International Communications in Heat and Mass Transfer 38 (2011) 1080–1085

Contents lists available at ScienceDirect

International Communications in Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ichmt

# Time resolved numerical modeling of oil jet cooling of a medium duty diesel engine piston☆

Avinash Kumar Agarwal *, Sandeep Kumar Goyal, Dhananjay Kumar Srivastava

*Engine Research Laboratory, Department of Mechanical Engineering, Indian Institute of Technology Kanpur, Kanpur 208016, India*

ARTICLE INFO

Available online 19 May 2011

*Keywords:*
Oil jet cooling
Thermal management
Internal combustion engines
CFD analysis
Oil mist

ABSTRACT

In medium to heavy duty diesel engines, ever increasing power densities are threatening piston's structural integrity at high engine loads and speeds. This investigation presents the computational results of the heat transfer between piston and an impinging oil jet, typically used to keep the pistons cool. Appropriate boundary conditions are applied and using numerical modeling, heat transfer coefficient (h) at the underside of the piston is predicted. This predicted value of heat transfer coefficient significantly helps in selecting right oil (essentially right oil grade), oil jet velocity, nozzle diameter (essentially nozzle design) and distance of the nozzle from the underside of the piston. It also predicts whether the selected grade of oil will contribute to oil fumes/mist generation. Using numerical simulation (finite element method), transient temperature profiles are evaluated for varying heat flux (simulating varying engine loads) to demonstrate the effect of oil jet cooling. The model, after experimental validation, has been used to understand the transient temperature behavior of the piston and the time taken in achieving steady state. High speed CCD camera is used to investigate the oil jet breakup, localized pool boiling and mist generation due to impinging jet on the piston's underside.



## 1. Introduction

The population of diesel fuelled vehicles is growing rapidly. This is possible due to the tremendous progress achieved in diesel engine technology in the areas of power, dynamics and ride comfort of the vehicles during the last two decades. Today's diesel vehicles play a vital role in reduction of fleet fuel consumption, leading to a significant decrease in greenhouse gas emissions. The increased number of diesel vehicle registrations however also has an effect on NOx and particulate emissions. The development of more efficient and powerful internal combustion engines requires the use of new and advanced technologies. These advanced engine technologies and emission requirements for meeting very stringent global emission norms have increased the power density range of the contemporary engines. Increase in power density causes an increase in operating temperature of engine components, especially in the combustion chamber. In the combustion chamber, cylinder head and liner are normally cooled using engine coolant however the piston is not cooled, making it susceptible to disintegration/thermal damage due to prolonged heating, especially at higher engine loads/speeds. Material constraints restrict the increase in thermal loading of piston. High piston temperatures may lead to engine seizure because of piston warping. The temperature of critical areas in the piston therefore needs to be kept below the material design limit. In most of the engines, pistons are made of aluminum alloy. Aluminum alloy begins to melt/loose its structural strength at temperature beyond 775 K. It is therefore important to determine the piston temperature profile so that the thermal stresses and deformations can be controlled and could be kept within the prescribed limits. This information can be helpful in designing the piston appropriately. Piston cooling also reduces the chances of carbon deposition on the piston crown. Carbonization of piston crown leads to formation of hot spots, which may cause pre-ignition of the combustible gasses (especially in SI engines), which is an undesirable combustion phenomenon. Piston cooling also has significant effect on tail-pipe emissions. It has been found that an increase in piston temperature (from 189 to 227 °C) leads to significant reduction in unburned hydrocarbon emissions, increase in smoke opacity with no change in the emission of oxides of nitrogen [1].

In modern medium/heavy duty engines, pistons are cooled by oil jet impingement from the underside of the piston. The associated high heat transfer rate is due to the stagnating mass that impacts hot impingement surface at high speed. However, if the temperature at the underside of the piston, where the oil jet strikes the piston, is above the boiling point of the oil, it may contribute to oil mist and smoke generation. This mist significantly contributes to non-tail pipe emissions (non-point source emissions) in the form of unburnt hydrocarbons (UBHC's). Another disadvantage of oil jet cooling of piston is that since lubricating oil comes in direct contact with very hot piston surface, temperature of lubricating oil increases. This results in

☆ Communicated by W.J. Minkowycz.

* Corresponding author.
*E-mail address:* akag@iitk.ac.in (A.K. Agarwal).

0735-1933/$ – see front matter © 2011 Elsevier Ltd. All rights reserved.
doi:10.1016/j.icheatmasstransfer.2011.05.006

deterioration of physical and chemical properties of lubricating oil and the efficiency of lubrication in an engine. This may also reduce the residual useful life of the lubricating oil.

There is a wide range of temperatures, pressures and heat flux encountered in an internal combustion engine depending on load and speed conditions of the engine operation. The value of local transient heat flux varies by an order of magnitude depending on the spatial location in the combustion chamber and the crank angle position. When an engine is running in a steady state, the heat transfer throughout most of the engine structure is steady. The heat flux increases with increasing engine load and speed. The maximum heat flux in the engine components occurs at wide open throttle and maximum speed. The heat flux is highest in the cylinder head, exhaust valve, valve seat, and center of the piston. The piston and valves are difficult to cool since they are always in dynamic state and are exposed to very high temperatures. The temperatures of the piston and valves depend on their thermal conductivity. Higher thermal conductivity leads to lower surface temperatures.

The pistons of engines can be cooled either by oil, water or air. Air cooling is simpler from design point of view, but lower specific heat per unit volume of air requires very large quantities of air to be directed towards the piston. This involves bulky ducting arrangement and an additional air compressor, which makes it less practical. Water cooling was applied to heavy, low speed engines for some time; but later it was abandoned because of serious design and maintenance difficulties with piping and sealing. However, this type of cooling has merit because water has significantly higher specific heat and lower viscosity than oil leading to higher heat transfer and effective piston cooling. Oil jet piston cooling is another way to cool the piston. In this method, the lubricating oil drawn from the oil sump is released at high pressure in the form of an oil jet ensuing from a nozzle mounted on the cylinder block and the nozzle is directed towards the underside of the piston. The oil jet splashes the oil on to the underside surfaces of the piston, thus removing the heat from the piston and effectively cooling it.

Significant amount of information is available in open literature [2–4] on heat transfer coefficient under impinging jets, which are widely used for variety of heating and cooling applications. Martin [5] and Jambunathan et al. [6] conducted a thorough review of heat transfer research related to impinging jets for such applications. Most of the research referenced was performed using a single jet or a group of jets impinging on a flat plate. Correlations were presented for average Nusselt number (Nu). The research also referred to the effect of jet interaction, jet angle, and Nusselt number distribution. Steven and Webb [7,8] experimentally investigated the effect of jet inclination on the local heat transfer coefficient on an obliquely impinging, round, free liquid jet striking a constant heat flux surface. The problem parameters investigated were jet Reynolds number in the range 6600–52000 and jet inclination ranging from 40° to 90°, measured from the horizontal. Experiments were carried out for nozzle sizes, d = 4.6 and 9.3 mm. It was found that the point of maximum heat transfer along the x-axis (the line of intersection of the jet inclination plane with the impingement surface) is shifted upstream (with respect to the jet flow) as a function of jet inclination with a maximum observed shift of 0.5 times nozzle diameter. In addition, it was found that the shape of local Nusselt number profile along the x-axis changed as the jet was inclined. One of the changes was sharpening of the peak in the profile at the point of maximum heat transfer. Another change was an increasing asymmetry around the point of maximum heat transfer with the upstream side of the profile dropping off more rapidly than the downstream side.

The objective of this paper is to develop a computational model for oil jet cooling of an actual production grade piston to predict the time resolved temperature distribution of piston and Investigating the conditions under which the oil jet cooling of the piston starts contributing significantly towards the non-tail pipe emissions through mist generation. The results obtained from the computational model are validated by performing experiments of the oil jet cooling of the actual piston using a thermal imaging camera. This model is used for predicting the time resolved temperature profile of the piston. In the end, experiments are conducted to show the generation of oil mist/smoke from the oil jet cooling at different piston surface temperatures.

## 2. Computational model development and validation

In order to understand the oil jet cooling of automotive pistons, digitization of piston geometry needs to be done. Then general heat transfer equation in cylindrical coordinates with appropriate boundary conditions is applied. Some of the boundary conditions are taken directly from the real-time problems. This is because they have time dependent factors hence a transient model has been developed in the present study. Piston is taken as axisymmetric. A Finite element analysis of the governing differential equation has been developed for given geometry using "Ansys" software. The coordinate system and the notations used in this computational model are given in Fig. 1.

The governing differential equation for the piston in cylindrical coordinates is given by Eq. (1).

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = \frac{1}{\alpha}\frac{\partial T}{\partial t}. \tag{1}$$

Eq. (1) is derived from the fundamental heat transfer equation by considering following assumptions.

a) The two dimensional governing differential equation is taken, as from physical and geometrical considerations, the flat plate is axisymmetric i.e. $\frac{\partial T}{\partial \phi} = 0$ where $\phi$ is the azimuth angle.
b) Material is considered to be homogeneous and isotropic.

The necessary boundary conditions are the temperature and heat transfer coefficient of the medium in contact with the piston surfaces. There are four boundary conditions and one initial condition for cooling of piston:

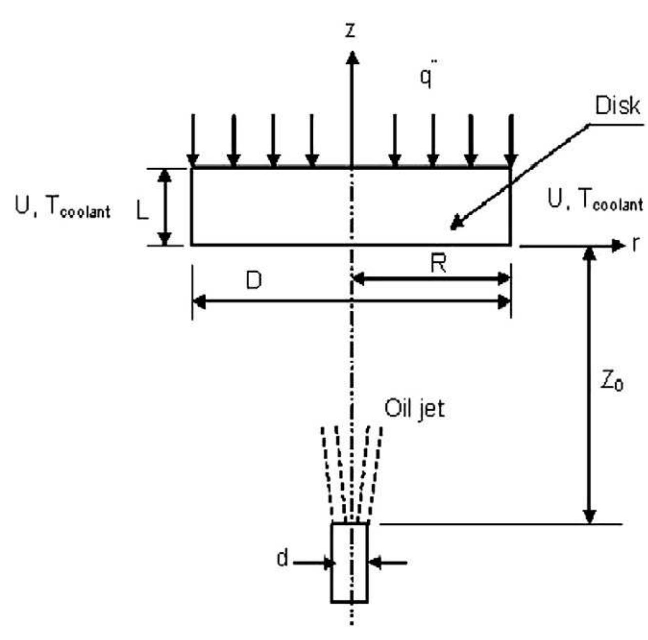


**Fig. 1.** Coordinate system and notations used for oil jet cooling [9,10].

At the starting point of the test, the piston is at ambient temperature hence the initial condition for piston is

At time $t = 0, T = T_{ambient}$.

1. The top surface (crown) of the piston is in contact with the hot combustion gasses, i.e.

$$+k\frac{\partial T}{\partial z} = q'' \tag{2a}$$

2. The sliding lubricated surfaces of the piston are in contact with the liner and the rings and participate in heat transfer to the coolant, i.e.

$$-k\frac{\partial T}{\partial r} = U(T - T_{Coolant}). \tag{2b}$$

Where $T_{coolant} = f(t)$.

3. The under side of the piston is exposed either to the crankcase atmosphere or to the cooling oil jet, i.e.

$$+k\frac{\partial T}{\partial z} = h\left(T - T_{surrounding}\right) \tag{2c}$$

In absence of the oil jet cooling $T_{surrounding} = T_{crankcase}$. If oil jet cooling is being used then $T_{surrounding} = T_{oil\ jet}$. In case of oil jet cooling, h is convective heat transfer coefficient between piston surface and cooling oil jet. In absence of oil jet cooling h represents the convective heat transfer coefficient between piston surface and air in the crankcase.

4. From the physical and geometrical considerations, only half portion of the axisymmetric piston is taken for analysis, i.e. we can assume insulated conditions for the straight edge of the half cut piston.

$$\frac{\partial T}{\partial r} = 0 \tag{2d}$$

The localized heat transfer coefficient at the point of jet impingement was calculated from correlation given by Stevens and Webb for axisymmetric, single-phase free round liquid jets impinging normally against a flat uniform heat flux surface [8]. The correlation for localized heat transfer coefficient at the underside of the piston surface, $h = f(r)$, is given by following Eqs. (3a)–(4).

$$\frac{Nu}{Nu_0} = \left(1 + f(r/d)^{-9}\right)^{-1/9} \tag{3a}$$

Where

$$f(r/d) = ae^{b(r/d)}. \tag{3b}$$

The values of a and b in Eq. (3b) are given in Table 1. These values are found out by Steven and Webb [8] experimentally.

**Table 1**
Values of a and b used in equation [7].

| d (mm) | 2.2 | 3 | 4.1 | 5.8 | 8.9 |
|---|---|---|---|---|---|
| a | 1.13 | 1.141 | 1.34 | 1.48 | 1.57 |
| b | −0.23 | −0.2395 | −0.41 | −0.56 | −0.7 |

Stagnation point Nusselt number for Eq. (3a) is given by the following equation. Eq. (4) is valid for Re = 4000–52000.

$$Nu_0 = 2.67\text{Re}^{0.567}\text{Pr}^{0.4}(z_0/d)^{-0.0336}(v/d)^{-0.237}. \tag{4}$$

The input parameters for this model are given in Table 2. These parameters represent the inputs for the actual properties of material, oil used in an engine under investigation and the geometry of the actual oil jet.

For the validation of computational model for the complex piston geometry, piston temperature profile has been captured by thermal imaging (IR) camera with oil jet cooling for different heat flux. Fig. 2 represents the computational and experimental results of temperature profile of the piston. Results suggest that the computational model developed predicts temperature profile for complex piston geometry with reasonable accuracy. There is a variation of approximately 10 °C in upper limit of temperature and 20 °C for lower limit of temperature. The variation in computationally predicted temperature profile is within 10% of the experimental results. This variation may possibly be due to mist generation which might generate some error in IR camera due to interference with radiative heat transfer.

## 3. Computational results

In an engine, the piston undergoes different heat flux under varying engine load and speed conditions. This computational model is used for prediction of temperature profile of production grade piston, where a typical heat flux to the piston surface varies from 30 to 50 kW/m$^2$. The computational simulator is used for predicting steady-state temperature profile for 3/4th of axisymmetric piston segment. To understand the effect of varying power/fuel input on oil jet cooling of pistons, the normal heat input at the top surface of the piston was varied. In the present study, steady state temperature profile of the piston for 45 kW/m$^2$ heat flux incident on the top surface of the piston (without and with oil jet cooling of piston) is shown in Fig. 3.

The maximum temperature occurs at the center of the piston top surface. The maximum temperature at the piston top surface is approximately 325 °C without cooling and 290 °C with oil jet cooling. The temperature at the underside of the piston varies from 295 to 315 °C without cooling and from 260 to 280 °C with oil jet cooling. The temperature in the first compression ring groove varies from 312 to 315 °C without cooling and from 273 to 275 °C with oil jet cooling. The temperature in the skirt region varies from 243 to 290 °C without cooling and from 205 to 250 °C with oil jet cooling. Here, oil jet cooling of piston reduces the temperature by approximately 35–40 °C all over the surface.

For experimental investigations, it is required to find out, when the piston is going to attain steady state temperature. This is calculated using the computational model. Five point of observation on piston surface are selected (Fig. 4). These five points can be considered as

**Table 2**
Input parameters for numerical simulation.

| | |
|---|---|
| Piston diameter (D) | 89 mm |
| Oil jet impingement distance from BDC (z) | 55 mm |
| Diameter of jet (d) | 3 mm |
| Oil temperature | 100 °C |
| Oil type | SAE 15W40 |
| Oil flow rate (Q) | $8\times10^{-5}$ m$^3$/s (4.8 LPM) |
| Specific heat ($C_p$) | 2.219 kJ/kgK |
| Oil thermal conductivity ($k_{oil}$) | 0.137 W/mK |
| Density of oil ($\rho$) | 847 kg/m$^3$ |
| Kinematic viscosity of oil ($\nu$) | $14.1\times10^{-6}$ m$^2$/s |
| Aluminum thermal conductivity (k) | 137 W/mK |
| Specific heat of aluminum | 900 J/kgK |
| Density of aluminum | 2700 kg/m$^3$ |
| Jet velocity (v) | 20 m/s |

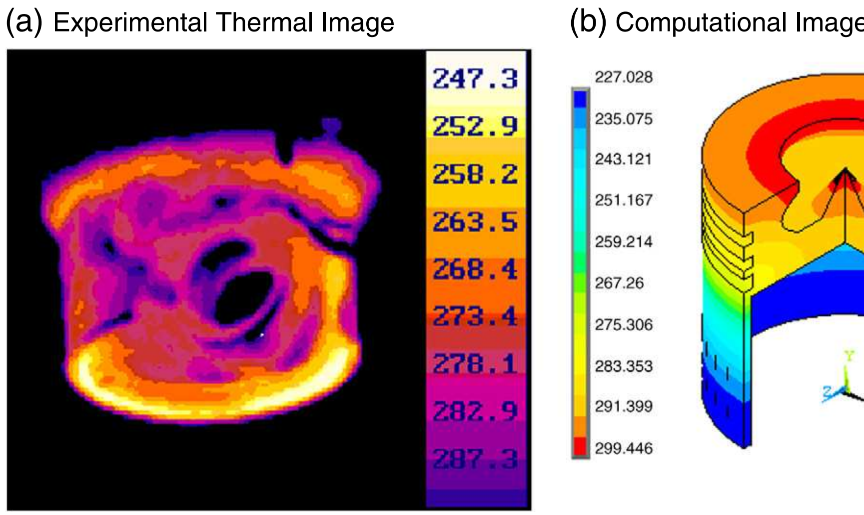


**Fig. 2.** Steady state temperature distribution for piston geometry with oil jet cooling ($q''=40$ kW/m$^2$).

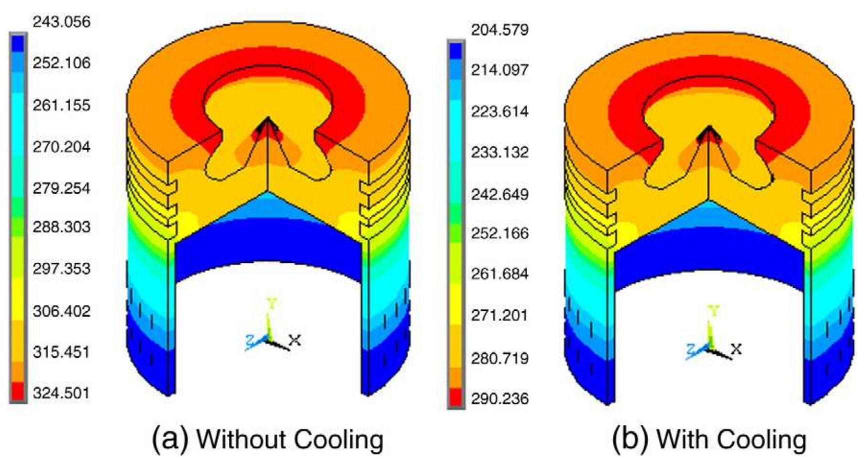


**Fig. 3.** Steady state temperature profile of the piston without/with oil jet cooling ($q''=45$ kW/m$^2$).

critical points for given piston geometry. Points 1 and 3 correspond to maximum and minimum temperature points, and Points 2 and 5 correspond to maximum and minimum temperature on the underside surface of piston where oil jet cooling is applied. Mist generation mainly depends on these temperatures. Point 4 is corresponding to maximum temperature in the ring region.

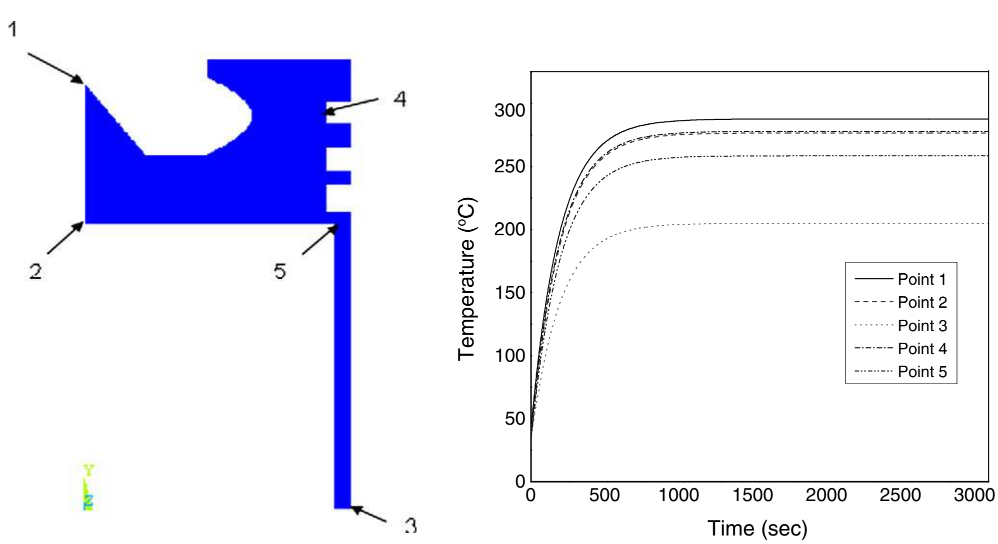


**Fig. 4.** Piston temperature variation with time ($q''=45$ kW/m$^2$).

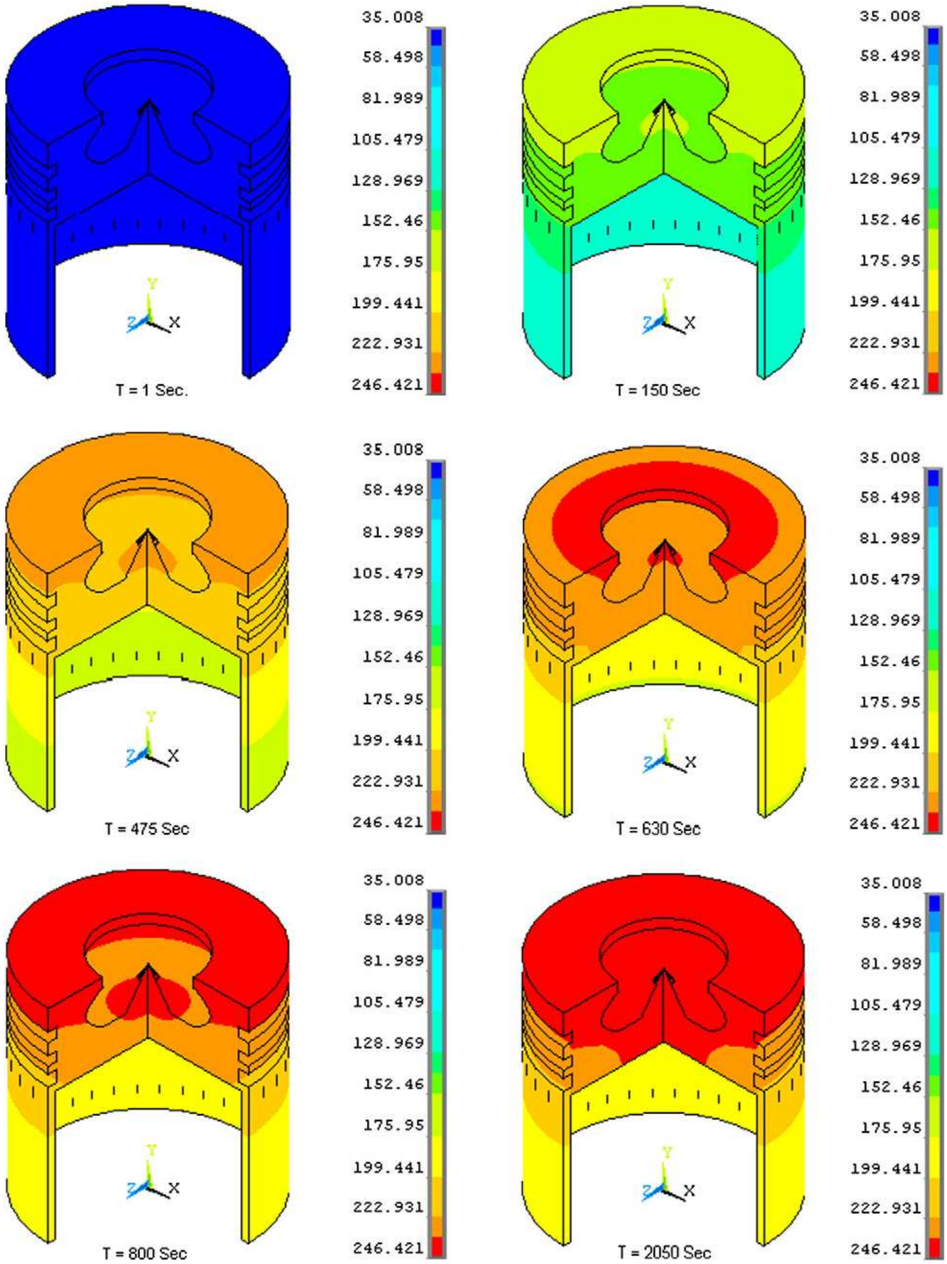


**Fig. 5.** Transient piston temperatures ($q'' = 35$ kW/m$^2$).

For understanding the transient behavior of the piston temperature profile with time, the temperature variation at these five different points is computed from this model and is shown in Fig. 4 w.r.t. time while the heat flux input to the piston is 45 kW/m$^2$. It shows that after 1400 s from starting (when combustion starts), piston achieves steady state and the temperatures do not vary with time thereafter.

## 4. Transient piston temperature

Transient analysis of piston is necessary to accommodate the real time boundary conditions. Initially when heat flux is applied to piston (when combustion starts), its temperature ramp rate is very high, which decreases with time before finally stabilizing.

For 35 KW/m$^2$ heat flux, 5 min after starting, maximum temperature of piston reaches approximately 208 °C. In next 5 min, it reaches approximately 240 °C. Simulation results show that piston reaches its steady state after approximately 25 min. Temperature profiles of the piston at different times are shown to reflect this time dependence of temperatures rise for a given heat flux of 35 kW/m$^2$ (Fig. 5). The time taken in achieving steady state is given in Table 3 for different heat fluxes.

This table suggests that the time taken for stabilizing temperature profiles of the piston at various locations is lower for higher load conditions and if the pistons are cooled.

**Table 3**
Time taken in attaining steady state piston temperatures.

| S. No. | Heat flux (kW/m$^2$) | Time for attaining steady state (seconds) | |
|---|---|---|---|
| | | Without cooling | With cooling |
| 1 | 30 | 1700 | 1300 |
| 2 | 35 | 1620 | 1225 |
| 3 | 40 | 1540 | 1151 |
| 4 | 45 | 1450 | 1106 |
| 5 | 50 | 1400 | 1075 |

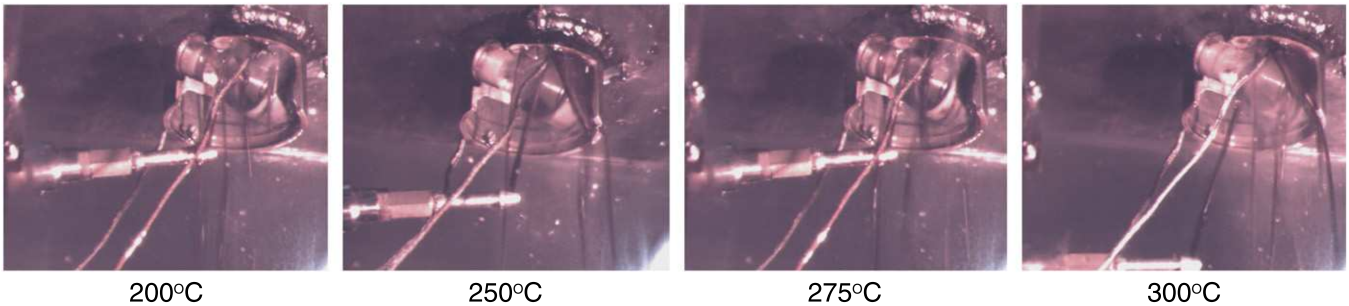


**Fig. 6.** Oil impingement on heated piston.

## 5. Mist formation

Mist generation studies were also carried out on the same piston. The underside of the piston was maintained at different temperatures and mist generation was captured using a high speed camera.

Initially, the piston was maintained at temperature of 200 °C and then the piston temperature was increased to 250, 275, and 300 °C for subsequent investigations and visualization. The oil jet impingement on piston is shown in Fig. 6. When piston temperature was around 200 °C, there was no mist generation. The oil jet cooling was effective under this condition and there was no oil jet breakup. As the underside of piston temperature increased to 250 °C, the oil jet started breaking into bigger oil droplets after impingement on the heated underside surface and some mist generation was observed out of the impingement region. When the jet was directed at the piston maintained at 275 °C, localized boiling of oil at the point of impingement was observed. There was however no sign of a film boiling at the point of impingement. At this temperature, there were some changes in the spray pattern also. Oil changes its viscosity due to change in temperature. Changes in oil properties was the reason for change in spray pattern. At 300 °C, large quantities of white smoke and oil mist generation were observed to be coming out of the impingement region on the piston's underside. There was localized boiling of oil at the point of impingement. These investigations confirm that the impingement of high velocity oil jet on the hot piston surface actually leads to oil mist generation and emission of unburnt hydrocarbon (UBHC) from the crankcase region, which potentially leaks out of the engine in the form of non-point source emissions.

## 6. Conclusions

A computational model for temperature prediction of piston under oil jet cooling has been developed. Heat transfer coefficient was predicted using Steven-Webb relation. This model predicts time resolved temperature profile for different heat fluxes applied to the piston by varying heat input at the top of the piston. Validation of the model was done by piston temperature profile experiment using IR Camera for varying heat fluxes. Average difference between numerical and experimental values was around 10–20 °C. Temperature obtained from computational model varies from 240 to 325 °C without oil jet cooling and 205–290 °C with oil jet cooling for piston (for 45 kW/m$^2$ heat flux applied). Oil jet cooling reduces the piston temperature by 35–40 °C.

Once the engine is started, it takes significant amount of time for the temperature profile of the piston to achieve steady state. The time taken for achieving steady state of temperature profiles of the piston at various locations is lower for higher load condition and this further goes down if the pistons are cooled by an oil jet. In case of oil jet cooling piston temperature remains within 200–300 °C hence this temperature limit was also examined for investigation of mist generation. The mist generation due to oil jet cooling of heated surface starts when surface temperature is above 250 °C and increased further. When surface temperature reaches 300 °C, large quantities of oil mist and smoke is generated due to localized pool boiling of the oil jet generated.

## References

[1] N. Ladommatos, Z. Xiao, H. Zhao, The effect of piston bowl temperature on diesel exhaust emissions, Proc. IMechE Part D: J. Auto. Eng. 219 (2005) 371–388.
[2] C. Cristina, A.S. Fleiscer, M. Rounds, R.J. Goldstein, Jet impingement cooling of a convex semi-cylindrical surface, Int. J. Therm. Sci. 40 (2001) 890–898.
[3] C. Gau, C.M. Chung, Surface curvature effect on slot-air-jet impingement cooling flow and heat transfer process, Trans. ASME 113 (1991) 858–864.
[4] D.H. Wolf, F.P. Incropera, R. Viskanta, Jet impingement boiling, in: J.P. Hartnett, et al., (Eds.), Advances in Heat Transfer, 23, Academic Press, New York, 1993, p. 1132.
[5] H. Martin, Heat and mass transfer between impinging gas jets and solid surfaces, Adv. in Heat Transfer, 13, 1977, pp. 1–60.
[6] K. Jambunathan, E. Lai, M.A. Moss, B.L. Button, A review of heat transfer data for single circular jet impingement, Int. J. Heat Fluid Flow 13 (2) (1992) 106–115.
[7] J. Steven, B.W. Webb, The effect of inclination on local heat transfer under an axisymmetric, free liquid jet, Int. J. Heat Mass Transfer 34 (4/5) (1991) 1227–1236.
[8] J. Steven, B.W. Webb, Local heat transfer coefficient under and axisymmetric, single-phase liquid jet, ASME J. Heat Transfer 113 (1991) 71–78.
[9] A.K. Agarwal, S.K. Goyal, Experimental and numerical investigations of jet impingement cooling of piston of heavy-duty diesel engine for controlling the non-tail pipe emissions, SAE Paper No 2007-01-0763, 2007.
[10] M.B. Varghese, A.K. Agarwal, Numerical investigations of piston cooling using oil jet in heavy duty diesel engines, J. Engine Res. 7 (5) (2006) 411–422.