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

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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

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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:

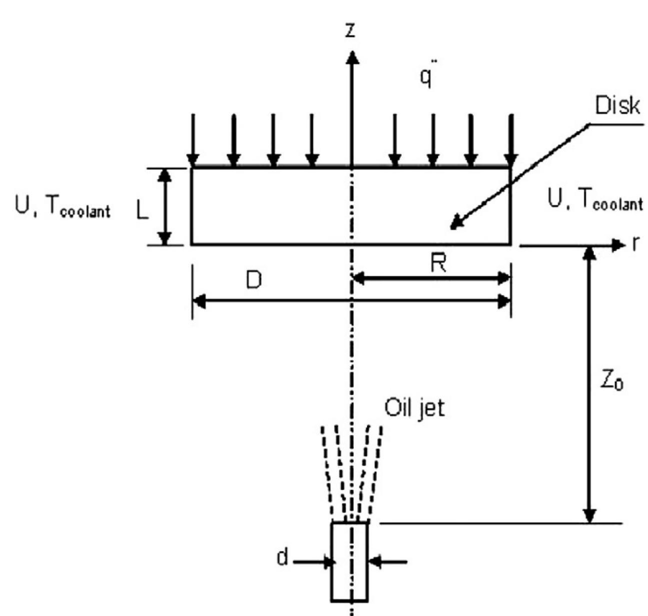


**Fig. 1.** Coordinate system and notations used for oil jet cooling [9,10].