Proceedings of ICES2006:
Internal Combustion Engines Division, Spring Technical Conference 2006
May 7-10, 2006, Aachen, Germany

ICES 2006-1434

EXPERIMENTAL AND NUMERICAL INVESTIGATIONS OF JET IMPINGEMENT COOLING OF FLAT PLATE FOR CONTROLLING THE NON-TAIL PIPE EMISSIONS FROM HEAVY DUTY DIESEL ENGINES

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ABSTRACT

The continuous increase in power density has led to higher thermal loading of pistons of heavy duty diesel engines. Material constraints restrict the maximum operating temperature of a piston. High piston temperature rise may lead to engine seizure because of piston warping. To avoid this, pistons are usually cooled by oil jet impingement from the underside of the piston in heavy duty diesel engines. Impingement heat transfer has been used extensively because of the high rates of cooling it provides. The associated high heat transfer rate is due to the oil jet 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 the mist generation. This mist significantly contributes to non tail-pipe emission (non-point source) in the form of unburnt hydrocarbons (UBHC's).

This paper presents and discusses the results of a numerical and experimental investigation of the heat transfer between a constant heat flux flat plate and an impinging oil jet. Piston boundary conditions are applied to the flat plate. Using the numerical modeling, heat transfer coefficient (h) at the underside of the piston is calculated. This predicted value of heat transfer coefficient significantly helps in selecting right oil grade, oil jet velocity, nozzle diameter and distance of the nozzle from the underside of the piston. It also helps to predict whether the selected grade of oil will contribute to mist generation. Using numerical simulation (finite element method) temperature profiles are evaluated by varying heat flux. Infrared camera is used to investigate and validate the temperature profile of the flat plate. High speed camera is used to capture the mist generation and oil jet breakup due to impinging jet.

INTRODUCTION

Impinging jets are characterized by unusually high heat transfer coefficient and consequently have been widely used for cooling and drying in a variety of industries. Several studies have characterized the heat transfer coefficient under various configurations of impinging jets.

Impinging jets can be grouped according to several broad characteristics. In general, an impinging jet is termed as a submerged jet if it is a gas jet issuing into a gas or a liquid jet issuing into a liquid, and will be termed as a free jet if it is a liquid jet issuing into a gas. In addition, impinging jets can be classified by their shape, their temperature relative to the surface against which they are impinging, regardless the jet is oriented normally or obliquely with respect to the impinging surface, and whether the impinging surface is flat or curved [1]. To optimize the heat transfer, an understanding of the temperature field is essential, in particular near the impingement surface where the flow characteristics dominate the heat transfer process.

A significant amount of literature exists on heat transfer coefficient of the impinging jets. Yang *et al*. [1] presented the heat transfer results of jet impingement cooling on a semi-circular concave surface and clearly explained the significant effects of the nozzle geometry and the curvature of the plate. A wide variety of nozzles were tested by Garimella and Nenaydykh [2], and Colucci and Viskanta [3] to determine the effect of the nozzle geometry on local convective heat transfer rate for jet impingement cooling on a flat plate. Gauntner *et al*.

[4] presented a review report of the flow characteristics of a turbulent jet impinging on a flat plate. Martin [5] carried out heat and mass transfer studies on impinging jets and presented results which are based on single round nozzle, arrays of round nozzles, single slot nozzle, and arrays of slot nozzles. Hwang and Cheng [6] utilize a transient liquid crystal technique to evaluate the detailed heat transfer coefficient on two principal walls of a triangular duct with a swirling flow. Three different angles between the jet and the duct axial direction were examined. The wall average Nusselt number for the bottom and target walls of the triangular duct was insensitive to different jet inlet angles.

The continuing increase of power densities has led to the high thermal loading of the heavy duty diesel engine pistons. Cylinder wall and cylinder head have water cooling which s not possible for pistons hence these become most thermally stressed part of the engine. Material constraints restrict the increase in thermal loading of piston. High piston temperature rise may lead to engine seizure because of piston warping. Pistons are cooled by oil jet impingement from the underside of the piston in heavy duty diesel engines. Heavy-duty diesel engine pistons are cooled by oil jet fired at the underside of the piston by a nozzle mounted on the cylinder block. 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 from the engine. The problem of non tail pipe emission has unfortunately not being looked into so seriously, as the current stress of all the automobile manufacturers is on meeting the tail pipe emission legislative limits.

Significant amount of research has been done on piston temperature simulation and prediction of heat transfer coefficient for oil jet cooling of piston. Kajiwara *et al*. calculated the heat transfer coefficient in the cooling gallery of the oil jet cooled piston directly using CFD code [7]. Martins *et al*. analyzed the cooling conditions of articulated piston and their impact on the piston performance in an effort to optimize articulated piston cooling [8]. Pimenta *et al*. used numerical simulation (finite element method) to study cooling of automotive pistons by liquid cooling jets [9]. 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) [10]. Stotter carried out experimental investigations on heat transfer for various methods of piston cooling, which were applied in calculations and predictions of piston temperatures [11]. Flynn *et al*. found oil cooling as one of the efficient means for piston temperature control [12]. Otto Kruggel measured piston temperatures in an air cooled two-stroke gasoline engine [13]. 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 [14]. However, all these studies were limited to controlling the piston temperature only. In the present investigation, the mist generation potential of the piston cooling is investigated experimentally and computationally.

NOMENCLATURE

C_p = Specific heat of the oil, J/kgK
d = nozzle diameter, m
T= temperature, ^{0}C
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 = $\frac{\mu C_p}{k_{jet}}$
r = distance from the left hand side of piston.
Re = jet Reynolds number based on the nozzle diameter $= \frac{vd}{\gamma}$
$v = v_{jet(absolute)} - v_{piston}$ = 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)
γ = Kinematic viscosity of the oil, m^2/s
μ = Dynamic viscosity of oil, kg/ms
$\Psi_j(r,z)$ = shape function

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.

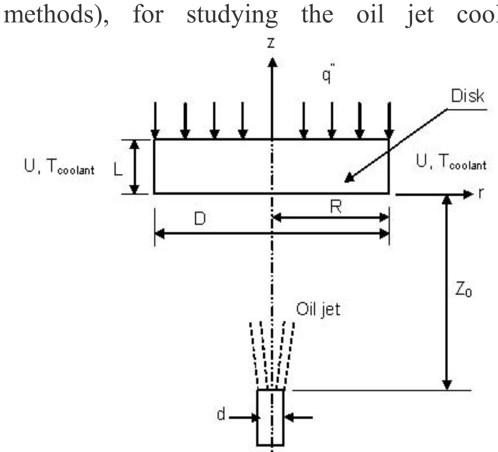


Figure 1: The Coordinate System and Pictorial View of the Notations Used for Oil Jet Cooling

Using the numerical model developed by Stevens and Webb [15, 16], 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 nozzle diameter etc.

In order to understand the oil jet cooling of automotive piston, a computational model was developed for flat plate, Heat equation has been solved in cylindrical coordinates with proper boundary conditions. Finite element method has been applied to find the temperature profile for appropriate heat flux. It is known from the literature that the temperature of the underside of the piston where oil jet impinges, vary between 250 to 350^0 C [17]. Required temperature has been obtained at flat plate by varying heat flux in numerical model and then this is applied in experimental setup to find temperature profile and then investigate the mist generation phenomenon.

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

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

Equation (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 ϕ is the azimuth angle.
b) Heat transfer phenomenon in piston is considered to be steady i.e. $\frac{\partial T}{\partial t} = 0$ where t is the time.

There are four boundary conditions (as in the case of piston) for our computational domain.

1. The top crown in contact with the hot combustion gases, i.e.

$$+k\frac{\partial T}{\partial z} = q^{||} \qquad (2a)$$

2. The sliding lubricated surface in contact with the liner, including the rings, i.e.

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

3. The inside of the piston exposed either to the crankcase atmosphere or to a coolant, usually oil, i.e.

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

4. From the physical and geometrical (axisymmetric)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. The negative sign on the R.H.S. of equation (2b) arise because temperature will be decreasing with increasing r in the disk.

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 [15,16]. 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 1. For other values of d except given in Table 1, a and b are calculated by using linear interpolation between just lower and higher values of d given in Table 1.

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 1: Values of a and b in equation (3b) [15, 16]

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

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

Flat plate Diameter (D)	190 mm
Oil jet impingement distance from BDC (z)	55 mm
Diameter of jet (d)	3 mm
Oil temperature ($T_{oil\ jet}$)	85^0 C
Coolant temperature ($T_{coolant}$)	100^0 C
Oil type	SAE 15W40
Oil flow rate (Q)	8×10^{-5} m^3/s (4.8 litres/minute)
Specific heat (C_p)	2.219 Kj/kgK
Oil thermal conductivity (k_{oil})	0.137 W/mK
Density of oil (ρ)	847 kg/m^3
Kinematic viscosity of oil (v)	14.1×10^{-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
Heat flux (q”)	35 kW/m^2

Table 2: Input Parameters for Numerical Simulation

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

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

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

$$w = \Psi_i \qquad (7)$$

$$\left[\mathbf{K}^e_{ij} + \mathbf{H}^e_{ij}\right]\left\{\mathbf{T}^e_j\right\} = \left\{\mathbf{Q}^e_i\right\} + \left\{\mathbf{P}^e_i\right\} \qquad (8)$$

$$K^e_{ij} = 2\Pi \int_{\Omega^e} \left(k_r \frac{\partial \Psi^e_i}{\partial r}\frac{\partial \Psi^e_j}{\partial r} + k_z \frac{\partial \Psi^e_i}{\partial z}\frac{\partial \Psi^e_j}{\partial z}\right) rdrdz$$

$$H^e_{ij} = 2\Pi \oint_{\Gamma^e} h^e \Psi^e_i \Psi^e_j rds$$

$$Q^e_i = 2\Pi \oint_{\Gamma^e} q_n \hat{\Psi}^e_i rds$$

$$P^e_i = 2\Pi \oint_{\Gamma^e} h^e T^e_\infty \Psi^e_i rds$$

EXPERIMENTAL SETUP

The objective was to investigate the conditions, under which the oil jet cooling of the flat plate start contributing significantly towards the non-tail pipe emissions through mist and smoke generated because of oil jet break up and localized pool boiling. Schematic of the experimental setup is shown in Figure 2.

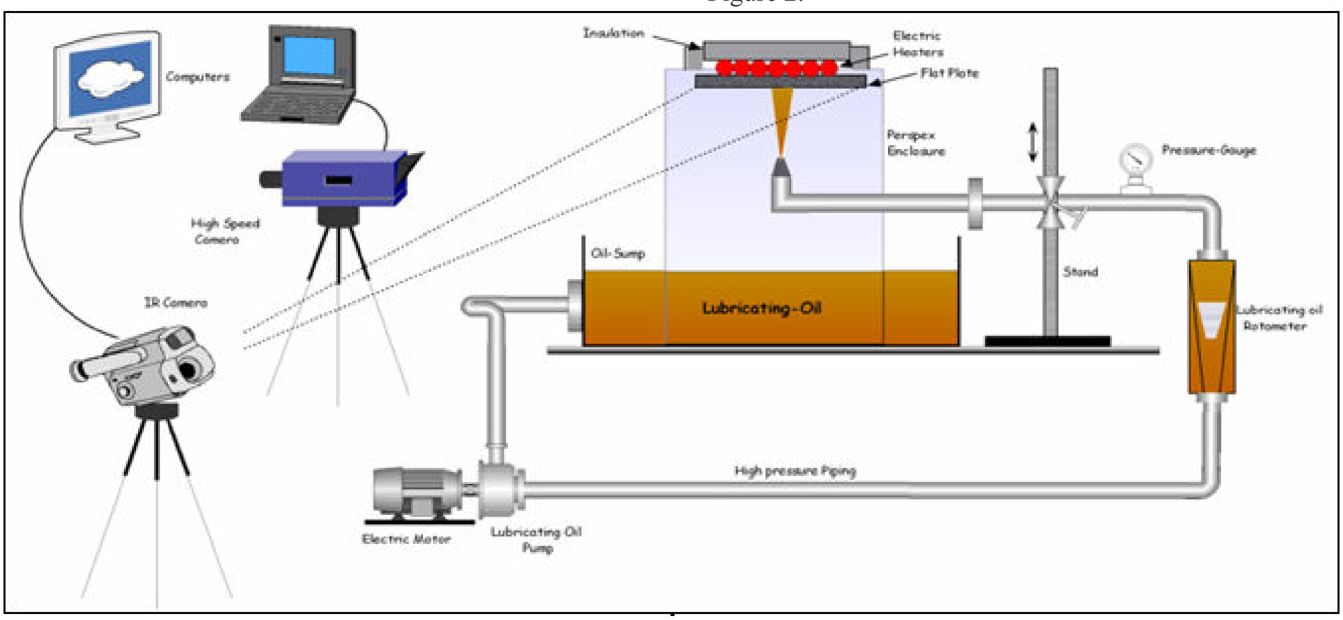


Figure 2: Schematic of Experimental Setup

A high speed CCD camera (BASLER Inc. BCAM v 1.7.0015), has been used to investigate the mist generation and oil jet breakup study the camera is colored CCD camera with a frames rate of 60 frames per second (fps).

All materials, which are above 0 K (-273^0 C), emit infrared energy. The infrared energy emitted from the object is converted into an electrical signal by the imaging sensor (micro bolometer) in the camera and displayed on a monitor as a color or monochrome thermal image. In our experimental setup a thermal infrared imaging camera (Infrared Inc. USA, Model/ Type: CAM/IR 2000) has been used to find the temperature profile of flat plate.

MODEL VALIDATION

For the validation and investigation of the model, the temperature profile of the flat plate is being captured by Infrared camera. Using the model described earlier, isotherms of the predicted temperature profile in the piston have been plotted. The results of simulation and experimental investigations are shown in Figure 3-5. Maximum variation between numerical and experimental value of temperature is around 5%.

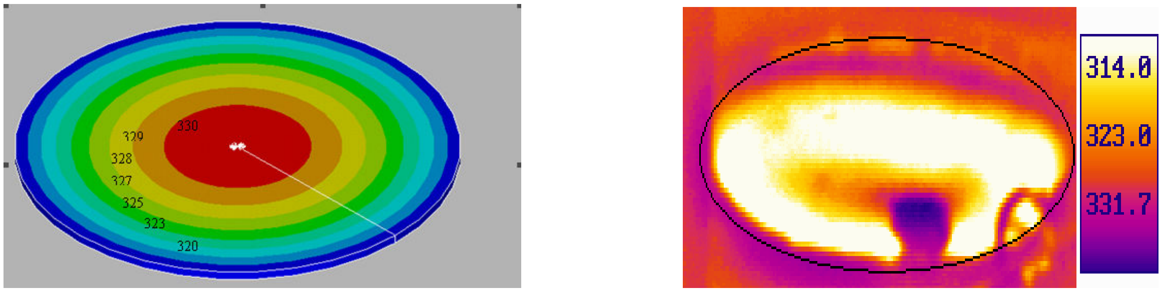


Figure 3: Numerical and Experimental Steady State Temperature Distribution in the Flat Plate with Oil Jet Cooling (Jet Velocity v= 30m/s)

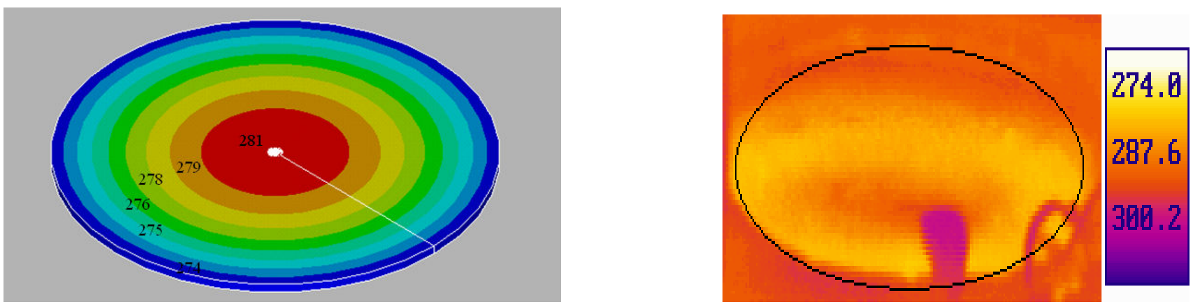


Figure 4: Numerical and Experimental Steady State Temperature Distribution in the Flat Plate with Oil Jet Cooling (Jet Velocity v= 40m/s)

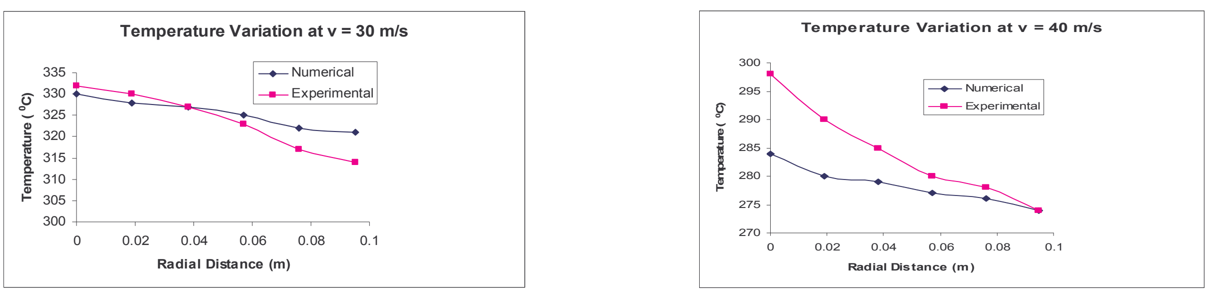


Figure 5: Temperature Variation Radially Outward from Centre (Velocity v= 30m/s and v= 40m/s)

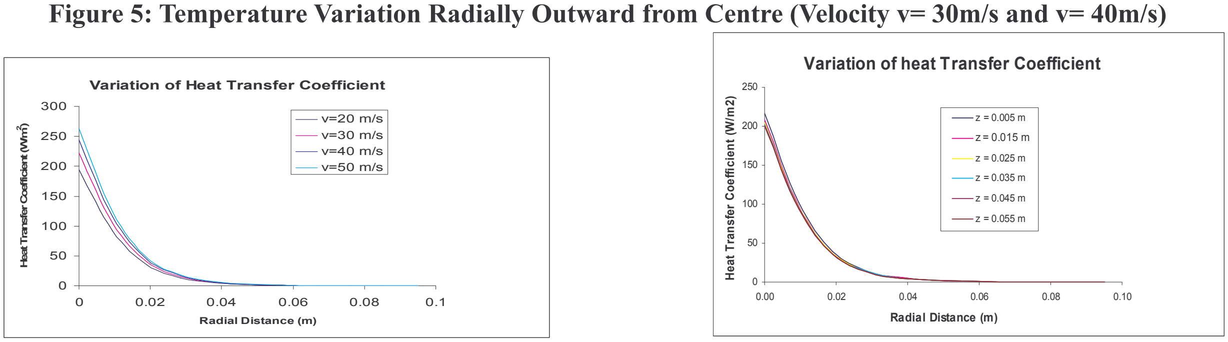


Figure 6: Variation of Heat Transfer Coefficient with Varying jet Velocity and Nozzle Distance from Flat Plate

Heat transfer coefficient at the underside of the flat plate, for the configuration of oil jet cooling is a function of distance of the point under consideration from the point of impingement and relative jet velocity. Heat transfer coefficient at the underside of the flat plate is calculated using Steven and Webb correlation. Variations of heat transfer coefficient from the center in the radially outwards direction for different relative jet velocity and nozzle distance from the flat plate are shown in Figure 6. This Figure reflects that the heat transfer coefficient depends more on relative jet velocity than nozzle distance from the plate. These results validate the simulated model.

EXPERIMENTAL RESULTS AND DISCUSSION

Various experiments have been done and for different nozzle distance from the flat plate and relative jet velocity, temperature profiles captured by Infrared camera, are shown below. Relative jet velocity is varying from 30m/s to 50 m/s in the interval of 10m/s while nozzle distance from flat plate is varying from 0.015m to 0.055m in the interval of 0.020m.

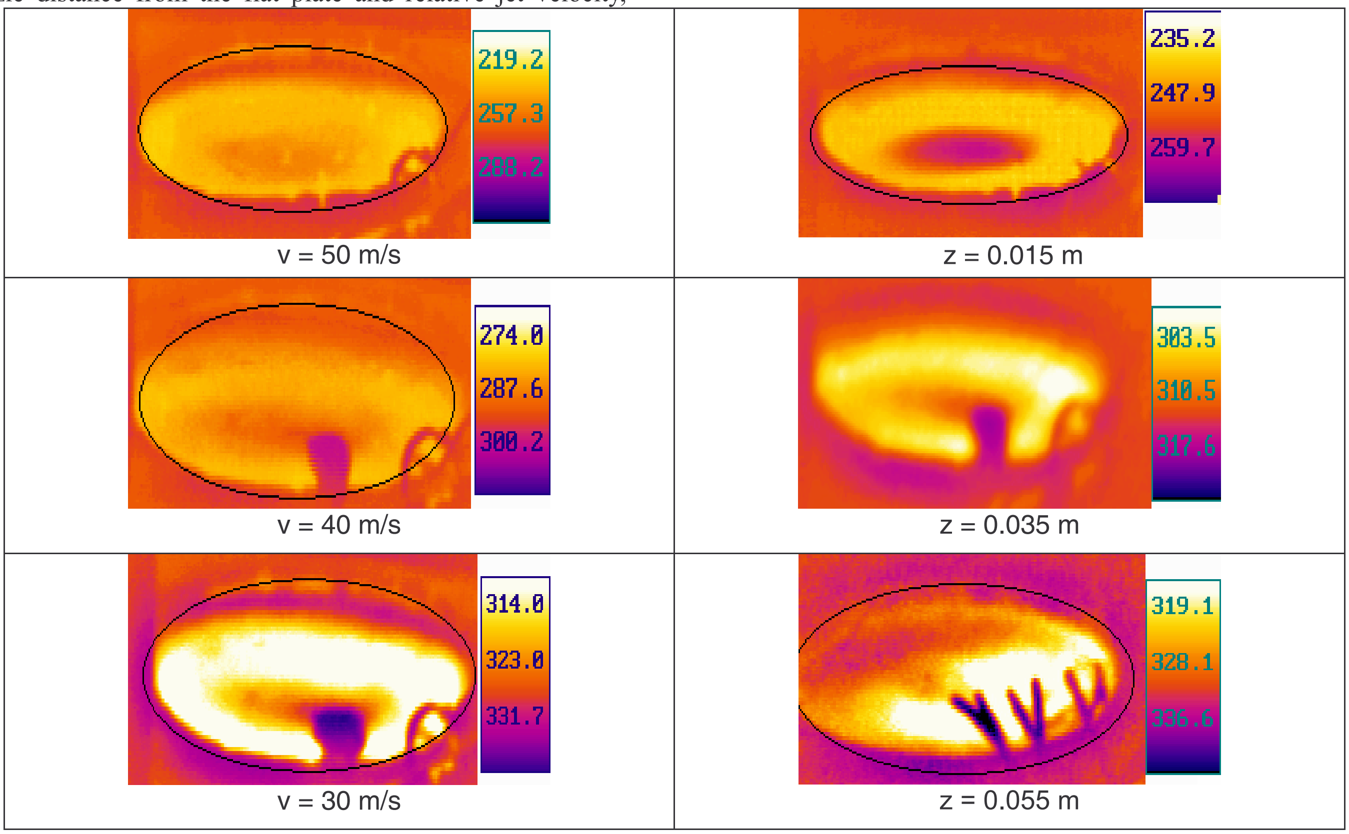
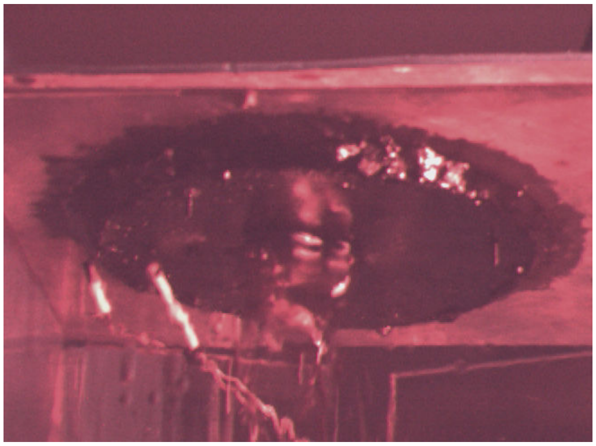


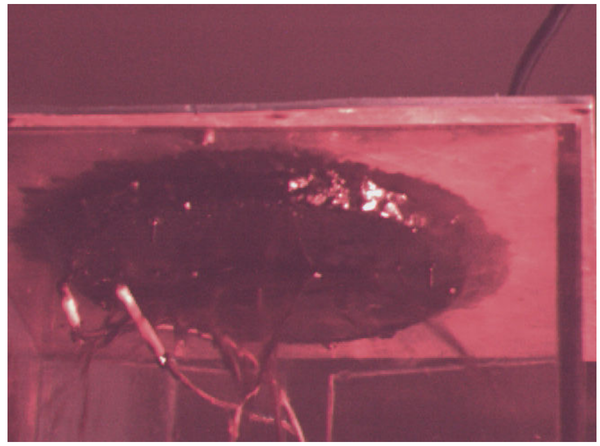
Figure 7: Steady State Temperature Profile at Different jet Velocity and Nozzle Distance from Flat Plate with Constant Heat Flux

Mist generation studies were carried out on the flat plate that was used for experimental validation of the numerical simulator. Various high speed images were taken at different jet velocity and different nozzle distance from flat plate with time.

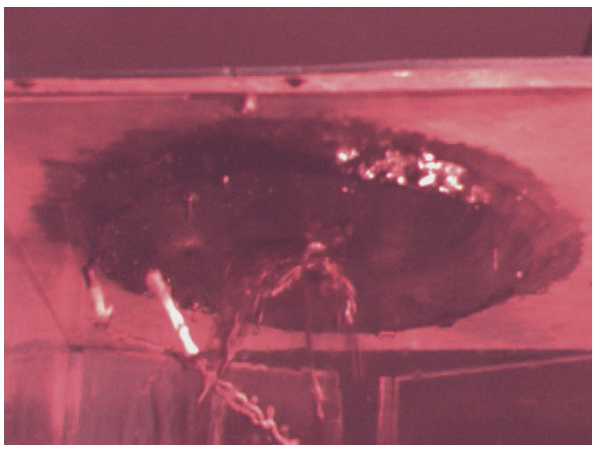
As shown in Figure 8, up to 200^0 C the oil jet was cooling the plate effectively and there is no sign of oil jet breakup and mist. As the plate temperature increased to 250^0 C, the oil jet started breaking into bigger oil droplets after impingement on the hot plate and there was some amount of mist generation observed out of the impingement region on the hot plate. When the jet was fired to the plate at 275^0 C, the jet started breaking into fine droplets and localized boiling of oil at the point of impingement was observed. At this temperature there is change in spray pattern also. The jet is more sprayed out due to the change in viscosity of oil with increase in oil temperature. The temperature of the hot plate was further increased to 300^0 C and a slightly white smoke started coming out of the impingement region along with the fine oil mist. At 325^0 C, large quantity of smoke and oil mist was observed to be coming out of the impingement region on the hot plate. These investigations confirm that the impingement of velocity oil jets on hot piston surface may lead to generation and emission of unburnt hydrocarbon (UBHC) from the crankcase region.



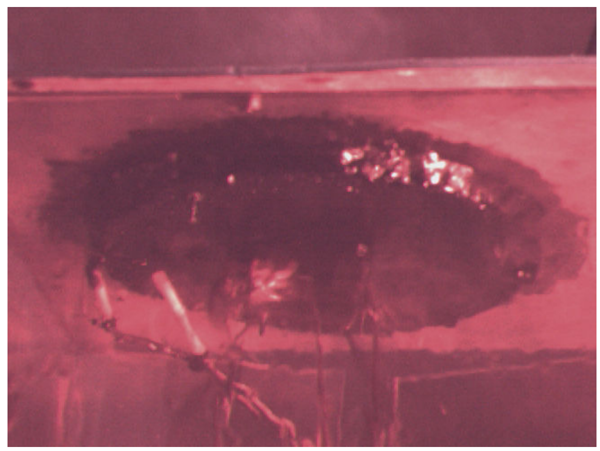
Surface Temperature = 150^{0} C



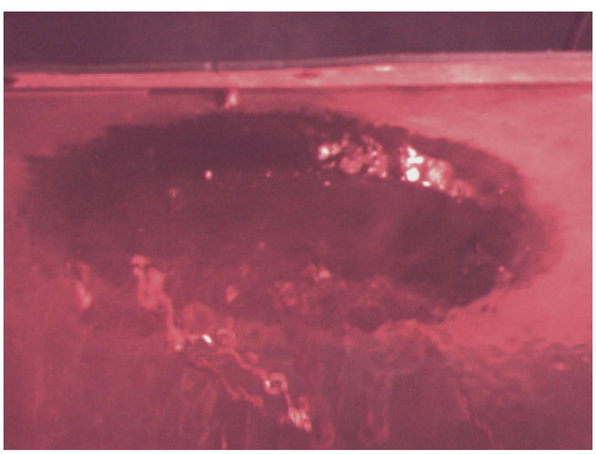
Surface Temperature = 200^{0} C



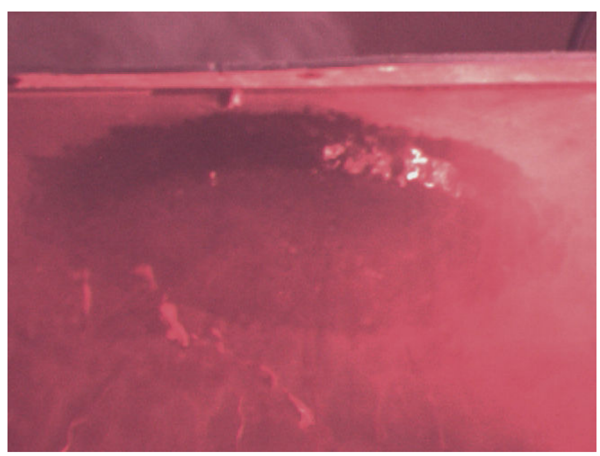
Surface Temperature = 250^{0} C



Surface Temperature = 275^{0} C



Surface Temperature = 300^{0} C



Surface Temperature = 325^{0} C

Figure 8: Mist Generation Study and oil jet breakup at Different Surface Temperature

CONCLUSION

A computational model for temperature prediction of flat plate under oil jet cooling has been developed. Heat transfer coefficient was predicted using Steven Webb relation. This model predicts temperature for different heat flux and desired temperature (between 250^0 C to 350^0 C) is obtained below the flat plate by varying heat flux. Validation of the model is done by flat plate temperature profile using IR Camera. Variation of heat transfer coefficient has been investigated with change in jet velocity and nozzle distance from the flat plate.

Although oil jet cooling leads to increase in power density from the engine yet 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 control the piston temperature, where the mist generation with smoke starts. Mist generation starts at piston temperature 250^0 C and increase with increase in piston temperature. Hence piston temperatures above 250^0 C are undesired and can contribute to emission problem.

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