Numerical Investigations of Piston Cooling Using Oil Jet

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Abstract

Thermal loading of diesel engine pistons has increased dramatically in recent years due to applications of various technologies to meet low emission and high power requirements. Control of piston temperatures by cooling of these 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 due to piston warping. 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 being used, it may contribute to the mist generation. This mist may significantly contribute to the non-tail pipe emissions in the form of unburnt hydrocarbons (UBHC). The problem of non-tail pipe emissions 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 computational fluid dynamics (CFD) tools, such as finite elements methods for studying the oil jet cooling of pistons. Using the numerical model developed by Stevens and Webb (1991), the heat transfer coefficient (h) required at the underside of the piston is predicted. This predicted value of heat transfer coefficient significantly helps in selecting right oil type, jet velocity, jet diameter and distance of the jet from the underside of the piston. It also helps to predict whether the oil selected will contribute to mist generation or not and if it contributes to mist generation then it helps in selecting the oil which does not contribute to mist generation. Grid generation for a production grade M & M DI 2500 engine piston has been done using GNUPLOT. Isotherms of the predicted temperature profiles in the piston have been plotted using TECPLOT.

Introduction

Direct Injection (DI) Diesel engines have an advantage in fuel economy compared with gasoline engines. Diesel engines are eco-friendly and have high potential to meet future exhaust emission regulations because of their lower carbon dioxide (CO_2) emissions. Diesel engines however suffer from the problem of the emission of nitrogen oxide (NO_x) and particle matter (PM). Requirements of higher power and lower fuel consumption still remain in the case of diesel engines installed on commercial vehicles such as trucks and buses. In order to meet these requirements, current diesel engines are required to have boosted turbo-charging, high pressure fuel injection and improvement of airflow in the piston combustion bowl. The current trend in the automobile industry is towards increasing the power density of the engine and making lighter engines. These requirements lead to higher thermal load on the engine, especially on the pistons. In internal combustion engines, the thermal energy is released in the combustion chamber by combustion of the fuels. The combustion gases supply work to the piston, and the residual heat is transferred to the piston body, cylinder liner, valves, cooling fluid, lubricating oil, and exhaust gases. In the piston, the heat flux traverses the rings zone, oil gallery and under crown. Typical experimental values assigned to the rings zone 34.0% of the piston heat transfer, 44.9% to the oil gallery and the remaining to the under crown [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 1. The oil jets hit the hot piston at a very high relative velocity ranging from 5 m/s to 50 m/s. The oil jet breaks into mist, if the temperature at the underside of the piston is above the boiling point of the oil being used to cool the piston. This piston cooling generated mist contributes significantly towards the non-tail pipe emissions from the engine.

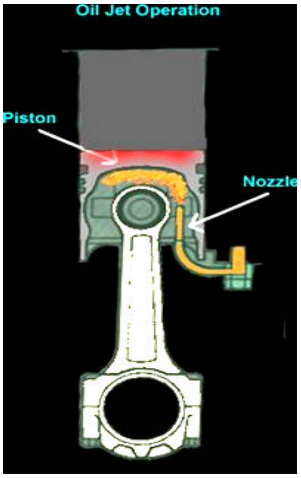

Figure1: Oil Jet Cooled Piston

Historical Perspective

The problem of air pollution created by automotive engines in metropolitan cities has become very severe and requires urgent corrective action. Unburnt hydrocarbons (UBHC) are important pollutants, which are contributed by the following three sources in a petrol engine.

Evaporative Losses	15-25% of HC
Crankcase blow-by	20-35% of HC
Tailpipe exhaust	50-60% of HC

In diesel engines, evaporative losses do not exist, crankcase blow-by is present but contribution made by it is not clearly known. It should be quite significant as pressures developed during combustion and power stroke is quite high. Also, in general, diesel engines are poorly maintained as compared to petrol engines. Most of the big engines are diesel powered, therefore the contribution of hydrocarbon emission from diesel engines is quite large. The control of blow-by emission is quite simple and inexpensive and results in 15-35% reduction in total UBHC emission together with increased lubricating oil change period and decreased deterioration of lubricating oil [2].

UBHC emissions from the diesel engines are mainly contributed by blow-by and the mist generated by oil jet cooling in modern high powered internal combustion engines. The oil jet cooling is an effective way of keeping the piston under-crown surface temperatures under control.

The studies of surface cooling by means of jets were originally conducted aiming the thermal protection of stator and rotor blades of gas turbines. Thus, extensive reviews presented in the literature such as by Martin (1977) refer to gas jets or air jet cooling in air surroundings [3]. Besides the problem of single jet, this study showed results with array of jets, discrete hole injection, and slot injection. Down and James (1987) presented experimental correlations for liquid jets in quiescent air. They presented results from different works for many jet and flow conditions (Reynolds and Prandtl numbers), heating or cooling, liquid or gaseous medium, plane or concave surfaces, and circular array or slot jet. Hrycak (1988) presented studies on the impingement of round jets on flat and concave surfaces with models for turbine blades [4]. Beltaos (1976) analyzed the fluid dynamic behaviour of circular turbulent jet impingement [5]. Sparrow and Lovell (1980) obtained experimental data on jet impingement on surfaces at oblique angles (90° – 30°) [6]. They have observed that the point of maximum Nusselt number (Nu) moves upwards against the flow. However the mean value of the heat transfer coefficient (h) is not affected significantly.

Chang H. Oh et. al designed liquid jet array cooling modules for operation at very high load fluxes and used them to remove fluxes as high as 17 MW/m^2 [7]. The cooling was entirely convective, without boiling. Wen et. al used impingement cooling on a flat surface by using circular jet with longitudinal swirl strips for cooling [8]. Smoke-flow visualization is also used to investigate the behaviour of the complicated flow phenomenon under the swirling-flow jet for this impingement cooling. Oliphant et. al compared liquid jet array and spray impingement cooling in the non-boiling regime experimentally [9]. Cornaro et. al used jet impingement cooling for convex semi-cylindrical surface [10].

On the other hand, studies on cooling of internal combustion engine started in 1960's. Bush has worked for Prof. London at Stanford University (USA) and introduced the term "cocktail shaker" [11]. His interest was on reciprocating pistons with partially filled cavities. After long tests, he presented heat transfer models and governing parameters. His experimental correlations were obtained for liquids with Pr > 0.5 and Pr << 1. Further results were presented by French with several different rig and engine test configurations, and an expression for the heat transfer coefficient was presented [12].

Evans (1977) conducted a more thorough study of the “cocktail shaker” piston cooling concept [13]. Movies of a flow visualization apparatus were taken. This time, an open gallery was used. His main observation was the detection of different flow regimes in the off gallery. Considering the full 360^0 cycle of the piston (crank angle) six regimes were identified. He has modelled the six regimes using known correlations and a numerical method is presented to evaluate the average value of h for each cycle.

Kajiwara et. al calculated the heat transfer coefficient in the cooling gallery of the oil jet cooled piston directly using CFD code [14]. Piston temperature distribution has also been predicted quite accurately by this approach. In order to realize the clean exhaust emission and the customer’s requirements, such as higher power and fuel economy, one of the most effective designs in combustion bowl optimization is the re-entrant shape design. The active airflow and the lower thermal capacity together increase the bowl edge temperature. Therefore, it becomes difficult to secure sufficient reliability and durability of the pistons that have the re-entrant combustion bowl.

Spray impingement cooling research is still being used to a great extent in achieving high heat transfer rates from heating surfaces and are not being extensively used in automobiles currently.

Martins et. al (1993) analyzed the cooling conditions of articulated piston and their impact on the piston performance in an effort to optimize articulate piston cooling [12]. Pimenta et. al used numerical simulation (finite element method) temperature profiles and heat fluxes to study cooling of automotive pistons by investigating liquid cooling jets [1]. Dhariwal investigated blow-by emission and lubricating oil consumption in I. C. engine and tried to control blow-by losses using Positive Crankcase Ventilation (PCV) [2].

Model Development

A numerical model has been developed using computational fluid dynamics (CFD) tools, such as finite elements methods for studying the oil jet cooling of pistons. Using the numerical model developed by Stevens and Webb (1991), the heat transfer coefficient (h) required at the underside of the piston is predicted [15-16]. 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, oil flow, engine speed, oil hole diameter etc. The higher boosted turbo-charging enables the combustion in the high excess air region. It is effective to reduce the NO_x and PM, and can realize high power and fuel economy. However, it causes increase of peak firing pressures and temperature of the parts that compose the power cell such as pistons, piston rings and cylinder liners. Therefore, it becomes difficult to secure sufficient reliability and durability when boosted turbo-charging is used. The increase of the piston temperature causes cavity edge cracking due to increase in the thermal load and reduction in material strength in the case of the aluminium alloy pistons. An excessive temperature of the piston results in piston scuffing, increased blow-by gas by sticking of the piston rings to grooves and an increased oil consumption caused by wear of piston rings and ring grooves. These in turn decrease the reliability and durability of the engine significantly. Therefore, the control of piston temperature by piston cooling becomes important. There are basically two approaches to cope with the increase of thermal load of pistons. One is the improvement of piston cooling ability through the redesign of piston structure. The other is the improvement of material strength in the high temperature region. Figure 2(a) shows the structure of a spray-cooled piston, which is generally used in diesel engines. In this type, cooling oil is sprayed from an oil jet nozzle mounted on the lower deck of the cylinder block, to the backside of the piston crown. Figure 2(b) shows the structure of a cooling gallery type piston, which has higher capacity in piston cooling than those without cooling gallery described above. The latter approach uses high strength materials in the high temperature region of a piston. High strength aluminium alloys formed by changing the chemical ingredients or by using composites with ceramic fibres are some examples. In heavy-duty diesel engines, many ferrous pistons that have higher strength than aluminium alloys are used. Such examples include the articulated pistons, which combine a steel crown with an aluminium skirt, and the nodular cast iron mono-block pistons. Recently, articulated pistons came into serial production for high speed, high output direct injection diesel engines. This piston configuration appears as today’s most suitable design to withstand the new engine performance requirements. Most of the state-of-art of piston cooling technique came out of the need to keep the aluminium pistons structurally suitable to resist the rising engine power, as well as, to control the carbon building up. This fact led piston manufacturers to develop appropriate inner designs suitable to improve piston cooling conditions. Special nozzles were adapted on the engine block to throw cooler oil against the piston under crown and to increase heat exchange. Closed galleries were projected around the combustion bowl in order to remove heat and to decrease the combustion bowl rim and the ring groove zone temperatures [14].

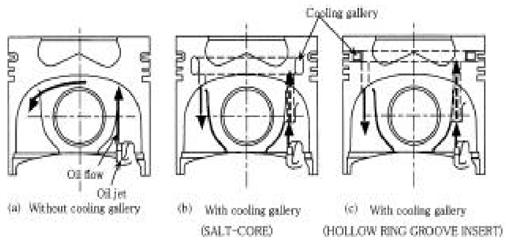

Figure 2: Comparison of Piston Structure [14]

Finding the optimum 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 designs.

The governing differential equation for the piston in cylindrical coordinates is: [17]

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

The boundary conditions (abbreviated as B.C. from now onwards) are:

B. C. 1: at r=0, $\frac{\partial T}{\partial r} = 0$ (axisymmetric) (2a)

B. C. 2: at r=R, $-k\frac{\partial T}{\partial r} = U(T - T_{coolant})$ (2b)

B. C. 3: at z=0, $+k\frac{\partial T}{\partial z} = h(T - T_{oiljet})$ (2c)

B. C. 4: at z=l, $+k\frac{\partial T}{\partial z} = q^{\parallel}$ (2d)

The local heat transfer coefficient at the bottom surface of the disk, h=f(r) which is calculated from equations (3) and (4). 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 [15-16]. The correlations are 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.

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: Values of a and b in equation (3b)

Where Nu = local Nusselt number = hD/k_{jet}

Nu_0 = Stagnation point Nusselt number

h = local heat transfer coefficient (W/m^{2}K) at the bottom surface of the disk

D = diameter of the disk, m

k_{jet} = thermal conductivity of the oil jet (W/mK)

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

where Re = jet Reynolds number based on the nozzle diameter = $\frac{vd}{g}$ (1)

Pr = Prandtl number of the oil jet = $\frac{mC_p}{k_{jet}}$

z_o = vertical distance of the disk from the nozzle exit, m (Figure 3)

d = nozzle diameter, m (Figure 3)

v = $v_{jet(absolute)} - v_{piston}$ = relative jet velocity (averaged over a cycle), m/s

Equation 4 is valid for Re = 4000-52000.

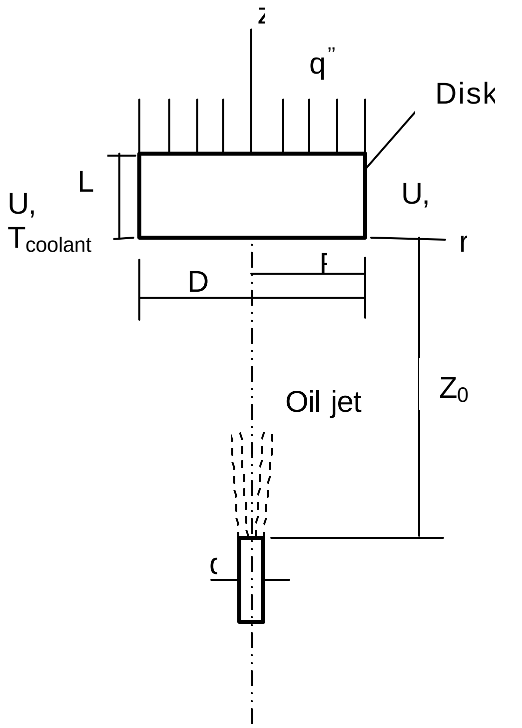

Figure 3: The Coordinate System and Pictorial View of the Notations Used

Numerical Simulation

The variational statement of the governing differential equation is:

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

Using Lagrange interpolation function

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

where $\Psi_j(r,z)$ is the shape function and using the Rayleigh-Ritz method of approximation

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

the final form of the equation in matrix form is

$$\left[K_{ij}^e + H_{ij}^e\right]\left\{T_j^e\right\} = \left\{Q_i^e\right\} + \left\{P_i^e\right\} \quad (8)$$

where

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

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

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

$$P_i^e = 2\Pi\oint_{\Gamma^e} h^e T_\infty^e\Psi_i^e\, r\,ds$$

Input Parameters

The piston used for present investigation is production grade piston from M & M DI 2500 diesel engine. The input parameters for the simulation are as follows:

- Piston Diameter (D): 89 mm
- Impingement vertical distance to BDC (z): 54.65 mm
- Diameter of jet (d): 3 mm
- Oil temperature: 100^0 C
- Oil type: 15W40
- Oil flow rate (Q): $8X10^{-5}$ m^3/sec
- Specific heat (C_p): 2.219 kJ/kg K
- Oil thermal conductivity (k): 0.137 W/mK
- Density of oil: 847 kg/m^3
- Kinematic viscosity: $14.1X10^{-6}$
- Aluminium thermal conductivity: 137 W/mK
- Jet velocity: 20 m/s
- Specific power: 0.35 kW/cm^2

A structured mesh was generated within the piston profile using Transfinite Interpolation method [18]. GNUPLOT is used for mesh generation for the axisymmetric segment of the piston.

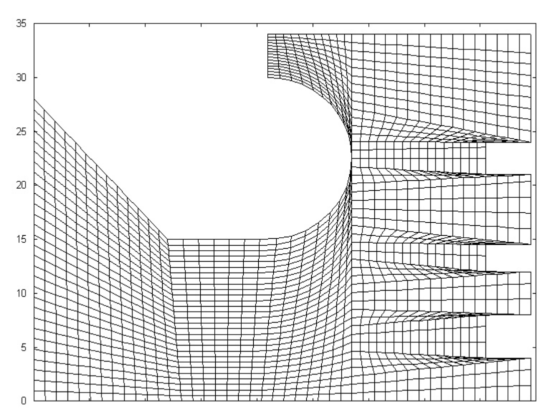

Figure 4: Axisymmetric Mesh Generated within the Piston Profile

Using the model described earlier, isotherms of the predicted temperature profile in the piston have been plotted using TECPLOT. The results of this simulation are shown in Figure 5.

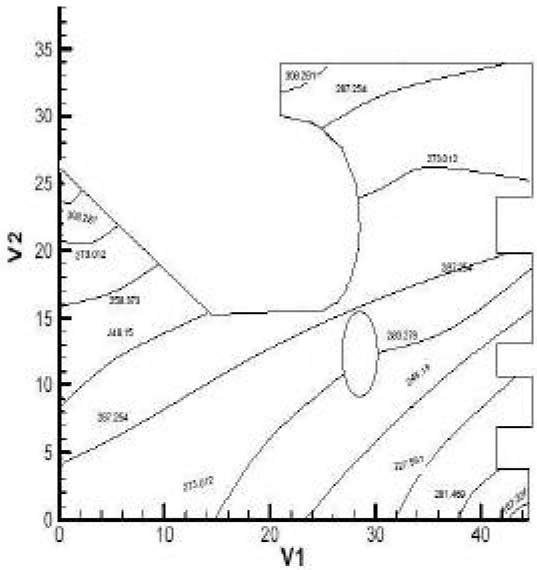

Figure 5: Steady State temperature distribution within the piston

The maximum temperature occurs at the piston top edge. The temperature at the piston top edge is 309^0 C. The temperature at the underside of the piston varies from 288^0 C to 189^0 C. The temperature in the first compression ring groove varies from 273^0 C to 287^0C. The temperature in the second ring groove varies from 248^0 C to 228^0 C. The temperature in the oil ring groove varies from 227^0 C to 201^0 C.

Conclusions

It is clear from the results that the maximum temperature occurs at the edge of the combustion chamber. A CFD code in C language is developed for temperature prediction. Using this numerical simulator, one can back calculate the heat transfer coefficient once temperature profile is known. IIT Kanpur is also involved in experimentally validating the numerical simulation results of oil jet cooling of pistons. This versatile CFD simulator can generate mesh for commercial production grade engines and predict temperature profiles with reasonable accuracy. More testing and further model refinement are presently being conducted.

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