

Experimental Investigations Into Free-Circular Upward-Impinging Oil-Jet Heat Transfer of Automotive Pistons

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ABSTRACT

The purpose of this research was to measure and correlate the area-average heat transfer coefficients for free, circular upward-impinging oil-jets onto two automotive pistons having different undercrown shapes and different diameters. For the piston heat transfer studies, two empirical area-average Nusselt number correlations were developed. One was based on the whole piston undercrown surface area with the Nusselt number based on the nozzle diameter, and the other was based on the oil-jet impingement area with the Nusselt number based on the oil-jet effective impingement diameter. The correlations can predict the 95% and 94% of the experimental measurements within 30% error, respectively. The first correlation is simpler to use and can be employed for cases in which the oil jet wets the whole piston undercrown. The latter may be more useful for larger pistons or higher Prandtl number conditions in which the oil jet wets only a portion of the undercrown. Furthermore, these correlations are suggested to be applicable in numerical studies of piston oil-jet cooling. Validations by using steady-state numerical simulations have also been performed and showed very good agreements between the simulated and measured results.

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INTRODUCTION

In recent years, in order to improve the fuel economy of passenger cars and light trucks, the trend of automotive engine development has been to reduce the average engine displacement size and/or incorporate turbocharging. Both the downsizing and turbocharging engine designs can effectively improve fuel efficiency, however, applying these methods will significantly increase the power density of an engine. This increase in the engine power density increases component thermal loading (i.e. higher combustion pressures and temperatures) in these more compact design engines. Each engine component has a metallurgical temperature limit, especially pistons, which are mainly made of lightweight aluminum alloys with relatively low critical-temperature limits. The hottest area inside an engine is the combustion chamber, where the piston is located at the bottom of the combustion chamber and is one of few components surrounding the combustion chamber that is not directly cooled by the standard engine cooling system. As a result, piston durability/reliability issues are directly impacted by the increasing engine thermal loading conditions.

In traditional automotive engines, about 80% of the piston heat exits through the sides of the pistons through the rings, skirts and oil-films to the cylinder walls, and the rest of the heat transfers from the piston undercrown components (undercrown surface, wrist pin, connecting rod, etc.) to the splashing of engine oil [1]. In order to remove the

additional heat generated by modern high specific power engines, a supplemental cooling mechanism is needed. Pachernegg [2] noted that applying a forced fluid convection to the underside of the piston can efficiently reduce the overall piston temperature. Consequently, impinging oil-jet cooling to the underside of a piston was the topic of several research investigations.

Although this relatively new piston cooling technique is becoming common in automotive applications, there are parasitic pumping losses that need to be optimized. A better understanding of the convective heat transfer between the piston undercrown surface to the impinging oil-jet is necessary to improve this cooling technique. Many studies have presented the convective heat transfer coefficients of low Prandtl number ($Pr < 10$; such as air and water) fluid jets downward-impinging onto flat plates; a few worked on high Prandtl number ($Pr > 100$; such as lubrication oil and coolant) fluid jets. However, these convective heat transfer correlations cannot be adequately used for piston cooling under engine operating conditions due to the mismatch of boundary conditions or parameter ranges, especially the jet impingement direction, the Prandtl number range and the complex geometry of the piston undercrown surface. To address that lack of appropriate heat transfer correlations for upward oil jets on automotive pistons, this study covered a wide range of parameters appropriate to a piston oil-jet cooling at typical engine operating conditions, and tested two different production pistons. The following section gives an overview of some history of piston oil-jet cooling.

PISTON OIL-JET COOLING

History

As one of the engine components facing the highest thermal loading, engineers and researchers have looked for and studied better methods for piston cooling. Given the limited ability to cool through the piston rings and skirt areas, engine producers have relied on oil splash and, more recently direct oil-jet cooling from a fixed nozzle underneath the piston and targeting at the piston undercrown. [Figure 1](#) illustrates oil-jet cooling and also shows the complexity of the piston undercrown surface and the off-center oil-jet targeting area that is necessary to avoid jet interruptions by the connecting rod.

One of the earlier studies on the oil-jet piston cooling was done by Sanders and Schramm [3], who estimated the piston undercrown convective heat transfer coefficient at one engine speed using temperatures estimated from piston hardness measurements. An experimental study was performed by Stotter [4], in which the effectiveness of oil jet cooling was demonstrated and the heat transfer coefficient was plotted against a function of the oil mass flow rate, thermal conductivity and viscosity. The first available heat transfer correlation specific to piston oil-jet cooling, shown in [Equation 1](#), dates back to 1970, and was suggested by Seale and Taylor [5]. See also Rakopoulos et al. [6], for medium-size direct-injection diesel engines. The heat transfer coefficient, h , was given as

$$h \left(\frac{W}{m^2 K} \right) = 5.3 \left\{ \left[\frac{B(m)}{2} \right] \left[\frac{N(rpm)}{60} \right] \left[\frac{d(m)}{v(m^2/s)} \right] \right\}^{\frac{1}{2}} \quad (1)$$

where B is the engine bore, N is the engine speed, d is the oil-jet nozzle diameter and v is the oil kinematic viscosity at the oil bulk temperature. Unfortunately, the Seale-Taylor correlation is very limited for general engine operating conditions because it did not account for all the factors that affect the jet impingement heat transfer process (e.g., Reynolds number, Prandtl number, nozzle-to-surface distance and viscosity ratio based on piston and sump temperatures).

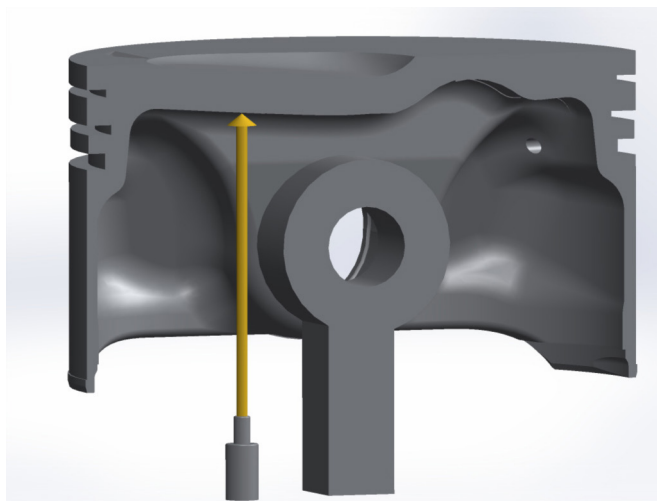


Figure 1. Oil-Jet Impingement to a Piston Undercrown Surface

Another study, by Mavropoulos [7], also provided a correlation for the convective heat transfer coefficient between the oil-jet and the piston undercrown surface and is given in [Equation 2](#),

$$h \left(\frac{W}{m^2 K} \right) = 68.17 \left\{ [r_c(m)] \left[\omega \left(\frac{rad}{s} \right) \right] \left[\frac{d(m)}{v(m^2/s)} \right] \right\}^{\frac{1}{2}} \quad (2)$$

where r_c is the crank radius and ω is the engine angular speed. In [Equation 2](#), the crank radius is related to the nozzle-to-surface spacing, and the rest of terms appear to be very similar to the Reynolds number. However, some important parameters are still missing such as the Prandtl number, diameter of the target surface and viscosity ratio of the oil and piston and sump temperatures.

The effectiveness of oil-jet piston cooling in engine operating conditions was experimentally investigated by the following researchers, Sanders and Schramm [3], Minami and Mukohara [8], Seale and Taylor [5] and Schelling [9]. Additionally, Luff et al. [10] recently investigated both the influence of piston jet cooling on engine emissions and fuel consumptions. Correspondingly, numerical studies of the impinging oil-jet piston cooling were performed by Pimenta and Filho [11], Varghese et al. [12, 13], Agarwal et al. [14, 15, 16], Izadi et al. [17], Mikhaylov et al. [18] and Najafabadi et al. [19]. Given the lack of a heat transfer correlation appropriate for pistons that covers the important parameter ranges, most of these numerical studies used the stagnation Nusselt number correlation developed by Stevens and Webb [20] for a downward-directed water-jet impinging on a flat surface, given in [Equation 3](#),

$$Nu_o = 2.67 Re_d^{0.567} Pr^{0.4} \left(\frac{Z}{d} \right)^{-0.0336} \left(\frac{U}{d} \right)^{-0.237} \quad (3)$$

where Z is the distance between nozzle and surface and U is the mean jet velocity at nozzle exit. [Equation 3](#) was developed for a low viscosity liquid-jet downward-impinging onto a relatively small flat surface, and so the study lacked the proper ranges of parameters that would be relevant to piston oil-jet cooling at engine operating conditions. Additionally, there is a dimensional jet velocity term in this correlation, which could have significant effect when applying it to a higher viscosity fluid impinging with different sizes of nozzles.

Metzger et al. [21] are believed to be the first to report the effects of high Prandtl number (ranging from 3 to 151) on liquid-jet impingement heat transfer. Their tests were run at steady-state conditions for nominally constant values of the adiabatic wall temperatures (discussed later), which were measured by impinging fully developed fluid-jets onto the insulated target surfaces ($D = 12.7, 25.4, 50.6$ and 96.3 mm). Metzger et al. [21] presented an area-average Nusselt number correlation for synthetic-base lubrication oil-jets downward-impinging on heated flat surfaces, as shown in [Equation 4](#),

$$Nu_d = 2.65 Re_d^{0.47} Pr^{0.24} \left(\frac{D}{d} \right)^{-0.68} \left(\frac{\mu_w}{\mu_o} \right)^{-0.37} \quad (4)$$



Despite the large amount of research on free-circular liquid-jets, many of the parameters that had been studied and correlated do not lend themselves to the range of parameters that are relevant to the oil-jet piston cooling in an engine, and no studies were performed on oil-jets upward-impinging to the piston undercrown surfaces of real engines. That is due to the difficulty of measuring the surface convective heat transfer from a complex piston undercrown surface to an upward oil-jet impingement. In this study, laboratory experiments were performed to calculate the piston undercrown heat transfer coefficients on two stationary production engine pistons with three different oil-jet nozzles.

Correlations of Steady State Upward Oil-Jet Impingement Heat Transfer

According to the authors' earlier publication [22], which was based on the first portion of experimental results from this study, Equation 5 can be used to represent the average Nusselt number correlation for the upward-impinging oil-jets striking onto a heated piston undercrown surface, for experimental test conditions relevant to firing conditions of a production engine.

$$Nu_{d,B}^p = 13.932 Re_{d,e}^{0.596} Pr^{0.239} \left(\frac{Z}{d}\right)^{0.089} \left(\frac{B}{d}\right)^{-1.398} \left(\frac{\mu_w}{\mu_{aw}}\right)^{0.312} \quad (5)$$

Figure 2 shows a comparison between the predicted and measured area-average Nusselt numbers by using Equation 5, which shows a very good result for a single production piston ($B = 92$ mm) and two commercial nozzles ($d = 1.13$ mm and 1.70 mm). Equation 5 is able to predict 100% of 130 experimental data points within $\pm 30\%$ error and 97% within $\pm 15\%$ error.

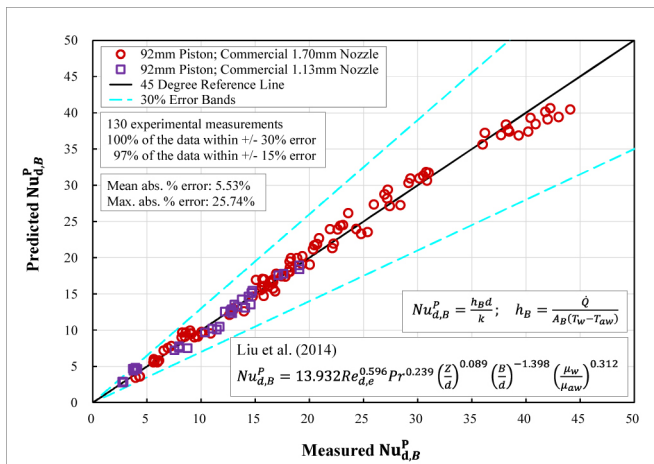


Figure 2. Experimental Results: Comparison of the Predicted to Measured Area-Average Nusselt Number by Using the Correlation of Liu et al. (Equation 5)

In the above study, the product of the heat transfer coefficient and the area, ($h_B \cdot A_B$), was calculated from the measured piston heat transfer rate and the oil-to-piston temperature difference. The piston undercrown impingement area was represented by the piston bore area ($A_B = \pi B^2/4$), and the piston undercrown temperature was taken to be

an average temperature calculated from four thermocouple measurements around the jet impingement area. (See Figure 7.) Also, the oil properties were based on oil film temperatures [$T_f = (T_w + T_j)/2$].

The area-average Nusselt number is a dimensionless temperature gradient at the surface, which can also be interpreted as the ratio of conductive thermal resistance to the convective thermal resistance of the fluid. For the previous results, the Nusselt number for the piston, based upon nozzle diameter and bore area, $Nu_{d,B}^p$, can be expressed as Equation 6, where d is the nozzle diameter and k is the thermal conductivity of the fluid.

$$Nu_{d,B}^p = \frac{h_B d}{k} \quad (6)$$

The Reynolds number of free-circular fluid-jet is a dimensionless parameter which can be defined as the ratio of the momentum forces to viscous forces and presented in Equation 7, where U_e is the mean fluid-jet velocity near the impingement plane and ν is the kinematic viscosity of the fluid.

$$Re_{d,e} = \frac{U_e d}{\nu} \quad (7)$$

The Prandtl number represents the relative importance of momentum and energy transport by the diffusion process, which can be shown in Equation 8, where c_p is the specific heat and μ is the dynamic viscosity of the fluid.

$$Pr = \frac{c_p \mu}{k} \quad (8)$$

Also, since the viscosity of motor oil is strongly dependent upon temperature, a Prandtl number based on film temperature may not capture the effect of viscosity in both the fluid jet and the wall film. To account for the strong dependence on temperature, a viscosity ratio was used in Equation 5, in which μ_w is the dynamic viscosity based on wall temperature and μ_{aw} is the dynamic viscosity based on adiabatic wall temperature. The adiabatic wall temperature is an estimate of the oil temperature near the impingement area, taking into account viscous dissipation in the high Prandtl oil jet as it approaches the stagnation zone at the piston surface. The adiabatic wall temperature is explained in the next section, below.

Following upon the earlier work, a smaller production piston ($B = 80$ mm) and a custom oil-jet nozzle, manufactured from a straight section of precision drawn tubing ($d = 1.70$ mm, $l/d = 40$), were added to test matrix, and the comparison of the predicted $Nu_{d,B}^p$ to the measured $Nu_{d,B}^p$ was plotted in Figure 3. In comparison to Figure 2, Figure 3 shows that, for the same piston (92 mm) now with the custom nozzle (1.70 mm), Equation 5 offers a slightly less accurate prediction of the Nusselt number, especially in the range of $8 < Nu_{d,B}^p < 18$. This suggests that the more intact jet produced by the custom nozzle resulted in a slightly lower Nusselt number than the commercial nozzle for the same piston. For

with the same commercial nozzle (1.70 mm) used earlier, Equation 5 does not predict the piston diameter effect very well, especially at higher measured $Nu_{d,B}^P$ values.

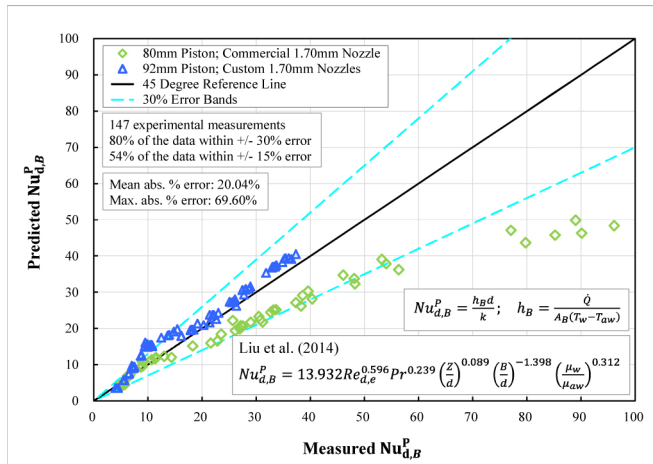


Figure 3. Experimental Results with Additional Piston and Custom Nozzle: Comparison of the Predicted to Measured Area-Average Nusselt Number by Using the Correlation of Liu et al. (Equation 5)

In the previous correlation, Equation 5, the oil jet properties were evaluated at the oil film temperature. Since engineers would typically have a better estimate of oil sump temperature, it was felt that any revised or improved correlation should use the sump (jet) temperature to evaluate properties. Secondly, the previous correlation used piston temperatures local to the jet stagnation point region even though the correlation was for an overall heat transfer coefficient. In this study the authors decided to use the area-weighted piston undercrown temperatures in the wetted area to compute the convective heat transfer coefficients. Lastly, the oil does not always wet the whole piston undercrown area, especially at lower oil-jet temperature and flow rate conditions. Accordingly, two different area-average Nusselt number correlations were developed; one is based on the whole piston undercrown area (A_{PU} ; measured from a piston CAD model), and the other is based on the effective heat transfer area, A_{ϵ} , which was measured on a flat plate and in another study by Liu [23].

Adiabatic Wall Temperature and Recovery Factor

Under high velocity [$U_j > 25(m/s)$] and/or high Prandtl number [$Pr \approx O(100)$] conditions, as in the case of the present study with engine lubricating oils, in which significant viscous energy dissipation occurs in the boundary layer, the fluid temperature rises in part due to viscous dissipation as it decelerates in the stagnation zone. For such conditions, the surface heat flux can be expressed by Equation 9 [24, 25],

$$q = h(T_w - T_{aw}) \quad (9)$$

where h is convective heat transfer coefficient, T_w is the local wall temperature and T_{aw} is the adiabatic wall temperature, which is used in place of the free stream fluid temperature. The adiabatic wall

temperature can be determined using Equation 10, and is represented schematically in Figure 4.

$$T_{aw} = T_j + \frac{r^* U_j^2}{2c_p} \quad (10)$$

where T_j is the liquid jet temperature evaluated at the nozzle exit condition, and U_j is the fluid-jet velocity. The recovery factor, r^* , represents the fraction of the kinetic energy of the free stream fluid, which is recovered as thermal energy in the fluid adjoining to the wall. Note that, the recovery factor depends on the velocity distribution, and analysis and experiments show that the recovery factor for a laminar boundary layer on a flat plate is $r^* \approx Pr^{1/2}$, while the recovery factor for a turbulent boundary layer on a flat plate is $r^* \approx Pr^{1/3}$ [24, 26, 27]. Thus, we can rewrite Equation 10 as Equation 11.

$$r^* = \frac{2c_p(T_{aw} - T_j)}{U_j^2} \cong Pr^n \quad (11)$$

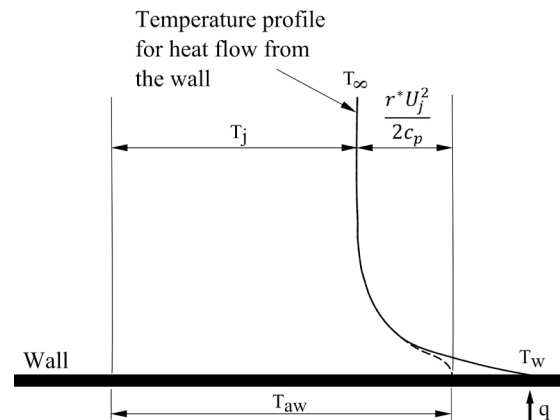


Figure 4. Schematic of Temperature Profiles in High Speed Free Stream Flow over a wall

In a recent oil jet cooling study, Soltis [28] investigated a high Prandtl number ($Pr \approx 150$) fluid-jet impacting an insulated surface and found a recovery factor relationship to be $r^* = Pr^{0.463}$. Soltis' results agree with the recovery factor relationship obtained by Leland and Pais [29]. Therefore, all the adiabatic wall temperatures in this study will be calculated by Equation 11 with $n = 0.463$.

EXPERIMENTAL APPARATUS AND PROCEDURE

Oil Circuit System

The oil circuit system, as shown in Figure 5, can be divided into several key components, including an oil reservoir, an adjustable speed dry sump pump, two oil filters, two immersion heaters with a PID controller, an inline backup heater, a compact chiller, pressure gages, a pressure transducer, a positive displacement flow meter and insulation.