Numerical and Experimental Investigation of Oil Jet Cooled Piston

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NUMERICAL AND EXPERIMENTAL INVESTIGATION OF OIL JET COOLED PISTON

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ABSTRACT

Thermal loading of diesel engine pistons has increased dramatically in recent years due to applications of various advanced technologies to meet low emission and high power requirements. Control of piston temperatures by cooling of pistons has become one of the determining factors in a successful engine design. The pistons are cooled by oil jets fired at the underside from the crankcase. Any undesirable piston temperature rise may lead to engine seizure because of piston warping. However, if the temperature at the underside of the piston, where oil jet strikes the piston, is above the boiling point of the oil being used, it may contribute to the mist generation. This mist significantly contribute to the non-tail pipe emissions in the form of unburnt hydrocarbons (UBHC's), which has unfortunately not been looked into so seriously, as the current stress of all the automobile manufacturers is on meeting the tail pipe emission legislative limits.

A numerical model has been developed using finite elements method for studying the oil jet cooling of pistons. Using the 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 type, oil jet velocity, oil jet diameter and distance of the nozzle from the underside of the piston. It also helps predict whether the selected grade of oil will contribute to mist generation. Experimental validation of the numerical modeling was carried out on a flat plate. Problem of mist generation was also investigated on a flat plate using high speed camera.

NOMENCLATURE

C_p = Specific heat of the oil, J/kgK

d = nozzle diameter, m
D = diameter of the disk, m
h = local heat transfer coefficient (W/m^{2}K) at the bottom surface of the disk
k_{jet} = thermal conductivity of the oil jet (W/mK)
k_r = thermal conductivity of the material in r direction
k_z = thermal conductivity of the material in z direction
Nu = local Nusselt number = hD / k_{jet}
Nu_0 = Stagnation point Nusselt number
Pr = Prandtl number of the oil jet = $\frac{mC_p}{k_{jet}}$
r = distance from the left hand side of piston.
Re = jet Reynolds number based on the nozzle diameter
$= \frac{vd}{g}$
v = $v_{jet(absolute)} - v_{piston}$ = relative jet velocity (averaged over a cycle), m/s
w = weight function
z = distance from the underside of the piston.
z_o = vertical distance of the disk from the nozzle exit, m (Figure 3)
g = Kinematic viscosity of the oil, m^2/s
m = Dynamic viscosity of oil, kg/ms
$\Psi_j(r, z)$ = shape function

INTRODUCTION

The current trend in the automobile industry is towards increasing the power density of the engines and making lighter engines. These requirements lead to higher thermal load on the engine, especially on the pistons. In the heavy duty diesel engines, combustion chamber and cylinder head are normally water cooled. The piston however cannot be cooled using water jacket because of logistics problem. The piston temperature is one of the limiting factors in high-powered internal combustion engines. This problem is particularly severe in transportation engines, where the space-weight to power ratio is of prime importance. In this case, the bore and stroke have to be kept to a minimum and the engine speed is limited by piston velocity. The only factor, which can be improved, is the mean cylinder pressure. An increase in mean cylinder pressure necessarily means additional heat input to the cylinder, a part of which has to be rejected through the walls of the combustion chamber and piston. Thus, the increase in density of the heat flux causes rise in cylinder wall and piston temperatures.

Figure 1: Piston Scuffing Due to Excessive Piston Temperatures

The temperatures of certain critical areas in piston need to be kept low because of material constraints. Aluminium alloys begin to melt at temperatures greater than 500^{0} C as shown in figure 1.The liner and the cylinder head can be however be maintained at a reasonably high temperatures by suitably adjusting the water or air cooling. The total heat flow through the piston crown amounts to about 2 percent of the energy released by the fuel. The direct effect of the piston cooling on thermal efficiency of the engine is therefore miniscule. A large part of this heat loss represents heat transferred to the piston during the exhaust process; therefore the direct loss of what would otherwise be available energy is probably considerably less than 1 percent of total heat input. Typical heat balance for the piston cooling load in a heavy duty diesel engine is given in table 1.

Location of heat dissipation	Dissipation %
undercrown surface	71
under crown surface behind ring-groove pad	11
upper and lower skirt sections	10
rings and lands	8

Table 1: Heat Balance for Piston Cooling-Load [1]

The piston is usually cooled by oil jets fired to the underside from the crankcase in a heavy-duty diesel engine as shown in figure 2. The oil jets hit the hot piston

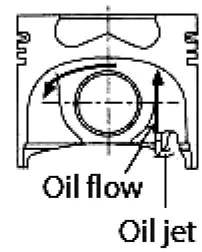

Figure 2: Oil Jet Cooling of Pistons

at a very high relative velocity ranging from 5 m/s to 40 m/s. The oil jet breaks into mist because of high temperature at the underside of the piston and high relative velocity. This piston cooling generated mist contributes significantly towards the non-tail pipe emissions in the form of unburnt hydrocarbons (UBHC's) from the engine, this unfortunately has not been looked into seriously by automobile manufacturers. So investigating the conditions, under which the oil jet cooling of the piston start contributing significantly towards the non-tail pipe emissions through mist generation. Finding optimum cooling condition by testing is both expensive and time-consuming. Therefore, piston temperature predictions with sufficient accuracy at the design stage become important. In engine designs, accurate prediction of piston temperatures is required because oil pump capacity and lubricating system are decided by the amount of piston cooling oil. Therefore, the prediction is needed at the first stage of engine designing.

METHODS OF PISTON COOLING

There are various methods of piston cooling either cooled by oil, water or air. Water cooling was applied to heavy, low speed engines for some time; but more recently it is abandoned because of serious design and maintenance difficulties with piping and sealing. However, this type of cooling has advantages, since water has a higher specific heat and lower viscosity than oil and thus better heat transfer takes place.

Air cooling is simpler from design point of view, but lower specific heat per unit volume requires very large quantities of air to be supplied to the piston. This involves bulky pipes and ducts and an additional air compressor, which makes it less practical.

The usual method is oil cooling, in which oil can be supplied from the main lubrication system along the connecting rod to the piston or from a separate oil supply. After piston cooling, the oil returns to the crankcase. There are six different types of method of oil cooling of pistons.

1. In splash lubrication system, there is no direct oil supply to the piston. The oil retained in the crankcase is churned and splashed up by the internal parts of the engine (connecting rod big end and crankshaft) into a combination of liquid and mist. The oil mist is sprayed

over the interior of the engine i.e.:- on the cylinder walls and on the underside of the piston crown.

2. Cooling due to oil emerging from the small end bearings, which are pressure lubricated. Pistons of marine engines are cooled with the help of pressure lubricated oil emerging from the small end bearings.

3. The third method uses nozzle plate and nozzles. The oil goes up the annular space formed between the oil tube and the bore in the piston rod, and returns down the centre. The oil is sprayed up into the matching bores on the underside of the crown. This allows the crown to be made very thin as possible, in order to allow maximum heat transfer while maintaining strength.

4. “Cocktail shaker” is a cooling method applied particularly to engine pistons. The oil is brought into a closed chamber below the piston crown and released from the chamber to the crankcase through overflow holes or baffles. The oil is agitated violently by the piston reciprocating movement and the turbulence of the oil motion produces a high heat transfer coefficient between oil and the piston.

5. Oil jet cooling, where the oil is released at high pressure from a nozzle mounted on the cylinder block and the nozzle directed towards the underside of the piston. The oil jet hits the undercrown of the piston and is then splashed on to the surrounding walls of the piston.

6. The methods 4 and 5 are often combined, that is, the oil is injected from the top of the connecting rod onto the piston undercrown and then cooled in the ``cocktail shaker'' chamber from which it is released through the overflow to the crankcase.

Out of the six methods, oil jet cooling method is most practical for cooling of pistons and is used extensively in commercial heavy duty diesel engines.

HISTORICAL PERSPECTIVE

The studies of surface cooling by means of jets were originally conducted aiming the thermal protection of stator and rotor blades. Thus, extensive reviews presented in the literature to analyze the behaviour of different types of jets in different operating conditions [2-10]. Studies on cooling of internal combustion engine started in 1960's. Bush and London introduced the term ``cocktail shaker'' [11]. The effect of piston reciprocating movement on the oil cooling gallery heat transfer coefficient was also analyzed by Bush and London [12], presenting basic design information for ``cocktail shaker'' cooled pistons.

Advanced results were presented by French using different experimental rigs and engine test configurations and an expression for the heat transfer coefficient was presented [13]. Evans (1977) conducted a thorough study of the ``cocktail shaker'' piston cooling concept [14]. Movies of flow visualization were taken using an open gallery. Kajiwara et al. calculated the heat transfer coefficient in the cooling gallery of the oil jet cooled piston directly using CFD code [15]. Martins et al. (1993) analyzed the cooling conditions of articulated piston and their imapct on the piston performance in an effort to optimize articulated piston cooling [16]. Pimenta et al. used numerical simulation (finite element method) to study cooling of automotive pistons by liquid cooling jets [17]. Dhariwal investigated blow-by emission and lubricating oil consumption in an IC engine and tried to control blow-by losses using Positive Crankcase Ventilation (PCV) [18]. Stotter (1966) carried out experimental investigations on heat transfer for various methods of piston cooling at English Electric Co. Ltd., England, which were applied in calculations and predictions of piston temperatures [19]. Flynn (1945) et al. found oil cooling as one of the efficient means for piston temperature control [20]. Otto Kruggel (1971) measured piston temperatures in an air cooled two-stroke gasoline engine [21]. Gerhard Woschni et al. found out the local heat transfer coefficients in the piston of a high speed diesel engine from experimentally measured piston temperature distribution [22].

Huebotter (1925) et al. analyzed the flow of heat in pistons [23]. Janeway carried out quantitative analysis of heat transfer in engines [24]. Paschkis et al. used electrical analogy method for determining unsteady state heat transfer through the piston [25]. A rough check of the piston cooling affected by the crankcase air and oil was also made [1]. Willis (1944) et al. found out the operating temperatures and stresses in aluminium aircraft engine parts [26]. Sanders et al. analyzed the variation of piston temperature with piston dimensions and undercrown cooling [27]. Baker (1932) et al. analyzed the piston temperatures and their relation to piston design [28]. Wang et al. developed a simplified annular heat-pipe cooled piston crown (AHPCC) [29]. The annular heat pipe is an extension of reciprocating heat pipe.

MODEL DEVELOPMENT

A numerical model has been developed using computational fluid dynamics (CFD) tools (finite elements methods), for studying the oil jet cooling of pistons. Using the numerical model developed by Stevens and Webb (1991) [9, 10], the heat transfer coefficient (h) required at the underside of the piston is predicted. The heat transfer coefficient in the cooling gallery has great effect on the piston temperature. However, it is hard to predict with sufficient accuracy because it is influenced by various factors, e.g.: oil flow, engine speed, oil jet diameter etc.

There are basically two approaches to cope with the increase in thermal load of pistons. One is the improvement of piston cooling ability through redesigning of piston structure. The other is improvement of material strength in the high temperature region.

In order to understand the oil jet cooling of automotive piston, a computational model was developed. For this, general heat transfer equation in cylindrical coordinates with appropriate boundary conditions needs to be solved. A weak formulation of the governing differential equation need to be developed for given geometry. The geometry of the surface in question is converted to grid for finite element analysis. Thereafter Steven and Webb correlation is applied to this grid in order to find out the heat transfer coefficient at the jet cooled surface. A code is developed using weak formulation and isotherms are drawn on the surface of piston. The effect of various operating parameters on piston cooling is investigated in detail.

The path of heat flowing from the hot combustion gases through the piston body to the surroundings is conditioned by the shape of the piston body and by the boundary conditions, consisting of the sliding surfaces in contact with the lubricated liner and the inside of the piston, which may be cooled or not. The heat flow through the piston can be considered as a steady flow, because the amplitude of the periodic changes of combustion gas temperature is almost lost in the boundary gas film and in a very thin “skin” layer of the piston crown [19].

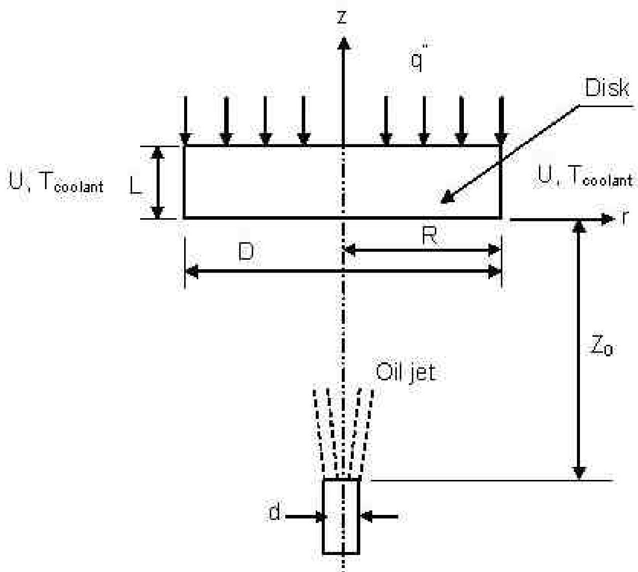

Figure 3: The Coordinate System and Pictorial View of the Notations Used for Oil Jet Cooling.

The governing differential equation for the piston in cylindrical coordinates is given by following equation

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = 0 \qquad (1)$$

The two dimensional governing differential equation is taken, as from physical and geometrical considerations the piston is axisymmetric.

Knowing the boundary conditions, this equation may be solved by numerical methods. 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:

a) The top crown in contact with the hot combustion gases, i.e.:-

$$+k\frac{\partial T}{\partial z} = q'' \qquad (2a)$$

b) The sliding lubricated surface in contact with the liner, including the rings, i.e.:-

$$-k\frac{\partial T}{\partial r} = U(T - T_{coolant}) \qquad (2b)$$

c) The inside of the piston exposed either to the crankcase atmosphere or to a coolant, usually oil, i.e.:-

$$+k\frac{\partial T}{\partial z} = h(T - T_{oiljet}) \qquad (2c)$$

d) From the physical and geometrical conditions only half portion of the piston can be taken for analysis, i.e.:-

$$\frac{\partial T}{\partial r} = 0 \qquad (2d)$$

The positive sign on the L.H.S. of equations (2c) and (2d) arise because temperature will be increasing with increasing z in the disk. ‘k’ (W/mK) is the thermal conductivity of the piston, which is made of aluminium. The local jet heat transfer coefficient was calculated from the correlations given by Stevens and Webb (1991) for axisymmetric, single-phase free round liquid jets impinging normally against a flat uniform heat flux surface [9,10]. The correlation for local heat transfer coefficient at the piston underside surface h = f(r) is given by equation (3) and (4) given below:

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

Where $f(r/d) = ae^{b(r/d)}$ (3b)

The values of a and b are listed in Table 2.

d (mm)	2.2	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

Table 2: Values of a and b in equation (3b) [8, 9]

$$Nu_0 = 2.67 Re^{0.567} Pr^{0.4} (z_0/d)^{-0.336} (v/d)^{-0.237} \qquad (4)$$

Equation (4) is valid for Re = 4000-52000.

NUMERICAL SIMULATION

The governing differential equation (1) is solved using FEA methods. The variational statement of the governing differential equation is:

$$2\Pi\int_{\Omega^e} w\left(\frac{1}{r}\frac{\partial}{\partial r}(k_r r\frac{\partial T}{\partial r})+\frac{\partial}{\partial z}(k_z\frac{\partial T}{\partial z})\right)rdrdz=0 \qquad (5)$$

$$T=\sum_{j=1}^{n} T_j\Psi_j(r,z) \qquad (6)$$

$$w=?_i \qquad (7)$$

$$\left[\mathbf{K}_{ij}^{e}+\mathbf{H}_{ij}^{e}\right]\left\{\mathbf{T}_j^{e}\right\}=\left\{\mathbf{Q}_i^{e}\right\}+\left\{\mathbf{P}_i^{e}\right\} \qquad (8)$$

$$K_{ij}^{e}=2\Pi\int_{\Omega^e}(k_r\frac{\partial\Psi_i^e}{\partial r}\frac{\partial\Psi_j^e}{\partial r}+k_z\frac{\partial\Psi_i^e}{\partial z}\frac{\partial\Psi_j^e}{\partial z})rdrdz$$

$$H_{ij}^{e}=2\Pi\oint_{\Gamma^e}h^e\Psi_i^e\Psi_j^e rds$$

$$Q_i^{e}=2\Pi\oint_{\Gamma^e}\hat{q}_n\Psi_i^e rds$$

$$P_i^{e}=2\Pi\oint_{\Gamma^e}h^eT_\infty^e\Psi_i^e rds$$

INPUT PARAMETERS

The piston used for present investigation is production grade piston from Mahindra and Mahindra Direct injection 2500 diesel engine. The input parameters for the simulation are as follows:

Piston Diameter (D):	89 mm
Oil jet distance from BDC (z):	55 mm
Diameter of jet (d):	3 mm
Oil temperature:	1000° C
Oil type:	15W40
Oil flow rate (Q):	8X10-5 m3/ sec
Specific heat (Cp):	2.219 kJ/ kgK
Oil thermal conductivity (k):	0.137 W/mK
Density of oil (r):	847 kg/m3
Kinematic viscosity (g):	14.1X10-6 m2/s
Aluminum thermal conductivity (k):	137 W/mK
Jet velocity (v):	20 m/s
Specific power ($q^{\|\|}$):	35 kW/m2

A structured mesh was generated for the piston profile using Transfinite Interpolation method [30]. 770 quadrilateral elements were taken. Tecplot v 8.0 was used for viewing the mesh within the half axisymmetric segment of the piston. The structured mesh generated for the piston profile is shown in figure 4.

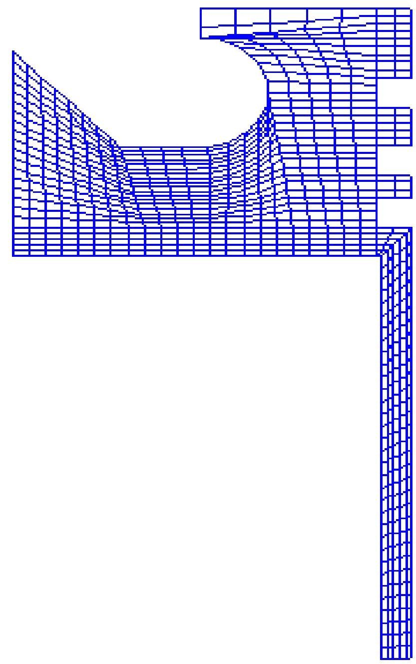

Figure 4: Axisymmetric Mesh Generated within the Piston Profile.

EXPERIMENTAL SETUP

The objective of setting up an experiment was to validate the computational model for oil jet cooling of flat plate experimentally.

The second objective was to investigate the conditions, under which the oil jet cooling of the flat plate/piston start contributing significantly towards the non-tail pipe emissions through mist and smoke generated because of oil jet break up and localized boiling. Schematic of the experimental setup is shown in figure 5.

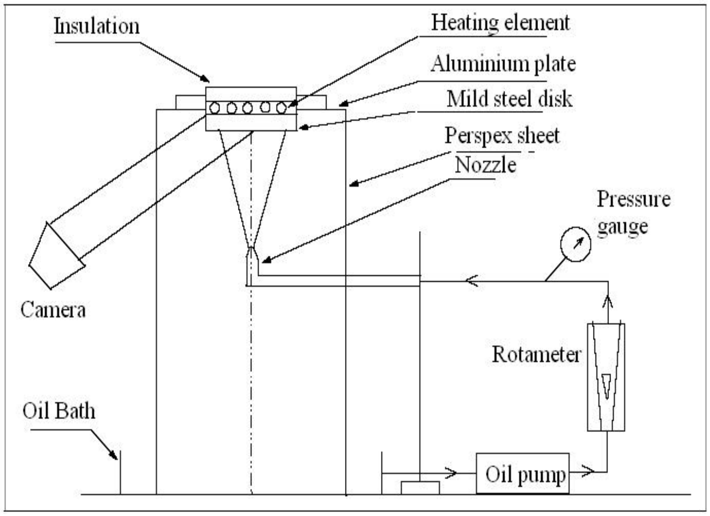

Figure 5: Schematic of Experimental Setup

Experimental set up consists of a square cross-sectional perspex enclosure. A hot plate is mounted on the enclosure. A throttle valve is used to control the flow pressure and thus velocity of the jet. A rotameter is connected in the line to measure the oil flow rate. Rotameter is capable of measuring a flow rate of 3 - 30 litres per minute. Pressure gauge was connected to measure the oil pressure.

EXPERIMENTAL VALIDATION

The nozzle location was kept constant and validation of the model for flat plate was carried for temperature at various distances from the centre of the flat plate is shown in Figure 6.

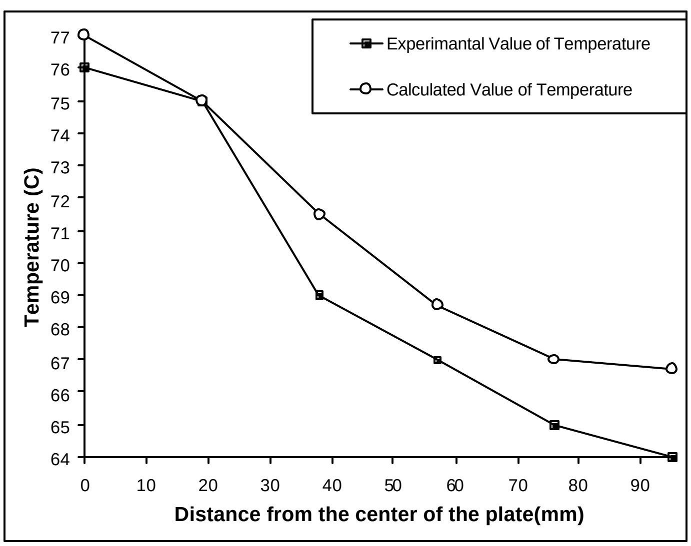

Figure 6: temperature variation along the distance from the centre of the plate while keeping nozzle location constant.

The oil jet velocity was kept constant and validation of the model for flat plate was carried for temperature at various distances from the centre of the flat plate is shown in Figure 7.

Maximum percentage difference between numerical and experimental value is 5 % where as the minimum percentage difference between numerical and experimental value is 0.09 %. Most of the differences were of the order of 1 to 2 %. The average percentage difference between numerical and experimental value is 1.6%.

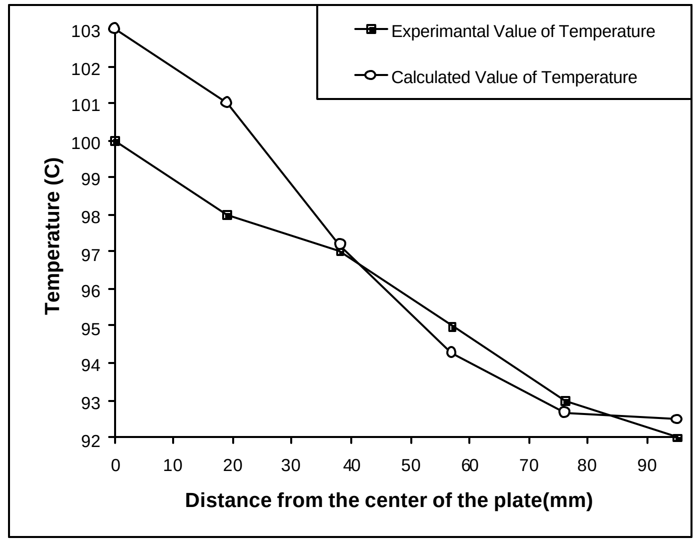

Figure 7: temperature variation along the distance from the centre of the plate while keeping oil jet velocity constant.

NUMERICAL AND EXPERIMENTAL RESULTS

Using the model described earlier, isotherms of the predicted temperature profile in the piston have been plotted using Tecplot v 8.0 for both cases, without oil cooling at the underside of the piston and with oil cooling at the underside of the piston. The results of this simulation are shown in Figure 8 and 9.

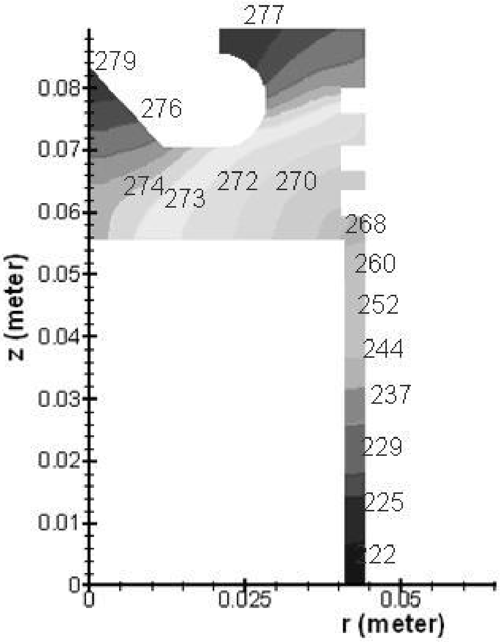

Figure 8: Steady state temperature distribution in the piston without oil jet cooling.

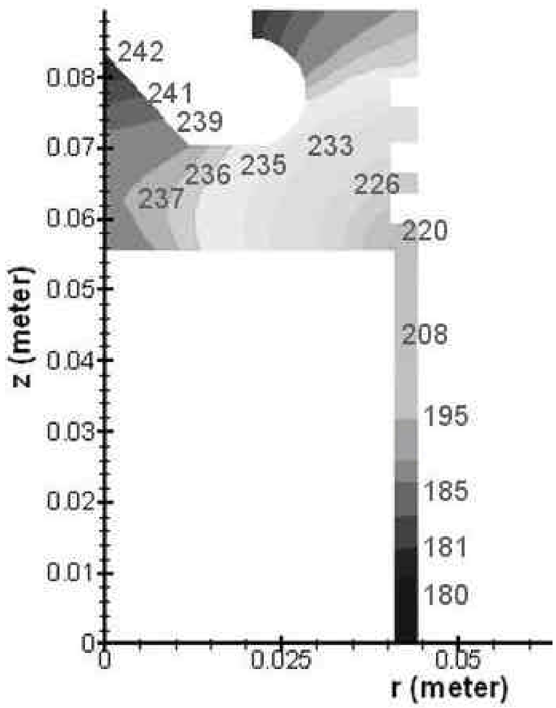

Figure 9: Steady state temperature distribution in the piston with oil jet cooling.

It is observed that the piston temperatures generally get lowered by approximately 40^0 C by using oil jet cooling. These results match reasonably well with the experimental data available in literature [22, 27 and 30].

Heat transfer coefficient at the underside of the piston, for the configuration of oil jet cooling is a function of the distance of the point under consideration from the point of impingement. Heat transfer coefficient at the underside of the piston is calculated using Steven and Webb correlation. Variation of heat transfer coefficient from the center in the radially outwards direction for the selected input parameters is shown in figure 10.

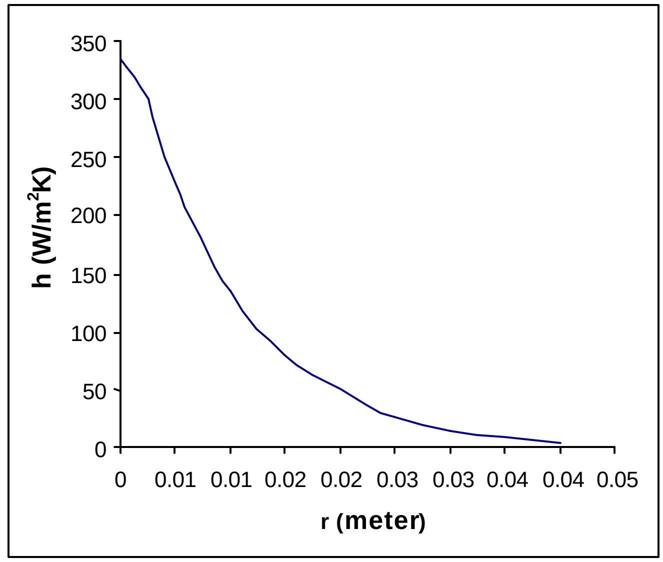

Figure 10: Variation of Heat Transfer Coefficient from the Centre Radially Outwards.

The temperature is highest at the centre, when the oil jet cooling is not employed and it gets lowered radially outwards towards the skirt as shown in figure 11.

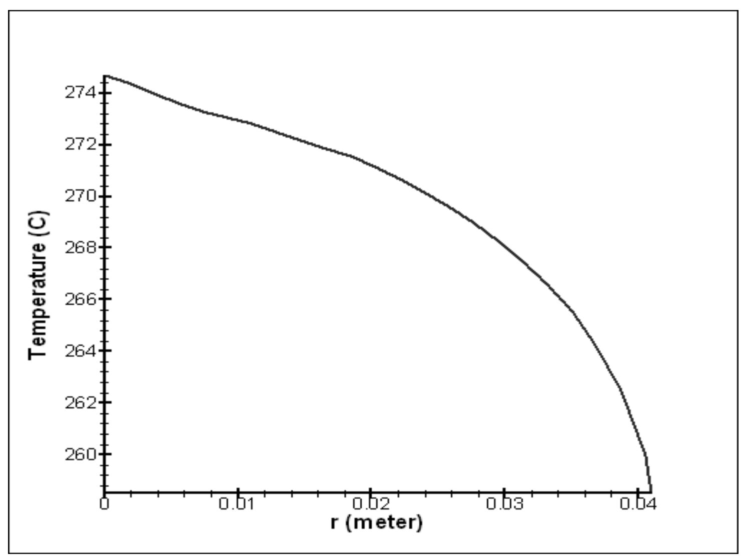

Figure 11: Variation in temperature from the centre radially outwards without oil jet cooling.

The nozzle distance from the BDC was varied and its effect on the piston temperature is examined. The results are represented in figure 12. The result shows that the piston cooling gets improved with decreasing nozzle distance from BDC.

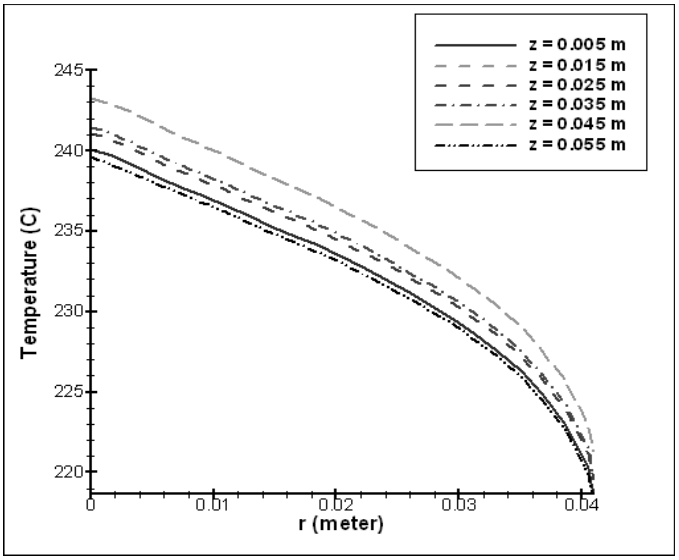

Figure 12: Variation in temperature from the centre radially outwards for different nozzle distance from the BDC.

The effect of oil jet velocity on the piston temperature is also investigated and the results are shown in figure 13. The results show enhanced cooling with increasing jet velocity.

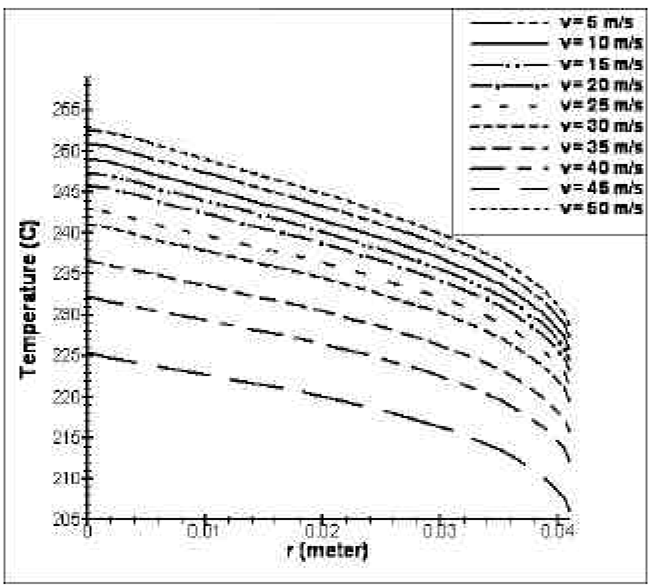

Figure 13: Variation in temperature from the centre radially outwards for different relative jet velocity.

The effect of oil type on the piston temperature is also investigated and the results are shown in figure 14. The results show enhanced cooling with decrease in oil viscosity.

The maximum temperature occurs at the centre of the piston top surface. The maximum temperature at the piston top is approximately 281^0 C without oil cooling at the underside of the piston while the temperature with oil cooling is 243^0 C. The temperature at the underside of

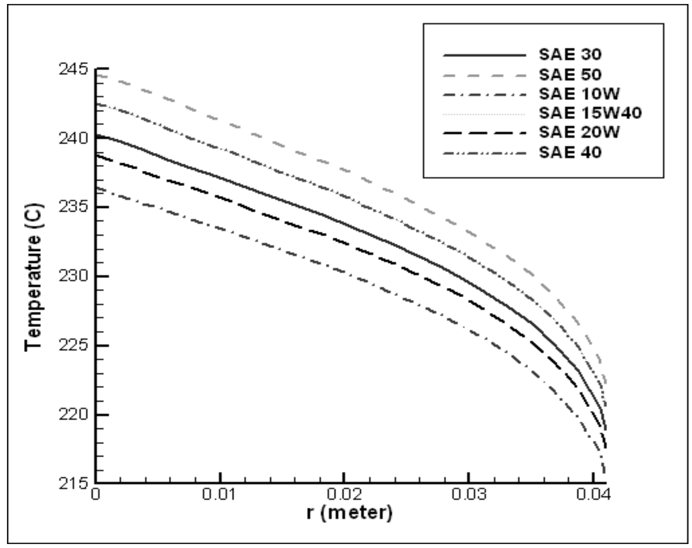

Figure 14: Variation in temperature from the centre radially outwards for different oil.

the piston varies from 275^0 C to 257^0 C without oil jet cooling and varies from 237^0 C to 212^0 C with the first compression ring groove varies from 275^0 C to 271^0 C without oil jet cooling while the temperature with oil jet cooling varies from 235^0 C to 231^0 C. The temperature in the skirt varies from 252^0 C to 222^0 C without oil jet cooling while the temperature with oil jet cooling varies from 216^0 C to 180^0 C.

Mist generation studies were also carried out on the flat hot plate that was used for experimental validation of the numerical simulator. Mist generated when high velocity jet striking a hot plate, were recorded using a camera. The plate was heated upto 250^0 C and oil jet was fired from below (z = 85 mm) and oil jet diameter 3 mm using SAE 40 with a jet velocity of 50 m/s. The oil jet was cooling the plate effectively and was not breaking. The plate temperature was increased to 300^0 C and it was observed that the oil jet was broken into bigger oil droplets after impingement on the hot plate. When the jet was fired to the plate at temperature 325^0 C, the jet started breaking into fine droplets and localized boiling of oil at the point of impingement.

The temperature of the hot plate was further increased to 345^0 C and a slightly white smoke started coming out of the impingement region along with the fine oil mist. At 355^0 C, huge quantity of white smoke and oil mist was observed to be coming out of the impingement region on the hot plate.

CONCLUSION

A computational model for oil jet cooling of piston of heavy duty diesel engine was developed using Finite Element Analysis (FEA) method. Transfinite interpolation method is used for grid generation of piston geometry. Heat transfer coefficients were predicted using Steven-Webb correlation. For numerical integration Gauss-Legendre Quadrature two-point formula was used.

Solution of the linear simultaneous equations was obtained by Gauss Elimination technique. For further refinement of the solution obtained by Gauss Elimination technique, Gauss-Siedel Iterative technique was used. Temperature profiles of piston are predicted. A CFD code in C language is developed for temperature prediction. The versatile CFD simulator can generate mesh for commercial grade engine pistons and predict temperature profiles with reasonable accuracy. This simulator is used for predicting temperature and effect of oil jet cooling on production grade piston. Maximum temperature on piston surface occurs at the center of the top surface of the piston. The numerical investigations of heat transfer conditions with oil jet cooling of piston have produced quantitative results of piston temperatures. An increase in power density of the engine by 10 to 12 % can be achieved by oil jet cooling of the piston.

Validation of the numerical simulator was carried out on flat plate. Average difference between numerical and experimental value is around 1.6 %. The effect of various parameters on oil jet cooling of pistons such as jet diameter, jet velocity, jet impingement distance and oil type is also investigated in details. Mist generation studies were also carried out on the flat plate using a camera. Oil jet cooling leads to increase in power density from the engine but it may also lead to increase in non-tail pipe unburnt hydrocarbon emissions, which are not easy to detect/measure. Hence it is extremely important to avoid the piston temperature, where the mist generation with smoke starts.

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