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Numerical investigations of piston cooling using oil jet in heavy duty diesel engines

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Abstract: Thermal loading of diesel engine pistons has increased dramatically in recent years owing to applications of various technologies to meet low emission and high power density 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 excessive piston temperature rise may lead to engine seizure because of piston warping. However, if the temperature at the underside of the piston, where the oil jet strikes the piston, is above the boiling range of the oil being used, it may contribute to the generation of mist. This mist significantly contributes to the non-tailpipe emissions in the form of unburnt hydrocarbons (UBHCs). The problem of non-tailpipe emissions has unfortunately not been looked into so seriously, as the current stress of all automobile manufacturers is to meet the tailpipe emission legislative limits.

A numerical model has been developed using finite element methods for studying the oil jet cooling of pistons and has been validated experimentally on a flat plate cooled by oil jet. Using numerical modelling, the heat transfer coefficient (h) at the underside of the piston is predicted. This predicted value of heat transfer coefficient significantly helps in selecting the correct oil type, oil jet velocity, oil jet diameter, and distance of oil nozzle from the underside of the piston. It also helps to predict whether the selected grade of oil will contribute to mist generation. Isotherms of the predicted temperature profiles in a production grade piston have been plotted.

Keywords: piston cooling, oil jet, non-tailpipe emissions, cooling nozzle, modelling, pool boiling, piston temperature profile, oil grade

1 INTRODUCTION

The current trend in the automobile industry is towards increasing the power density of the engine and making lighter engines. These requirements lead to a higher thermal load on the engine, especially on the pistons. The piston temperature is one of the limiting factors in high-powered internal combustion (IC) engines. This problem is particularly severe in mobile engines, where the space–weight–power ratio is of primary importance [**10**]. In this case, the bore and stroke have to be held to a minimum and the engine speed is limited by piston velocity. The only factor that can be improved is the mean cylinder pressure. An increase in the mean cylinder pressure necessarily means additional heat input into the cylinder, a part of which has to be rejected through the piston and cylinder walls of the combustion chamber. Thus increase in heat flux causes a rise in wall and piston temperatures. The temperatures of certain critical areas in the piston need to be kept below material design limits, e.g. aluminium alloy begin to melt at temperatures greater than 775 K and the melting point of iron is about 1800 K [**2**]. The total heat flow through the piston crown amounts to roughly 2 per cent of the energy released by the fuel consumed. The direct effect of the piston cooling on thermal efficiency of the engine is therefore small. 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 per cent

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Table 1 Heat balance for the piston-cooling load

Location	Dissipation (per cent)
Heat dissipation from the undercrown surface	71
Heat dissipation from the undercrown surface behind the ring–groove pad	11
Heat dissipation from the upper and lower skirt sections	10
Heat dissipation from the rings and lands	8

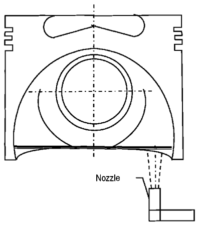


Fig. 1 Oil jet cooled piston

of the total heat input. The heat balance for the piston-cooling load is given in Table 1 [**3**, **4**].

The piston is usually cooled by oil jets fired to the underside from the crankcase in a heavy-duty diesel engine, as shown in Fig. 1. The oil jets hit the hot piston at a very high relative velocity ranging from 5 to 50 m/s. As oil comes in contact with a high-temperature piston, its temperature increases further. High temperature leads to thermo-oxidative stressing of the oil, resulting in deterioration of physical and chemical properties of the oil in the lubrication systems of combustion engines, thus reducing its residual useful life. It not only leads to excessive wear and corrosion but also may cause catastrophic failure of lubricated components. On the other hand, if the surface temperature reaches the oil boiling range, localized pool boiling of the oil around the high-temperature region takes place. The oil jet breaks into mist because of high temperatures at the underside of the piston and high relative jet velocity. This piston-cooling-generated mist contributes significantly towards the non-tailpipe emissions from the engine.

2 METHODS OF PISTON COOLING

The piston can be cooled by oil, water, or air. Water cooling has been applied to heavy, low-speed engines for some time, but more recently it seems to have been abandoned because of the serious design and maintenance difficulties with piping and sealing [**5**]. 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 the design point of view, but the 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. 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, which then can be freely released to return to the crankcase.

There are five distinct methods of oil cooling [**5**].

1. No direct oil supply to the piston. The heat from the piston underside is transferred to the crankcase atmosphere. This method applies to engines having no forced lubrication of the small end of the connecting rod.
2. Cooling due to oil emerging from the small-end bearings, which are pressure lubricated.
3. 'Cocktail shaker', a cooling method applied particularly to 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 overflow is arranged in such a manner that the volume of the chamber is only partly filled with oil. The oil is agitated violently by the pistons reciprocating movement, and the turbulence of oil motion produces a high heat-transfer coefficient between the oil and the piston.
4. Jet cooling, where the oil is released at high pressure from a nozzle at the top of the small end of the connecting rod. The oil jet hits the undercrown of the piston and is then dispersed to splash on the surrounding walls of the piston.
5. Methods 3 and 4 are often combined; i.e. the oil is injected from the top of the connecting rod on to the piston undercrown and then collected in the 'cocktail shaker' chamber, from where it is released through the overflow to the crankcase.

3 HISTORICAL PERSPECTIVE

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

Evaporative losses: 15–25 per cent of HCs
Crankcase blow-by: 20–35 per cent of HCs
Tailpipe exhaust: 50–60 per cent of HCs

In diesel engines, evaporative losses do not exist, crankcase blow-by is present but its contribution is not clearly known. It should be quite significant, as pressures developed during combustion and power stroke are quite high. Also, in general, diesel engines are poorly maintained as compared with 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 a 15–35 per cent reduction in total UBHC emission together with increased lubricating oil change period owing to decreased deterioration of lubricating oil [**6**].

UBHC emissions from 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 undercrown surface temperatures under control.

The studies of surface cooling by means of jets were originally conducted using the thermal protection of stator and rotor blades of gas turbines. Thus, extensive reviews presented in the literature, such as by Martin, refer to gas jets or air jet cooling in air surroundings [**7**]. Besides the problem of a single jet, this study showed results with an array of jets, discrete hole injection, and slot injection. Down and James (1987) presented experimental correlations for liquid jets in quiescent air [**8**]. They presented results from different works for many jet flow conditions (Reynolds and Prandtl numbers), heating or cooling, liquid or gaseous medium, plane or concave surfaces, and circular array or slot jet. Hrycak presented studies on the impingement of round jets on flat and concave surfaces with models for turbine blades [**9**]. Sparrow and Lovell obtained experimental data on jet impingement on surfaces at oblique angles (90–30°) [**10**]. They 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.

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 [**11**]. The cooling was entirely convective, without boiling. Wen and Jang used impingement cooling on a flat surface by using a circular jet with longitudinal swirl strips for cooling [**12**]. 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 [**13**]. Cornaro *et al.* used jet impingement cooling for a convex semi-cylindrical surface [**14**].

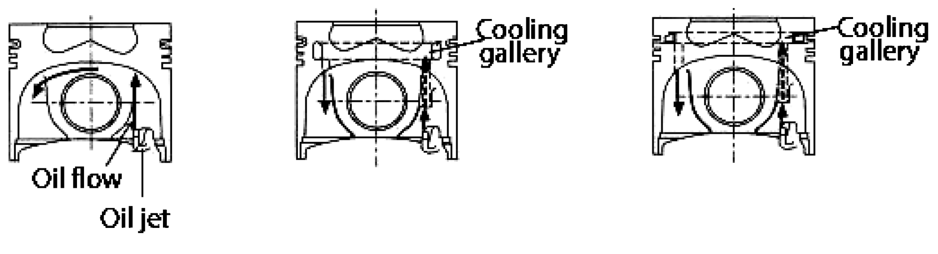
On the other hand, studies on cooling of the internal combustion engine started in the 1960s. Bush worked for Professor London at Stanford University (USA) and introduced the term ‘cocktail shaker’ [**15**]. His interest was on reciprocating pistons with partially filled cavities. After long tests, he presented heat transfer models and governing parameters. His experimental correlations were obtained for liquids with $Pr > 0.5$ and $Pr \ll 1$. Further results were presented by French, with several different rig and engine test configurations, and an expression for the heat transfer coefficient was presented [**16**]. Evans conducted a more thorough study of the ‘cocktail shaker’ piston-cooling concept [**17**]. Film of a flow visualization apparatus was 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° cycle of the piston, six regimes were identified. He modelled the six regimes using known correlations, and a numerical method was 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 a computational fluid dynamics (CFD) code [**18**]. Piston temperature distribution was also predicted quite accurately by this approach. In order to realize clean exhaust emissions and the customers’ 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 re-entrant combustion bowl. Spray impingement cooling research is still being used to a great extent in achieving high heat transfer rates from heated surfaces and is not extensively used in automobiles currently. Leites and De Camargo analysed the cooling conditions of the articulated piston and their impact on the piston’s performance in an effort to optimize articulate piston cooling [**19**]. Pimenta used numerical simulation (finite element method) for predicting temperature profiles and heat fluxes of automotive pistons using liquid cooling jets [**20**]. Dhariwal investigated blow-by emission and lubricating oil consumption in an engine and tried to control blow-by losses using positive crankcase ventilation (PCV) [**6**].

4 MODEL DEVELOPMENT

A numerical model has been developed using CFD tools (finite element methods) for studying the oil jet cooling of pistons. Using the numerical model developed by Stevens and Webb [**21–22**], the heat transfer coefficient (h) required at the underside of the piston is predicted. The heat transfer coefficient in the cooling gallery has a 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 hole diameter, etc. The higher boosted turbocharging enables the combustion in high excess air regions also. It is effective to reduce the NO_x and particulate matter (PM), and can realize high power and fuel economy. However, it causes an increase in 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 have sufficient reliability and durability when boosted turbocharging is used. The increase in the piston temperature causes cavity edge cracking due to an increase in the thermal load and reduction in material strength, in the case of aluminium alloy pistons. An excessive temperature of the piston results in piston scuffing, increased blow-by owing to 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 in the thermal load of pistons. One is the improvement in piston cooling ability through redesign of the piston structure. The other is 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, cooling oil is sprayed from an oil jet nozzle mounted on the lower deck of the cylinder block to the back side of the piston crown. Figure 2(b) shows the structure of a cooling gallery-type piston, which has a higher capacity in piston cooling than those without the cooling gallery described above. The latter approach uses high-strength materials in the high-temperature region of a piston. High-strength aluminium alloys are formed by changing the chemical ingredients or by using composites with ceramic fibres. In heavy-duty diesel engines, ferrous pistons having a 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 monoblock 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 state-of-the-art piston-cooling techniques came from the need to keep the aluminium pistons structurally suitable to resist the rising engine power, as well as to control carbon build up. This fact led piston manufacturers to develop appropriate inner designs suitable for improving piston-cooling conditions. Special nozzles were adapted on the engine block to throw cooler oil against the piston under crown to increase heat transfer. 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 [**18**].

Finding an 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 systems are decided by the amount of piston cooling oil. Therefore, the prediction is needed at the first stage of engine design.

The path of heat flow 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



a) Without cooling gallery b) With cooling gallery (Salt-Core) c) With cooling gallery (Hollow Ring Groove Insert)

Fig. 2 Comparison of piston structures [**18**] 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 [**5**].

The governing differential equation for the piston in cylindrical coordinates (Fig. 3) is given by the following equation [**23**]

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

This two-dimensional governing differential equation is taken from physical and geometrical considerations, assuming that the piston is axisymmetric.

Knowing the boundary conditions, equation (1) 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 (see Fig. 3).

1. The top crown is in contact with the hot combustion gases

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

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

$$-k\frac{\partial T}{\partial z}=U(T-T_{\text{coolant}}) \tag{2b}$$

3. The inside of the piston is exposed either to the crankcase atmosphere or to lubricating oil

$$+k\frac{\partial T}{\partial z}=h(T-T_{\text{oil jet}}) \tag{2c}$$

4. From the physical and geometrical conditions only a half-portion of the piston can be taken for analysis

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

The positive sign on the left-hand side of equations (2c) and (2d) is taken because temperature increases with increasing z on the disc; k (W/m K) 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 for axisymmetric, single-phase, free, round liquid jets impinging normally against a flat uniform heat flux surface [**21**, **22**]. The correlation for local heat transfer coefficient at piston's underside surface,

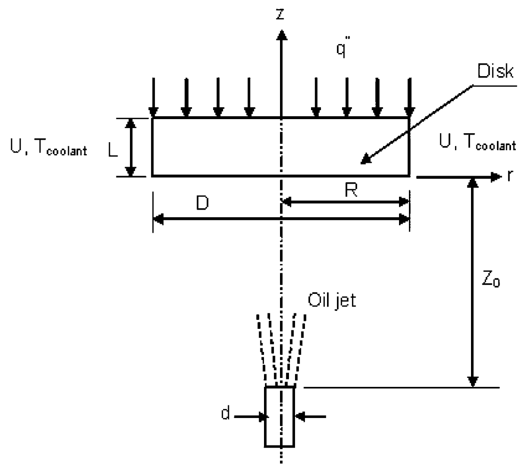


Fig. 3 The coordinate system and pictorial view of the notations used for oil jet cooling

$h=f(r)$, is given by equations (3) and (4)

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

where $f(r/d)=a\mathrm{e}^{b(r/d)}$. The values of a and b are listed in Table 2.

$$Nu_0=2.67\ Re^{0.567}\ Pr^{0.4}\left(\frac{z_0}{d}\right)^{-0.336}\left(\frac{v}{d}\right)^{-0.237} \tag{4}$$

Equation (4) is valid for the range $Re=4000$–$52\,000$.

5 NUMERICAL SIMULATION

The governing differential equation (1) is solved by use of finite element analysis (FEA) methods. The variational statement of the governing differential equation is

$$2\Pi\int_{\Omega^e} w\left[\frac{1}{r}\frac{\partial}{\partial r}\left(k_r r\frac{\partial T}{\partial r}\right)+\frac{\partial}{\partial z}\left(k_z\frac{\partial T}{\partial z}\right)\right]r\,\mathrm{d}r\,\mathrm{d}z=0 \tag{5}$$

Using the Lagrange interpolation function gives

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

Using the Rayleigh–Ritz method of approximation

Table 2 Values of a and b in equation (3) [**21**, **22**]

d (mm)	a	b
2.2	1.13	−0.23
2.3	1.141	−0.2395
4.1	1.34	−0.41
5.8	1.48	−0.56
8.9	1.57	−0.7

gives

$$w = \Psi_i \tag{7}$$

The final form of equation (5) in matrix form [**24**] is

$$[\mathbf{K}_{ij}^{\mathrm{e}} + \mathbf{H}_{ij}^{\mathrm{e}}]\{\mathbf{T}_j^{\mathrm{e}}\} = \{\mathbf{Q}_i^{\mathrm{e}}\} + \{\mathbf{P}_i^{\mathrm{e}}\} \tag{8}$$

where

$$\mathbf{K}_{ij}^{\mathrm{e}} = 2\Pi \oint_{\Omega^{\mathrm{e}}} \left(k_r \frac{\partial \Psi_i^{\mathrm{e}}}{\partial r} \frac{\partial \Psi_j^{\mathrm{e}}}{\partial r} + k_z \frac{\partial \Psi_i^{\mathrm{e}}}{\partial z} \frac{\partial \Psi_j^{\mathrm{e}}}{\partial z} \right) r \, \mathrm{d}r \, \mathrm{d}z$$

$$\mathbf{H}_{ij}^{\mathrm{e}} = 2\Pi \oint_{\Gamma^{\mathrm{e}}} h^{\mathrm{e}} \Psi_i^{\mathrm{e}} \Psi_j^{\mathrm{e}} r \, \mathrm{d}s$$

$$\mathbf{Q}_i^{\mathrm{e}} = 2\Pi \oint_{\Gamma^{\mathrm{e}}} q_n \hat{\Psi}_i^{\mathrm{e}} r \, \mathrm{d}s$$

$$\mathbf{P}_i^{\mathrm{e}} = 2\Pi \oint_{\Gamma^{\mathrm{e}}} h^{\mathrm{e}} T_\infty^{\mathrm{e}} \Psi_i^{\mathrm{e}} r \, \mathrm{d}s$$

6 INPUT PARAMETERS

The piston used for the present investigation is a production grade piston from a medium duty direct injection, transportation diesel engine (make: Mahindra and Mahindra, India; model: MDI2500). The input parameters for the simulation are as follows:

Piston diameter (D): 89 mm
Oil nozzle distance from BDC (bottom dead centre) (z): 55 mm
Diameter of oil nozzle (d): 3 mm
Oil temperature: 100 °C
Oil type: 15W40
Oil flowrate (Q): 8×10^{-5} m^3/s
Specific heat of the oil (C_p): 2.219 kJ/kg K
Oil thermal conductivity (k): 0.137 W/m K
Density of oil (ρ): 847 kg/m^3
Kinematic viscosity (γ): 14.1×10^{-6} m^2/s
Aluminium thermal conductivity (k): 137 W/m K
Jet velocity (v): 20 m/s
Specific heat loss from the piston (q''): 35 kW/m^2

The flowchart of the numerical model is shown in Fig. 4.

A structured mesh was generated for the piston profile using the transfinite interpolation method [**25**]; 770 quadrilateral elements were taken for analy-

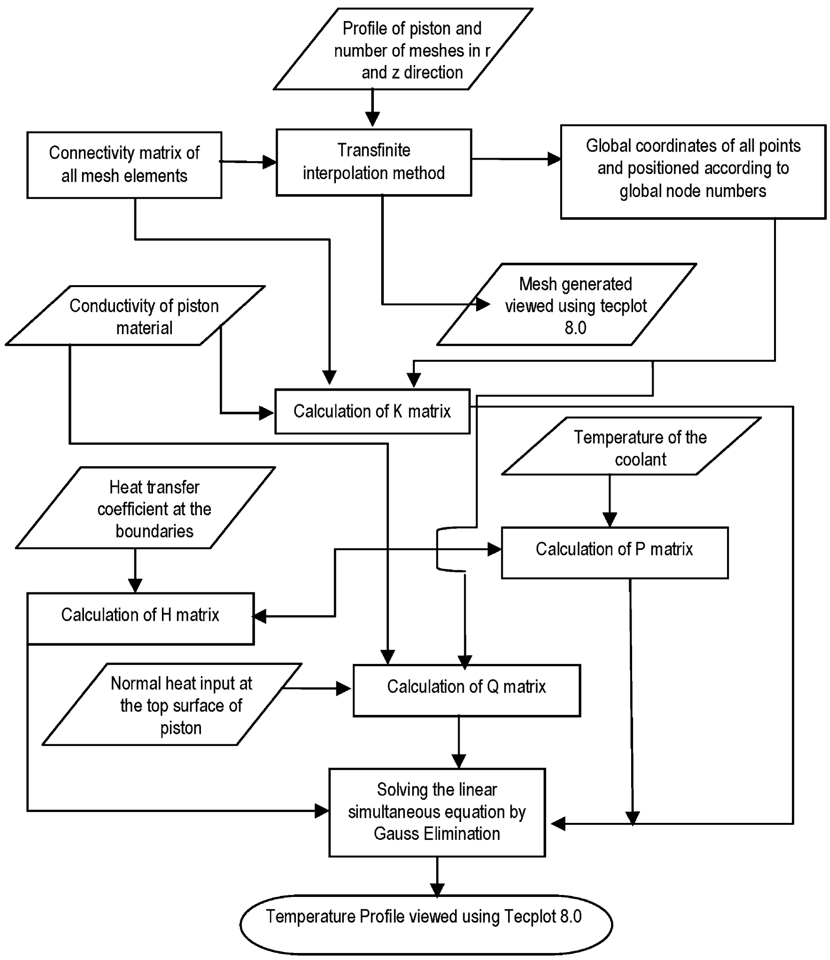


Fig. 4 Schematic of numerical model

sis after conducting grid-independence test. 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 Fig. 5.

7 EXPERIMENTAL SET-UP

The objective of setting up an experiment was to validate experimentally the computational model for oil jet cooling of a flat plate. The schematic of the experimental set-up is shown in Fig. 6. The experimental set-up is housed in 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 the velocity of the jet. A rotameter is connected in the line to measure the oil flowrate. The rotameter is capable of measuring a flowrate of 3–30 litres per minute. A pressure gauge was connected to measure the oil pressure.

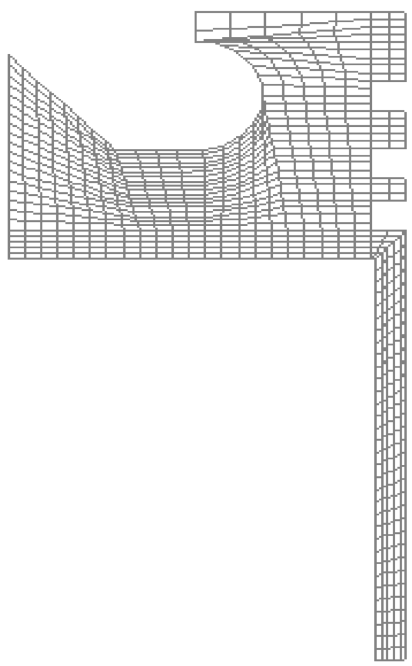


Fig. 5 Axisymmetric mesh generated for the piston profile

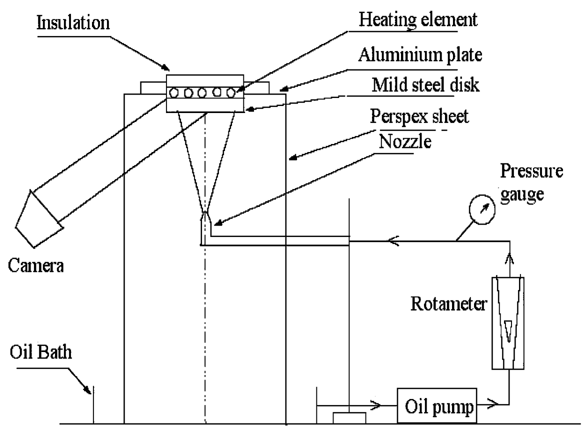


Fig. 6 Schematic of the experimental set-up

8 EXPERIMENTAL VALIDATION OF MODEL

The nozzle location was kept constant, and validation of the model for a flat plate was carried out for different temperatures at various distances from the centre of the flat plate, as shown in Fig. 7.

The oil jet velocity was kept constant, and validation of the model for the flat plate was carried out for different temperatures at various distances from the centre of the flat plate, as shown in Fig. 8. A maximum percentage difference between the numerical and experimental values is 5 per cent, whereas

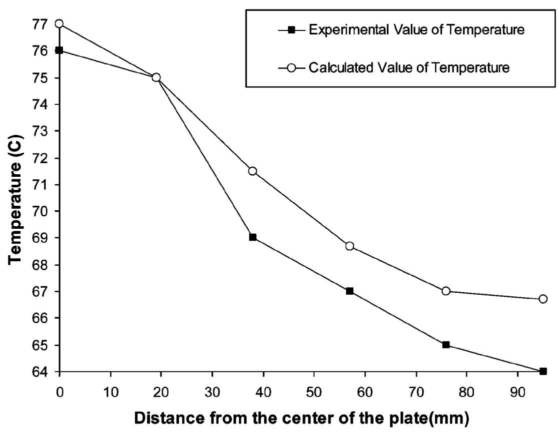


Fig. 7 Temperature variation along the distance from the centre of the plate while keeping the nozzle location constant

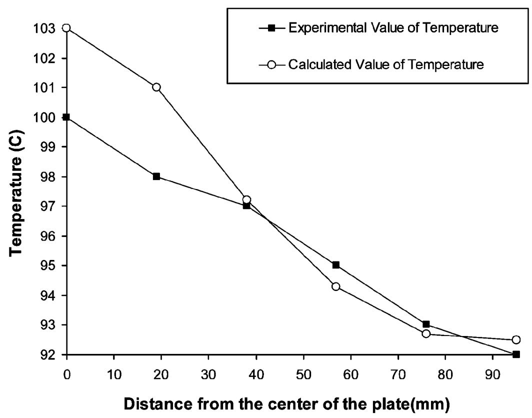


Fig. 8 Temperature variation along the distance from the centre of the plate while keeping oil jet velocity constant

the minimum percentage difference between the numerical and experimental values is 0.09 per cent. Most of the differences are of the order of 1–2 per cent. The average percentage difference between the numerical and experimental values is 1.6 per cent.

This experiment validated the model and confirmed that the model is able to predict the heat transfer coefficient and the temperature profile of the piston with reasonable accuracy. Hence this CFD simulator can be used for further analysis of various engines and piston conditions using parametric variations.

9 RESULTS AND DISCUSSIONS

After successful validation of the model, isotherms of the predicted temperature profile in the piston have been plotted using Tecplot v. 8.0 for both cases, both without oil cooling and with oil cooling at the underside of the piston. The results of this simulation are shown in Figs 9 and 10.

It is observed that the piston temperatures are generally reduced by approximately 40 °C by using oil jet cooling. These results match reasonably well with the experimental data available in the literature [**1**, **3**, **5**]. The temperature is highest at the piston centre (Fig. 11) when oil jet cooling is not employed, and reduces radially outwards towards the skirt.

The nozzle distance from the BDC is varied, and its effect on the piston temperature is examined. The results are represented in Fig. 12. The result shows that the piston cooling is improved as the nozzle distance decreases from BDC.

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

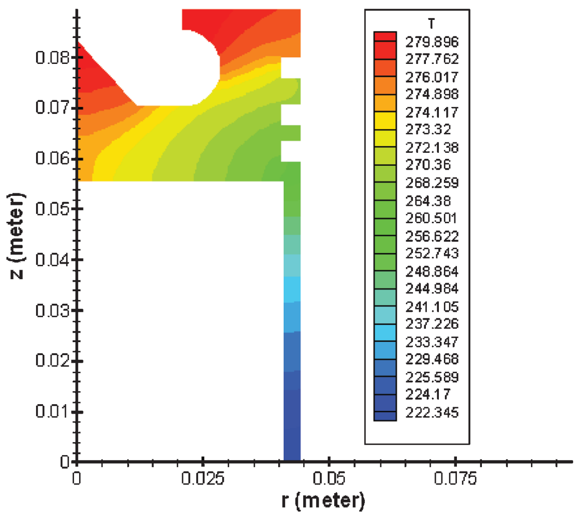


Fig. 9 Steady state temperature distribution in the piston without oil jet cooling

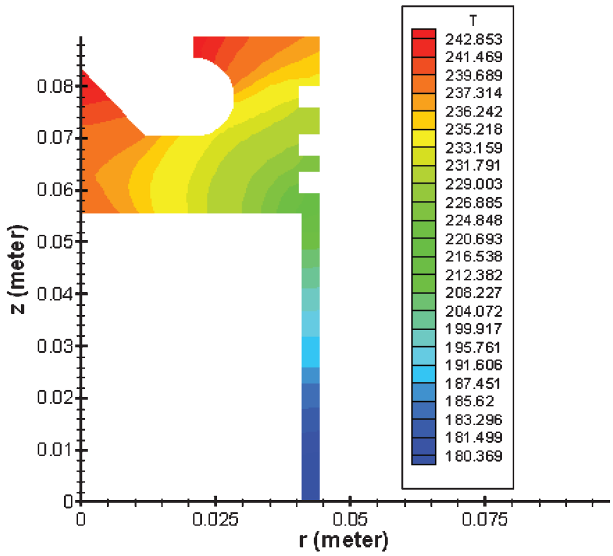


Fig. 10 Steady state temperature distribution in the piston with oil jet cooling

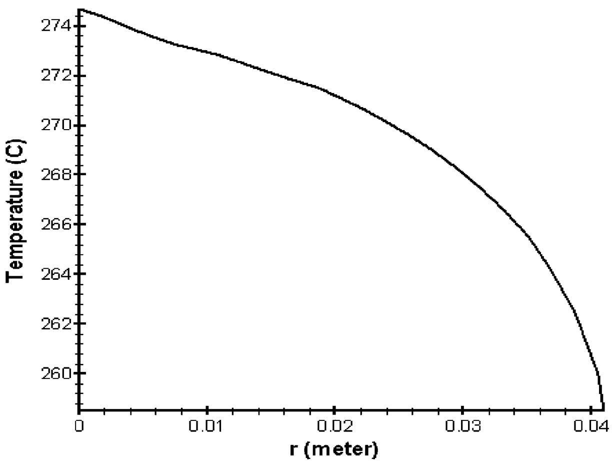


Fig. 11 Variation in temperature from the centre radially outwards without oil jet cooling

In addition, the effect of oil grade on the piston temperature is investigated and the results are shown in Fig. 14. The results show enhanced cooling with decrease oil viscosity (lower grade oil).

The maximum temperature occurs at the centre of the piston top surface. The maximum temperature at the piston top is approximately 281 °C without oil cooling while the temperature with oil cooling is 243 °C at the same location. The temperature at the underside of the piston varies from 275 to 257 °C without oil jet cooling and from 237 to 212 °C with oil jet cooling. The temperature in the first compression ring groove varies from 275 to 271 °C without oil jet cooling, while the temperature with oil jet

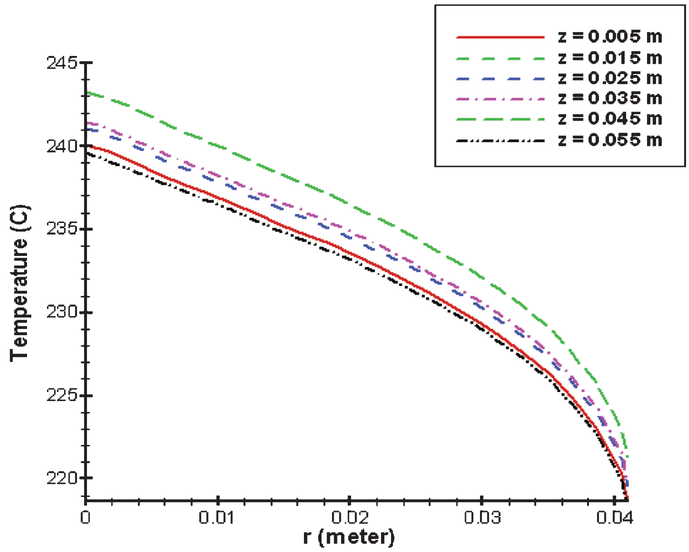


Fig. 12 Variation in temperature from the centre radially outwards for different nozzle distances from BDC

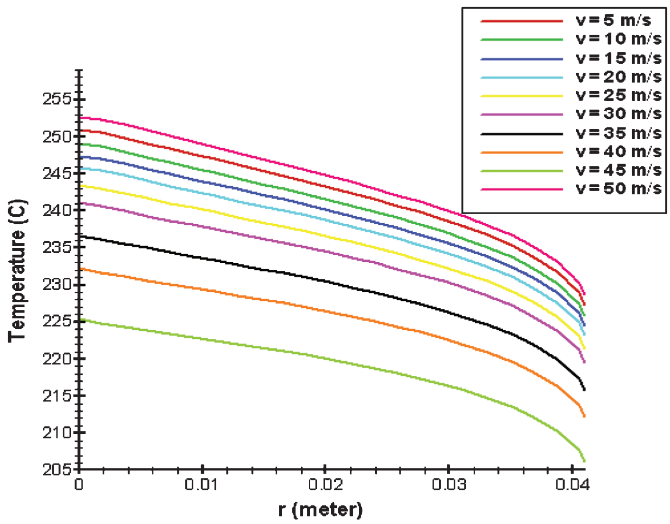


Fig. 13 Variation in temperature from the centre radially outwards for different relative jet velocities

cooling varies from 235 to 231 °C. The temperature in the skirt varies from 252 to 222 °C without oil jet cooling, while the temperature with oil jet cooling varies from 216 to 180 °C.

10 CONCLUSION

A model for oil jet cooling of the pistons from a heavy-duty diesel engine was developed using the FEA method and this model was validated using a flat plate cooled by oil jet. The transfinite interpolation method is used for grid generation of the piston. Heat transfer coefficients were predicted using a Steven-Webb correlation. Temperature profiles of the piston were predicted. A CFD code in the C language was developed for temperature prediction. This versatile CFD simulator can generate mesh for commercial production grade engines and predict temperature profiles with reasonable accuracy. The maximum temperature on the piston surface occurs at the centre of the top surface of the piston. The numerical investigations of heat transfer conditions with oil jet cooling of the piston have produced quantitative results of piston temperatures. A temperature difference of approximately 40 °C was predicted between oil jet cooled and non-cooled pistons. This numerical investigation confirms that

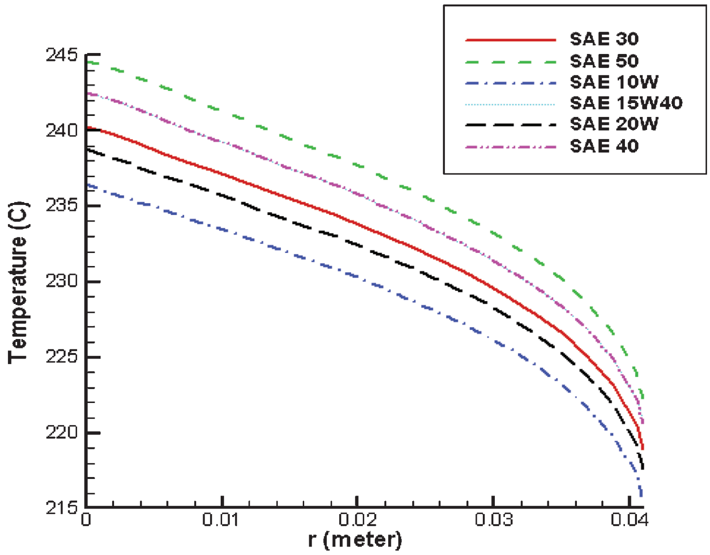


Fig. 14 Variation in temperature from the centre radially outwards for different oils

the effective control of piston temperature is essential for performance improvement and non-tail pipe emission control from heavy-duty diesel engines.

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APPENDIX

Notation

C_p	specific heat of the oil (J/kg K)
d	nozzle diameter (m)
D	diameter of the disc (m)
h	local heat transfer coefficient (W/m^2 K) at the bottom surface of the disc
k_{jet}	thermal conductivity of the oil jet (W/m K)
k_r	conductivity of the material in the r direction
k_z	conductivity of the material in the z direction
Nu	local Nusselt number $= hD/k_{\text{jet}}$
Nu_0	stagnation point Nusselt number
Pr	Prandtl number of the oil jet $= \mu C_p/k_{\text{jet}}$
r	distance from the left-hand side of the piston
Re	jet Reynolds number based on the nozzle diameter $= vd/\gamma$
v	relative jet velocity (averaged over a cycle) (m/s) $= v_{\text{jet(absolute)}} - v_{\text{piston}}$
w	weight function
z	distance from the underside of the piston
z_0	vertical distance of the disc from the nozzle exit (m) (Fig. 3)
γ	kinematic viscosity of the oil (m^2/s)
μ	dynamic viscosity of oil (kg/m s)
$\Psi_j\ (r, z)$	shape function

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