bowl depth.
- Lesser is the surface to volume ratio lesser the heat losses. Greater the bowl depth more is the heat flux and more is the heat losses. Thus, the piston with a lesser surface to volume ratio and increased ceramic coating (15 mm bowl depth with 1mm coating) can be potentially stable in transient thermal conditions.

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