A study of upward oil jet impingement on flat and concave heated surfaces and the application to IC engine piston cooling

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A Study of Upward Oil Jet Impingement on Flat and Concave Heated Surfaces and the Application to IC Engine Piston Cooling

by

Ting, Yew Siang
DipEng, BEng (Hons), AMIMechE

A Doctoral Thesis Submitted in Partial Fulfilment of the Requirements for the Award of Doctor of Philosophy of Loughborough University

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ABSTRACT

This thesis presents research on upward pointing oil jets that provide cooling of downward facing heated surfaces. The specific purpose of this research is to improve understanding of the oil jet cooling of internal combustion engine pistons.

In this research, the cooling of heated blocks with flat and concave surfaces was investigated. Temperature measurements were obtained using an array of thermocouples embedded inside the heated blocks. A flash illumination and high resolution CCD camera system was used to observe the liquid jet impingement. Observations identified a ‘bell-sheet’ flow pattern, jet interference, jet splatter and jet breakup which provided insights into the liquid jet impingement processes normally encountered on downward-facing surfaces.

Bespoke contracting-type nozzles were used to produce the jet flow structure. The data from these nozzles were used to generate new empirical correlations for oil jet cooling of downward-facing flat surfaces and for predicting the size of impingement. The results obtained from these tests were also used for comparison with cooling jets from production automotive piston cooling nozzles.

The research has demonstrated that the effectiveness of oil jet cooling can be affected by preheating the oil and varying the injector size to alter the targeted cooling efficiency, and liquid loss due to jet breakup and splatter. Local heat transfer coefficients were observed to increase when the jet Reynolds number increased.

Piston undercrown cooling was studied using a range of oil jet configurations. The cooling rates improved with optimised targeted jets. The
results also indicated that the undercrown geometry designs such as cross-hatched surfaces, undercrown-skirt and gudgeon-pin boss, were significant for enhancing the local rate of forced convective heat transfer.

New empirical correlations were developed from the experimental results that enabled prediction of the heat transfer coefficient and jet impingement size for high Prandtl number liquid jets impinging onto downward-facing surfaces. The heat transfer correlations were developed for normal ($\theta = 90^\circ$) and inclined ($\theta = 75^\circ, 60^\circ$ and $45^\circ$) jet impingements.
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# NOMENCLATURE

**Roman Letters**

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<th>Symbol</th>
<th>Definition</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a)</td>
<td>crank throw</td>
<td>m</td>
</tr>
<tr>
<td>(A)</td>
<td>area</td>
<td>m²</td>
</tr>
<tr>
<td>(A_w)</td>
<td>area of heated surface or cooled target</td>
<td>m²</td>
</tr>
<tr>
<td>(c_p)</td>
<td>specific heat capacity</td>
<td>J.kg⁻¹.K⁻¹</td>
</tr>
<tr>
<td>(C)</td>
<td>coefficient</td>
<td>-</td>
</tr>
<tr>
<td>(d)</td>
<td>diameter of free surface jet</td>
<td>m</td>
</tr>
<tr>
<td>(D)</td>
<td>diameter of heated surface or disc</td>
<td>m</td>
</tr>
<tr>
<td>(D_i)</td>
<td>diameter of cooling area at targeted surface</td>
<td>m</td>
</tr>
<tr>
<td>(D_n)</td>
<td>diameter of nozzle orifice</td>
<td>m</td>
</tr>
<tr>
<td>(g)</td>
<td>acceleration due to gravity</td>
<td>m.s⁻²</td>
</tr>
<tr>
<td>(h)</td>
<td>heat transfer coefficient</td>
<td>W.m².K⁻¹</td>
</tr>
<tr>
<td>(h(r))</td>
<td>thickness of local liquid film in radial direction</td>
<td>m</td>
</tr>
<tr>
<td>(k)</td>
<td>thermal conductivity</td>
<td>W.m⁻¹.K⁻¹</td>
</tr>
<tr>
<td>(l)</td>
<td>connecting rod length</td>
<td>m</td>
</tr>
<tr>
<td>(l_o)</td>
<td>distance of nozzle to target plate for onset of splattering</td>
<td>m</td>
</tr>
<tr>
<td>(L)</td>
<td>length of channel; average length of the wall jet region</td>
<td>m</td>
</tr>
<tr>
<td>(m)</td>
<td>exponent</td>
<td>-</td>
</tr>
<tr>
<td>(n)</td>
<td>exponent</td>
<td>-</td>
</tr>
<tr>
<td>(Nu)</td>
<td>Nusselt number</td>
<td>-</td>
</tr>
<tr>
<td>(Oh)</td>
<td>Ohnesorge number ((\mu(\rho.a.d)^{0.5}))</td>
<td>-</td>
</tr>
<tr>
<td>(Pr)</td>
<td>Prandtl number ((\nu/\alpha = \nu.\rho.c_p/k))</td>
<td>-</td>
</tr>
<tr>
<td>(Q)</td>
<td>heat transfer rate</td>
<td>W or J.s⁻¹</td>
</tr>
<tr>
<td>(q)</td>
<td>heat flux</td>
<td>W.m⁻²</td>
</tr>
<tr>
<td>(r)</td>
<td>radial distance measured from the point of jet impact</td>
<td>m</td>
</tr>
<tr>
<td>(r^*)</td>
<td>recovery factor</td>
<td>-</td>
</tr>
<tr>
<td>(r_h)</td>
<td>radius at which downstream flow is fully turbulent</td>
<td>m</td>
</tr>
<tr>
<td>(r_m)</td>
<td>radius at which neither (\delta) nor (\delta_i) reaches the film surface</td>
<td>m</td>
</tr>
<tr>
<td>(r_o)</td>
<td>radius at which (\delta) reaches the surface of the liquid sheet</td>
<td>m</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>-----------------------------------------------------------------------------</td>
<td>--------</td>
</tr>
<tr>
<td>( r_t )</td>
<td>radius at transition point from laminar to turbulent flow</td>
<td>m</td>
</tr>
<tr>
<td>( r_T )</td>
<td>radius at which ( \delta ) reaches the surface of the liquid sheet</td>
<td>m</td>
</tr>
<tr>
<td>( r_v )</td>
<td>radius at which ( \delta ) reaches the surface of the liquid sheet</td>
<td>m</td>
</tr>
<tr>
<td>( r_1 )</td>
<td>radius at which ( \delta ) reaches the surface of the liquid sheet</td>
<td>m</td>
</tr>
<tr>
<td>( r_2 )</td>
<td>radius at which ( T_w ) reaches the liquid ( T_{sm} )</td>
<td>m</td>
</tr>
<tr>
<td>( R )</td>
<td>radius of curvature</td>
<td>m</td>
</tr>
<tr>
<td>( R_{rc} )</td>
<td>ratio of connecting rod length to crank radius, ((l/a))</td>
<td></td>
</tr>
<tr>
<td>( \text{Re} )</td>
<td>Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td>( \text{Re}_d )</td>
<td>Reynolds number of circular liquid jet ((\rho u d/\mu))</td>
<td>-</td>
</tr>
<tr>
<td>( \text{Re}_l )</td>
<td>Reynolds number based on the average length of the wall jet region ((\rho u L/\mu))</td>
<td>-</td>
</tr>
<tr>
<td>( s )</td>
<td>max. heat transfer point from the jet impingement point</td>
<td>m</td>
</tr>
<tr>
<td>( S_p )</td>
<td>instantaneous piston speed</td>
<td>m.s(^{-1})</td>
</tr>
<tr>
<td>( \text{St} )</td>
<td>Stanton number ((h/\rho u c_p))</td>
<td>-</td>
</tr>
<tr>
<td>( T )</td>
<td>temperature</td>
<td>°C or K</td>
</tr>
<tr>
<td>( T_f )</td>
<td>temperature, liquid jet at the nozzle exit</td>
<td>°C or K</td>
</tr>
<tr>
<td>( T_{film} )</td>
<td>film temperature or ((T_f + T_w)/2)</td>
<td>°C or K</td>
</tr>
<tr>
<td>( T_{\infty} )</td>
<td>temperature of fluid far removed from heat source</td>
<td>°C or K</td>
</tr>
<tr>
<td>( T_0 )</td>
<td>stagnation temperature</td>
<td>°C or K</td>
</tr>
<tr>
<td>( u )</td>
<td>mean jet velocity</td>
<td>m.s(^{-1})</td>
</tr>
<tr>
<td>( u_f )</td>
<td>velocity of impinging jet or liquid film</td>
<td>m.s(^{-1})</td>
</tr>
<tr>
<td>( u_m )</td>
<td>local film velocity in boundary layer region</td>
<td>m.s(^{-1})</td>
</tr>
<tr>
<td>( u_\infty )</td>
<td>free-stream velocity</td>
<td>m.s(^{-1})</td>
</tr>
<tr>
<td>( V_n )</td>
<td>mean velocity at nozzle exit</td>
<td>m.s(^{-1})</td>
</tr>
<tr>
<td>( w )</td>
<td>splattering parameter</td>
<td>-</td>
</tr>
<tr>
<td>( w_n )</td>
<td>width of rectangular (planar or slot) nozzle</td>
<td>m</td>
</tr>
<tr>
<td>( \text{We} )</td>
<td>Weber number ((\rho u^2 d/\sigma))</td>
<td>-</td>
</tr>
<tr>
<td>( x )</td>
<td>distance from the central axis in x-axis direction</td>
<td>m</td>
</tr>
<tr>
<td>( y )</td>
<td>distance in y-axis direction</td>
<td>m</td>
</tr>
<tr>
<td>( Z )</td>
<td>nozzle-to-surface separation</td>
<td>m</td>
</tr>
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</table>
Greek Letters

<table>
<thead>
<tr>
<th>Greek Letter</th>
<th>Description</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>$\alpha$</td>
<td>thermal diffusivity, $k/\rho c_p$</td>
<td>m².s⁻¹</td>
</tr>
<tr>
<td>$\delta$</td>
<td>boundary layer thickness, viscous</td>
<td>m</td>
</tr>
<tr>
<td>$\delta_t$</td>
<td>boundary layer thickness, thermal</td>
<td>m</td>
</tr>
<tr>
<td>$\theta$</td>
<td>jet inclination angle or crank angle, $\pi/180^\circ$ (in radians)</td>
<td>°</td>
</tr>
<tr>
<td>$\pi$</td>
<td>$\pi$, approximately 3.14159...</td>
<td>-</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>surface tension of liquid/gas</td>
<td>N.m⁻¹</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density of liquid</td>
<td>kg.m⁻³</td>
</tr>
<tr>
<td>$\nu$</td>
<td>kinematic viscosity of liquid</td>
<td>m².s⁻¹</td>
</tr>
<tr>
<td>$\mu$</td>
<td>dynamic viscosity of liquid</td>
<td>kg.m⁻¹.s⁻¹</td>
</tr>
<tr>
<td>$\xi$</td>
<td>splattered fraction of incoming jet's liquid</td>
<td>-</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>temperature difference</td>
<td>°C or K</td>
</tr>
<tr>
<td>$\Delta y$</td>
<td>distance between the two points in y-axis direction</td>
<td>m</td>
</tr>
</tbody>
</table>

Superscripts
- average or mean

Subscripts
- ambient
- adiabatic wall
- aluminium
- copper
- stagnation
- heater or heating element
- jet
- liquid
- maximum
- engine oil
- saturation
- solid surface or wall
## Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Full Form</th>
</tr>
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<tbody>
<tr>
<td>API</td>
<td>American Petroleum Institute</td>
</tr>
<tr>
<td>CCD</td>
<td>Charge Coupled Device</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>CNC</td>
<td>Computer Numerical Control</td>
</tr>
<tr>
<td>EPA</td>
<td>Environmental Protection Agency</td>
</tr>
<tr>
<td>F</td>
<td>Freon</td>
</tr>
<tr>
<td>FC-77</td>
<td>Fluorocarbon Liquid</td>
</tr>
<tr>
<td>FE</td>
<td>Finite Element</td>
</tr>
<tr>
<td>HSDI</td>
<td>High Speed Direct Injection</td>
</tr>
<tr>
<td>ICE</td>
<td>Internal Combustion Engine</td>
</tr>
<tr>
<td>MAPE</td>
<td>Mean Absolute Percentage Error</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional-Integral-Derivative</td>
</tr>
<tr>
<td>R</td>
<td>Refrigerant or sometimes called Freon (F)</td>
</tr>
<tr>
<td>R-113</td>
<td>Trichlorotrifluorethane</td>
</tr>
<tr>
<td>RMS</td>
<td>Root-Mean-Square</td>
</tr>
<tr>
<td>SAE</td>
<td>Society of Automotive Engineers</td>
</tr>
</tbody>
</table>
ACKNOWLEDGEMENTS

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CHAPTER 1

INTRODUCTION

Internal Combustion (IC) engines represent the major source of motive power for many applications and are expected to continue to do so well into the 21st Century. The majority of vehicles today use IC engines because they are:

- Relatively efficient and inexpensive, compared to other forms of prime-mover e.g. gas turbines
- Relatively easy to refuel, compared to electric or fuel cell cars.

Over the past decades, engineers have been striving to increase the speed and power rating of high-speed diesel engines. The demand for improved engine performance has imposed greater demands upon engine cooling systems. In addition, many exhaust emission control strategies increase thermal loading in IC engines.

Ever since the IC engine was invented in the 19th Century, engine designers have been aware that, in order to maximise engine durability and efficiency, cooling of engine components is essential. As engine power increases it
becomes more difficult to limit the increase in thermal loading. In 1969, C. C. J. French wrote an influential paper entitled "Taking the Heat Off the Highly Boosted Diesel". This publication highlighted the importance of cooling IC engines and the need for more work to ensure that it would be possible to develop them to their full potential.

The current limit of power provided by the engine is partially due to the fact that the temperature of most engine pistons is close to the design limit. If the temperature of the piston can be reduced, then engines can run at higher power densities and hence the power output of the engine can be increased correspondingly. Hence, in order to be able to enhance the power output of the engine, a cooling technique is required which can maintain the piston within its thermal operating limits.

The research reported in this thesis concerns the study of cooling oil jets for enhancing convective heat transfer from piston undercrowsns to improve piston durability and limit piston temperatures. Effective design of oil jets can provide significantly enhanced piston cooling.

In this first chapter, the importance of the cooling systems on engine performance is introduced and areas influenced by the cooling system such as thermal issues and performance of engines are discussed. The benefits associated with optimising an IC engine cooling system design are also reviewed. Finally, the chapter describes the structure of the thesis and the main contributions resulting from this research.

1.1 Importance of Cooling Systems in IC Engines

Although the IC engine has been improved significantly over the last century it has still to reach its maximum potential in effective conversion of fuel chemical energy into mechanical work. In typical current engines, about 60% of the energy from the delivered fuel is converted into heat, which is
dissipated in the exhaust gas flow and engine cooling system, as shown in Figure 1-1.

![Fuel energy distribution diagram](image)

**Figure 1-1:** Typical fuel energy distribution in an IC engine – adapted from Taylor (1998) with added annotation

Inside the engine, fuel is intermittently burned and a significant proportion of the heat from this combustion process is exhausted from the engine, but some of it penetrates into the engine solid surfaces and heats them up. However, engine materials have a finite maximum temperature at which they can operate without failing and this is low relative to burning gas temperature. According to Angus *et al.* (1970), the flame face surface temperatures occurring in diesel engines are generally in the range of 300 to 680 °C, while in highly rated diesel engines with thick flame decks it can be in the range of 780 to 800 °C.

Engines of high power density and low emission are subject to high thermal loading and peak firing pressures and require increased cooling of their pistons due to high thermal load. This is because the control of piston temperatures has become one of the determining factors in a successful engine design (Kajiwara *et al.*, 2003). The following subsections summarise
the key motivations for improving cooling of IC engine pistons. The first subsection in this chapter covers the temperature related reasons for providing cooling. The second subsection discusses the heat transfer effects and their implications on engine performance.

1.1.1 Thermal Issues

a) Lubrication breakdown: When lubricating oil is exposed to an elevated surface temperature, its viscosity starts to decrease with temperature due to the weakening of molecular bonds. Therefore, it is good practice to provide continuously replenished oil flows over the cylinder hot surfaces to maintain good lubrication in the piston assembly (i.e. the piston rings and skirt). This flow of oil also removes heat from the hot surfaces.

b) Thermal fatigue: All piston materials expand and contract with temperature. Once a piston is subjected to an elevated temperature, depending on its thermal properties (i.e. thermal conductivity and thermal expansion coefficient), a thermal gradient due to different temperature distributions may arise in it, causing different thermal expansions. This in turn produces stress gradients within the piston and with time this may introduce flaws that can lead to failure, hence the occurrence of thermal fatigue. If the effect of these temperature gradients can be reduced then the amount of fatigue experienced by the piston will be similarly reduced. If the high temperature gradients can be reduced then the detrimental effects of thermal cycles and material expansion in the piston will be similarly reduced.

c) Creep: When a piston is exposed to an elevated temperature and is under an applied stress of a certain magnitude during the compression and combustion process, the piston material will creep and may eventually rupture. Creep is a material deformation
mechanism dependent on material properties, exposure temperature, applied stress and exposure time. Rather than failing suddenly with fracture, the material permanently strains over a longer period of time until it finally fails. However, creep deformation can happen in relatively short time frames under very high temperature conditions. Therefore, reducing the piston thermal loading by effective cooling is important to minimise the onset of creep.

1.1.2 Engine Performance

a) **Combustion**: Hotspots on the combustion chamber walls (specifically in spark ignition engines) can cause pre-ignition of the fuel-air mixture during the compression stroke. The likelihood of hotspots is due to overheating of the fuel-air mixture during compression. This is undesirable because it impairs the engine performance whilst also increasing the component mechanical and thermal stresses, which eventually cause major engine damage. This effect can be prevented by eliminating the hotspots on the piston crown by using an effective piston cooling system to maintain the piston at its ideal operating temperature. Hotspots can also promote 'knocking' after the spark. Knocking can lead to severe piston damage due to the high pressure and temperature waves.

b) **Emissions**: Stringent global environmental legislation determines the levels of emissions such as NOx, CO, HC and PMs that an engine is allowed to produce. In many emissions reduction strategies cylinder and piston temperatures need to be controlled.

c) **Power Consumption and Fuel Economy**: If an oversized oil pump is used for piston oil jet cooling purposes, the engine performance is reduced since the oil pump consumes power from the engine which could be used as output power. Also, an oversized oil pump capacity
(volumetric flow rate) could potentially reduce the piston to below its ideal operating temperature. Therefore, an effective oil jet cooling system design can minimise the power consumed by this oil pump, thereby maximising engine power output and improving fuel economy.

1.2 Cooling System Design

In recent decades, French (1972) and Munro et al. (1973) highlighted the potential benefits for engine if the piston cooling system is optimised. The benefits include reducing the cost, size, weight and noise of cooling components, and improving engine thermal efficiency.

In 1998, White et al. developed a PC based computer program for evaluating the cooling system performance so that it could be optimised at the vehicle concept stage or prior to major updates to a model. This could include, for example, the vehicle front end styling or a change of engine. For the latter, Lawrence and Kortekaas (2001) also developed a PC based computer simulation package that could be used to model cooling system behaviour and establish system operating characteristics. In addition, the main challenge of the EPA/02, Euro 4 and future (i.e. Euro 5) cooling system development has been to identify and develop the optimal cooling systems with regard to both performance and cost.

The following examples discuss some of the key benefits for designers and manufacturers, beside engine's durability and performance, associated with effective oil cooling system design:

1. **Design Freedom**: If an IC engine oil cooling system could be reduced in size it could minimise design compromises in the oil gallery. For example, an optimised oil jet size could determine the flow characteristics (oil velocity, volumetric flow rate and pressure) in the
oil circuit, which are required for satisfactory cooling of the piston throughout the engine cycle.

2. **Oil Pump Design**: If a compact oil cooling system could deliver a higher specific heat rejection per unit volume, then a smaller oil pump (e.g. positive-displacement internal gear pump) would lead to a lower power consumption, which improves fuel economy. This could also potentially reduce the system noise and weight. In addition, avoiding unnecessary high oil pressures in the system would also reduce excessive wear of the oil pump.

3. **Package and Cost**: If the capacity of an oil cooling system can be reduced then it would yield a smaller packaging requirement for the oil pump and improve the parasitic losses and the thermal characteristics of the piston. Also, an optimised oil cooling system size would lead to a price reduction, for example, reducing oil gallery sizes would reduce the significant impact on manufacturing and packaging of the engine.

This section has shown, in a general way, how the improvement to oil cooling system is beneficial with respect to design. As with all automotive manufacturers, the design of the oil cooling systems is an area to address for possible improvements to engine production costs.

1.3 **The Future of Engine Cooling**

The study performed by Ricardo (Morgan *et al.*, 1999), described the key market drivers in the future for the 2005 and beyond heavy-duty engine market. These are summarised as:

- Performance
- Engine emissions
• Vehicle legislation
• Durability, reliability and maintenance
• Packaging size, weight and cost.

In order to meet the above key requirements, it is necessary to increase the engine fuel conversion efficiency, and in doing so the engine walls must be maintained at suitable operating temperatures. In the engine combustion chamber, since heat transfer is proportional to surface area and excess temperature, it is likely that the piston contributes more to charge heating during the induction and compression strokes than the cylinder wall, cylinder head or valves in isolation (Robinson et al., 1999). It therefore seems reasonable that more effort should be focused at reducing the piston crown temperature.

Therefore, the control of piston temperature by effective piston cooling becomes important, and the research reported in this thesis seeks to contribute to this by studying and modelling piston oil jets.

1.4 Thesis Overview

In Chapter 1 of this thesis, the reasons for cooling IC engines have been introduced. The potential areas associated with optimising the design of piston and the increased demands on engine performance are highlighted, hence the needs for research work in this area. Finally, the contributions of the work presented in this thesis are stated.

Chapter 2 addresses the thermal loading in IC engines when operating at high power densities and the main problems that can arise in heavy-duty pistons due to excessive thermal loading. Piston cooling oil jets are introduced as a potential method for significantly improving piston cooling systems, and this is followed by a description of the oil jet system found in IC engines.
Chapter 3 presents a review of previous literature, outlining the reported findings in all areas of liquid jet heat transfer that are relevant to this thesis. The chapter begins with an overview of the fundamentals of liquid jet impingement which highlights the particular features of free-surface liquid jet impingement and its thermo-fluids characteristics. Heat transfer models of cooling liquid jets from different types of jet flows (i.e. laminar, transitional and turbulent) are reviewed in the literature study and relevant knowledge that can be applied to piston oil jet cooling is identified. The chapter then discusses the use of deliberately configured liquid jets of different profiles. Finally the effects of nozzle designs are discussed.

Chapter 4 describes an experimental system developed for the investigation of impinging oil jet heat transfer. The experimental apparatus designed to enable the heat transfer measurements is described. A description is given of the optical access to the test chamber allowing visualisation of the liquid jet impingement behaviour with digital image analysis. The system calibration and validation tests are documented.

Chapter 5 describes the test methodology used for assessing a wide range of oil jet impingement configurations used in the experiments. The oil jet configurations were deliberately chosen in order to produce a sufficient quantity of data from the experimental rig for correlating new models of liquid jet impingement on downward-facing surfaces. Particular piston operating conditions investigated are discussed.

Chapter 6 discusses the fluid dynamics of liquid jet impingement on downward-facing surfaces, and the main focus of this chapter is to present the heat transfer data for each experimental set-up. The results of this experimental work are analysed, and analysis shows how the oil jets developed in this research are potentially beneficial if used in an engine cooling system. It was found that they offer the combined advantages of
increasing the heat transfer coefficient and potentially minimising crankcase aerosol generation with a continuous oil flow on the piston undercrown.

Chapter 7 presents new empirical equations correlated from the experimental data in this research. These correlations are capable of predicting the heat transfer coefficient values for cooling jet impingement of high Prandtl number liquids and to estimate the size of oil jet impingement areas for downward-facing surface. The correlated equations are also presented for the prediction of local Nusselt numbers on concave surfaces and piston undercrows.

Chapter 8 summarise the major conclusions of this research and outlines potential areas of future work.

1.5 Contributions from the Work Presented in this Thesis

The PhD work reported in this thesis has made four novel contributions towards the design and understanding of cooling oil jet systems (i.e. jet temperature, velocity and size) for downward-facing hot surfaces and their use in designing piston cooling systems for IC engines. These are:

1. Extensive experimental data were obtained to generate a new correlation of cooling oil jet of high Prandtl number liquid impingement on a downward-facing surface.

2. Experiments of oblique jet impingement on a tilted downward-facing test-surface at different angles were conducted. The experimental data were used to generate a new correlation, which enables the prediction of heat transfer coefficients of oblique cooling jet impingement of high Prandtl number liquids.
3. Investigations of free-surface liquid jet impingement on a downward-facing concave surface have generated new data. The experimental results from the concave test-surface were used to identify the effect of different surface geometries.

4. A method was developed for using liquid jets impinged onto downward-facing test-surfaces under isothermal conditions without changes in liquid film properties by adjusting the test-surface temperature to be the same as liquid jet temperature. The experimental data were used to generate a correlation that capable to predict the size of impingement areas for the design of liquid jet cooling systems on downward-facing flat surfaces.

1.6 Concluding Remarks

In this first chapter, the motivation for the research work has been outlined and has given an overview of the thesis. Chapter 2 will address the thermal loading in modern heavy-duty IC engines and the need for oil jet systems for piston cooling.
CHAPTER 2

IC ENGINE THERMAL LOADING AND COOLING SYSTEMS

As stated in Chapter 1, oil cooling systems play an important role in the operation of modern IC engines, which is likely to increase in the future due to a market drive for higher engine specific power output and lower emissions. Chapter 1 also explained that an optimised oil cooling system is beneficial for cost and packaging of engine assemblies in the automotive industry.

This chapter will focus on the thermal loading of IC engine pistons. The thermal loading on the piston is generally a function of the engine load. As one of the main components in an IC engine, the piston is expected to continue in its development and they are expected to become stronger, lighter, thinner and more durable as design understanding improves (Silva, 2006).

The current limit of power provided by the engine is partially due to the fact that the temperature of most engine pistons is close to the design limit.
Therefore, in order to be able to enhance the power output of the engine, a cooling technique is required which will maintain the piston within its thermal operating limits.

Oil jets can be used to cool the underside of IC engine pistons with significantly enhanced heat transfer coefficients. However, to-date there has been limited research into their behaviour and hence they are studied in detail in this research.

2.1 Thermal Loading Phenomena in IC Engines

Thermal loading is a vital factor in the design and operation of heavy-duty engines. The peak burned gas temperature in the cylinder of an IC engine can rise up to 2227 °C (Heywood, 1988) with a flame face surface temperature up to 800 °C (Angus et al., 1970). A proportion of the total heat is rejected to radiator coolant and flow by cooling air; the rest of the high level of thermal loads stays within the engine block and heats it up. These local heat flows are very important where the temperature gradient throughout the metal can be high enough to cause high thermal stress levels, leading to thermal fatigue. The temperature differences experienced are depending on the local heat flux, thermal conductivity and thickness.

Figure 2-1 shows how different parts of the engine operate at different ranges of temperatures. So, some regions inside the enclosed cylinder are more prone to overheating than others (Heisler, 1999).
The material properties of the engine parts can change when they reach high temperatures. Thermal stress can occur leading to cracks, deformation and weaknesses in the material. Therefore, the heat from the combustion of fuel has to be taken away continuously otherwise the metal components will become damaged. The engine temperatures have to be maintained at an optimum level and also as steady as possible to ensure its performance.

As shown in Figure 2-1, the moving piston inside the engine has a high surface temperature due to the combustion process. It is therefore important to understand the thermal loading in IC engine’s piston in order to control the thermal stress within acceptable levels.

### 2.1.1 Thermal Loading in IC Engine’s Piston

Thermal loading in pistons of diesel engines installed on commercial and military vehicles, construction, mining and agricultural equipment has been increased in recent years due to application of technologies to meet low
emission and higher specific power output requirements. As the highly boosted engine is operated with high air density, the piston operating temperature rises. The piston crown is the region where the heat flux is highest, the peak temperature occurs close to the centre of the piston. However, the temperature trend can be different for pistons made with different materials, for example aluminium, which is a much better conductor of heat than cast iron. Aluminium can conduct 3.2 times more heat away than cast-iron pistons in a given period and alloy pistons have thicker sections (Heisler, 1999). With flat-topped pistons (typical of spark-ignition engines) the centre of the crown is hottest and the outer edge cooler by 20 to 50 °C. However, the maximum piston temperatures in DI diesel engine pistons are at the lip of the bowl, and the piston crown surface temperatures are about 50 °C higher than SI engine equivalent temperatures (Heywood, 1988). Figure 2-2 shows an illustration of the measured temperature distribution in high-speed DI diesel engine piston, where the highest temperature occurs at the lip of the bowl and centre of the crown, and the heat flows from crown to undercrown and ring grooves.

![Figure 2-2: Measured temperature distribution in piston of DI diesel engine at full load - Heywood (1988)](image)

The thermal analysis of a piston is important from different perspectives. Firstly, the aluminium alloy-based pistons, which have a pronounced
temperature dependence of their mechanical properties, must not exceed the allowed operating temperature. The rapid deterioration of the mechanical properties of the commonly used aluminium-silicon (Al-Si) alloy at temperatures above 200 °C is responsible for piston ring sticking and piston material transfer due to contact adhesion and wear (Wang et al., 1995). In spite of their poor high-temperature strength and fatigue resistance compared to cast iron, Al-Si alloy is used extensively for pistons in the automobile industry because of its low expansion coefficient, low density (and hence mass), good wear resistance, good castability, good thermal conductivity and excellent corrosion resistance.

The temperature distribution leads to thermal deformations and thermal stresses. Hence, a high temperature in the piston results in an increase in the designed clearance between the piston and cylinder liner, which causes noise due to piston slapping. Cast iron pistons have been proposed to solve some of the problems. However, the temperature of the cast iron piston can be higher, usually about 40 to 80 °C than that of Al-Si alloy pistons (Heywood, 1988), so that coking and degradation of lubricating oil may be causing a serious problem (Wang et al., 1995). This is further discussed in Section 2.1.2.

In highly boosted diesel engines, the heavily thermally loaded pistons have reached their temperature limit and hence distortion can lead to piston scuffing. Any undesirable temperature rise inside the piston may lead to engine seizure due to piston warping. Figure 2-3 shows the example of overheated pistons.
Figure 2-3: Engine pistons with damaged head: (a) cracks on localised areas on the rim; (b) cracks all around the piston rim - Silva (2006)

2.1.2 Piston Assembly Friction

Designing a diesel engine piston assembly system that meets the requirement for improved emission, higher power delivery, and longer service life becomes increasingly important. A higher temperature of charge in an engine combustion chamber may be preferred for a better engine performance. However, this high temperature would affect the design of engine elements because it might cause considerable thermal-tribological problems (Wang et al., 1998).

Piston scuffing in diesel engines, particularly the piston/piston-ringing/cylinder-liner interface failure, is a serious problem that degrades engine performance and shortens engine life. Many causes, such as piston thermal deformation and distortion, lubricant coking, excessive wear and debris deposition, as well as ring sticking, have been proven to induce scuffing (Wong et al., 1993). Figure 2-4 shows an example of damaged ring grooves. These causes are closely related to high temperatures in the piston assembly, especially the piston ring area, as a result of the excessive combustion heat (Wong et al., 1993; Heywood 1988). Among the assembly
elements, the piston is the most vulnerable because heat may accumulate there due to the difficulty in accessing cooling sources (Wang et al., 1998). Hence, oil cooling and lubrication of the piston assembly system are necessary to prevent such problems.

Figure 2-4: Engine piston with damaged grooves: (a) piston; (b) detail of damaged grooves – Silva (2006)

The piston assembly is the dominant source of engine rubbing friction. The majority of internal engine friction (or mechanical losses), as much as 45% of the total engine mechanical losses, is generated by the piston ring assembly (Taylor, 1998), as shown in Figure 2-5. The mechanical losses in the piston ring assembly are contributed by piston rings and piston skirt.

Figure 2-5: Mechanical losses distribution in an IC engine – Taylor (1998)
Most IC engine pistons are made of an aluminium alloy which has a thermal expansion coefficient that is 80% higher than the cylinder bore material made of cast iron. This leads to differences between running and the design clearances (Esfahanian et al., 2006). Therefore, the increased piston temperature would lead to thermal expansion, causing the piston friction force to increase significantly.

The frictional heating from piston rings and piston skirt is also a source of heat, and the friction force of surface contact between both metals (piston ring/skirt and cylinder liner) determines the amount of heat generated. However, the piston crown conducts higher heat from the combustion of fuel, and then distributes this heat to other parts such as cooling oil, rings, and skirt. Hence, the piston rings and piston skirt are passing the intense heat to the lubricant film (or oil film), as shown in Figure 2-6.

Figure 2-6: Heat transfer path from the rings - adapted from Esfahanian et al. (2006) with coloured arrows.

The conditions in the lubricant film between the piston ring and cylinder liner are also affected by heat transfer through the piston crown. Piston scuffing is mainly caused by the breakdown of lubrication in the region of contact sliding, as supported by many experimental observations (Wang et al., 2005). The viscosity of the lubricant at the running temperature has a
pronounced effect. A lubricant film used to fill the clearance between the surfaces may suffer a reduction in viscosity at high temperatures owing to additive/polymer degradation, which can cause friction on the metal surfaces. Continued use at elevated temperatures can result in premature component wear and eventually engine seizure. Therefore, the lubricant must be chemically stable in order to keep the engine clean and to prevent wear.

Therefore, the cooling of heavily thermally loaded pistons is extremely crucial to keep the lubricant film performing well, which allows the piston ring assembly system to run smoothly with minimum friction force within the sliding surfaces and with less carbon deposits on the surfaces (i.e. ring grooves, skirt and liner).

2.2 Piston Cooling Process in Present Study

Work in this research study considers the configuration of an oil jet to provide forced convection heat transfer and its application to cooling of the hot piston where water coolant is not practical.

The use of lubricating oils as coolants is increasing in current designs of fast moving and high powered machinery, especially in sophisticated systems such as electrical drive systems in aircraft, integrated starter/generator turbine engines, stationary engine/generator sets, highly loaded bearings, power transmission gearing, and cylinder heads and reciprocating pistons in internal combustion engines. Lubricating oil can be used in the cooling of hot surfaces where the temperature is higher than the boiling point of liquid water. It also provides lubrication for sliding surfaces and reduces friction losses in piston assemblies.

A conventional method for cooling pistons is crankcase oil splash or mist undercrown cooling where the crankshaft 'whisks' the oil from the oil sump
and splashes it all over the surfaces underneath the piston. However, due to reciprocating motion of the mechanism, efficient piston cooling is difficult to achieve using crankcase oil splashing for heavy duty diesel engines. Because of the non-uniformity of the cooling conditions and the lack of axial symmetry, the temperature in the piston is circumferentially non-uniform. This non-uniformity is responsible for the structural distortion and variation of the piston working clearance, which eventually affects the tribological performance of the piston (Wang et al., 1998). Therefore, above a certain engine rating, additional oil cooling is necessary and this is required for highly rated engine pistons.

The key to successfully implementing an oil jet in IC engine cooling systems is the determination of the correct oil flow rate and supply conditions. Piston undercrown oil jet impingement ideally must occur at all times during the piston operation so that excessive surface temperatures associated with dry-out are avoided.

### 2.2.1 Cooling Oil Jet for Pistons

Piston cooling is critical for achieving the designed engine performance especially for heavy-duty internal combustion engines. Control of piston temperatures by cooling has become one of the determining factors in a successful engine design. The pistons are cooled by oil jets impinged onto the piston undercrown.

The oil jet piston cooling system has been proven to be a significant part of a new engine’s design, where the tests have been carried out inside a real engine, by McKissick Jr. and Schmidt (2003) in their development of a turbocharged 2.4L engine for the Chrysler PT Cruiser. They developed a package that included an integrated turbocharger/exhaust manifold, oil jets for piston cooling and numerous other upgrades to satisfy the demanding performance, emissions and durability requirements for the powertrain. Due
to the high thermal loading expected on the piston, block-mounted oil jets were added to help in cooling the pistons. Targeting the underside of the piston crown for the full length of its stroke, tests have demonstrated that the cooling jets reduced piston temperatures enough so that higher cost piston materials were not required (McKissick Jr. and Schmidt, 2003).

Table 2-1 shows a comparison of piston cooling structure types and dissipated heat distribution predicted using measured temperature data. In the case of piston cooling using an oil splash or mist cooled type, the majority of the heat flows from combustion bowl to piston ring as the heat flows into the lubricant film and cylinder walls. For the undercrown impingement type, the cooling oil is delivered by a block-mounted jet and it impinges directly onto the underside of the piston, thus the majority of the heat flows from combustion bowl to piston undercrown. In the cooling gallery type, the cooling oil enters the channel built inside the upper section of the piston and near the ring grooves, thus the majority of the heat flows from combustion bowl to cooling gallery.

Table 2-1: Piston cooling structure and dissipated heat distribution – redrawn from Kajiwara et al. (2003).
Although, the gallery cooling type mentioned in Table 2-1 is advantageous for reducing the heat flows out through the piston ring grooves, it has disadvantages that should be noted in its application. The disadvantages of gallery cooled (or "cocktail shaker") piston are as follows:

a) The gallery has a reputation of causing stress concentration and reducing piston strength in the ring groove region (Cao and Wang, 1995). To avoid problems the gallery size can be reduced but this limits the volume of cooling oil flow (Law and Day, 1969).

b) Since the galleries are located in the upper section of the piston it is not easy for cooling oil to reach the gallery from the crankcase. This requires accurate jet alignment and capture efficiency, which can give problems due to the rapid reciprocating motion of the piston (Cao and Wang, 1995).

c) The gallery must be adequately but partially filled with oil flow in order to promote a cooling action and to make efficient use of the oil. The influence of the amount of cooling oil is significant (Kajiwara et al., 2003).

d) The gallery cooling technique relies on a high reciprocating frequency (or known as "cocktail shaker" effect) of piston in order to achieve high forced convection to remove heat from the internal surface, and a sufficient size of gallery for the required amount of oil to flow inside (French, 1972). Hence, this method appears to be more feasible for high-speed engines.

e) The gallery cooled piston also costs more in production and is more suitable in the manufacturing of bigger bore size pistons (Law and Day, 1969).
Therefore, in the case of undercrown impingement type, if using an effective cooling oil jet, the heat dissipation from piston can be further improved at the underside of the crown (i.e. undercrown). Thus, the heat dissipation through the ring grooves region can be reduced significantly since the heat flow from combustion bowl to undercrown is increased. As a result, the temperature in the ring grooves and top skirt regions can be decreased significantly, and the piston can work at a better thermal condition. The degradation of the lubricant and the aluminium alloy properties will be greatly reduced, and hence, both the performance of the piston assembly and the life of the piston-cylinder interface will be improved significantly.

In addition, the amount of oil flow for piston cooling is also a significant part of the total oil flow in the engines. In engine designs, accurate use of a cooling oil jet is important because oil pump capacity and lubricating system are influenced by the amount of piston cooling oil. Hence, an optimised cooling oil jet can optimise oil pump's power consumption which leads to improved fuel economy.

For the reasons mentioned above it is necessary to search for improved piston cooling that would tolerate a higher temperature in the combustion chamber but retain reasonably low temperature levels at tribological interfaces, and at the same time, improve the temperature uniformity. Development of pistons by incorporating effective cooling oil jets suggests a direction to realise these goals because of the heat transfer capability of oil jet impingement.

### 2.3 Oil Jet Assembly in IC Engine Cooling Systems

Piston cooling oil is supplied from a standing jet installed inside the crankcase chamber. The oil jet is usually installed next to each piston cylinder inside the engine block. These jets are positioned in the crankcase and mounted on the engine block along with the main oil gallery which supplies
cooling oil to the jets, as shown in Figure 2-7. In such cooling systems these openings (oil gallery) can be closed with bolt-on steel plates when engine blocks are used in lower specific power output engines.

Figure 2-7: Underside view of jet nozzles mounted on engine block in a 3.0L turbocharged diesel engine – from Schambeck (2006) and further annotated.

As shown in Figure 2.2 (see Section 2.1.1), the heat flux on the piston top surface is uneven. Asymmetries of piston bowl, exhaust valves, intake valves, and spark plug (only in SI engine) locations and combustion chamber shape cause the heat flux to be asymmetric. Furthermore, the piston bowl at the crown is usually off-centre in two-valve per cylinder engines and the heat flux at the piston bowl surface is considered to be highest. The piston bowl is used to improve the fuel-air mixing rates inside the combustion chamber, thus the highest combustion rates occur nearly. Therefore, oil jets for piston cooling should be installed on the side where the piston bowl is located because highest temperatures in the upper section of the piston are surrounding the bowl, as shown in Figure 2-8. Hence, jet impingement directly below this side is needed to absorb more heat from the piston undercrown.
The uneven heat flux on the piston top surface can cause the temperature profiles on the crown to vary significantly, especially at high engine speeds and loads, and this would lead to thermal fatigue. Hence, cooling oil jets with a good alignment can maintain more uniform temperatures of the piston surfaces.

Therefore, it is the objective of present study to show that the use of oil jet impingement cooling selectively applied in regions of high heat flux will support the enhancement in piston cooling.

### 2.3.1 Oil Jet Nozzles for Piston Cooling

A number of designs of oil jet nozzles have been produced for the automotive industry. Most of these nozzle designs are filed as patents, and therefore their exact design is not typically explained in detail. The general design objectives for an oil jet nozzle are to achieve a smooth solid oil jet stream which avoids liquid loss prior the impingement on the targeted surface, to improve the efficiency of piston cooling.
Figure 2-9 shows an example of different types of oil jet nozzles that are available for a range of specific engine sizes. They all have their relative merits depending on the practical application. Typically, the nozzle orifice is machined to 1 mm nominal internal diameter, or larger. If the size of orifice is too small, the viscous losses in the nozzle will be high. However, using a large orifice size would cause a low velocity oil jet exiting from the jet nozzle, because the jet velocity depends on the amount of oil that can be driven by the positive displacement oil pump. Therefore, the size of nozzle orifice must be selected carefully for specific engine designs in order to achieve a good oil jet flow which meets the piston speed and size of cooling area.

Figure 2-10 shows a schematic of jet nozzles with different designs. There are two types of orifices, the converging-type and pipe-type orifices. Each has its own purpose to control the flow structures of cooling jet at the orifice outlet. All the piston cooling jets have different ways of bending of the nozzle body, some with one bend only but others have two bends. They may be bent at different angles to specify the direction of an oil jet which impinges on the most critical point of piston undercrown.
Figure 2-10: Schematic of piston cooling jet nozzles.

Figure 2-10(a) and Figure 2-10(b) show jet nozzles with ball valve housing at the inlet. At a specific oil pressure (typically 2 to 2.5 bar) generated by the oil pump during the engine running, the spring-loaded ball valve, which is housed inside the ball valve housing, opens and oil squirts at a precisely defined angle onto the underside of the piston crown. The purpose of the ball valve is to ensure that there is no cooling oil squirting at the piston underside when the engine is running at low load or idle condition, so that the piston temperature is still maintained close to its ideal thermal operating levels. However, when an engine is running at high load, high pressure cooling oil driven by the oil pump will pass through the jet nozzle by pressing in the ball valve, and the cooling oil flows out from the orifice.
In Figure 2-10(c), no ball valve housing is provided at the inlet of the jet nozzle. This design ensures that the jet nozzle keeps squirting cooling oil to the piston underside surface whenever the engine is running. Therefore, this type of jet nozzle is used for heavily thermally loaded engines that require piston cooling at all load conditions.

2.4 Concluding Remarks

This chapter has stated that piston undercrown impingement cooling type has the potential to be improved from its existing use by incorporating effective oil jets to increase the heat transfer coefficient significantly. Therefore, in order to apply oil jet cooling most effectively it is necessary to understand the relationship between the heat transfer rate, and oil flow rate and supply conditions.

Previous investigations have suggested that gallery cooled techniques were found to be more effective in cooling the area near the ring groove region than undercrown impingement cooling. However, the gallery cooling technique relies on a high reciprocating frequency of the piston in order to achieve high forced convection, and a sufficient gallery size for the required amount of oil to flow inside. The gallery cooled piston also costs more in production and is only suitable in conjunction with big piston sizes. Therefore, undercrown piston cooling is still the best choice for most IC engine producers as relatively standard pistons may be used and an effective cooling oil jet can remove heat effectively from the targeted hot points.

Chapter 3 considers the fundamentals of liquid jet impingement and the factors affecting heat transfer.
CHAPTER 3

COOLING LIQUID JET IMPINGEMENT

As stated in Chapter 2, cooling oil jet impingement has great advantages over conventional cooling methods in IC engine. This chapter reviews published work on the characteristics of liquid jet impingement on flat surfaces and is followed by a discussion of the fundamentals of liquid jet impingement heat transfer. Focus is on jet parameters and nozzle configurations that control and enhance cooling effects on hot surfaces, while minimising the splattering of liquid droplets and formation of aerosol emission.

3.1 Fundamentals of Liquid Jet Impingement

Liquid jet impingement offers an effective removal of locally concentrated heat. Liquid jets can be created using a simple straight tube or a contracting nozzle, and this nozzle can be aimed directly toward the region of a heat load. A liquid is fed at a suitable pressure into the nozzle so that it exits at a suitable velocity. When the jet strikes the hot surface, it forms thin hydrodynamic and thermal boundary layers that offer low thermal resistances. Therefore, the heat transfer coefficients are large and well suited
for cooling very localised high heat flux sources and maintaining acceptable surface temperatures.

![Figure 3-1: Schematic of a free-surface jet impingement.](image)

The jets of interest in the present research are single-phase, free-surface jets (unsubmerged jets), where a jet of liquid discharges through an ambient gas and impinges onto a target surface (Figure 3-1). In free-surface liquid jets the effect of surface shear stress and entrainment on the flow of the jet-gas interface is often negligible, due to the large difference in density and viscosity between the liquid and the surrounding gas. The shape of the free surface depends on the gravitational, surface tension and inertial forces. The jet speed, size and orientation determine the magnitude of these forces. Usually, the nozzle exit velocity distribution and jet shape are not greatly affected by the gravitational acceleration for small nozzle-to-surface separations, thus the jet velocity is preserved and virtually unchanged until impinged onto the target surface (Womac et al., 1993).

### 3.2 Stagnation Nusselt Number

As the jet approaches an impingement surface, it is concurrently decelerated and accelerated in directions normal and parallel to the surface, respectively.
These changes occur in the stagnation region, which is also characterised by a strong favourable pressure gradient parallel to the surface and accelerates the flow in streamwise directions. The pressure gradient also acts to suppress the turbulence that is usually associated with the pre-impingement turbulent jets. Within this region, the thermal boundary layer is extremely thin, thereby providing a large convection coefficient across the region. Later on, in the downstream flow the pressure gradient decays to zero and transition to turbulence may occur.

The stagnation zone characteristics of an impinging liquid jet are of great interest because the maximum heat transfer coefficient occurs in that region. The general form of empirical formulae is to correlate a maximum Nusselt number in the stagnation region as a function of Prandtl (Pr) and Reynolds (Re) numbers, i.e.

\[
\text{Nu}_{\text{max}} = C \text{Re}^m \text{Pr}^n
\]  

(3-1)

where the exponent \( n \) is a function of liquid viscosity and the coefficients \( C \) and \( m \) are determined empirically.

Much previous research has been reported for unsubmerged laminar and turbulent liquid jets, and all these jets are impinging onto upward-facing flat surfaces of either a uniform wall temperature or uniform heat flux. These are now discussed.

### 3.2.1 Laminar Jet (Uniform Profile)

Zhao and Ma (1989) presented analytical results for the local heat transfer with a circular impinging jet of uniform laminar profile. Their stagnation Nusselt number was expressed below for the different range of Prandtl numbers at uniform heat flux condition:
\[
\begin{align*}
\text{Nu}_d &= 0.7212 \text{Re}_d^{1/2} \text{Pr}^{0.4} \quad (0.7 < \text{Pr} < 3) \\
\text{Nu}_d &= 0.7212 \text{Re}_d^{1/2} \text{Pr}^{0.37} \quad (3 < \text{Pr} < 10) \\
\text{Nu}_d &= 0.8597 \text{Re}_d^{1/2} \text{Pr}^{1/3} \quad (\text{Pr} > 10)
\end{align*}
\]

Liu et al. (1991) presented analytical results for the heat transfer in stagnation zone with a circular laminar jet impinged onto a uniform heat flux surface.

\[
\begin{align*}
\text{Nu}_d &= 0.715 \text{Re}_d^{1/2} \text{Pr}^{0.4} \quad (0.15 \leq \text{Pr} \leq 3) \\
\text{Nu}_d &= 0.797 \text{Re}_d^{1/2} \text{Pr}^{1/3} \quad (\text{Pr} > 3)
\end{align*}
\]

Liu et al. (1993) investigated the convective heat removal of uniform profile laminar jets impinged onto uniformly heated surfaces. They represented the stagnation Nusselt number correlation for a sharp-edged orifice nozzle based on the experimental results with cold water for Pr \geq 3 and Reynolds numbers below \(1.3 \times 10^5\).

\[
\text{Nu}_d = 0.745 \text{Re}_d^{1/2} \text{Pr}^{1/3}
\]

### 3.2.2 Laminar Jet (Parabolic Profile)

Scholtz and Trass (1970) measured the stagnation Nusselt number of a fully developed (parabolic profile) laminar jet, which issued from a sufficiently long tube at Reynolds numbers below the transitional value of 2000 to 4000. They showed that

\[
\text{Nu}_d = 1.648 \text{Re}_d^{0.5} \text{Pr}^{0.361}
\]

was in good agreement with the experimental data for Pr = 2.45 and 500 < \text{Re}_d < 1960, and it appeared to be unaffected by the ratio of nozzle-to-surface separation to jet diameter for 0.05 < Z/d < 6.0.
Ma et al. (1993) collected a significant amount of data from their experimental investigation of the stagnation point heat transfer. They derived a correlation based on various data for free-surface circular jets from a pipe-type nozzle for a Reynolds number range of 50 to 23000 onto a uniformly heated surface. R-113 (Pr = 7 - 8), Kerosene (Pr = 20 - 21) and transformer oil (Pr = 260) were used as coolants in their experiment. Their empirical equation was found to be

\[ \text{Nu}_d = 1.29 \text{Re}^{1/2} \text{Pr}^{1/3} \quad (3-9) \]

### 3.2.3 Turbulent Jet

A liquid jet issued from a long tube has a fully developed turbulent jet for Reynolds numbers above a transitional value of 2000 to 4000 (Lienhard, 1995). Stevens and Webb (1991) investigated the stagnation Nusselt number using a pipe-type nozzle of inside diameter ranging from 2.2 to 8.9 mm, and a nozzle length of 380 mm long to ensure fully developed turbulent water jets for a Reynolds number range of 4 \times 10^3 to 5.2 \times 10^3 at the nozzle outlet. The correlation

\[ \text{Nu}_d = 1.51 \text{Re}^{0.44} \text{Pr}^{0.4} (Z / d)^{-0.11} \quad (3-10) \]

matched their experimental data with an average error of 15% and a maximum error of 60%.

At higher Reynolds number Gabour and Lienhard (1994) reported a stronger dependence of Nusselt number on Reynolds number. They investigated the stagnation point heat transfer of unsubmerged fully developed turbulent jet impinged onto a smooth surface. The smooth wall Nusselt number was well represented by
\[ \text{Nu}_d = 0.278 \text{Re}_d^{0.633} \text{Pr}^{1/3} \quad (3-11) \]

to an accuracy of about 3%, for jet diameters of 4.4 to 9.0 mm over a Reynolds number range of \(2 \times 10^4\) to \(8.4 \times 10^4\), and the Prandtl number of cold water was held nearly constant at 8.2 to 9.1. The effect of nozzle-to-surface separation on the stagnation point was found to be negligible for a \(Z/d\) range of 0.9 to 19.8.

Faggiani and Grassi (1990), however, represented the Reynolds number dependence in terms of two exponents, for fully developed turbulent jets impinged onto a uniformly heated surface with \(Z/d = 5\), and yielded correlations of the form

\[
\begin{align*}
\text{Nu}_d & = 1.10 \text{Re}_d^{0.473} \text{Pr}^{0.4} \quad \text{Re}_d < 7.7 \times 10^4 \quad (3-12) \\
\text{Nu}_d & = 0.229 \text{Re}_d^{0.615} \text{Pr}^{0.4} \quad \text{Re}_d > 7.7 \times 10^4 \quad (3-13)
\end{align*}
\]

to an accuracy of within 30% for \(\text{Re}_d < 9 \times 10^4\) at \(\text{Pr} = 8\).

### 3.2.4 Comparison of Stagnation Nusselt number Correlations

A graphical comparison of stagnation Nusselt number correlations for free-surface circular jets summarised in Sections 3.2.1, 3.2.2 and 3.2.3 is given in Figure 3-2 for jets at large Reynolds number range (i.e. \(100 < \text{Re}_d < 10^5\)) and low Prandtl number liquid (i.e. \(\text{Pr} = 8\)).
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Figure 3-2: Comparison of stagnation Nusselt number correlations for free-surface circular jets of low Prandtl number liquid (i.e. Pr = 8).

As seen in Figure 3-2, the jet velocity distribution remains laminar at low Reynolds numbers below the transitional value of 2000 to 4000. The stagnation Nusselt number with parabolic laminar jets is higher than those with uniform velocity laminar jets. The higher stagnation pressure causes large pressure gradient near the stagnation point for the flow with parabolic profile than uniform profile. This causes rapid acceleration in the stagnation region, thus offering higher heat transfer coefficients.

In the turbulent zone, the stagnation Nusselt number values are higher than in the laminar zone. This is because a turbulent jet at high Reynolds number will tend to disrupt the very thin boundary layers, which are the thermal resistances, and thus raising the heat transfer coefficient during impingement.
Figure 3-3 shows the comparison of stagnation Nusselt number correlations summarised in Sections 3.2.1 and 3.2.2 for free-surface circular laminar jets (i.e. $Re_d \leq 2000$) of high Prandtl number liquid (i.e. $Pr = 260$). In the graph, the jets of parabolic profile exited from the pipe-type nozzle showed higher stagnation Nusselt numbers than jets of uniform profile exited from the sharp-edge orifice nozzle.

![Figure 3-3: Comparison of stagnation Nusselt number correlations for free-surface circular laminar jets of high Prandtl number liquid (i.e. $Pr = 260$).](image)

3.3 Local Heat Transfer at Downstream Locations

3.3.1 Flow Field

When a liquid jet strikes a flat surface, it spreads radially in a thin film. This thin liquid film is responsible for the convective heat removal from a hot surface. Watson (1964) previously studied theoretically the hydrodynamics of this thin liquid film and he obtained solutions for the boundary layer and viscous similarity regions. Azumo and Hoshino (1983, 1984a, 1984b, 1984c, 1984d, ...)
1984d) subsequently experimentally confirmed Watson's laminar results using the Laser-Doppler Velocimetry measurements and their velocity profiles showed good agreement to Watson's theory in the laminar regions of the flow. Watson also noted that the liquid film flow would be terminated by a hydraulic jump at a location independently controlled by the downstream conditions. The thickness of the liquid film initially decreased and then increased with radius as the viscous wall effects slowed the spreading of liquid film in radial direction.

As suggested by Liu et al. (1991), the downstream flow field is divided into the following regions (as illustrated in Figure 3-4):

Region 1 \((0 - r_m)\): The stagnation zone.
Region 2 \((r_m - r_0)\): The laminar boundary layer region. In this region, the viscous boundary layer thickness is less than the liquid sheet thickness, and that the rest of the liquid on the surface is unaffected by the wall friction. The free surface liquid velocity is equal to the incoming jet speed.
Region 3 \((r_0 - r_l)\): The viscous similarity region. In this region, the boundary layer thickness is the same as the liquid film thickness. The viscous effects extend through the entire liquid film, and the surface speed decreases with increasing radius.
Region 4 \((r_l - r_h)\): The region of developing turbulent flow.
Region 5 \((r_h \text{ onwards})\): The region of fully turbulent flow. This region may turn to laminar at farther downstream as the film speed decreases.
3.3.2 Heat Transfer

Liu and Lienhard (1989) investigated the convective heat transfer to a circular liquid jet impinging onto a uniformly heated surface. The cooling jet impingement was investigated analytically and experimentally for subcooled, unsubmerged, uniform velocity laminar jets in the absence of phase change. They used an integral method to predict the Nusselt number for constant heat flux surface and to examine the distance required to reach the liquid saturation temperatures.

The downstream heat transfer phenomena beneath an impinging circular jet can be divided into different regions by taking account of the development of viscous and thermal boundary layers and the possible occurrence of nucleate boiling or hydraulic jump (as indicated in region 6 in Figure 3-5). These regions may appear in different combinations, depending on the jet Reynolds number, temperature levels, wall heat flux and liquid physical properties (Liu and Lienhard, 1989).
In their research, they assumed that Prandtl number was greater than unity \((\text{Pr} > 1)\), as for most liquids. The downstream heat transfer regions (as illustrated in Figure 3-5) can then be subdivided as follows:

**Region 1.** *The stagnation zone.*

**Region 2.** \(\delta < h\) region: Neither the thermal boundary layer nor viscous boundary layer reaches the free surface. The velocity and temperature outside the viscous boundary layer is undisturbed and approximately equal to the jet velocity and jet temperature.

**Region 3.** \(\delta = h\) and \(\delta_t < h\) region: The viscous boundary layer has reached the free surface, and the velocity outside the viscous boundary layer decreases with radius. The temperature outside the thermal boundary layer is not affected by heat transfer but the thermal boundary layer is affected by viscous retardation of the momentum boundary layer.

**Region 4.** \(\delta = h, \delta_t = h\) and \(T_w < T_{sat}\) region: The thermal boundary layer has reached the surface of the liquid sheet and the temperature of the liquid surface increases with radius. Liu and Lienhard’s analysis showed that this region cannot exist for \(\text{Pr} > 4.859\), (as shown in
Figure 3-6) it means that the thermal boundary layer can never reach the surface of the liquid sheet before subcooled boiling occurs. This is because the growth of the thermal boundary layer is slower than the thickening of liquid film caused by the viscous retardation.

Region 5. *The boiling region:* This may include regions of nucleate boiling, burnout and dry surface if the heat flux is high enough. Since boiling might occur at any wall temperature beyond $T_{sat}$ depending on the heater surface finish and other factors, which refer to the entire region for which $T_w > T_{sat}$ as the boiling region.

Region 6. *The hydraulic jump:* A sudden increase of liquid sheet thickness occurs, and the heat transfer deteriorates significantly as the liquid velocity is much lower than in the upstream region. If the jet is directed upward as shown in Figure 3-7, the hydraulic jump is different than for the downward-directed jet. In that case, the jump is replaced with a liquid fall due to gravity effect.

![Region map for Pr > 4.859 - Liu and Lienhard (1989)](image-url)
In their research, Region 5 or Region 6 was not treated and it must be noted that all of the above regions may not occur at the same time, and that the last two may occur in a different sequence. For example, the boiling incipience may take place in any region of high wall superheat.

The radial distribution of Nusselt number was found accurately predicted using Liu et al. (1991) formulae in Table 3-1 for Prandtl numbers above unity.

![Figure 3-7: Region map for the upward-flowing jet - Liu and Lienhard (1989)](image)

<table>
<thead>
<tr>
<th>Region</th>
<th>Range</th>
<th>Nu</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Stagnation zone</td>
<td>$0 \leq r/d &lt; 0.787$</td>
<td>$0.715 \text{Re}_{d}^{1/3} \text{Pr}^{1/3}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$(0.15 \leq \text{Pr} \leq 3)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$0.797 \text{Re}_{d}^{1/3} \text{Pr}^{1/3}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$(\text{Pr} &gt; 3)$</td>
</tr>
<tr>
<td>2. Transition:</td>
<td>$0.787 &lt; r/d &lt; 2.23$</td>
<td>$0.715 \text{Re}_{d}^{1/3} \text{Pr}^{1/3}$</td>
</tr>
<tr>
<td>Stagnation to</td>
<td></td>
<td>$(0.15 \leq \text{Pr} \leq 3)$</td>
</tr>
<tr>
<td>boundary layer</td>
<td></td>
<td>$(\text{Pr} &gt; 3)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$0.797 \text{Re}_{d}^{1/3} \text{Pr}^{1/3}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\left( \frac{27 \text{Re}_{d} \text{Pr} (r)}{80} \right)^{1/3}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\frac{1}{2} \left( \frac{r}{d} \right)^2 - 0.2535$</td>
</tr>
<tr>
<td>3. Boundary layer</td>
<td>$2.23 &lt; r/d &lt; r_{b}/d$</td>
<td>$0.715 \text{Re}_{d}^{1/3} \text{Pr}^{1/3}$</td>
</tr>
<tr>
<td>region</td>
<td>where $r_{b}/d = 0.1773 \text{Re}_{d}^{1/3}$</td>
<td>$0.632 \text{Re}_{d}^{1/3} \text{Pr}^{1/3} \left( \frac{d}{r} \right)^{1/2}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$0.407 \text{Re}_{d}^{1/3} \text{Pr}^{1/3} \left( \frac{d}{r} \right)^{2/3}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\left[ 0.171 \left( \frac{d}{r} \right)^{3} + \frac{5.147 (r)}{\text{Re}_{d} (d)} \right]^{2/3}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\left[ \frac{1}{2} \left( \frac{r}{d} \right)^2 + C_{3} \right]^{1/3}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$0.267 \left( \frac{d}{r_{b}} \right)^{1/2}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$C_{3} = \left[ 0.171 \left( \frac{d}{r_{b}} \right)^{3} + \frac{5.147 (r_{b})}{\text{Re}<em>{d} (d)} \right]^{1/3} \left( \frac{r</em>{b}}{d} \right)^{2}$</td>
</tr>
</tbody>
</table>

42
4 to 5. Transition: Laminar to turbulent
\[
\text{Nu}_{\text{lim}}(r_i) + \left[ \text{Nu}_{\text{lim}}(r_h) - \text{Nu}_{\text{lim}}(r_i) \right] \left( \frac{r_h - r_i}{r_h - r_i} \right)
\]
- \text{Nu}_{\text{lim}} from equation in similarity region
- \text{Nu}_{\text{lim}} from equation in turbulent region

5. Turbulent region \( r/d > 28600 \, \text{Re}_d^{0.48} \)
\[
0.0052 \, \text{Re}_d^{3/4} \left( \frac{d}{h} \right) \left( \frac{d}{r} \right)^{3/4} \times \left( \frac{\text{Pr}}{1.07 + 12.7 \left( \text{Pr}^{2/3} - 1 \right) \sqrt{C_f/2}} \right)^{1/4}
\]
- \( C_f = 0.073 \left( \frac{\text{Re}_d}{r} \right)^{1/4} \)
- \( \frac{h}{d} = \frac{0.02091 \left( \frac{r}{d} \right)^{5/4}}{\text{Re}_d^{1/4} \left( \frac{r}{d} \right)^{1/4} + C \left( \frac{d}{r} \right)} \)
- \( C = 0.1713 + 5.147 \left( \frac{r}{d} \right) - 0.02091 \left( \frac{r}{d} \right)^{1/4} \)

Table 3-1: Suggested formulae for local Nusselt number of laminar jets with \( \text{Pr} > 1 \), uniform heat flux - Liu et al. (1991)

Zhao and Ma (1989) also developed formulae for local heat transfer of uniform profile laminar jets with \( \text{Pr} > 1 \), impinged onto a uniformly heated surface. Their local Nusselt numbers were formulated from the stagnation zone until the similarity region (in Table 3-2). When a jet impinges onto a uniformly heated surface, the surface temperatures change with jet flow rate. Typically, the surface temperatures are lowest in stagnation zone and then increases with radius in downstream region.
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4. Similarity region

\[ \frac{r}{d} > \frac{r_s}{d} \]

\[ \left\{ \frac{4}{(Pr Re_d)} \right\} \left( \frac{r}{d} \right)^3 + \left( \frac{272}{525} \right) \left( \frac{R}{d} \right)^2 \]

\[ R = 5.147 \frac{r^2}{(d/Re_d)} + 0.17132 \left( \frac{d^2}{r} \right) \]

Table 3-2: Suggested formulae for local Nusselt number of laminar jets with \( Pr > 1 \), uniform heat flux – Zhao and Ma (1989)

Webb and Ma (1995) developed formulae for local Nusselt numbers from the stagnation zone until the boundary layer region (see Table 3-3), for uniform profile laminar jets with \( Pr > 1 \), impinged onto an isothermal surface. When a jet impinges onto an isothermal surface, the surface temperatures remain constant but the surface heat losses change with jet flow rate. Typically, the heat flux loss is highest in the stagnation zone and then decreases with radius in downstream region.

<table>
<thead>
<tr>
<th>Region</th>
<th>Range</th>
<th>Nu</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Stagnation zone</td>
<td>( r/d &lt; 1 )</td>
<td>( 0.878 \frac{Re_d}{d}^{1/3} Pr^{1/3} )</td>
</tr>
<tr>
<td>2. Transition:</td>
<td>( 1 &lt; r/d &lt; \frac{r_s}{d} )</td>
<td>( 0.619 \frac{Re_d}{d}^{1/3} Pr^{1/3} (\tilde{r})^{1/2} ) where ( \tilde{r} = \frac{r}{d} \frac{1}{Re_d} )</td>
</tr>
<tr>
<td>Stagnation to boundary layer</td>
<td>where ( \frac{r_s}{d} = 0.141 \frac{Re_d}{d}^{1/3} )</td>
<td></td>
</tr>
<tr>
<td>3. Boundary layer region</td>
<td>( r/d &gt; \frac{r_s}{d} )</td>
<td>( \frac{2Re_d^{1/3} Pr^{1/3}}{(6.41 \tilde{r}^2 + 0.161/\tilde{r})(6.55 \ln(35.9 \tilde{r}^3 + 0.899) + 0.881)^{1/3}} )</td>
</tr>
</tbody>
</table>

Table 3-3: Suggested formulae for local Nusselt number of laminar jets with \( Pr > 1 \), uniform surface temperature – Webb and Ma (1995)

3.3.3 Comparison of Local Nusselt number Correlations

A graphical comparison of local Nusselt number correlations summarised in Tables 3-1, 3-2 and 3-3 is shown in Figure 3-8 for free-surface circular jets of high Prandtl number liquid (i.e. \( Pr = 260 \)), discharging from a nozzle of 2 mm diameter for \( Re_d = 2000 \). The local Nusselt numbers are plotted based on equations developed by Zhao and Ma (1989), Liu et al. (1991), and Webb and Ma (1995).
As seen in Figure 3-8, the local heat transfer in the stagnation zone is significantly higher than the radial flow region. Outside this zone, the local heat transfer decreases rapidly as the distance from the impingement point increases. The predicted local Nusselt number in the stagnation zone is higher for uniform wall temperatures than uniform wall heat flux. In the radial flow region, the local Nusselt number profiles exhibited a sharp drop for \( r/d > 0.787 \), followed by a slower decrease thereafter. All the local heat transfer predictions exhibit a similar trend, except for the sudden decrease in the line (near \( r/d = 2.5 \)) representing Liu et al. (1991) equations. The sudden decrease in Nusselt number coincides with transition to boundary layer region.
3.4 Other Aspects of Jet Impingement

3.4.1 Liquid Jet Impingement Using Numerical Method

Rahman et al. (2000) investigated the free-surface jet impingement of high Prandtl number liquids using a numerical method. Their work produced valuable insights for local heat transfer coefficient due to the effect of jet Reynolds number and heated disc of different thickness and materials. Galerkin finite element method was chosen for these calculations, where the dependent variables, i.e. velocity, pressure and temperature were interpolated to a set of nodal points that defined the finite element.

In their numerical model, they considered an axisymmetric jet discharging from a nozzle and impinging perpendicularly at the centre of a solid circular disc heated by discrete sources as shown in Figure 3-9.
Figure 3-10 shows the free surface height distribution for different Reynolds numbers when a liquid jet impinged at the centre of the disc. It can be seen that the fluid spreads radially as a thin film, and that the minimum film height occurs at a radius larger than the radius of the nozzle and the film height gradually increases with radius after that point. A smaller thickness is observed at higher Reynolds number because of larger impingement velocity, which translates to a larger fluid velocity in the film. Due to the high viscosity of the working fluid (i.e. high Prandtl number liquid), the boundary layer develops rapidly and the velocity of the fluid decreases as it spreads radially along the disc (Rahman et al., 2000).

Figure 3-10: Free surface height distribution for different Reynolds numbers - Rahman et al. (2000)

Figure 3-11 shows the effect of Reynolds number on the interface temperature. The spatial distribution is indicated in the figure. The temperature at the stagnation point was lowest, since the heat transfer is
highest in this region. As the fluid moves downstream the temperature increases. It can be seen that as the Reynolds number increases the temperature at the interface decreases. This is because of the higher local fluid velocity, which results in larger rate of heat transfer between the disc and the liquid (Rahman et al., 2000).

Figure 3-11: Temperature at interface for different Reynolds numbers - Rahman et al. (2000)

Figure 3-12 shows the effect of Reynolds number on the local Nusselt number. The maximum is located at the stagnation point. Outside the stagnation zone, the local heat transfer coefficient decreases rapidly. As the fluid reaches the location where the boundary layer starts to develop, a uniform Nusselt number zone is attained. After that point, the heat transfer coefficient continues to decrease until the edge of the disc, presenting the
pattern of an external flow over a flat surface. Far downstream, the heat transfer coefficient slightly increases, which is due to transition to turbulence. However, it should be noted that a fully turbulent flow regime could not be attained even at the outer radius in this simulation (Rahman et al., 2000).

Figure 3-12: Local Nusselt number variation for different Reynolds numbers - Rahman et al. (2000)

Figure 3-13 shows the local and average Nusselt numbers for different disc thickness. It is important to notice that the main differences are at the stagnation region, and as soon as the fluid enters into the boundary layer region, the distributions of Nusselt number for different values of disc thickness are close to each other. As the thickness of the disc increases, the temperature distribution at the solid-fluid interface becomes more uniform.
due to thermal spreading by radial conduction within the disc. Therefore, the heat transfer coefficient at the centre of the disc decreases (Rahman et al., 2000).

![Diagram of Local Nusselt number variation for different disc thicknesses](image)

(Re = 550, Τ_ι = 348 K, H_n = 0.0085 m, r_n = 0.00085 m, constantan plate, q_w = 63 kW.m⁻²)

Figure 3-13: Local Nusselt number variation for different disc thickness - Rahman et al. (2000)
The effect of disc thickness on temperature distribution within the solid is seen in Figure 3-14. When the disc thickness is negligible, the interface temperature is controlled by the heat flux condition at the heater. However, when an adequate thickness is provided the interface temperature becomes more uniform because of the radial spreading of heat within the solid. It can
be seen that the temperature becomes more uniform as the disc thickness increases. It is interesting to note that when the disc thickness was large, the temperature remained uniform over the bottom part of the disc. The isothermal lines start growing around the heat sources and become parallel to the bottom of the disc indicating almost one-dimensional heat conduction. However, in regions near the solid-fluid interface, the isothermal lines tend to be concentric around the stagnation point. The effect of non-uniform heat transfer is found only across the upper part of the disc. When the disc thickness is small, the temperature contours have entirely different orientation because of strong conduction over the entire disc thickness (Rahman et al., 2000).

Figure 3-15 shows the effect of different disc materials. The local Nusselt number increases slightly as thermal conductivity of the disc material decreases. At large radius, the Nusselt number for different materials is practically identical. A material with a lower thermal conductivity achieves a higher average Nusselt number because it produces a lower temperature at the stagnation region, which gives rise to a higher heat transfer coefficient (Rahman et al., 2000).
Figure 3-15: Local Nusselt number variation for different materials - Rahman et al. (2000)

Figure 3-16 shows the temperature distribution inside the disc for three different materials. For copper (thermal conductivity of $k = 386 \text{ W.m}^{-1}\text{K}^{-1}$) and silicon (thermal conductivity of $k = 148 \text{ W.m}^{-1}\text{K}^{-1}$), the temperature is more uniform and the maximum-to-minimum temperature difference is smaller, 1.3 and 4.2 respectively. For constantan (thermal conductivity of $k = 22.7 \text{ W.m}^{-1}\text{K}^{-1}$) the maximum-to-minimum temperature difference is fairly large (i.e. 18.2 K) because of much lower thermal conductivity. The shapes of the isotherms are not significantly affected by thermal conductivity (Rahman et al., 2000).
Figure 3-16: Isothermal lines within the disc for different materials - Rahman et al. (2000)
3.4.2 Jet Impingement of High Pr Liquid with Recovery Factor

In a high Prandtl number liquid, which corresponds to the small thermal diffusivity ($\alpha = \frac{\text{ratio of thermal conductivity to volumetric heat capacity, } k}{\rho c_p}$) compared with viscosity, the thermal boundary layer is much thinner than the viscous boundary layer. Thus, the temperature of the fluid layer near the wall must increase, due to viscous heating (or heat generated by viscous friction) as the kinetic energy of the flow is converted to internal energy while the flow decelerates through the boundary layer (Kreith and Bohn, 2001).

For flow over an adiabatic surface, such as perfectly insulated wall, Figure 3-17 shows schematically the velocity and temperature distributions. The actual shape of the temperature profile depends on the relation between the rate at which viscous shear work increases the internal energy of the fluid and the rate at which heat is conducted toward the free stream.

![Figure 3-17: Velocity and temperature distribution in high-speed flow over an insulated plate - from Kreith and Bohn (2001) with changed annotation](image)

The influence of heat transfer to and from the surface on the temperature distribution is illustrated in Figure 3-18. In high speed flow, heat can be transferred to the surface even when the surface temperature is above the free-stream temperature. This phenomenon is the result of viscous shear and
is often called aerodynamic heating. For most practical purposes the rate of heat transfer can be calculated with the same relations used for low-speed flow, if the convective heat transfer coefficient is redefined by the relation

\[ q = h(T_w - T_{aw}) \] (3-14)

where \( q \) is the conduction heat flux carried inside the metal, \( h \) is the calculated convective heat transfer coefficient, and \( T_w \) and \( T_{aw} \) represent the local wall temperature and the adiabatic wall temperature, respectively (Kreith and Bohn, 2001). Hence, the Nusselt number is obtained by

\[ \text{Nu} = \frac{hd}{k_l} \] (3-15)

where \( d \) is the liquid jet diameter and \( k_l \) is the thermal conductivity of liquid.

Figure 3-18: Temperature profiles in a high-speed boundary layer for heating and cooling - from Kreith and Bohn (2001) with changed annotation

In the calculation of local or average impingement heat transfer, the static jet temperature \( (T_j) \) should be replaced by adiabatic wall temperature \( (T_{aw}) \) and is defined by
\[ T_{aw} = T_j + r^* \frac{u^2}{2c_p} \]  \hspace{1cm} (3-16)

where \( T_j \) is the jet temperature at the nozzle exit, \( u \) is the average jet velocity and \( c_p \) is the liquid specific heat capacity. The recovery factor \( (r^*) \) is a function of Prandtl number. Figure 3-19 shows the higher the Prandtl number is, the thinner the thermal boundary layer, and the higher the recovery factor is, because the Prandtl number represents the thickness ratio of viscous boundary layer to thermal boundary layer and the consequent difficulty in removing the heat generated by viscous dissipation. The recovery factor as function of Prandtl number can be written in the form

\[ r^* = Pr^n \]  \hspace{1cm} (3-17)

where the exponent \( n \) is a function of liquid viscosity.

---

Figure 3-19: Increasing of recovery factor with Prandtl number - plotted from Equation (3-17) and \( n = 0.5 \) from Ma et al. (1990).
As reported in most of the literature (e.g. Ma et al., 1997, and Li and Guo, 1997), the Reynolds number has little effect on the recovery factor for the free impinging jet. However, Equation (3-16) shows that the jet velocity (and hence the Reynolds number) has a large impact on the adiabatic wall temperature, as shown in Figure 3-20.

Figure 3-20: Reynolds number effect on adiabatic wall temperature - plotted from Equation (3-16) and \( n = 0.5 \) from Ma et al. (1990).

Webb and Ma (1995) stated that using a high Prandtl number liquid as a working fluid in liquid jet impingement, the effect of viscous dissipation in the heat transfer process could become significant. For air flows, the practical importance of the viscous dissipation effect arises only in the case of high velocities of about half the speed of sound (Kreith and Bohn, 2001). However, for high Prandtl number liquids, this effect could be significant even at moderate velocities.

Metzger et al. (1974) are believed to be the first to report the effects of Prandtl number on jet impingement heat transfer for a single circular liquid jet impinging normally onto isothermal heated circular surfaces. Water and
synthetic lubricating oil were used as coolants. Jets of oil were discharged from a pipe-type nozzle with internal diameters ranging from 3.84 to 8.18 mm. The fully developed turbulent liquid jets were created from tube nozzles within the ranges of $2.2 \times 10^3 \leq \text{Re}_d \leq 138.2 \times 10^3$, $3.0 \leq \text{Pr} \leq 151$, and $1.75 \leq D/d \leq 13.2$. In their tests, the maximum values of the jet velocity ($u$) were approximately 23 m.s$^{-1}$ for both the water ($3.0 \leq \text{Pr} \leq 4.2$) and oil ($85 \leq \text{Pr} \leq 151$) that flows through the smallest jet supply tube.

Figure 3-21 shows a typical set of results (in Stanton number form) obtained with oil by varying the mass flow rate at each of the three different values of Pr, and conducted with $Z/d = 1$. The majority of their heat transfer data were obtained for $T_w - T_{aw} = 3.3$ K to avoid the problem of variable properties. The stagnation Stanton number is, like the Reynolds number, based on the mass velocity of the jet in the supply tube (Metzger et al., 1974).

![Figure 3-21: Typical stagnation Stanton Numbers (Oil) - Metzger et al. (1974)](image)

The physical significance of Stanton number is that it can be related to the ratio of the convective heat transfer to the thermal capacity of the flowing fluid.
The stagnation Stanton number is also equal to Nusselt number divided by Reynolds number and Prandtl number (or \( \text{Nu}_d = \text{St}.\text{Re}_d.\text{Pr} \)). In their study, the recovery factor was evaluated as

\[ r^* = \text{Pr}^{0.6} \]  

(3-19)

The following correlation was proposed for the lubricating oil:

\[
\text{Nu}_d = 2.65 \text{Re}_d^{0.47} \text{Pr}^{0.24} (D/d)^{-0.68} \left( \frac{\mu_{aw}}{\mu_w} \right)^{0.37}
\]  

(3-20)

where all properties are based on \( T_{aw} \) except \( \mu_{aw} \), which is based on \( T_w \). Increasing the temperature difference had the effect of increasing the heat transfer coefficient. The ratio, \( \mu_{aw}/\mu_w \), specifically accounts for the variable properties and Equation (3-20), was stated to be valid for \( 2.2 \times 10^3 \leq \text{Re}_d \leq 12.1 \times 10^3, 85 \leq \text{Pr} \leq 151, 1.75 \leq D/d \leq 25.1, 1.0 \leq \mu_{aw}/\mu_w \leq 1.7 \).

For water it was recommended that the correlation below be used:

\[
\text{Nu}_d = 2.74 \text{Re}_d^{0.348} \text{Pr}^{0.487} (D/d)^{-0.774} \left( \frac{\mu_{aw}}{\mu_w} \right)^{0.37}
\]  

(3-21)

Equation (3-21) was stated to be valid for \( 6.4 \times 10^4 \leq \text{Re}_d \leq 13.82 \times 10^4, 3.0 \leq \text{Pr} \leq 4.2, 1.75 \leq D/d \leq 13.2, \mu_{aw}/\mu_w \approx 1.0 \).

Figure 3-22 shows the Metzger et al. results in Nusselt number form. The correlation lines were plotted using Equation (3-20) and with variable
properties (i.e. \( \text{Re}_d \), Pr, \( D/d \) and \( \frac{\mu_{\text{jet}}}{\mu_o} \)) from Figure 3-21. As can be seen, the \( \text{Nu}_d \) increased as the \( \text{Re}_d \), Pr and \( d \) increased.

Figure 3-22: Correlation lines of \( \text{Nu}_d \) from Metzger et al. (1974).

Ma et al. (1990) later also studied the effects of Prandtl number on local heat transfer. Their experiments were performed with liquid jet impinging onto a vertical constant-heat-flux surface from a pipe-type nozzle of internal diameter \( d = 0.987 \) mm in the range of \( \text{Re}_d = 357 \) to 667, as shown in Figure 3-23.
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Figure 3-23: Details of test chamber and instrumentation used by Ma et al. (1990).

Figure 3-24: Detail of electrically heated test section (item 5 in Figure 3-23) used by Ma et al. (1990).
The working fluid was transformer oil (Pr = 202 to 260). The main part of the test section was a strip of 20 μm thick constantan foil with a heated section of 5 mm x 5 mm (nominal), as shown in Figure 3-24. Their results were obtained for a free jet at Pr = 260. In their study, the recovery factor was evaluated as

\[ r^* = Pr^{0.5} \]  (3-22)

and all their data for stagnation Nusselt number were well correlated by

\[ Nu_d = 1.29 \text{ Re}_d^{0.5} Pr^{1/3} \]  (3-23)

Ma et al. (1997) continued to study the local heat transfer and recovery factor with impinging free-surface circular jets of transformer oil. Their test-surface set-up was the same as in Ma et al. (1990), and liquid jet impinged from a small pipe and orifice nozzles of 1 mm diameter in the ranges of \( \text{Re}_d = 183 \) to \( 2600 \), \( \text{Pr} = 82 \) to 337, \( Z/d = 4 \) to 20. The main part was a strip of 10 μm thick constantan foil with a heated section of 5 x 10 mm (nominal) exposed to the coolant. In their study, the experimental data can be well correlated by the following recovery factor:

\[ r^* = 5.53 Pr^{0.24} \]  (3-24)

Their stagnation Nusselt number was expressed by the following correlations:

\[ Nu_d = 1.27 \text{ Re}_d^{0.495} Pr^{1/3} \]  (for pipe-type)  (3-25)

\[ Nu_d = 0.967 \text{ Re}_d^{0.51} Pr^{1/3} \]  (for orifice-type)  (3-26)
Figure 3-25: Comparison of correlation lines for Ma et al. (1990) and (1997).

Figure 3-25 shows a comparison of the stagnation Nusselt number from Ma et al. (1990) to Ma et al. (1997). For a pipe-type nozzle the two data sets are fairly close to each other for high Prandtl number (Pr = 705) but for low Prandtl number (Pr = 131) the two data sets deviate away. Whilst the correlation lines for the orifice-type showed stagnation Nusselt number that is generally lower than the pipe-type nozzle.

Leland and Pais (1999) investigated the effects of high Prandtl number and property variation due to large wall-to-fluid temperature differences on free jet impingement heat transfer. This is because a limited number of data were obtained for small values of \( T_w - T_{aw} \) (i.e. 8.9 and 17.8 K and these were shown to correspond to \( \mu_{aw}/\mu_w = 1.29 \) and 1.68 respectively, by Metzger et al., 1974) to study the effects of variable properties. Jets of lubricating oil were discharged from a contracting-type nozzle with orifice sizes of 0.508, 0.838 and 1.702 mm, and the heat transfer surface was a 12.95 mm diameter oxygen-free copper block. The nozzle body was supplied by an 8.12 mm internal-diameter tube and 16 diameters long. Large numbers of data points were obtained for a wide range of conditions (i.e. \( 109 \leq \text{Re}_d \leq 8592, 48 \leq \text{Pr} \leq \))
445, and surface-to-jet diameter ratios, $D/d$, of 7.61, 15.45 and 25.49), and highly varying properties (a wall-to-jet temperature difference, $\Delta T = T_w - T_j = 50$ to $120 \text{ K}$). The adiabatic wall to wall viscosity ratio ($\mu_{\text{wall}}/\mu_{\text{jet}}$) which ranged from 1 to 14.28. The nozzle-to-surface distance was varied from 1 to 8.5 mm which yielded $Z/d$ ratios of 1.19 to 7.87.

Figure 3-26: Comparison of stagnation Nusselt number data to correlation for laminar jet and turbulent jet, respectively - Leland and Pais (1999).

Their data were correlated with Equations (3-27) and (3-28), with mean absolute errors of 5.4% and 7.0%, respectively, as shown in Figure 3-26. The use of the film temperature ($T_{\text{film}}$) for the calculation of the properties was critical to obtaining such an excellent fit. To predict $\text{Nu}_{d}$ for $800 \leq \text{Re}_d \leq 1100$, Leland and Pais suggested that Equations (3-27) and (3-28) be averaged. Thus, for $\text{Re}_d < 800$:

$$\text{Nu}_d = 0.28 \text{Re}_d^{0.84} \text{Pr}^{0.32} \left[ \left( \frac{D}{d} \right)^{-0.64} - 0.1 \right] \left( \frac{\mu_{\text{wall}}}{\mu_{\text{jet}}} \right)^{0.04}$$  \hspace{1cm} (3-27)

and for $\text{Re}_d > 1100$:

$$\text{Nu}_d = 0.28 \text{Re}_d^{0.84} \text{Pr}^{0.32} \left[ \left( \frac{D}{d} \right)^{-0.64} - 0.1 \right] \left( \frac{\mu_{\text{wall}}}{\mu_{\text{jet}}} \right)^{0.04} \left( \frac{1}{\text{Re}_d} \right)^{0.27}$$  \hspace{1cm} (3-28)
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\[ \text{Nu}_d = 1.78 \text{Re}_d^{0.58} \text{Pr}^{0.22} \left[ \frac{D}{d} \right]^{0.32} - 0.30 \left( \frac{\mu_{\text{ave}}}{\mu_w} \right)^{0.27} \]  

(3-28)

The recovery factor was correlated by

\[ r^* = \text{Pr}^{0.47} \]  

(3-29)

The Prandtl number was based on the first approximation of the film temperature \( T_{\text{film}} \) or \( \frac{T_j + T_w}{2} \) rather than \( T_j \) alone. This was done because \( \text{Pr} \) will change under increased \( T_w \), and viscous dissipation should reduce as the film heats up across the surface. This approach also led to better correlations for \( \text{Nu}_d \).

For high Prandtl number fluids and very low wall-to-adiabatic wall temperature differences, the correlation of Metzger et al. (1974), Equation (3-20), was recommended in place of Equations (3-27) and (3-28) (Leland and Pais, 1999).

\[ \text{Figure 3-27: Comparison of Nu}_d \text{ for different nozzle diameters - Leland and Pais (1999). Correlation lines adapted from Figure 3-26.} \]
Figure 3-27 shows the comparison of Leland and Pais (1999) correlation lines and results for stagnation Nusselt number for different nozzle diameters. The correlation lines were adapted from Figure 3-26. For small jets, the correlation line compares very well with the data, and slightly over predicts for larger jet. There are two jumps in each of the correlation line, and it is due to the averaged Nu_d values from laminar and turbulent equations, particularly for $800 \leq \text{Re}_d \leq 1100$ only, as recommended by the authors.

The literature has thus far described the studies of free-surface liquid jet impingement with high Prandtl number liquids and the importance of using adiabatic wall temperature for calculating the local heat transfer coefficient.

Ma et al. (1990 & 1997), have conducted their study using a very thin test-surface (i.e. few tens of µm only) made from constantan foil, and they have recorded high local Nusselt numbers in their measurements. Their results were obtained from an experimental set-up with nozzle-to-surface distances of smaller than 20 mm and a vertical test-surface. Whereas, Metzger et al. (1974) and Leland and Pais (1999) have conducted a similar type of experiment but they were using a thicker test-surface of greater than 10 mm and an upward-facing test-surface made from cylindrical copper. The measured stagnation Nusselt number (Nu_d) values from Metzger et al. (1974) and Leland and Pais (1999) are rather lower compared to the results by Ma et al. (1990 & 1997), this is due to the effect of surface thickness on temperature distribution within the solid, as been discussed by Rahman et al. (2000) in Section 3.4.1.

### 3.4.3 Average Heat Transfer

Several studies have correlated the average heat transfer coefficients for axisymmetric impinging jets. In Section 3.2 and 3.3, these jets were shown to have a strong variation in local heat transfer coefficient ($h$) as the radial distance from the stagnation point increases. Also, the average heat transfer
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coefficient ($\bar{h}$) depends on the ratio of jet diameter ($d$) to the target diameter ($D$) over which the average is taken. Changes in nozzle conditions to get different velocity profile or turbulence levels will also affect the $\bar{h}$ strongly, especially when the target area is not much larger than the jet. In addition, increases in the liquid bulk temperature with radius can introduce variable property effects into correlations for $\bar{h}$.

The average heat transfer coefficient (in Equation 3-30) over the heated surface was determined from measurement of total power supplied to the heater ($Q$), area of the heated surface ($A_w$), average surface temperature ($T_w$) and jet temperature from the nozzle ($T_j$).

$$\bar{h} = \frac{Q}{A_w(T_w - T_j)} \quad (3-30)$$

Womac et al. (1993) presented in detail the correlation for average heat transfer coefficients. Their research specific interest was in electronics cooling applications. The following correlation was proposed, which is a weighted average of the stagnation zone and downstream contributions to $\bar{h}$:

$$\frac{\bar{Nu}_d}{Pr^{0.4}} = 0.516 \text{Re}_d^{0.5} \left( \frac{D}{d} \right) A_r + 0.491 \text{Re}_L^{0.32} \left( \frac{D}{L} \right) (1 - A_r) \quad (3-31)$$

Here, the area ratio is

$$A_r = \frac{\text{jet cross sectional area}}{\text{cooled target area}} = \frac{d^2}{D^2}$$

The average length of the wall jet region is $L = 0.5(D - d)$ and $d$ is the diameter of the jet just prior to impact. The correlation was derived from
data for low turbulent jets impinged onto a nearly isothermal surface at $1.25 \times 10^3 < \text{Re}_d < 51 \times 10^3$ for $2.3 \leq D/d \leq 9.2$ and Pr = 7 (for water) or 25 (for FC-77). It fits all their experimental data to within ±15% and is not affected by the bulk warming. Therefore, the correlation predicted the intermediate Prandtl numbers (Pr ≈ 7, Pr ≈ 25) with a reasonable accuracy but somewhat over predicted the high Prandtl number (Pr ≈ 85, Pr ≈ 113) data.

### 3.5 Nozzles for Liquid Jets

A relatively large body of literature, e.g., Rouse et al. (1951), Murakami and Katayama (1966), Leach and Walker (1966), McCarthy and Molloy (1974), Hoyt and Taylor (1985), addressed the design of nozzles for liquid jets. Most of this literature was directed toward the application of fire-fighting nozzles and jet-cutting nozzles, where the general design objectives were to achieve a high discharge coefficient and to inhibit jet breakup and spreading. Those aims were generally best met by minimising turbulence in the liquid leaving the nozzle (Lienhard, 1995). In engine cooling systems, an oil jet below the reciprocating piston is also designed to achieve a smooth solid jet stream, which avoids the amount of liquid lost and oil mist formation prior the impingement on the target surface.

However, a nonuniform velocity profile (e.g. parabolic) and turbulence are of great significance in jet impingement, because they can markedly increase the stagnation zone heat transfer. These effects are illustrated in Figure 3-2 (see Section 3.2.3). The nozzle that produces a jet determines its velocity profile and turbulence characteristics, as well as its contraction coefficient and its propensity for splattering or breakup. Each of these features can have a significant impact on the jet heat transfer characteristics, particularly in the stagnation zone.

The velocity profile and turbulence intensity must usually be determined for each individual nozzle configuration. McCarthy and Molloy (1974) described
the key features of a nozzle that affect these properties: (1) the contraction ratio from upstream diameter to outlet diameter of the nozzle; (2) the boundary layer development within the nozzle, generally a function of the length of contracted portion of the nozzle and of the interior roughness of the nozzle; and (3) the interior streamlining or contraction angle of the nozzle. Lienhard (1995) have reviewed the literature of nozzle types and their design effects on heat transfer. Many designs are possible, although three basic configurations are widely used as shown in Figure 3-28:

![Figure 3-28: Basic nozzle configurations](image)

Figure 3-28: Basic nozzle configurations: (a) sharp-edged orifice, (b) circular tube or parallel-plate channel, and (c) converging nozzle - Incropera (1999)

The three configurations shown in Figure 3-28 are described as follow:

- **Sharp-edged orifice** can be used to produce laminar jets of nearly uniform velocity profile provided that the orifice diameter is small comparatively to the size of large upstream plenum. The orifice sharp edge, as shown in Figure 3-28(a), provides a contraction of the liquid jet after passing through the orifice, which causes the jet diameter (or width) to be less than the orifice diameter, $D_n$ ($w_n$). A strong contraction will tend to damp turbulence in the jet. However, the high contraction ratio and stable upstream flow needed to produce uniform laminar jets are not usually achieved in practice. For example, nozzle contraction ratios are limited by the jet diameter required and the largest feasible supply line size, and turbulence or secondary flows...
upstream of the nozzle are seldom fully damped (Lienhard, 1995). Ramamurthi and Nandakumar (1999) studied experimentally the characteristics of flow through a small sharp-edged cylindrical orifice, and they presented the liquid jet structure formed from the orifice in photographs. Sharp-edged cylindrical orifices are commonly used for metering flows and injecting liquid fuels into combustion chambers at high velocities.

- **Long circular tube or parallel-plate channel.** Here the boundary layer development along the walls of the nozzle is a primary cause of velocity nonuniformity. The extent of the boundary layer growth depends on both the length of the nozzle and the flow acceleration within the nozzle. For a sufficiently long nozzle, the fluid will exit the nozzle with a fully developed laminar or turbulent velocity profile. The long straight tube nozzle, as shown in Figure 3-28(b), has been used in many impingement cooling experiments; it has no contraction and a length-to-diameter ratio of $L/D_n \approx 60$ is sufficient to provide a fully developed (parabolic) profile for laminar flow ($Re_d < 2300$), whereas $L/D_n \approx 20$ is sufficient to achieve a fully developed turbulent flow with blunter nonuniform profile.

- **Converging nozzles** are designed with a strong contraction that suppresses boundary layer growth along the nozzle wall, as well as turbulence generation within the nozzle. These effects increase with increasing the contraction ratio and streamlining of the interior surface. The interior streamlining of a nozzle contraction will prevent flow separation, thus reducing head loss (friction between the fluid and the walls of the pipe) and turbulence. However, a terminating straight section beyond the contraction, as shown in Figure 3-28(c), can promote the boundary layer development. Conversely, roughening the interior surface of the nozzle and/or attaching screen or grid to the nozzle exit may enhance turbulence in the flow. Nozzle discharge conditions vary with Reynolds number, and generalised
relations between nozzle design and turbulence are lacking. Adding flow straighteners upstream of the contraction can reduce further undesired secondary flows or turbulence (Incropera, 1999). Hoyt and Taylor (1985) developed a contracting nozzle without a terminating straight section beyond the contraction.

Lienhard (1995) stated that, in general, the performance of a nozzle will vary with its Reynolds number, owing to changes in the boundary layer development, changes in the position of laminar-turbulent transition, and surface tension at high Reynolds number. For slow, small diameter jets, the surface tension effects can have some influence, i.e., when jet Weber number ($We_d < 100$), the jet contraction increases, and for $We_d$ below 8, the nozzle flow ‘chugs’ rather than being a steady. In addition, small irregularities on the lip of a nozzle can cause nonuniformities in the liquid jet surface that promote unsteadiness and breakup.

Stevens et al. (1992) studied the effect of selected nozzle configurations on the flow structure and heat transfer characteristics for turbulent, free-surface liquid jets. Measurements were made with four nozzle configurations, as illustrated in Figure 3-29, which are the fully developed pipe-type nozzle, contoured orifice and sharp-edged orifice with and without damping screens in the plenum chamber. A more detailed description of these nozzles may be found in their paper, Part 1. In both of their experiments for flow structure and heat transfer studies, a $Z/d = 1$ was used to ensure that the gravitational effects influenced the jets impinging velocity by less than 2%.
For flow structure characterisation, Stevens et al. (1992) observed the distinctly different mean radial velocity gradient and turbulence behaviour for each of the nozzle types. Close to the plate, the magnitude of the radial velocity gradient was highest for the sharp-edged orifices, followed by the pipe-type nozzle, and finally, the contoured orifice. The turbulence was a strong function of the distance from the impingement plate. They found that the addition of the turbulence-damping screens to the plenum chamber of the sharp-edged orifice resulted in only minor changes in the mean radial velocity structure and associated gradient, but yielded substantial differences in the turbulent fluctuations of the radial velocity component.
Stevens et al. (1992) also studied the effect of selected nozzle configurations on heat transfer for Reynolds numbers of $22 \times 10^3$ and $43.7 \times 10^3$ (in Figure 3-30), where the sharp-edged orifice without turbulence-damping screens provided the highest heat transfer coefficients, followed by the sharp-edged orifice with screens, the fully developed pipe-type nozzle, and finally, the contoured orifice. The difference in stagnation Nusselt numbers among the nozzle configurations studied is approximately 40% at both Reynolds numbers shown. They concluded that the nozzle configurations affected the heat transfer depending on hydrodynamic conditions, in terms of velocity gradient and turbulence level. In the impingement region, the heat transfer coefficient appears to be influenced strongly by the velocity gradient.

### 3.6 Effects of Nozzle-To-Surface Separation

In the context of cooling nozzles, the jet stability is important mainly for the large nozzle-to-surface separations where jet breakup may disperse the coolant, and jet splattering can reduce the flow along the wall. When the
nozzle-to-surface separation is small (a few nozzle diameters or less), jet breakup and splattering are insignificant.

The nozzle-to-surface separation ($Z$) can influence the jet behaviour in other ways. Although the shear stress on a liquid-gas interface is small, the jet turbulence and interfacial shear can combine to induce interfacial waves or roughness, which in turn, can lead to atomisation of the liquid (Grassi and Magrini, 1991) and splattering of liquid droplets from the impingement surface (Lienhard et al., 1992, and Bhunia and Lienhard, 1994). Both processes become more influential with increasing $Z$ and have deleterious effect on the surface heat transfer, which leads to rapid decay in Nu with radius. The flow at large radial distances from the stagnation zone is more seriously affected than the flow near to the stagnation zone (Liu et al., 1989). This phenomenon is discussed further in Section 3.6.2.

Several experiments of downward jet impingement by Faggiani and Grassi (1990), Gabour and Lienhard (1994), and Stevens and Webb (1991), have shown that the nozzle-to-surface separation has little net influence on the turbulent jet stagnation-zone heat transfer. Their studies have measured the stagnation point Nusselt number of fully developed turbulent water jets for $Z/d$ between 1 to 33 and Reynolds number in between $9 \times 10^3$ to $130 \times 10^3$. The results showed that there are no significant effects of surface shear stress and entrainment on the flow of the jet-gas interface for $Z/d = 5$ to 10. The data suggest that, for Reynolds numbers below about $25 \times 10^3$, and decreasing $Z/d$ down to unity may raise the Nusselt number by up to 15% (Lienhard, 1995). However, for low speed jets, gravitational forces can cause speed and diameter to vary with $Z/d$.

If the jet emerges from the nozzle with a nonuniform velocity profile, a large nozzle-to-surface separation allows time for viscosity and turbulence to eliminate the velocity nonuniformity in the jet. The rate of velocity profile
relaxation (fully developed profile) depends on whether the jet is laminar or turbulent. Laminar jets may require up to 20 nozzle diameters to achieve a nearly uniform profile, and turbulent jets relax much faster than laminar jets, reaching a nearly uniform profile within 3 to 5 nozzle diameters. Therefore, the rate of relaxation is not simply a function of Reynolds number, but also depends on the jet speed and size (Lienhard, 1995). This phenomenon is further discussed in Section 3.6.1.

3.6.1 Liquid Jet Characteristics

Unusual effects observed in past experimental work on jet stability arise from differences in the velocity profile and turbulence properties of the jet as it emerges from the nozzle. The liquid jet emerges from a nozzle as a continuous body of cylindrical form can be in either a laminar or turbulent state.

Laminar flow is promoted by the absence of any disturbances to the flow, a rounded entrance to the tube, and high liquid viscosity, whereas turbulent flow is promoted by a high flow velocity, large tube size, surface roughness, rapid changes in tube-cross-sectional area, and protuberances projecting into the flowing stream. In a smooth tube with no flow disturbances, an initially laminar flow will remain laminar up to Reynolds numbers much higher than the critical value, but if the Reynolds number exceeds the critical value, only a small disturbance to the flow is required to initiate a transition to turbulent flow. Conversely, an originally turbulent flow will remain turbulent if its Reynolds number remains above the critical value (Lefebvre, 1989).

The critical Reynolds number, in a long straight cylindrical tube, may be defined as the Reynolds number below which, any disturbances in the flow will damp out. Above the critical Reynolds number, disturbances in the flow never damp out, no matter how long the tube is, as described by Schweitzer (1937). Generally, the value of critical Reynolds number was defined to be
around 2320. Schweitzer also pointed out that, there is a general tendency to associate the turbulent flow with a Reynolds number higher than critical, although in reality the flow is sometimes laminar at Reynolds numbers higher than critical and may be turbulent or semi-turbulent at Reynolds numbers below the critical value (Lefebvre, 1989).

Giffen and Muraszew (1953) have suggested various velocity distributions in jets as illustrated in Figure 3-31:

- **Laminar jet**: The velocity distribution in the jet immediately downstream of the orifice varies in a parabolic manner, rising from zero at the outer surface to a maximum at the jet axis. If the jet is injected into quiescent or slow-moving air, there is no appreciable velocity difference between the outer skin of the jet and the adjacent air. However, after a certain distance, the combined effects of air friction and surface tension forces create surface irregularities that ultimately lead to disintegration of the jet.

- **Semi-turbulent jet**: As illustrated in Figure 3-31, the annulus of laminar flow surrounding the turbulent core tends to prevent the liquid particles in the core from reaching and disrupting the jet surface. At
the same time, the influence of air friction is minimal, due to the very low relative velocity between the jet surface and the surrounding air. Thus, jet disintegration does not occur close to the orifice exit. However, farther downstream, the faster turbulent core outpaces its protective laminar layer and then disintegrates in the normal manner of a turbulent jet. Alternatively, and to some extent simultaneously, a redistribution of energy takes place between the laminar and turbulent components of the total flow, which produces a flattening of jet velocity profile. This process brings to the jet surface liquid particles with radial components of velocity. These particles disrupt the jet surface so that ultimately it disintegrates into drops.

- **Turbulent jet**: If the flow at the orifice is fully turbulent, the radial velocity component soon leads to disruption of the surface film followed by general disintegration of the jet. It should be noted that when the jet is fully turbulent, no aerodynamic forces are needed for breakup. Even when injected into a vacuum, the jet will disintegrate solely under the influence of its own turbulence (Lefebvre, 1989).

In the attempt to standardise a velocity profile emerging from the nozzle, many workers have used long tubes as nozzles to ensure that the jet velocity profile is initially fully developed when emerged from the orifice exit. The jet velocity is either a laminar (e.g. parabolic) velocity profile or turbulent velocity profile.

The changes in velocity profile or velocity profile "relaxation" that occurs at the downstream of the nozzle exit can have a significant influence on the jet stability and on its subsequent breakup into drops (McCarthy and Molloy, 1974). Once the jet leaves a nozzle and as soon as the physical constraint of the nozzle wall is removed, the process of velocity profile relaxation occurs by a mechanism of momentum transfer between the transverse layers within the jet. In addition, there exists an additional disruptive mechanism arising
from the internal motions such as turbulence and inertial effects (Lefebvre, 1989).

The different roles played by liquid turbulence and air friction allow the distinction to be made between primary and secondary atomisation. Primary atomisation is related to jet breakup by the action of internal forces, such as turbulence, inertial effects, or those arising from velocity profile relaxation and surface tension. Secondary atomisation always involves the action of aerodynamic forces in addition to those present in primary atomisation (McCarthy and Molloy, 1974). For high velocity jets it is generally believed that the action of the surrounding air or gas is the cause of atomisation, although jet turbulence is a contributing factor because it ruffles the surface of the jet, making it more susceptible to aerodynamic effects. Aerodynamic forces promote atomisation by acting directly on the surface of the jet and by splitting the drops formed in primary atomisation into smaller droplets (Lefebvre, 1989).

According to Reitz (1978), the following four regimes of breakup (as shown in Figure 3-32) are encountered as the liquid injection velocity is progressively increased. Each regime is characterised by the magnitudes of the jet Reynolds number (Reₐ) and Ohnesorge number (Oh). Ohnesorge number represents the ratio of viscous force to a surface tension force.
Figure 3-32: Breakup regimes of round liquid jets in quiescent air – from Liu (1999) with changed annotation

Depending on jet velocity, nozzle diameter and properties of liquid and air, the disintegration of a round liquid jet in a quiescent gas may occur in the following four regimes, as shown in Figure 3-32 and summarised as follows (Liu, 1999):

1. **Rayleigh jet breakup**: This is caused by the growth of axisymmetric oscillations of the jet surface, induced by surface tension. The jet breakup length is linearly proportional to the jet velocity, typically many nozzle diameters downstream of the discharge nozzle exit. Droplet diameters are larger than the jet diameter.
II. *First wind-induced breakup*: The surface tension effect is now augmented by the relative velocity between the jet and the ambient gas, which produces a static pressure distribution across the jet, thereby accelerating the breakup process. Liquid breakup occurs a few diameters downstream of the discharge nozzle exit. Average droplet diameters are about the same as the jet diameter.

III. *Second wind-induced breakup*: Droplets are produced by the unstable growth of short-wavelength surface waves on the jet surface caused by the relative motion of the jet and the ambient gas. This wave growth is opposed by surface tension. Liquid breakup occurs several diameters downstream of the discharge nozzle exit. Average drop diameters are smaller than the jet diameter.

IV. *Atomisation*: The jet breaks up completely within a short distance from the discharge nozzle exit in an irregular manner and the spray formed is conical. Average droplet diameters are much less smaller than the jet diameter.

In an oil cooled engine the jet velocity is typically equal to the maximum piston velocity which is around 20 m.s\(^{-1}\), which gives \(Re_d = 4300\) and \(Oh = 0.032\) for \(d = 2\) mm and \(T_j = 120\) °C. Therefore, Figure 3-32 shows that in such a system the oil jet would be in the second or third breakup regime when injected at highest speeds for the case of highly rated diesel engine piston.

### 3.6.2 Splattering Effects

In liquid jet impingement, splattering is associated with jet impingement with liquids of lower surface tension or higher Weber number and when the jets are longer, because the disturbances reaching the liquid sheet on the surface are then larger. When a jet splatters, much of the incoming liquid can become airborne, as droplets, within few jet diameters from the point of impact. Airborne droplets no longer contribute significantly to cooling the downstream surface and hence cooling is far less efficient than it could be if
splattering were suppressed. Therefore, understanding the causes and scaling of splattering is an essential element in the jet cooling system design (Lienhard et al., 1992). In clean-room situations (e.g. silicon chip manufacture), where impinging jets are used for post-etching debris removal, the splattered liquid can produce airborne contamination which introduces flaws on the chip surface. In metal forming operations involving toxic chemicals, the splattered droplets can create a hazardous aerosol, and containment may necessitate significant air filtration costs (Bhunia and Lienhard, 1994).

![Splattering during impingement of a turbulent liquid jet](image)

(a) Splattering jet at $Re = 28,400$  
(b) Splattering turbulent jet at $Re = 48,300$

Figure 3-33: Splattering during impingement of a turbulent liquid jet – Lienhard et al. (1992)

Errico (1986) has described the basic physical mechanism of splattering, where disturbances to the surface of the incoming jet are strongly amplified as the jet spreads into a liquid film along a wall normal to the axis of the jet, and can cause droplets to break free from the liquid sheet (Figure 3-33).

The associated flow regimes along the surface can be characterised in an average sense as follows (Figure 3-34), Lienhard et al. (1992):

- **Stagnation Zone**: A very thin wall boundary layer with a turbulent free stream above it.
- **Region Before Splattering:** Disturbances to the liquid sheet are strongly amplified in this region. As in the stagnation zone, the wall boundary layer is affected by turbulent and capillary disturbances to the flow above it.

- **Region of Splattering:** A portion of the liquid sheet breaks free as droplets, owing to the instability of the disturbed liquid sheet. The effective radial size of this zone is fairly small.

- **Region After Splattering:** Having lost both mass and momentum in the splattering process, the remaining liquid sheet continues to flow outward. The liquid sheet is then fully turbulent.

![Diagram](image)

**Figure 3-34:** Regions for turbulent incoming jet - Lienhard *et al.* (1992)

Errico (1986)'s experiments on the splattering of impinging jets demonstrated that the jet splatter is directly tied to the surface roughness and deformation of the incoming liquid jet. Jet splatter can be reduced by making the nozzle-to-surface separation shorter (so that the disturbances to the liquid jet have
less time to develop). Furthermore, jet stability is related to the specific nozzle design. Lienhard et al. (1992) also found that the onset of splattering in jets varies with Re, We, and ratio of jet length to jet diameter.

Splattering and turbulence both produce additional mixing in the liquid sheet which will tend to enhance the heat transfer relative to a laminar sheet, but the increasing amount of mass splattered will reduce the heat transfer, and thus creates a rapid decay in local Nusselt number with radius. As more mass is splattered away from the surface, the remaining film has less momentum and slows more quickly. Conversely, the wall friction will be generally lower for laminar flow, which should result in larger velocities at a given downstream radius. The relative cooling efficiency of these cases is not obvious a priori, apart from the expectation that turbulence enhances heat transfer in the stagnation zone (Lienhard et al., 1992). As observed by Liu and Lienhard (1989), splattering has a strong effect on the wall temperature, it may cause decrease in the Nusselt number of up to 20%, and the downstream flow is more significantly affected than the upstream flow. In most real systems, which are likely to have somewhat irregular flow structure and thus have splatter, this effect will reduce the efficiency of the jet in cooling the surface.

Lienhard et al. (1992) and Bhunia and Lienhard (1994) experimentally and analytically investigated splattering and heat transfer phenomena during impingement of an unsubmerged fully turbulent liquid jet. They developed predictive results for the local Nusselt number along a uniform heat flux surface, onset of splattering and total mass splattered. Their experiments cover Weber numbers between 130 to $31 \times 10^3$, Reynolds number between $2.7 \times 10^3$ to $98 \times 10^3$, and nozzle-to-surface separations of $0.2 \leq Z/d \leq 125$.

Lienhard et al. (1992) observed that the distribution of droplet diameters typically ranges from a few μm to almost a mm. Most droplets passing a
particular point have essentially the same velocity irrespective of their size. Very small droplets (less than about 20 μm) suffer significant viscous drag and move slower, but these droplets contribute little to the mass-averaged velocity. Bhunia and Lienhard (1994) presented the correlation in Equation (3-32) for the splattering ratio ($\xi$) in the range of $4.4 \times 10^3 < \omega < 10 \times 10^3$, for turbulent jets produced by fully developed turbulent pipe-flows, where $Re_d$ exceeds 2000. The parameter, $\xi$, is the ratio of splattered-liquid volume flow rate to incoming jet volume flow rate, and a scaling parameter (or splattering parameter), $\omega$, characterised by the root mean square (rms) amplitude of disturbances that reach the target:

$$\xi = -0.258 + 7.85 \times 10^{-5} \omega - 2.51 \times 10^{-9} \omega^2$$

(3-32)

$$\omega = \text{We}_d \exp \left( \frac{0.971 Z}{\sqrt{\text{We}_d d}} \right)$$

(3-33)

where $\text{We}_d$ is the liquid jet Weber number, $Z$ is the nozzle-to-surface separation and $d$ is the liquid jet diameter.

Figure 3-35: The fraction of incoming mass splattered ($\xi$) as function of $\omega$ for $Z/d < 50$ - Bhunia and Lienhard (1994)
Figure 3-35 shows the total fraction of liquid splattered from the sheet as a function of $\omega$. The ratio of splattered-liquid flow to total incoming flow increases monotonically with $\omega$. When $\omega \leq 2120$, the amount of splattering is very small or zero, with $\xi \leq 2.5\%$. In many engineering applications, splattering may be neglected in this range. However, for $\omega \geq 2120$, splattering increases rapidly and splattering of as much as 30% of the incoming fluid is observed at Weber numbers of 5628 and Reynolds number of $47.8 \times 10^3$.

![Graph showing Onset of splattering - Bhunia and Lienhard (1994)](image)

Figure 3-36: Onset of splattering – Bhunia and Lienhard (1994)

Bhunia and Lienhard (1994) also defined the onset of splattering as the point where 5% of the incoming fluid is splattered. In view of their earlier observation, the amount of splattering at a fixed $Z/d$ depends strongly on the jet Weber number and not on Reynolds number. The onset point is uniquely identifiable by its $Z/d$ and $We_d$. In other words, for a jet of a given Weber number, the onset point is reached at a certain $Z/d$. Figure 3-36 shows the data for onset points, and a correlation for the onset point data is derived as

$$\frac{l_o}{d} = \frac{130}{1 + 5 \times 10^{-7} We_d^2} \quad (3-34)$$
For low Weber numbers, near 100, the turbulent disturbances are strongly damped by the surface tension, and the observed onset lengths are close to the capillary breakup lengths. In this range, the splattering is essentially of drop impingement type.

Lienhard et al. (1992) concluded that splattering is found to occur in proportion to the magnitude of surface disturbances to the incoming jet and it only occurs within a certain radial range rather than along the entire film surface. The breakup radius is about $5.7d$ from the point of impingement, although further study of the scaling of both breakup radius and splattered droplets profiles was needed. They observed that the jet splatters when $We_d > 2120$ for any $Z/d$.

Bhunia and Lienhard (1994) concluded that the amount of splattering at a given nozzle-to-surface separation depends principally on the jet Weber number, which was the same observation as Lienhard et al. (1992). Over the range of Reynolds numbers in their investigation, no significant effect of jet Reynolds number was identifiable. The presence of surfactant detergent (soap-water mixtures) in the jet did not alter the amount of splattering, presumably because the surfactant is unable to reach the surface at a concentration necessary to alter the surface tension, and only the surface tension of bulk fluid plays a role in splattering. For a turbulent jet, the amount of splattering is governed by the level of surface disturbances present on the surface of the jet. This observation is similar to those for laminar jets with externally imposed disturbances.

In an oil cooled engine the jet velocity is typically equal to the maximum piston velocity which is around 20 m.s$^{-1}$, which gives $Re_d = 4300$ and $We_d = 16500$ for $d = 2$ mm and $T_j = 120$ °C. Therefore, Figure 3-36 shows that in such a system more than 5% of the incoming fluid is splattered when the nozzle is
sitting at typical distances of 100 and 200 mm from the piston undercrown in the highly rated diesel engine.

3.7 Effects of Inclined Jet Impingement

Measurements for free-surface oblique jet impingements (circular jet impinged at inclined angles) were given by Stevens and Webb (1991), and Ma et al. (1997). The angle of inclination of the jet relative to the surface was varied from 90° (normal impingement) to 40°. Each paper reported an asymmetry of local Nusselt number about the stagnation point. Their results indicated a displacement of maximum heat transfer from the intersection of nozzle centre line and impact plane, and the displacement continued to increase as the jet angle becomes more inclined.

Stevens and Webb (1991) reported their experimental results for fully developed turbulent jet impinged obliquely onto a surface. They found the maximum Nusselt number increased with respect to the inclined jet flow, and the profile became increasingly asymmetric and shifted upstream. They suggested that it may be caused by a slight increase in the velocity gradient in the impingement region due to the required change in fluid flow direction of greater than 90° on the upstream side of the inclined jet.

![Figure 3-37: Schematic of inclined liquid jet – redrawn from Ma et al. (1997)](image-url)
Ma et al. (1997) measured the displacements of the location of maximum heat transfer from the intersection of nozzle centre line and impact plane at various Reynolds numbers for a parabolic profile laminar jet. They established the following correlation

\[
s / d = (0.119 + 0.00454\theta)\cos\theta
\]

where \(s\) is the maximum heat transfer point from the jet impingement point, \(d\) is the jet diameter, and \(\theta\) jet inclination angle (in radians) are defined in Figure 3-37.

Equation (3-35) was stated to be valid for \(d = 0.987\) mm, \(235 \leq \text{Re}_d \leq 1745, 202 \leq \text{Pr} \leq 260\). They observed that the maximum local Nusselt number reduced with increasing jet inclination, which was not the same as observed by Stevens and Webb (1991). This phenomenon is later discussed in Tong (2003) who measured distributions of heat transfer for uniform and parabolic profile jets impinging obliquely onto a flat surface.

Stevens and Webb (1991) and Ma et al. (1997) also found that the values of local Nusselt number on the upstream side of the maximum point tend to drop off more rapidly while those on the downstream side tend to decline more slowly. A more rapid decrease of Nusselt number along the upstream direction than the downstream direction is more noticeable at larger inclinations. This creates significant imbalance of cooling or heating capabilities on either sides of the stagnation point. This result is to be expected in view of the positive dependence of heat transfer on Reynolds number, and the much higher local flow rates on the downstream side of the profile for oblique impingement.
Tong (2003) has performed an interesting numerical study to understand the hydrodynamics and heat transfer mechanism of the free-surface impingement process of an oblique liquid plane jet of \( w_n = 2 \text{ mm} \). He has considered two different jet profiles, the uniform and parabolic jets, in identifying the locations of the maximum Nusselt number on the surface, as shown in Figure 3-38 for a jet Reynolds number of 10000 and 5000, respectively.

Figure 3-38: Computed Nusselt number distributions at various jet inclinations: (a) a uniform jet; (b) a parabolic jet - from Tong (2003) with changed annotation.

In the case of uniform jet impingement (Figure 3-38a), the maximum Nusselt number increased and shifted to the left side (upstream) as the jet inclination increased, which gave the same observation by other authors of experimental works (i.e. Stevens and Webb, 1991). Also, when the jet inclination increased the peak Nusselt number was almost twice as large as that of a uniform jet impinging at a normal angle. The most interesting finding for a parabolic jet impingement (Figure 3-38b) was the initial decline (as observed by Ma et al., 1997) and subsequent rise of the maximum Nusselt number with changing jet inclination. Therefore, the inlet velocity profile evidently plays key role in the flow and subsequently in the heat transfer.
Comparing the Nusselt number distributions for both the uniform and parabolic jets, in Figure 3-38 at normal impingement ($90^\circ$), the maximum value of Nu for the uniform jet was significantly lower to the peak for parabolic jet. At $75^\circ$ inclination, beside a decline in the maximum Nusselt number for the parabolic jet, the upstream shift was also found to be less than that for the uniform jet. Unlike the uniform jet, the flow for the parabolic jet is more concentrated at the jet centre. As the inclination begins, the peak spatial velocity gradient, which has a strong influence on the heat transfer, reduced in magnitude as it shifted upstream due to the uneven turning of the flow as it approached the wall. In other words, the asymmetry of the oblique impingement, aided by the parabolic velocity profile, reduces the spatial velocity gradient and hence the heat transfer. This trend continues and the influence propagates upstream with declining magnitude as the inclination increased. Eventually, the trend reverses as more of the higher velocity fluid at the inner region is, as shown by an arrow in Figure 3-37, affected and required to turn. Note that the decline in maximum Nusselt number has reduced at $60^\circ$ and the shift is closer to that of the uniform jet. At $45^\circ$, the maximum Nusselt number for a parabolic jet is higher than that at $60^\circ$ and $75^\circ$, but remained below Nu for the normal impingement. In view of the numerical results obtained in Figure 3-38, the contradiction in findings previously mentioned between Stevens and Webb (1991) and Ma et al. (1997) might be a result of the jet inlet velocity profile (Tong, 2003).
3.8 Concluding Remarks

This chapter has reviewed the literature on research into free-surface liquid jet impingement on upward-facing surfaces. The literature has highlighted the previous research methodologies and investigations, many of which had industrial applications. These include simulations and experimental techniques employed.

The review of literature has revealed that:

- There are specific types of liquid jet flow (i.e. laminar, transitional and turbulent) which depend on jet nozzle designs and operating conditions.
- The jet diameter, velocity and temperature are specifically key contributing factors that affect heat transfer enhancement and control of jet behaviour (i.e. breakup and splatter).
- The reported jet behaviour, normal jet impingement and oblique jet impingement studies will provide useful information for the present study in interpretation of the data.
- The heat transfer models from previous research works are useful for the present study, especially for the stagnation zone of liquid jet impingement where the highest heat transfer coefficient is located.

A key conclusion is that no specific study has been found by the present author on upward free-surface flowing jets on downward-facing heated surfaces, which is the configuration relevant to IC engine piston cooling. Hence, this emphasises the importance of present research to study the upward liquid jet impingement on various type of test-surfaces (i.e. flat, concave and piston undercrown).

The next chapter of this thesis looks specifically at the experimental apparatus and equipment used for present study.
Studies of piston cooling reviewed in Chapter 2 discussed the importance of direct impingement of engine oil on piston undercrows. Oil jets provided the best choice of cooling the piston while at the same time lubricating the piston pin and cylinder wall. Chapter 3 showed that there appears to be no published literature on upward flowing jets on downward-facing heated surfaces. The specific areas where research is required are the effects of cooling oil jet temperature, velocity, size and quality.

This chapter discusses the apparatus that was used to gain knowledge of cooling jet impingement on downward-facing hot surfaces. The next sections describe the design of experimental equipment for the heat transfer measurements. It also describes the photographic methods used to capture jet and impingement images. The accuracy of the experimental equipment is discussed near the end of this chapter.
4.1 Experimental Arrangement

The experimental facility built for this study was designed to impinge a controlled, re-circulated liquid flow onto a heated surface. The purpose of the test is to:

1. measure the distributions of heat transfer coefficient on the test-surfaces;
2. explore the effects of oil jet velocity, temperature, impingement angle and nozzle type on the local heat transfers;
3. capture the images of jet impingements (i.e. breakup and splatter).

The key components of the apparatus, as shown in Figure 4-1, were a test chamber, nozzle holder assembly, test section, pipeline assembly, electrical heating unit, data acquisition unit and digital camera.

Figure 4-1: Schematic of experimental facility.
The experimental test-rig for the present research was installed on a metal frame with swivel wheels for easy transportation, as shown in Figure 4-2. The test-rig was used to generate experimental data for the study of oil jet cooling.

![Figure 4-2: Set-up of oil jet test-rig.](image)

The test chamber was constructed of stainless steel with four transparent windows made of Perspex to view the impingement process. A ventilation duct was attached at the roof of the chamber to ensure that the oil mist and any smoke generated inside the chamber were extracted from the room to meet the health and safety requirements.

The test fluid is circulated in a closed loop system. Liquid jets issue from contracting-type and pipe-type jet nozzles. The jet nozzle is aimed vertically upward or rotated about a horizontal axis. The inclination of the nozzle can
be read from the angle markings described on the surface of the rotator stage. The nozzle holder assembly is fixed on a two-dimensional coordinate rack and can be adjusted with respect to the test section with a resolution of better than ±0.1 mm.

The fluid inside a stainless steel reservoir was heated using a digital immersion heater (with temperature controlled to an accuracy of ±0.1 °C), and delivered by a positive displacement gear pump to the nozzle. Along the pipeline assembly a control valve, by-pass valve, flow meter and pressure gauge were also arranged to control the flow rate, and to take the readings of flow rate and pressure inside the duct. The by-pass valve was used for flow rate control. The flow meter (with accuracy ±2% of full scale, and repeatability ±1%) and the “C-shaped” Bourdon tube type pressure gauges (pressure range 1 - 16 bar, and with accuracy ±1.6% of full scale) were used to display the volumetric flow rate of liquid (in litre.min⁻¹) and liquid pressure (in bar) inside the duct, respectively.

The heat transfer test modules will be described in more detailed in Section 4.1.1 below. The test section assembly was fixed horizontally at one side of the test chamber wall and was mounted below the ventilation duct. The test section could be rotated about a horizontal axis and its inclination relative to the jet nozzle could be read by the angle markings described on the surface of the rotator stage. Inside the test section, the cartridge heaters for heating the test-block were controlled automatically using a PID digital temperature controller, and the temperature measurements were taken from thermocouples fitted inside the heater block. The resulting signals from up to fifteen thermocouples were fed back to a PC-based software program through a 15-channel thermocouple data logger box. Details of the test section assembly are given in Section 4.1.1.
Visualisation of the impinging oil jet was made using a digital camera and halogen light source. At specified positions, the digital camera focus length was adjusted manually and saved as default in its built-in program to ensure that the lens focus point is always co-ordinated with the object distance from the camera location. A 50W halogen light was used to illuminate the object through the clear Perspex window, in order to achieve sharp focused and well-exposed photos from the camera.

### 4.1.1 Heat Transfer Test Modules

The heat transfer test-blocks for the present study were made of aluminium having a thermal conductivity of 202.8 W.m\(^{-1}\).K\(^{-1}\) at temperature 20 °C (Kreith and Bohn, 2001). The aluminium block was precision machined with a CNC machine. The purpose of the test section was to make measurements of heat flux in specimens of simple shape, e.g. cylindrical, which resembles the shape of IC engine pistons.

In Figure 4-3, a cross-section of the test-block’s top surface, small holes were drilled at the centre point and in the radial direction with distance of 6.25 mm between holes from the centre point to radial distance of 43.75 mm. These small holes were drilled 2 mm in diameter, which were drilled within 1 mm above the test-surface and counterbored 3 mm in diameter at a depth of 20 mm. K-type thermocouples were fitted inside these small holes until the thermocouple junction touched the bottom end of each hole. The blocks also have a machined flange at the outer surface for mounting purposes. The bottom section is machined to 100 mm diameter to resemble a common bore size of IC engine pistons.
(a) Flat Test Surface

Measuring surface temperature

Two thermocouples inside the same hole for measuring heat flux

(b) Concave Test Surface

Counterbored Holes

Figure 4-3: Surface-temperature measurements with thermocouple holes inside the aluminium test-blocks.

The test-block was designed to ensure that surface-temperature measurements did not affect the liquid jet impinging on the test-surface. As discussed by Baker et al. (1961), the techniques for temperature measurement
at an interior point can be applied to the measurement of surface temperature by choosing the point sufficiently close to the surface if the thermal gradient in the block was small. See Appendix B for thermocouples measurement inside the present test specimens.

As shown in Figure 4-4, the top section of the aluminium block is machined to a rectangular shape with two horizontal holes to accommodate heating elements (cartridge heaters of 535W each).

![Figure 4-4: Heat transfer aluminium test-block.](image)

The test-block was secured and enclosed inside a stainless steel carriage, as shown in Figure 4-5, which was built to hold the weights of the aluminium block and all the heat transfer components and to withstand the heat generated by the heating elements inside the test-block. It also had an air gap insulation (as shown in Figure 4-5) to minimise the heat losses through the block surfaces, so that near uniform surface temperatures could be achieved. The only part that was not covered inside the box was the test-surface (or cooling surface).
Figure 4-5 shows the heat transfer test-block whose key features were:

- Realistic test-surface area comparable to piston bore size in IC engine.
- Downward-facing test-surface to study upward jet impingements on piston underside.
- Adequate height to ensure uniform heat distribution at the bottom surface.
- Temperature measurements inside the solid.
- Temperature limits in excess of 660 °C (1220 °F).
- Cartridge heater, which could be changed according to heat flux required.

The thickness of the test-block as well as the use of high purity, high thermal conductivity aluminium was dictated by a desire to approximate as nearly as possible a uniform surface temperature boundary at the test-surface. In Figure 4-6, the temperature distribution within the aluminium block was predicted by using a three-dimensional (3-D) finite-element solution modelled using GAMBIT and FLUENT programs, with boundary conditions of a constant surface temperature of 150 °C at the two horizontal holes and a constant ambient temperature (i.e. $T_a = 36.3$ °C, as measured inside the test chamber) at all other surfaces of the test-block. The ambient heat transfer coefficient ($h_a$) at the bottom of the test-block is 10 W.m$^{-2}$.K$^{-1}$ and at the rest of the surfaces is 20 W.m$^{-2}$.K$^{-1}$. 

Figure 4-5: Cross-sectional schematic of the test-block location
Figure 4-6 shows the predicted test-surface temperatures for the 3-D test-block. The result shows the temperature was highest at the centre-point and decreased slightly with radius. The maximum temperature difference between the centre-point \( (r = 0 \text{ mm}) \) and the edge-point \( (r = 43.75 \text{ mm}) \) was less than 1 °C. This is due to heat losses through the test-block side surfaces, hence lower temperatures near to the edge of the test-surface.

Figure 4-6: Prediction of temperature distribution across the test-block modelled using FLUENT and GAMBIT programs.

The predicted surface temperatures for the test-block (as illustrated in Figure 4-6) were also confirmed during the test programme and the observed maximum temperature difference between the centre-point \( (r = 0 \text{ mm}) \) and the edge-point \( (r = 43.75 \text{ mm}) \) was less than 1 °C, as shown in Figure 4-7.
heating elements inside the top section of the test-block were maintained at 150 °C and the measured ambient temperature inside the test chamber is 36.3 °C.

Figure 4-7: Near uniform surface temperatures in the experimental test.

**Heat Transfer Surface Geometry**

Since no publications were found in the literature for upward jet impingement on a downward-facing surface, the two generic surface geometries were selected for the present study: the smooth flat and concave surfaces, as shown in Figure 4-8.

The root-mean-square (RMS) roughness heights for these flat and concave surfaces are 0.627 μm and 1.083 μm respectively, characterised using non-destructive surface roughness test equipment, ZYGO’s NewView 5000, and having a vertical resolution up to 0.1 nm. The NewView 5000 uses scanning white-light interferometry to image and to measure the micro structure and topography of surfaces in three dimensions.
Figure 4-8: Cooling surface geometries (a) flat surface; (b) concave surface.

**Heat Transfer at Piston Undercrown**

With knowledge gained from tests on flat and concave surfaces, heat transfer tests were conducted during the final stages of this research on a piston underside, namely an Al-Si piston from a Perkins 1104C (4.4 litre) 4-cylinder diesel engine (as shown in Figure 4-9), with bore size of 105 mm. The undercrown surface was manufactured with a crossed surface structure and the RMS roughness is approximately 300 μm. The piston material is made of aluminium alloy containing 12% Silicon and the thermal conductivity is 135 W.m⁻¹.K⁻¹ (private communication Perkins Engines Ltd).
4.1.2 Oil Jet Nozzles

Experiments were performed using two types of jet nozzle:

a) The contracting-type nozzles with orifice diameter of 1.5 and 3 mm, as shown in Figure-4-10, produced by the in-house mechanical engineering workshop (see Appendix A for detailed technical drawings and digital microscope images). The design specification for these nozzles was described in Mian (1997).
b) The **automotive nozzles** with orifice diameter of approximately 2 mm, which are production automotive oil jet nozzles acquired from Perkins Engines Company Ltd and BING Power Systems GmbH, as shown in Figure 4-11 & Figure 4-12 respectively. The Perkins oil jet has a converging section near the end of its orifice (Figure 4-11). Whilst, the BING oil jet has a straight section up to the end of its orifice (Figure 4-12). The BING oil jet also has a better finishing at the orifice lip compared to Perkins oil jet (see Appendix A), and these two nozzles were tested in the present research to compare the quality of their oil jets with those of the idealised contracting-type nozzles manufactured in-house.

![Orifice's lip with irregular surface finishing](image)

![Figure 4-11: Converging-type automotive jet for Perkins 1104C (4.4 litre) 4-cylinder diesel engine.](image)
4.1.3 Test Fluid (Coolant)

The type of diesel engine oil chosen in the present study was *Shell Rimula X (SAE 15W-40)*. This oil is available from Shell UK Products Ltd and is formulated to meet the 1998/Euro 2 on-highway exhaust emission standards as well as being suitable for a wide range of heavy-duty off-highway applications. Rimula X at API Service Category CH-4 is a mixture of highly refined mineral oils and additives that will enhance the lubricating and cooling effects inside engines. More studies on heat transfer characteristics of some oils used for engine cooling were found in Hosny and Abou (2004).

Rimula X is a high performance and dedicated heavy-duty engine lubricant for use in modern high-speed turbocharged diesel engines. This engine oil is recommended for most engine types found in on-highway heavy duty trucks, construction, mining and agricultural equipments. It is particularly claimed to be suitable for Caterpillar, Cummins, Detroit Diesel and Komatsu engines. It is also formulated to provide continuous protection even where higher sulphur fuels are used. More information about Rimula X can be viewed in Appendix D.
4.2 Jet Impingement Analysis

The advantages and disadvantages of different experimental approaches for recording and analysing liquid jet impingement were briefly reviewed. Consideration was given to studies such as Ruch and Holman (1975), Monde and Katto (1978), and Monde and Okuma (1985) who used simple photographic techniques on upward jet impingement analysis.

The pros and cons found in the past investigations were highlighted, and the information gained was used when developing the tools for this study. The work of these authors is now discussed in more detail.

4.2.1 Previous Photographic Studies

Studies such as those by Katto and Kunihiro (1973), Monde and Furukawa (1988), Lienhard et al. (1992), Wolf et al. (1996), Naraghi et al. (1999), and Hall et al. (2001) used simple photographic techniques to observe the liquid behaviour and to measure boiling characteristics on the upward-facing heated surface. For downward-facing heated surface, Ruch and Holman (1975), Monde and Katto (1978), and Monde and Okuma (1985) also used the same photographic techniques in their experiments. In Vader et al. (1991), photographs of jet impingement were made with a strobe light (flash), which makes the surface structure more visible. Lee et al. (2004) advanced these techniques by combining the use of a halogen light to illuminate the focused object.

In the case of an upward-facing heated surface, liquid jet impingement was photographed and the authors were therefore able to observe key features such as formation of vapour bubbles, splattering and hydraulic jump on the surface. However, in the case of downward-facing heated surface, visibility limits the usefulness of this method of jet impingement analysis. The natural falling of liquid films by gravity may obstruct the view and the splattering droplets that depart from the surface at certain speeds and angles may hit the
wall and blur the transparent enclosure of the test-rig. Therefore, in order to study the upward jet impingement on the surface, the enclosure panel needs to stand at adequate distance from the droplet splattering zone, and the camera sitting behind the panel needs to operate at high speed and large distance from the focused object. Advances in camera technology over the last few years mean that equipment capable of operating at this distance is now much more affordable.

The analysis and measurement of the results of work undertaken by Ruch and Holman (1975), Monde and Katto (1978), and Monde and Okuma (1985) was carried out on a downward-facing superheated surface with impinging jets of water and Refrigerant (R-113) or Freon (F-113). Ruch and Holman (1975) photographed their jet impingement at large distance and the recorded image in their picture was quite small to observe. Monde and Katto (1978) photographed their jet impingement on an unheated downward-facing surface at close distance and the recorded image inside their boiling vessel was easily observed. At high surface heat flux, liquid splashed from the surface due to the strong ejection of vapour from the nucleate-boiling liquid layer, and thus blurred the view. Monde and Okuma (1985) studied the critical heat flux in saturated forced convective boiling on a heated surface with an impinging jet. Some of their pictures showed a thin liquid film on the surface, and falling droplets as well as a few splashed droplets from the surface due to violent nucleate boiling.

The use of film cameras has been popular for more than one hundred years, but recently, the 35mm cameras have been replaced by digital charge coupled devices (CCDs). These cameras are capable of capturing high quality images and recording at frame rates from 30 to 20000+ frames per second (fps). Therefore, it is now possible to gather high speed and quality images, which can then be reviewed and analysed using computer software.
Digital CCD cameras have a built-in computer, which records images in a digital format. It has a mini screen built into the back of a camera, used to compose and to view digital images immediately after they are taken, and the digital images can be electronically stored on a memory card.

4.2.2 Digital CCD Camera

The development of digital CCD cameras now enables photographic studies to become significantly less reliant on manual analysis, as digital image processes can be used in the analysis of measurements. The benefits of optical non-contact measurements, combined with the speed, accuracy and reliability of an automated system (e.g. noise reduction, white balance etc.) is an ideal combination. In this study a photographic technique including digital image processing and automated data analysis was chosen. The system comprised a halogen light to illuminate the region of interest and a high-resolution digital CCD camera for images recording (see Figure 4-13).

![Diagram of photographic set-up](image)

Figure 4-13: Photographic set-up for liquid jet impingement.

With the camera sitting in front of the focused object, a halogen light was used to illuminate the object through the clear sidewall. The wall on the other side of the object was wrapped with a white colour to reflect some of the
light source from the halogen lamp and filling out some of the shadow cast by the light source.

![Image](image_url)

**Figure 4-14:** Images of jet impingement (a) with and (b) without illumination light.

Figure 4-14 shows some of the samples of liquid jet impingement taken in the test programme with and without the illumination of halogen light. Figure 4-14(a) was taken with the available lighting source from the room, and the oil jet is not clearly visible as the object and background are receiving the same amount of light source. Figure 4-14(b) was taken using the same camera and the oil flow was more clearly in focused with a much brighter light source and the contrast is stronger.

### 4.3 Data Logging

In this study, K-type glass fibre sheath thermocouples with junction tip size of less than 0.5 mm diameter were used to measure temperatures throughout the experimental apparatus, including the wall temperature \((T_{\text{w}})\), temperature near to the heating elements \((T_{\text{h}})\), liquid jet temperature \((T_j)\) and ambient temperature \((T_a)\). Type K thermocouples are available in the range of -200 to +1200 °C, and its E.M.F./Temperature curve is virtually linear (Appendix C) and its sensitivity is 41μV/°C. Its Class 1 (or A) accuracy is typically ±1.5 °C. In the present measurement, the smallest temperature
difference at stagnation point for $T_H - T_m \approx 7^\circ C$ and $T_H - T_{aw} \approx 10^\circ C$, thus the expected error is 21.4% and 15%, respectively.

In the present study, temperature data were acquired using a 15-channel data acquisition unit and logger was used to record temperature values up to 19.5 Hz (per thermocouple). The tests involved records from fifteen thermocouples, which were sampled at an interval time of approximately 0.77 s, which is the maximum rate for the present data logger. Tests showed that the averaged temperature was not affected by the sampling rate for decreases of sampling rate below 1.3 Hz. So this frequency was used to acquire the largest number of measurements per unit time.

Using only the data acquisition unit and computerised spreadsheets eliminated any uncertainties due to human error. Temperature data was saved directly using the program Visual Basic and was entered into an MS Excel spreadsheet. An MS Excel program was constructed to process the data, which enabled immediate assessment of the data and presentation in graphical format for the observation of heat transfer regime under the liquid jet impingement.

### 4.4 Accuracy and Calibration

All the experiments were conducted at ambient room temperature and steady-state conditions. These tests were carried out with variation of test-surface geometry, nozzle-to-surface spacing, jet inclination angle, jet temperature, flow rate, and diameter of orifice. Jet impingement area on the surface was measured with the images recorded by the digital camera.

Every effort such as calibrating thermocouples in-situ, defining oil jet temperature at orifice outlet and identifying photographing errors was made to ensure data accuracy and to minimise errors. Particular attention was paid to show how the impingement images were gathered and analysed.
especially for tests to study the splattering of the jet. Other areas where inherent errors are likely to occur include the temperature readings measurement. Both these issues are discussed in the following sub-sections.

Each thermocouple hole was filled with a silicon thermal paste (stable within temperature range of -50 to +200 °C), to ensure full thermal contact between the thermocouple tip and heater surface. These thermocouples were electrically insulated with a thermally conducting material such like exhaust assembly putty, which could resist temperatures up to 1000 °C.

4.4.1 Thermocouples Calibration with Melting Ice-Water

The accuracy of the thermocouples within the test-block was established in the following procedure. The thermocouples inside the aluminium block for present experiment were calibrated using the ice-point technique. As discussed by McGee (1988), thermocouples can be calibrated by immersing a high-conductivity-metal block in the ice bath and inserting them into holes drilled in the block. The ice-point technique is suitable as a calibrating point for many applications requiring accuracy up to 1 mK. Ordinary distilled (deionised) water was used for this test.

In the present test programme, Figure 4-15 shows that the calibration was done by dipping the test-surface of the test-block in a melting ice and filled with an ordinary deionised water to ensure good thermal contact with the surface. Figure 4-16 shows the water triple point temperature at 0.01 °C (McGee, 1988).
Figure 4-15: Thermocouple calibration inside the aluminium block with melting ice.

Figure 4-16: Phase diagram for water showing the triple point A, a thermodynamic invariant point – McGee (1988)

Therefore, the melting ice temperature at atmospheric pressure in the present test programme was expected to be very close to water triple point. The aluminium test-blocks in the present study were made of high thermal conductivity material, and this suggested that the temperature difference within 1 mm (i.e. $\Delta y = 10^{-3}$ m) would be low and measurements were therefore assumed to be equivalent to surface temperature.
Figure 4-17: Measured temperatures between two points at central axis.

Figure 4-18: Measured temperatures between two points along a vertical axis at radial $r = 37.5$ mm.

Figures 4-17, 4-18 and 4-19 show the measured temperatures using thermocouples inside the test-block. In Figures 4-17 and 4-18, the test-block was initially at ambient temperature, and took approximately 90 seconds to
reach the steady-state. After 90 seconds, both the temperatures of these two points remain unchanged about 0 °C at atmospheric pressure. During cooling-state, the temperature of near surface point cooled faster than the interior point that 29 mm above it, thus there was a heat flux flows between these two points. Another important thing about this cooling test is there was no significant different of cooling trend in between Figures 4-17 and 4-18.

![Diagram showing measured surface temperatures between centre point and radial point r = 37.5 mm.]

Figure 4-19: Measured surface temperatures between centre point and radial point \( r = 37.5 \text{ mm} \).

In Figure 4-19, throughout the cooling process there was no significant of temperature difference between the two points that near to the surface and at steady-state, the worst case scenario of these thermocouples being 1 mm from the cooling surface returned an average measured temperature of approximately 0.03 °C, which is sufficiently close to the measured melting ice temperature of 0.026 °C at atmospheric pressure in the present setup. The melting point of ice is 0 °C (Cengal and Boles, 1998). The measured value of melting ice temperature from the present test was also dependent on water purity and ambient pressure. See Appendix E for a heat balance for the present test-block.
4.4.2 Oil Jet Temperature at Orifice Outlet

The temperature of the liquid jet coming out of the nozzle was measured with a K-type thermocouple placed near to the nozzle orifice, as shown in Figure 4-20, and a cable tie was used to hold the thermocouple firmly on the nozzle. Similar measurement method was adopted by Elison and Webb (1994). After the measurement of the oil temperature the thermocouple was always removed from the orifice outlet to ensure an undisturbed liquid jet flow.

![Diagram of measuring liquid jet temperature at orifice outlet](image)

Figure 4-20: Measuring liquid jet temperature at orifice outlet.

Prior to each test, the liquid was allowed to circulate in the system until the jet temperature was steady at the orifice outlet. The influence of flow rates on outlet jet temperature was examined in the test programme. The temperature changes were also determined with accuracy of ±0.5 °C from the thermocouple measurements. Therefore, liquid inside the heater reservoir in this study must be heated to temperatures as examined in the test programme, in order to compensate for heat losses throughout the piping system, and thus, achieving the required jet temperatures at the orifice outlet.
4.4.3 Distortion and Perspective Errors in Photographing

The jet radii prior to impingement point and impingement size on the test-surface were recorded with the CCD camera, and analysed by digital image processing for measurements. Every effort was made to ensure accuracy and to minimise errors in the recorded images.

The following photographic procedure was used:

1. **Accuracy**: During set-up, the camera lens was focused onto a known surface (the grid’s surface) which was aligned at the test-surface centre line, and the camera was mounted on the 3-D head that allows the camera to pan and tilt, and the tripod was mounted on a rail that allows the camera to move back and forth in tiny increments with reference to its specified zoom lengths.

2. **Quality**: The halogen lamp and tripod-rail were used in the photographing applications to illuminate the focused object and to hold the camera firmly to avoid vibrations.

3. **Repeatability**: The CCD camera lens had a fixed five-focus zoom, each zoom set by a marked position on the rail. Therefore, the camera zoom was always set in one of the five calibrated positions.

As the jet impingement point was situated at the centre point of the test-surface, the accurately machined cylindrical test-block \( D = 100 \text{ mm} \) was used as a physical scale for the measurement of impingement area. However, the captured size of cylindrical test-block in the image was not accurate for the measurement analysis, and this is due to the perspective errors as shown in Figure 4-21.
Figure 4-21: Perspective errors in photographing a cylindrical object.

Perspective error, also known as parallax error, is generally encountered in close-up photography where the viewing point or camera lens is at a short distance from the focused cylindrical object. This error/discrepancy increases as the camera moves closer to the object. A trigonometric correction method, as illustrated in Figure 4-22, was used to determine the errors provided the following parameters are known, i.e. radius of the cylindrical object, and distance in between the viewing point and the object centre point.
Figure 4-22: Trigonometry method for defining the viewed diameter.

The following formulae were used to define the viewed diameter ($L_C$) of a cylindrical object

$$\theta = \sin^{-1} \left( \frac{R_C}{L_A} \right)$$  \hspace{1cm} (4-1)

$$L_B = L_A \cos \theta$$  \hspace{1cm} (4-2)

$$L_C = 2(L_B \sin \theta)$$  \hspace{1cm} (4-3)

where $R_C$ is the radius of the cylindrical object, $L_A$ is the distance in between the viewing point and the object centre point, $L_B$ is the distance in between
the viewing point and the viewed edge, and \( \theta \) (in radians) is the angle of \( L_B \) with respect to \( L_A \). Hence, the perspective error can be determined by

\[
100 \times \left( \frac{D - L_C}{D} \right) \%
\]

(4-4)

Therefore, in the present study, attention was paid to how the perspective errors can be minimised for the recorded jet impingement images. A method for achieving accurate measurements of the image was developed in the test programme, using a grid printed to scale (grid size 1 mm) and held using an angle bar that aligned exactly at the test-surface centre line, as shown in Figure 4-23, and the grid surface was used as a correct focus point for the camera focus length setting.

![Test-Plate](image)

The grid was aligned at the test surface centre line and located at a distance of 35 cm from the camera lens.

![Enlarged digital image.](image)

Figure 4-23: Printed grids surface for camera focus point.

In the digital image, the printed grid was used to indicate the number of pixels per millimetre. The images were thresholded in order to distinguish between the black line and the illuminated background. The pixel area of the grids was then measured as shown in Figure 4-24. Hence, with the known size per pixel, the number of pixels in the object of interest could be measured accurately. In actual measurement, the number of pixels per millimetre length printed on the grid surface is 20 pixels and the number of
pixels for 100 mm (100 grids) length printed on the grid surface is 2000 pixels.

![Image of grid and camera setup]

Figure 4-24: Perspective error in present camera set-up.

Figure 4-24 shows the perspective error expressed as a number of pixels representing a 1 mm grid length. The maximum difference in the number of pixels per millimetre length was found to be 1 pixel (equivalent 50 μm), for the current camera arrangement. However, the number of pixels for 100 mm length was measured on the image to be 1920 pixels. Therefore, the observed perspective error for the current camera set-up in taking pictures of the test-block of 100 mm diameter was 4%, when the camera was placed at a distance of 35 cm away from the focus point. The maximum error of 4% applies when the size of a 100 mm diameter jet impingement region is measured. However, in the present work, the typical diameters of jet impingement were less than 100 mm (the test-surface diameter), so the expected errors were less than 4%.
4.5 Concluding Remarks

This chapter has described the design and development of a test apparatus for a heat transfer study of liquid jet impingement onto downward-facing surfaces. Past investigations in photographic techniques were discussed and the lessons learned were used to select the individual components of the apparatus used in this study.

The main focus of the design was on the generation of accurate, stable and repeatable results in the photographic and heat transfer measurements. Human error was minimised in the experimental work by selecting the following equipment:

- CCD digital camera for photographing digital images and recording short videos.
- Data logging system for recording the temperature values from fifteen thermocouples, and a single computer system for controlling the thermocouples measuring rate through out the apparatus as well as recording and saving the results.
- Automatic control ‘loop’ system for controlling the temperature of heating elements inside the test-block.
- Digital controller immersion unit for heating the test liquid inside the reservoir.

A 3-D finite element (FE) model was carried out in the heat transfer analysis for the test-block to test the heat distribution at the test-surface. Results from the practical test were taken, analysed and compared to the previous FE results, and were shown to give adequate uniformity and repeatability. The test apparatus was therefore shown to be appropriate for this research and test-blocks could be tested successfully with an impinging cooling liquid jet.

Chapter 5 illustrates the experimental methods and work for present study.
CHAPTER 5

EXPERIMENTAL METHODOLOGY AND MEASUREMENTS

This chapter describes the methodology of the experimental study. A number of parameters were selected to investigate the effects of fluid velocity and viscosity on local heat transfer and flow structure from an upward liquid jet impingement on downward-facing hot surfaces. The configurations of the experimental apparatus such as test-surfaces and jet nozzles are also described.

In addition, a case study of oil jet cooling on a real piston undercrown is described. The characteristics of the piston in engine operation were investigated in order to identify potential overheated regions and to show how improvements to piston cooling could be made.
5.1 Experimental Methodology

As discussed in Chapter 4, careful calibration of the instrumentation was required to ensure that good quality experimental data could be obtained from the test-rig. Also, the use of image analysis for the measurement of liquid jet impingement was described. In this chapter, the experimental methodology that was used to define sets of experimental conditions is explained. This includes the procedure of operating the test rig, selection of experimental parameters and arrangement of experimental tests for obtaining enough data for this study.

For the study of local heat transfer, the effects of several cooling oil jet parameters were examined, namely: jet Reynolds number, Prandtl number, nozzle-to-surface separation and inclination angle of jet impinging on the cooling surface. The control of these parameters will be discussed in more detail in the later sections in this chapter.

Another important criterion considered in the study was the engine running temperatures which can vary when an engine is running at low load and high load conditions, or during idle condition. The control of oil temperatures is important since oil viscosity varies strongly with temperature. To illustrate the difference in viscosity, for typical heavy-duty diesel engine oil (SAE 15W-40) the kinematic viscosity at 120 °C is 9 x 10⁻⁶ m².s⁻¹ and 7871 x 10⁻⁶ m².s⁻¹ at -20 °C. In most diesel engine piston cooling systems, the typical oil jet operating temperature is from 100 to 120 °C.

The principal factors affecting heat extraction by oil jet cooling are:

1. Surface geometry (or piston undercrown design)
2. Oil velocity
3. Oil jet nozzle design
4. Temperature differences between the hot surface and cooling oil.
For each series of tests with a particular nozzle, the nozzle-to-surface separation was adjusted to the desired distance. For each test run, the desired flow rate was set to obtain the desired jet velocity and the power to the reservoir heater was adjusted to obtain the desired jet temperature. The heater block was heated to a nearly uniform temperature as indicated by the thermocouples. Once test conditions were reached, the test-surface was allowed to reach steady-state and all temperature measurements were made. Therefore, every test run included only steady-state data. Jet Reynolds numbers and Prandtl numbers, local heat transfer coefficients, and local Nusselt numbers were then calculated from the oil flow and temperature, heat flux, and local heated surface temperature data. Jet Reynolds numbers and Prandtl numbers were calculated from the oil flow rate and temperature which were evaluated at the orifice exit. In the present study, the temperatures used to find the material properties needed to evaluate these dimensionless numbers are 55, 70, 85, 100 and 120 °C.

The following sub-sections describe the test methods used to obtain high-quality experimental data.

5.1.1 Test Procedure

A test procedure was developed for precise use of the experimental rig to obtain data of cooling oil jet impingement for the present study. Every test was conducted in an identical manner to ensure the consistency of data and also to allow the results to be compared.

Figure 5-1 shows a schematic of the testing profile used throughout this experimental study.
During the experiment, the test-surface was heated by cartridge heaters to a temperature of 150 °C for all tests in the present study. The power of the cartridge heaters was automatically set at a low level during the warm-up and equalising periods to maintain the test-surface temperature as close as possible to 150 °C. This was held for a period of time until the entire system reached a stable temperature. The test-surface remained in steady-state until the test was started, where the cooling oil jet started impinging onto the test-surface. The power of the cartridge heaters was set at a high level during the cooling oil jet impingement process. The test-surface temperature was seen to drop quickly before levelling at its next steady-state during the sampling period. During the sampling, all temperatures were held at steady-state (±0.5 °C) and all readings were taken. Approximately 100 samples were recorded in order to obtain a good averaged temperature reading, and the observed variation was up to only 0.3 °C. Averaged data was used in the subsequent analysis.
Each test cycle was completed over different periods of time, the shortest one was 11 minutes and the longest one was 20 minutes. This was due to the warm-up and flow-back periods for different test set-ups. Cold oil jets took a shorter time to heat up to the required temperatures in the reservoir, but after the end of the test cycle it took a longer time to flow back from the test chamber to the reservoir, due to its high viscosity. Whilst hot oil jets took a longer time to heat up in the reservoir, the hot oil returned back to the reservoir much faster than cold oil because it was less viscous.

Oil jets at temperatures of 55, 70, 85, 100 and 120 °C were delivered by a positive displacement pump to the jet nozzle. Oil inside the reservoir was heated using a digital immersion heater and the oil had to be heated to temperatures higher than the desired oil jet temperatures to compensate for the heat losses in the experimental rig piping system.
5.2 Experimental Test Conditions

Tables 5-1, 5-2 and 5-3 show the overall experimental tests arrangement in the present study for oil jet cooling of flat surface, concave surface and piston undercrown.

<table>
<thead>
<tr>
<th>Disk Type</th>
<th>Nozzle Type</th>
<th>Jet Temperature, $T_j (°C)$</th>
<th>Jet Velocity, $u (m.s^{-1})$</th>
<th>Nozzle Setup, $Z (mm); \theta (°); x (mm)$</th>
<th>Number of Tests</th>
</tr>
</thead>
<tbody>
<tr>
<td>Disk1 (Flat Surface)</td>
<td>Contracting $d = 1.5\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>10, 15, 20, 25, 30, 35</td>
<td>$Z = 1.5, 15, 150$ ($Z/d = 1, 10, 100$); $x = 0; \theta = 90$</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>Contracting $d = 3.0\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>5, 7.5, 10</td>
<td>$Z = 3, 30, 300$ ($Z/d = 1, 1.10, 100$); $x = 0; \theta = 90$</td>
<td>45</td>
</tr>
<tr>
<td></td>
<td>Pipe (BING) $d = 2.0\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>10, 15, 20, 25</td>
<td>$Z = 50, 100, 150, 200, 300$ ($Z/d = 25, 50, 75, 100, 150$); $x = 0; \theta = 90$</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>Converging (Perkins) $d = 2.0\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>10, 15, 20, 25</td>
<td>$Z = 50, 100, 150, 200, 300$ ($Z/d = 25, 50, 75, 100, 150$); $x = 0; \theta = 90$</td>
<td>95</td>
</tr>
<tr>
<td></td>
<td>Converging (Perkins) $d = 2.0\ mm$ (reamed)</td>
<td>55, 70, 85, 100, 120</td>
<td>10, 15, 20, 25</td>
<td>$Z = 100$ ($Z/d = 50$); $x = 0; \theta = 90$</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>Contracting $d = 1.5\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>10, 15, 20, 25, 30, 35</td>
<td>$Z = 2; x = 0; \theta = 75, 60, 45$</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>Contracting $d = 3.0\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>5, 7.5, 10</td>
<td>$Z = 2; x = 0; \theta = 75, 60, 45$</td>
<td>45</td>
</tr>
</tbody>
</table>

Table 5-1: Experimental tests arrangement for oil jet cooling at flat surface.

<table>
<thead>
<tr>
<th>Disk Type</th>
<th>Nozzle type</th>
<th>Jet Temperature, $T_j (°C)$</th>
<th>Jet Velocity, $u (m.s^{-1})$</th>
<th>Nozzle Setup, $Z (mm); \theta (°); x (mm)$</th>
<th>Number of Tests</th>
</tr>
</thead>
<tbody>
<tr>
<td>Disk2 (Concave Surface)</td>
<td>Contracting $d = 1.5\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>10, 15, 20, 25, 30, 35</td>
<td>$Z = 3, 100, 200$ ($Z/d = 2, 66.67, 133.33$); $x = 0; \theta = 90$</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>Contracting $d = 3.0\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>5, 7.5, 10</td>
<td>$Z = 6, 100, 200$ ($Z/d = 2, 33.33, 66.67$); $x = 0; \theta = 90$</td>
<td>45</td>
</tr>
<tr>
<td></td>
<td>Contracting $d = 1.5\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>10, 15, 20, 25, 30, 35</td>
<td>$Z = 3$ ($Z/d = 2$); $x = 12.5, 25, 37.5$; $\theta = 76, 60, 41$</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>Contracting $d = 3.0\ mm$</td>
<td>55, 70, 85, 100, 120</td>
<td>5, 7.5, 10</td>
<td>$Z = 6$ ($Z/d = 2$); $x = 12.5, 25, 37.5$; $\theta = 76, 60, 41$</td>
<td>45</td>
</tr>
</tbody>
</table>

Table 5-2: Experimental tests arrangement for oil jet cooling at concave surface.
Table 5-3: Experimental tests arrangement for oil jet cooling at piston undercrown.

In the later sections in this chapter the reasons for these test conditions are explained.

5.3 **Nozzle Configurations**

The effects of nozzle configurations in downward liquid jet impingement have been broadly investigated by a number of workers in the past, and these effects on the local heat transfer were discussed in Sections 3.5, 3.6 and 3.7. Three different orifice diameters were used in the present research and are listed in Table 5-4. These orifice diameters were selected because they most closely model what would be used in the IC engine piston cooling application.

<table>
<thead>
<tr>
<th>Orifice Diameter, ( d ) (mm)</th>
<th>Orifice Length (mm)</th>
<th>Nozzle Type</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>3</td>
<td>Contracting</td>
<td>Manufactured In-house</td>
</tr>
<tr>
<td>2</td>
<td>6</td>
<td>Converging</td>
<td>Automotive oil jet</td>
</tr>
<tr>
<td>2</td>
<td>40</td>
<td>Pipe</td>
<td>Automotive oil jet</td>
</tr>
<tr>
<td>3</td>
<td>6</td>
<td>Contracting</td>
<td>Manufactured In-house</td>
</tr>
</tbody>
</table>

Table 5-4: Nozzle dimensions and types.

More details for the design specification of jet nozzles in Table 5-4 are given in Section 4.1.2.
Nozzle configurations can be subdivided into three categories:

- Surface-to-jet diameter ratio \((D/d)\)
- Nozzle-to-surface separation to jet diameter ratio \((Z/d)\)
- Oblique jet impingement or jet impinging at inclined angle \((\theta)\).

The different combinations tested in the present study were shown in the tables in Section 5.2.

In the present study, the different orifice sizes and nozzle standoff distances were used to show the effects of different ratios of surface-to-jet diameter \((D/d)\) and ratios of nozzle-to-surface separation to jet diameter \((Z/d)\), respectively, on the local heat transfer. Small ratios of nozzle-to-surface separation to jet diameter (i.e. \(Z/d = 1\) or \(2\)) were used to ensure that the gravitational effects influenced the jet velocities by less than 1% and also minimised changes to the jet structure due to aerodynamic drag on the jet surface. The Bernoulli’s equation, shown in Equation (5-1), was used to approximate the deceleration of the jet as it leaves the nozzle at velocity \((u)\) and reaches the plate at nozzle-to-surface separation \((Z)\) with upward jet impinging velocity \((u_f)\):

\[
    u_f = \sqrt{u^2 - 2gZ}
\]  

(5-1)

Table 5-5 shows the upward jet impinging velocities for the present study.
<table>
<thead>
<tr>
<th>Jet Diameter, ( d ) (mm)</th>
<th>Jet Standoff Distance, ( Z ) (mm)</th>
<th>Jet Velocity, ( u ) (m.s(^{-1}))</th>
<th>Jet Impinging Velocity, ( u_f ) (m.s(^{-1}))</th>
<th>Velocity Reduction (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>1.5</td>
<td>9.9985</td>
<td>0.0147</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>14.9990</td>
<td>0.0065</td>
<td></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>19.9993</td>
<td>0.0037</td>
<td></td>
<td></td>
</tr>
<tr>
<td>25</td>
<td>24.9994</td>
<td>0.0024</td>
<td></td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>29.9995</td>
<td>0.0016</td>
<td></td>
<td></td>
</tr>
<tr>
<td>35</td>
<td>34.9996</td>
<td>0.0012</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>9.8518</td>
<td>1.4825</td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>14.9016</td>
<td>0.6562</td>
<td></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>19.9263</td>
<td>0.3686</td>
<td></td>
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<td>25</td>
<td>24.9411</td>
<td>0.2357</td>
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<td></td>
</tr>
<tr>
<td>30</td>
<td>29.9509</td>
<td>0.1636</td>
<td></td>
<td></td>
</tr>
<tr>
<td>35</td>
<td>34.9579</td>
<td>0.1202</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>9.9980</td>
<td>0.0196</td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>14.9987</td>
<td>0.0087</td>
<td></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>19.9990</td>
<td>0.0049</td>
<td></td>
<td></td>
</tr>
<tr>
<td>25</td>
<td>24.9992</td>
<td>0.0031</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>9.8018</td>
<td>0.1964</td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>14.8686</td>
<td>0.0872</td>
<td></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>19.9017</td>
<td>0.0491</td>
<td></td>
<td></td>
</tr>
<tr>
<td>25</td>
<td>24.9214</td>
<td>0.0314</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>9.7012</td>
<td>1.9816</td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>14.8025</td>
<td>0.8758</td>
<td></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>19.8523</td>
<td>0.4917</td>
<td></td>
<td></td>
</tr>
<tr>
<td>25</td>
<td>24.8820</td>
<td>0.3144</td>
<td></td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>7.4961</td>
<td>0.523</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>7.4961</td>
<td>0.0523</td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>7.4607</td>
<td>0.5246</td>
<td></td>
<td></td>
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<tr>
<td>20</td>
<td>7.4607</td>
<td>0.2947</td>
<td></td>
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</tr>
<tr>
<td>25</td>
<td>7.43720</td>
<td>12.5609</td>
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</tr>
<tr>
<td>300</td>
<td>7.0968</td>
<td>5.3765</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>7.0968</td>
<td>2.9876</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 5-5: Reduction of upward jet impinging velocities due to gravitational effect.
Although, most of the experimental tests were performed with small nozzle stand-off distances, there were also some tests performed with large ratios of nozzle-to-surface separation to jet diameter, i.e. up to $Z/d = 150$. This is more representative of oil jet piston cooling applications and the data from these tests were used to identify the effects of large nozzle-to-surface separation on local heat transfer. The maximum velocity reduction in the present study is 12.5% for jet impingement with $d = 3.0$ mm at $u = 5$ m.s$^{-1}$ and $Z = 300$ mm (see Table 5-5). Therefore, in many cases the reduction is much smaller.

In Chapter 3, the vast majority of the available information on the heat transfer characteristics of impinging jets is with normal impingement (i.e. $\theta = 90^\circ$). The study of heat transfer from a jet impinging at an oblique angle to a surface has received relatively little attention. This is perhaps a reflection of the fact that normal impingement has more widespread applications. Nevertheless, oblique jet impingement also occurs in many applications, owing to the shape of the surface or to constraints on the positioning of the nozzle, i.e. cooling oil jet impinging at IC engine’s piston undercrown, where surface geometries are rarely flat. The heat transfer effect which is attributable to the asymmetric jet geometry is also important because it defines the imbalance of local heat transfer rates when the jet is impinged at inclined angles. Figure 5-2 shows a schematic of the tilted heated hot plate for the present study of upward liquid jet impingement. The heat transfer measurements were taken at jet impingement (inclination) angles of $\theta = 90^\circ$ (normal), $75^\circ$, $60^\circ$ and $45^\circ$ for each nozzle size. Results for normal impingement ($\theta = 90^\circ$) were used as a reference.
5.4 Jet Mass Flow Rate Selection

The available jet Reynolds numbers for the present study were dependent on the test-rig equipment, as illustrated in Section 4.1, i.e.

- Size of reservoir, for containing the amount of liquid (oil) used for liquid jet impingement tests. A large reservoir is needed to provide a large enough liquid supply for the large jet flow rate to run in a reasonable time.
- Size and power of digital immersion heater, for heating up liquid inside the reservoir to the required temperatures in a reasonable time. A large high power heater is needed to heat up the liquid inside a large reservoir.
- Capacity of oil pump, for delivering viscous oil from the reservoir to the jet nozzle. A high capacity oil pump is needed to deliver the large volumetric flow rate of liquid.
Table 5-6 shows the approximate flow rates for three orifice sizes that were selected for this study.

<table>
<thead>
<tr>
<th>Orifice Diameter (mm)</th>
<th>Volumetric Flow Rate (litre.min⁻¹)</th>
<th>Jet Velocity (m.s⁻¹)</th>
<th>Pressure (bar gauge)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>approx. 0 - 4</td>
<td>0 - 35</td>
<td>approx. 0 - 11</td>
</tr>
<tr>
<td>2.0</td>
<td>approx. 0 - 5</td>
<td>0 - 25</td>
<td>approx. 0 - 18</td>
</tr>
<tr>
<td>3.0</td>
<td>approx. 0 - 4.5</td>
<td>0 - 10</td>
<td>approx. 0 - 2.5</td>
</tr>
</tbody>
</table>

Table 5-6: Flow rates for different orifice sizes.

The experimental data covered three different types of jet flow: laminar, transitional, and turbulent flows, over a Reynolds number range of 263 to 5655, corresponding to the jet velocities from 5 to 35 m.s⁻¹. The jet Reynolds number was defined as in Equation (5-2) and its properties were evaluated using conditions at the nozzle orifice outlet to find velocity and oil viscosity.

\[
\text{Re}_d = \frac{ud}{v} = \frac{\rho ud}{\mu}
\]  

(5-2)

where \(u\) is the mean jet velocity, \(d\) is the jet diameter, \(v\) is the oil kinematic viscosity, \(\rho\) is the oil density, and \(\mu\) is the oil dynamic viscosity.

A pipelike nozzle (or non-uniform jet profile) provides turbulent supply flow to the jet when the Reynolds number exceeded a small value of \(\text{Re}_d = 2000\) to 4000, Liu et al. (1991). Therefore, the jet flow types considered for the present study were laminar flow for \(\text{Re}_d < 2000\), transitional flow for \(2000 \leq \text{Re}_d \leq 4000\) and turbulent flow for \(\text{Re}_d > 4000\).

For an orifice size of \(d = 1.5\) mm, the jet Reynolds numbers that could be obtained using the present test-rig were from 263 to 5655 (in Figure 5-3).
diagram highlights the strong dependence of the attainable range of Reynolds numbers on oil viscosity as the jet temperature varies.

![Diagram showing jet Reynolds number as a function of jet velocity.](image)

Figure 5-3: Jet Reynolds number as function of jet velocity ($d = 1.5$ mm).

For an orifice size of $d = 2.0$ mm, the jet Reynolds numbers that could be obtained using the present test-rig were from 351 to 5386 (in Figure 5-4).
Figure 5-4: Jet Reynolds number as function of jet velocity ($d = 2.0$ mm).

Figure 5-5 shows the results for a nozzle orifice of $d = 3.0$ mm and the jet Reynolds numbers that could be obtained using the present test-rig were from 263 to 3231. Due to the larger orifice size and lower jet velocities, nearly all the conditions were in the laminar regime, and only a few tests were in the transitional regime when the liquid was heated up to 100 and 120 °C (in Figure 5-5).
**Figure 5-5:** Jet Reynolds number as function of jet velocity ($d = 3.0$ mm).

### 5.5 Oil Viscosity Selection

A selection of oil viscosity values as function of temperature is given in Table 5-7 below, which was obtained from the viscosity/temperature chart in Appendix D.

<table>
<thead>
<tr>
<th>Jet Temperature, $T_j$ ($^\circ$C)</th>
<th>Prandtl Number, $\text{Pr}$</th>
<th>Kinematic Viscosity, $\nu$ (m$^2$.s$^{-1}$)</th>
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</thead>
<tbody>
<tr>
<td>55</td>
<td>705</td>
<td>57.0007E-06</td>
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<tr>
<td>120</td>
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<td>9.28392E-06</td>
</tr>
</tbody>
</table>

Table 5-7: Oil viscosity variation with temperature.
Figure 5-6 shows the significant decrease of kinematic viscosity and liquid (oil) Prandtl number with temperature rise. The high variation of the oil viscosity allowed the experimental test to show the effect of liquid viscosity on local heat transfer and jet structure.

![Figure 5-6: Oil Prandtl number and kinematic viscosity as function of temperature.](image)

### 5.6 Heat Transfer Measurements

The convective heat transfer produced at any specified location over the surface of a test-block may be expressed in the customary manner by the local Nusselt number, $\mathrm{Nu}$, and defined experimentally as

\[
\mathrm{Nu} = \frac{h d}{k_l} = \frac{q d}{k_l (T_w - T_{av})}
\]  

(5-3)

where $q$ is the conduction heat flux carried inside the metal, $k_l$ is the thermal conductivity of coolant, $T_w$ and $T_{av}$ represent the local wall temperature and
the adiabatic wall temperature, respectively, and \( h = q/(T_w - T_{mw}) \) is the calculated heat transfer coefficient.

Heat flux, \( q \), was calculated using the one-dimensional approximation of the heat conduction inside the heater block. The same method has previously been used by Leland and Pais (1999) for their test-heater in free jet impingement heat transfer of high viscous fluid. Using the Fourier’s law of conduction in the form;

\[
q = -k \frac{\Delta T}{\Delta y}
\]  

(5-4)

where \( k \) is the thermal conductivity of test-block, \( \Delta T \) the temperature difference and \( \Delta y \) the distance between points.

In the present experiments, the heat flux distributions from the heating source to the wetted surface of the impingement area were computed from the temperature difference across a known length inside the heater block, i.e.

\[
q(r) = \frac{k(T_{Hi} - T_w)}{\Delta y}
\]  

(5-5)

where \( T_{Hi} \) is the temperature near to the heating elements, \( T_w \) is the local wall temperature and \( \Delta y \) is the distance between points in vertical axis.

Therefore, it was important that parameters such as heat flux, wall temperature and adiabatic wall temperature can be measured accurately as these are then used to calculate the local Nusselt number. For example: (1) the temperature difference inside a test-block was used to define the thermal gradient with a distance normal to the surface across which the heat flux was been measured; and (2) by using the heat flux measurement in conjunction
with temperature difference in between the surface and the liquid, the heat transfer coefficient can be obtained.

### 5.7 Piston Cooling Configurations

In the case study of oil jet cooling of a piston underside surface, the following information and calculations were used to find out the actual piston movement inside the engine cylinder. Data of the oil jet configuration in the particular engine model were also used to define the actual cooling oil contact in the piston undercrown.

The case study of heavy duty DI engine considers a Perkins 1104C engine (4.4 litre - max. power 105 kW). The crankshaft rotational speeds in this engine range from 800 to 2500 RPM. Figure 5-7 shows an example of a reciprocating piston.

![Figure 5-7: Assembly of cylinder, piston, connecting rod and crankshaft.](image-url)
Chapter 5 - Experimental Methods and Measurements

The following equations were used to find the mean piston speed ($\bar{S}_p$) and instantaneous piston speed at crank angle ($S_p$):

$$\bar{S}_p = 2LN$$  \hspace{1cm} (5-6)

$$\frac{S_p}{\bar{S}_p} = \frac{\pi}{2} \sin \theta \left[ 1 + \frac{\cos \theta}{\left( R_{rc}^2 - \sin^2 \theta \right)^{1/2}} \right]$$  \hspace{1cm} (5-7)

where $R_{rc} = l/a$ and $L$ and $N$ are defined in Table 5-8.

| Bore, $B$ (mm) | 105  |
| Stroke, $L$ (mm) | 127  |
| Crank throw, $a$ (mm) | 63.5 |
| Con rod length, $l$ (mm) | 219  |
| Speed range, $N$ (rpm) | min 800 - max 2500 |
| Mean piston speed, $\bar{S}_p$ (m.s$^{-1}$) | min 3.387 - max 10.583 |

Table 5-8: Perkins 1104C engine specification - data obtained from Perkins Engine Co. Ltd.

From the data listed in Table 5-8, the calculated ratio of connecting rod length to crank radius is $R_{rc} = 3.44882$.

Figure 5-8 shows the instantaneous piston speed against crank angle for rated speed of 800 RPM (at idle) and 2500 RPM (at rated power) for the Perkins 1104C engine. It also shows how the piston velocity varies over up-stroke and down-stroke.
Figure 5-8: Piston speed at crank angle under low/high load condition.

As shown in Figure 5-8, for each stroke, the piston velocity is zero at the beginning of the stroke, reaches a maximum near the middle of the stroke, and decreases to zero at the end of the stroke.

Table 5-9 shows the oil jet characteristics used in production Perkins 1104C engines for piston cooling.

<table>
<thead>
<tr>
<th>Orifice diameter (mm)</th>
<th>2.0 nominal</th>
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<tbody>
<tr>
<td>Oil flow rate (litre.min⁻¹)</td>
<td>min 2.2 - max 2.5</td>
</tr>
<tr>
<td>Mean jet velocity, ( u ) (m.s⁻¹)</td>
<td>min 11.67 - max 13.26</td>
</tr>
</tbody>
</table>

Table 5-9: Perkins 1104C engine oil jet - data obtained from Perkins Engine Co. Ltd.

The pump capacity (i.e. operating pressure and volumetric flow rate) determined the amount of oil flow passing through the jet. From the oil flow rate that exited from the jet orifice, the mean jet velocity could be calculated. However, during operation the oil flow rate may vary for a given pressure
depending on the oil temperature due to the viscosity changes. A typical operating temperature for Perkins 1104C engine oil jet is 120 °C. Oil viscosity would slightly vary with viscosity grades of oil, but Shell Rimula X (SAE 15W-40) was recommended for Perkins 1104C engine.

Finally, the oil flow rate for Perkins jet in 1104C engine was 2.2 (at low load) to 2.5 (at full load) litre.min⁻¹. Thus the calculated jet speed from the orifice of 2 mm nominal internal diameter is 11.67 m.s⁻¹ (at low load) and 13.26 m.s⁻¹ (at high load).

As shown in Figure 5-9, if a maximum mean jet velocity of 13.26 m.s⁻¹ is used for cooling the piston undercrown at full load condition (2500 RPM), about 21% of the contact time is lost (or no oil contact) in the cycle. This effect is shown diagrammatically in Figure 5-10, where the piston underside surface is receiving no oil contact at crank angle of 41 to 114 deg during the piston up-stroke, because the maximum piston speed at full load condition is faster than the oil jet speed. During piston down-stroke, there is a large mass of cooling oil retained in the piston undercrown space, allowing maximum heat extraction. This emphasises the importance of continuous jet contact during this period.
Figure 5-9: Jet impinging velocity during the piston cycle.

Figure 5-10: Oil contact at piston undercrown during up-down stroke, at full load condition (2500 RPM).
Chapter 5 - Experimental Methods and Measurements

For the Perkins engine, the nominal jet nozzle internal diameter was 2 mm. The maximum mean jet velocity that can be obtained in the present experimental rig is 25 m.s\(^{-1}\) when oil jet temperature is at 120 °C, contains typical engine oil jet velocities in the middle of the available range.

5.8 Concluding Remarks

This chapter has described the configurations of experimental apparatus to control the cooling oil jet parameters. The potential of poor or zero piston cooling was also found by plotting the piston speed curve at crank angle with the specific oil jet speed impinging at the piston undercrown.

The next chapter discusses all the experimental results obtained using the above experimental methods.
CHAPTER 6

EXPERIMENTAL RESULTS AND DISCUSSION

This chapter presents and discusses the experimental results obtained using the cooling oil jet apparatus methodology described in Chapters 4 and 5. The jet configurations outlined in Chapter 5, i.e. different oil jet properties, orifice sizes, nozzle-to-surface separations and impinging angles are used to examine the effectiveness of oil jet impingement in piston undercrown cooling applications. The photographic apparatus as described in Chapter 4 was used throughout this study. The equipment comprised an optical oil jet test chamber in which the digital camera was mounted on an adjustable tripod to capture and video images for jet impingement analysis.

The heat transfer results in this chapter are from experiments conducted with heated surfaces of the following orientations and types:

1. Horizontal flat surface
2. Inclined flat surface
3. Concave surface
4. Diesel engine piston.
6.1 Observation of Jet Impingement

The images of oil jet impingement on a downward-facing test-surface were recorded using a digital CCD camera in a digital format, as described in Section 4.2.2.

6.1.1 Observation of Jet Contraction and Interference

One interesting result in the present investigation is the appearance of the 'bell-sheet' flow pattern formed when the jet impinged at viscous conditions and the nozzle-to-surface separation was small, as shown in Figure 6-1.

![Bell-Sheet](image)

Figure 6-1: Bell-sheet flow pattern at impingement on a downward-facing flat surface.

The oil jet impinges onto the centre of the downward-facing plate and flows radially outward as a film. The film velocity decelerates on the surface due to friction between the oil and the plate, as the viscous boundary layer reaches the free surface. When the radial velocity is zero, the liquid film leaves the surface and forms a thin liquid sheet that falls under gravity. The thin liquid sheet starts shrinking in diameter as it falls as a consequence of surface tension effects. As the thin sheet descends, it releases heat to the surrounding (Appendix F), and hence, the surface tension is greater at the bottom. Compared to water, the oil surface tension (i.e. $\sigma = 0.035 \text{ N.m}^{-1}$ at 25 °C) is
smaller than water (i.e. \( \sigma = 0.072 \text{ N.m}^{-1} \) at 25 °C) and it is more strongly affected by temperature, as indicated in Appendix G, thus unlike an oil sheet, a water sheet will merge almost immediately and draw together by surface tension forces as it falls under gravity (Carvalho et al., 2002). The round shape of the bell-sheet is due to surface tension forces, which pulling from all directions and a round shape has the smallest surface area per unit volume as compared to other shapes (Appendix G). Therefore, the shape of the bell-sheet flow pattern is dependent on the gravitational and surface tension forces.

The size of bell-sheet was found to be dependent on the conditions of the impinging jet. When a jet impinges onto a flat plate at high speed or high temperature (i.e. less viscous oil), the impingement area is larger since the downstream liquid film travels further away from the stagnation point and leaves the surface as a large thin sheet that eventually shrinks to form a big bell-sheet, as shown in Figure 6-2. Therefore, the size of bell-sheet depends on the liquid film inertial force.

![Figure 6-2: Size of bell-sheet at different oil jet velocities and temperatures using \( d = 3.0 \text{ mm} \).](image)
Figure 6-3 shows the bell-sheet flow patterns when a laminar jet impinges onto the surface at inclined angles (90 deg - θ) of 15°, 30° and 45°. The basic principles of the thin liquid bell-sheet are the same as above, except a portion of the liquid on the lower side merges into a stream before it leaves the surface, as indicated by the arrows (in Figure 6-3). This is because the impingement shape on an inclined surface is elliptic, with a sharp edge and liquid surface tension that lead the liquid film to merge on the lower side of the surface. The circular edge of the surface also leads the downstream liquid flow to merge at the lowest part of the edge.

Figure 6-3: Bell-sheet flow patterns for oblique jet impingements on a flat surface.
When an oil jet impinges onto a concave surface (see Figure 6-4), the thin liquid sheet that forms a bell-sheet flow pattern disappears. The downstream liquid film on the curved surface merges into separate streams, which fall straight down, as shown in Figure 6-4(a). However, at certain jet configurations the downstream liquid film tries to form a bell-sheet flow pattern when it leaves the surface but the thin liquid sheet is very unstable and eventually merges into a stream as it descends, as shown in Figure 6-4(b).

![Figure 6-4](image-url)

**Figure 6-4**: Bell-sheet flow pattern disappears when the jet impinged onto a concave surface.

Figures 6-5 and 6-6 show that for a turbulent jet, the thin liquid sheet may not form a bell-sheet because the thin liquid wall becomes very unstable and then breaks into droplets further downstream. The downstream liquid velocity at the edge of the impingement plate has not reach zero (due to high momentum of the turbulent jet), flowing from the edge of the disc into the air, and hence the wall surface tension is too weak to cause the bell-sheet to shrink inward as it descends outward. Wakimoto and Azuma (1999) observed liquid atomisation of a radial liquid sheet, which was generated by downward water jet impingements on a flat surface. The liquid film spread radially outward on the disc as a sheet, and flows from the edge of the disc into the air. They observed a sudden laminar-turbulent transition in the
liquid sheet when the Reynolds number exceeded a critical value, resulting in both perforation and disintegration of the sheet into droplets. Therefore, no bell-sheet flow patterns appear in turbulent jet impingements. Figures 6-5 and 6-6 show that this is likely to have played a major part in the present tests, since the bell sheet flow pattern generally failed to form when the Reynolds number exceeded 2000.

Figure 6-5: Observations of jet impingement on a flat surface at different Pr (or \( T_j = 55, 70, 85, 100 \) and 120 °C), \( \text{Re}_d \) (or \( u = 10, 15, 20, 25, 30 \) and 35 m.s\(^{-1} \)), \( d = 1.5 \) mm and \( Z/d = 1 \).
Figure 6-6: Observations of jet impingement on a flat surface at different Pr (or $T_j = 55, 70, 85, 100$ and $120 \, ^\circ C$), Re$_J$ (or $u = 5, 7.5$ and $10 \, m.s^{-1}$), $d = 3.0 \, mm$ and $Z/d = 1$.

Another factor that may contribute to the explanation why the bell-sheet flow pattern is not always observed in the present investigation is related to large nozzle-to-surface separation, as shown in Figure 6-7, where a sequence occurs. Figure 6-7(a) shows the liquid jet impinges onto a downward-facing flat surface and the downstream liquid film leaves the surface as thin liquid sheet which forms a bell-sheet flow pattern. Figure 6-7(b) shows the thin liquid sheet shrinking in diameter at the bottom and cutting through the jet, a process that maybe called as 'jet interference'. During jet interference, the incoming jet impingement is briefly interrupted and no liquid flows on the surface, as shown in Figure 6-7(c). Figure 6-7(d) shows the liquid jet re-establishing itself and impinging again onto the surface and re-forming the conical liquid sheet that leaves the surface. After that, the impinged liquid travels in a radial direction and forms a liquid film that covers the surface, Figure 6-7(e). The whole sequence repeats again when the downstream liquid film leaves the surface and forms a thin liquid sheet that starts shrinking in diameter as it descends, Figure 6-7(f).
Figure 6-7: Repeating jet interference sequence (approximately 0.447 seconds per cycle) observed at large nozzle-to-surface separations. Converging-type automotive jet \((d = 2\text{mm})\) impinged at \(Pr = 705, u = 10\text{ m.s}^{-1}\) and \(Z = 50\text{ mm}\).

In this study, an attempt was also been made to measure the size of jet impingement diameter by impinging the oil jet normally onto a downward-facing flat surface, near isothermal condition. This was achieved by maintaining the test-surface at the same temperature as the jet. Tables 6-1 and 6-2 show the ratios of jet impingement diameter to jet diameter \((D_i/d)\) for various jet properties and standoff distances. In order to prevent jet interference at large nozzle standoff distances, a separate setup was made, as shown in Appendix H, using clear Perspex tubes that were machined to lengths which allowed the jet orifice to be located at \(Z = 100\) and \(200\text{ mm}\) without the top edge of the tube touching the test-surface. In this test, only \(D_i\) of less than 100 mm were recorded due to the size of the test-surface \((D = 100\text{ mm})\). In the present study, for the heat transfer cases, no jet impingement diameters are measured because of jet interference or whole test-surface was covered by jet impingement.
<table>
<thead>
<tr>
<th>Jet Diameter, $d$ (mm)</th>
<th>Jet Standoff Distance, $Z$ (mm)</th>
<th>Prandtl Number, Pr</th>
<th>Jet Reynolds Number, $Re_d$</th>
<th>Ratio of Jet Impingement Diameter to Jet Diameter, $D/d$</th>
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</thead>
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Table 6-1: Recorded ratios of jet impingement diameter to jet diameter ($d = 1.5$ mm).
### Chapter 6 - Experimental Results and Discussion

#### Jet Jet Standoff Prandtl Jet Reynolds Ratio of Jet Impingement Diameter, Number, Number, Diameter, Distance, Number, Number, Diameter, Diameter, $D/d$  
<table>
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<tr>
<th>Jet Diameter, $d$ (mm)</th>
<th>Jet Standoff Distance, $Z$ (mm)</th>
<th>Prandtl Number, Pr</th>
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<td>410</td>
<td>457</td>
<td>16.87</td>
<td></td>
<td></td>
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<tr>
<td>410</td>
<td>686</td>
<td>25.70</td>
<td></td>
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</tr>
<tr>
<td>410</td>
<td>915</td>
<td>32.53</td>
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<tr>
<td>276</td>
<td>706</td>
<td>19.57</td>
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</tr>
<tr>
<td>276</td>
<td>1059</td>
<td>32.39</td>
<td></td>
<td></td>
</tr>
<tr>
<td>194</td>
<td>1049</td>
<td>21.96</td>
<td></td>
<td></td>
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<tr>
<td>194</td>
<td>1616</td>
<td>32.41</td>
<td></td>
<td></td>
</tr>
<tr>
<td>705</td>
<td>263</td>
<td>14.24</td>
<td></td>
<td></td>
</tr>
<tr>
<td>705</td>
<td>395</td>
<td>20.50</td>
<td></td>
<td></td>
</tr>
<tr>
<td>705</td>
<td>526</td>
<td>31.71</td>
<td></td>
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<tr>
<td>410</td>
<td>457</td>
<td>17.63</td>
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<tr>
<td>410</td>
<td>686</td>
<td>25.78</td>
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<tr>
<td>276</td>
<td>706</td>
<td>21.58</td>
<td></td>
<td></td>
</tr>
<tr>
<td>194</td>
<td>1049</td>
<td>26.86</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 6-2: Recorded ratios of jet impingement diameter to jet diameter ($d = 3.0$ mm).

#### 6.1.2 Observation of Jet Splattering

As has been discussed in Section 3.6.2, the basic physical mechanism of splattering is where disturbances to the surface of the incoming jet are strongly amplified as the jet spreads into a liquid film along a wall normal to the axis of the jet, causing the droplets to break free from the liquid film. The
phenomenon depends on the nozzle-to-surface separation and the Weber number, which is a measure of the ratio of inertia to surface tension forces in the impinging jet.

Figure 6-8(b) shows the continuous splattering of liquid droplets from the downstream liquid film at a distance near to the stagnation zone, when the nozzle standoff distance was large. The splattering in this test was not due to jet interference but was due to disturbance of the incoming jet by aerodynamic forces acting on its surface when the jet discharged through an ambient gas over a large distance. However, when the nozzle standoff distance was small, the downstream liquid film flowed smoothly on the surface and left the surface to form a bell-sheet flow pattern, as shown in Figure 6-8(a).

![Figure 6-8: Downstream liquid film flow types at different nozzle-to-surface separations using the converging-type automotive jet.](image)

When the jet had a high Reynolds number, the disturbances to the surface of the incoming jet were even more pronounced, so jet instability could occur at small distances from the orifice outlet. Splattering of liquid droplets from the downstream liquid film had also been observed at small nozzle-to-surface separations for such jets. It is linked to disturbances in the free surface of a turbulent jet, which may be amplified as the jet forms a liquid film, causing the film to become unstable and droplets to be discharged (Incropera, 1999).
Figure 6-9 shows a flash photograph (0.5 ms flash duration) of oil jet impingement at modest Reynolds number, which makes the surface structure of the jet and flying oil droplets clearly visible. As the liquid jet impinged onto a surface, the downstream liquid film formed but disturbances developed on the sheet causing instability which fragmenting the sheet and leading to dispersed droplet formation of the order of sheet thickness. As seen in the picture, the size of splattered oil droplets ranges from a few microns to millimetres. However, this photographic technique cannot capture the images of oil jet impingement at higher Reynolds numbers, since the strong ejection of oil droplets from the surface liquid film blurred the optical rig window.

In the literature of liquid jet impingement (in Section 3.6), splattering and jet instability (turbulence) both produce additional mixing in the liquid sheet which will tend to enhance the heat transfer relative to a laminar sheet. However, the increasing amount of mass splattered deteriorated the performance of jet impingement heat transfer. This topic will be discussed in the next sections.
6.2 Heat Transfer Measurements

The most important objective of the present study was the measurement of heat transfer coefficient at downward-facing hot surfaces, and also how to interpret the observed phenomena in relation to the heat transfer from the upward liquid jet impingement. The temperature of jet ($T_j$), mean velocity of jet ($u$), and nozzle-to-surface separation ($Z$) can influence the jet behaviour as well as on the surface heat transfer.

The presently used surface configuration was selected because it most closely models the heat transfer situations of interest such as the cooling of aluminium alloy pistons, which is close to a thick cylindrical block. The heat transfer data (Nusselt number) for present study was calculated from the following definition

$$\text{Nu} = \frac{h d}{k_j} = \frac{q d}{k_j (T_w - T_{aw})}$$

(6-1)

where $q$ is the conduction heat flux carried inside the metal, $d$ is the jet diameter, $k_l$ is the thermal conductivity of liquid, $h$ is the calculated heat transfer coefficient, and $T_w$ and $T_{aw}$ represent the local wall temperature and the adiabatic wall temperature, respectively. Adiabatic wall temperature was obtained from the following equation:

$$T_{aw} = T_j + r^* \frac{u^2}{2 c_p}$$

(6-2)

where $T_j$ is the liquid jet temperature outside the orifice, $r^*$ is the recovery factor, $u$ is the jet velocity, and $c_p$ is the liquid specific heat of capacity.

As has been discussed in Section 3.4.2, using a high Prandtl number liquid as a working fluid in liquid jet impingement, the effect of viscous dissipation in
the heat transfer process could become significant. Therefore, the calculation of local or average impingement heat transfer for present study, the static jet temperature \((T_j)\) was replaced by adiabatic wall temperature \((T_{aw})\), in Equation (6-2). The parameter \(r^*\) is the recovery factor of jet flow:

\[
r^* = \text{Pr}^n
\]  

As shown in Table 6-3, the best match of the range of operating conditions for the present data set is provided by the data of Leland and Pais (1999). Therefore, their value of \(n = 0.47\) was chosen to evaluate the recovery factor \((r^*)\) for the calculation of adiabatic wall temperature \((T_{aw})\) in Equation (6-2).

<table>
<thead>
<tr>
<th>Data Source</th>
<th>Pr</th>
<th>Re_d</th>
<th>Wall-to-fluid ΔT</th>
<th>n</th>
</tr>
</thead>
<tbody>
<tr>
<td>Metzger et al. (1974)</td>
<td>85</td>
<td>2200</td>
<td>17.8 °C</td>
<td>0.6</td>
</tr>
<tr>
<td>Ma et al. (1990)</td>
<td>202</td>
<td>357</td>
<td>263</td>
<td>0.5</td>
</tr>
<tr>
<td>Leland and Pais (1999)</td>
<td>48</td>
<td>109</td>
<td>120 °C</td>
<td>0.47</td>
</tr>
<tr>
<td>Present data</td>
<td>131</td>
<td>263</td>
<td>95 °C</td>
<td>0.47</td>
</tr>
</tbody>
</table>

Table 6-3: Operating conditions in previous research works and present study.

The Prandtl number \((\text{Pr})\) was based on the first approximation of the film temperature \((T_{film})\) or \((T_j + T_w)/2\) rather than \(T_j\) alone. This was done because the Prandtl number will change under increased \(T_w\) and viscous dissipation should reduce as the film heats up across the surface. This approach also led to better correlations for the Nusselt number (Leland and Pais, 1999).

### 6.3 Experimental Results and Discussion

#### 6.3.1 Oil Jet Cooling on a Flat Surface

An experimental investigation was performed to determine the heat transfer rates for an upward impinging free-surface axisymmetric jet of diesel engine lubricating oil for a wide range of temperatures 55 to 120 °C (which gave rise
to a range of Prandtl numbers, Pr = 131 to 705). Heat transfer coefficients were obtained for jet Reynolds numbers (Re) from 263 to 5655, nozzle orifice diameters (d) of 1.5 and 3.0 mm, and a heated flat aluminium surface with diameter (D) of 100 mm. The effect of nozzle stand-off distance was also investigated, in conjunction with ratios of nozzle-to-surface separation to jet diameter (Z/d) ranging from 1 to 100.

**Stagnation Nusselt Number**

Figure 6-10 shows the results of upward liquid jet impingement, using d = 1.5 and 3.0 mm. The results illustrate the effect of varying jet speeds and temperatures.

![Figure 6-10: Stagnation Nusselt number against jet Reynolds number, at different jet Prandtl numbers for constant Z/d = 1 and θ = 90°.](attachment:image)

The stagnation Nusselt number increased as the jet Reynolds number increased. Unlike the results observed for downward jet impingement where the downstream liquid film covered all the upward-facing hot surface area, the present observations (as shown in Figures 6-5 and 6-6) for upward jet
impingement, showed the downstream liquid film for low temperature and velocity jets covered a small area of the downward-facing hot surface because the film radial velocity decelerated on the surface due to friction between the oil and the plate, and the film left the surface when its radial velocity was zero. When the jet impinged at high velocity and high temperature, the downstream liquid film covered a larger area of the hot surface due to the higher film momentum and an extended distance for viscous boundary layer to reach the film surface. Hence, the hot surface received less cooling effects when the impingement area is small and vice-versa. Therefore, in Figure 6-10, the Nu$_d$ values for jets impinged at high Pr and low velocity were low and then it increased rapidly for jets impinged at low Pr and high velocity. The effect of nozzle diameter was also shown in the figure, where higher Nusselt numbers were achieved for larger nozzle diameter at the same Re$_d$, this was as expected by Metzger et al. (1974) and Leland and Pais (1999) in Chapter 3.

The stagnation Nusselt number data for Figure 6-10 were re-plotted in Figure 6-11 with log-log plot to check the power ($m$) of Reynolds number. The functional dependence of Nu$_d$ on Re$_d$ changes near Re$_d$ = 2000, indicated by a change in the slope of the Nu$_d$ against Re$_d$ data there (in Figure 6-11). These two regimes correspond to the predominantly of laminar and turbulent flow, respectively. The experimental data were correlated separately for the ranges of Re$_d$ ≤ 2000 and Re$_d$ > 2000 using the equation

$$\text{Nu}_d = C \text{Re}_d^m$$ (6-4)

where coefficients $C$ and $m$ are determined empirically.
Figure 6-11: The dependence of stagnation Nusselt number on jet Reynolds number.

Table 6-4 indicates that the stagnation Nusselt number was more strongly dependent on Reynolds number for laminar jets ($Re_d \leq 2000$) than initially turbulent jets ($Re_d > 2000$). The subscripts lam and turb indicate the jet regime for which the associated correlation coefficient is listed in Equation (6-4).

<table>
<thead>
<tr>
<th>Jet configuration, $d$ (mm)</th>
<th>$C_{lam}/C_{turb}$</th>
<th>$m_{lam}/m_{turb}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>0.323/0.382</td>
<td>0.662/0.62</td>
</tr>
<tr>
<td>3.0</td>
<td>1.324/1.526</td>
<td>0.566/0.525</td>
</tr>
</tbody>
</table>

Table 6-4: Summary of laminar and turbulent correlation coefficients for Equation (6-4) in Figure 6-11.

Figures 6-12 and 6-13 show the $Nu_d$ for jet impingement with varying jet properties (i.e. $Pr$, $Re_d$) and $Z/d$ greater than 1. The results showed the variation in $Nu_d$ when the jet impinged at different nozzle stand-off distances. Compared with corresponding conditions at $Z/d = 1$, the jet impinged at $Z/d = 10$ showed the $Nu_d$ were found to have increased at $Re_d <$
3000 and then it decreased at $Re_d > 3000$. At $Z/d = 100$, the $Nu_d$ were found to continue to have increased at $Re_d < 1000$ and then it decreased at $Re_d > 1000$.

Figure 6-12: Stagnation Nusselt number against jet Reynolds number, at different jet Prandtl numbers for constant $Z/d = 10$ and $\theta = 90^\circ$.

Figure 6-13: Stagnation Nusselt number against jet Reynolds number, at different jet Prandtl numbers for constant $Z/d = 100$ and $\theta = 90^\circ$. 

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At large nozzle-to-surface separation, the combined effects of air friction and surface tension forces created surface irregularities on the jet. This led the formation of a turbulent jet which will tend to enhance the heat transfer relative to a laminar jet. However, when high Reynolds number jets impinged at large nozzle-to-surface separations, the jet started to breakup and created liquid splattered from the free surface in the vicinity of the impingement region, as discussed in Section 6.1.2. This is because jets of high $Re_d$ impinged at large $Z$ allow disturbances to develop on the jet surface causing the jet to disintegrate. Splattering has the effect of removing liquid mass from the radial flow, resulting in a reduction of the liquid film thickness, and thus, a reduction of local heat transfer coefficients, as discussed in Section 3.6.2.

**Local Heat Transfer at Downstream Locations**

Figures 6-14 and 6-15 show the local Nusselt number distributions as function of $r/d$. It can be seen that the peak Nusselt numbers were in the stagnation zone and it decreased in the downstream region with larger $r/d$. Hence, a symmetric bell-shape was seen of the local Nusselt number profiles due to a decrease of downstream local heat transfer coefficients. The overall local Nusselt numbers increased as the jet Reynolds number increased.
Figure 6-14: Local Nu as function of $r/d$, for $d = 1.5$ mm impinged onto a flat surface at different $u$ and $T_j$, $Z/d = 1$, $\theta = 90^\circ$. 
Figure 6-15: Local Nu as function of $r/d$, for $d = 3.0$ mm impinged onto a flat surface at different $u$ and $T_j$, $Z/d = 1$, $\theta = 90^\circ$.

The jet impingement observations in Figures 6-5 to 6-6 showed that the size of the impingement diameters do affect the local Nusselt number distributions in Figures 6-14 and 6-15. Although, the maximum heat transfer was mainly caused at stagnation point but the rest of the downstream heat
transfer distributions were due to radial liquid flows on the impingement plate.

Figure 6-16 shows the local Nusselt number distributions as function of radius for different jet temperatures, taken at three different nozzle stand-off distances, i.e. $Z/d = 1, 10$ and 100. At low temperature, the jet impingement with large $Z/d$ seemed to give a better cooling capability at the hot surface, see Figure 6-16(a). Although the shear stress of liquid-gas interface is small, the interfacial shear can induce interfacial waves or roughness, which in turn, can lead to jet instability. Hence, there is a jet profile change for large jet stand-off distances. Jet instability can produce additional mixing in the liquid film which will tend to enhance the heat transfer relative to a laminar film. Apart from the stagnation Nusselt number that was increased, the local Nusselt numbers in the downstream region were also increased at the same time, as shown in Figure 6-16(a), with the vigorous flow of liquid film which actively picking up more heat (or disrupting thermal boundary layer development) from the surface.
Figure 6-16: Local Nu as function of $r/d$, for $d = 1.5$ mm impinged onto a flat surface at different $T_j$ and $Z/d$, $\theta = 90^\circ$. 
However, as the jet temperature increased, the cooling capability for jets impinged at large \( Z/d \) decreased dramatically, especially near the stagnation zone, as shown in Figures 6-16(b) and 6-16(c). This seems logical since the jet surface disturbances are greater at high Reynolds number causing the ratio of splattered flow to total flow increases monotonically with jet speed, as discussed in Section 3.6.2, where the increase of jet Weber number by jet speed will increase the amount of splattering. Therefore, the increasing amount of mass splattered will deteriorate the convective heat transfer at the surface, and thus creates a rapid decay in local heat transfer.

The following conclusions were drawn:

1. The local Nusselt numbers increased as the jet Reynolds number increased. The stagnation Nusselt number was found to increase more rapidly for jet impinged at high Prandtl number (low temperature jet) because of the rapid increment of impingement area as the jet velocity increased.

2. For a fixed Reynolds number, the local Nusselt numbers increased with nozzle diameter due to larger impingement flow rate that translated to a larger inertia flow in the liquid film. However, for a fixed jet Reynolds number, increases in the nozzle diameter are associated with reductions in jet velocity at the nozzle, and thus it may not suitable in an application where high jet impinging velocity is required. A thermally heavily loaded piston which operates at higher velocity than the cooling jet will have no oil contact at the piston underside surface, thus can cause the piston to fail with thermal fatigue problems.

3. The stagnation Nusselt number was observed more strongly dependent on Reynolds number for laminar jets (\( Re_d \leq 2000 \)) than
initially turbulent jets (Re\textsubscript{d} > 2000), for both jet configurations (i.e. \( d = 1.5 \) and \( 3.0 \) mm).

4. When a jet impinging at large nozzle-to-surface separations, the aerodynamic effect created surface irregularities on the jet and led it to a turbulent flow which will tend to enhance the local heat transfer relative to a smooth jet. However, for a jet impinged at high Reynolds numbers, the jet started to breakup and created liquid splatters from the free surface in the vicinity of the impingement region. This is because a jet of high Re\textsubscript{d} impinged at large \( Z \) allows the surface disturbances occur at the jet causing the jet becomes unstable. Splattering also has the effect of removing liquid mass from the radial flow, resulting in a reduction of the liquid film thickness. Thus, a reduction of local Nusselt numbers in the downstream region is also observed in the results.

5. Whenever an upward low velocity jet impinged at large nozzle standoff distances, there is a decrease of jet impinging velocity due to the effect of downward gravitational acceleration. The largest decrease of jet impinging velocity encountered here was 12.6%, using the calculation method in Equation (5-1), when the jet (\( d = 3 \) mm) impinged at \( u = 5 \) m.s\textsuperscript{-1} and \( Z = 300 \) mm. Hence, there is a slight decrease in stagnation Nusselt number when the jet impinged at large nozzle-to-surface separation even though there was no significant jet splatter observed.

6. It should be noted that for the experimental conditions (involving a highly viscous liquid) considered in the present investigation, a transition to turbulent flow regime in the downstream region could not be attained with the present size of surface measurement. This
was checked using the analytical formulae in Table 3-1, by Liu et al. (1991).

6.3.2 Oblique Oil Jet Cooling Impingement on a Flat Surface

Oblique oil jet impingement study for present research were conducted using $d = 1.5$ and $3.0$ mm and impinged at $Z/d = 2$. The ratio of nozzle-to-surface separation to jet diameter of 2 was used instead 1, is to leave an enough space between the nozzle and the surface which avoids the edge of the nozzle tip from touching the test-surface when it is tilted. Heat transfer coefficients were obtained for $Re_d = 263$ to 5655, $Pr = 131$ to 705, and a heated flat aluminium surface ($D = 100$ mm) tilted at different angles ($90^\circ - \theta$) of $15^\circ$, $30^\circ$ and $45^\circ$.

Figures 6-17 to 6-22 show the results of oblique liquid jet impingement conducted at various jet angles ($\theta = 75^\circ$, $60^\circ$ and $45^\circ$). The profiles of local Nusselt number distributions along the solid test-surface are also shown in the figures. The maximum Nusselt numbers were in the stagnation zone and decreased in the downstream region with radial distances from the point of jet impact. The local Nusselt numbers were generally found to increase as the jet Reynolds number increased.
Figure 6-17: Local Nu as function of $r/d$, for $d = 1.5$ mm impinged onto a flat surface at different $u$ and $T_j$, $Z/d = 2$, $\theta = 75^\circ$. 
Figure 6-18: Local Nu as function of \( r/d \), for \( d = 1.5 \) mm impinged onto a flat surface at different \( u \) and \( T_j \), \( Z/d = 2\), \( \theta = 60^\circ \).
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Figure 6-19: Local Nu as function of $r/d$, for $d = 1.5$ mm impinged onto a flat surface at different $u$ and $T_j$, $Z/d = 2$, $\theta = 45^\circ$. 
Figure 6-20: Local Nu as function of r/d, for d = 3.0 mm impinged onto a flat surface at different u and T_j, Z/d = 2, θ = 75°.
Figure 6-21: Local Nu as function of r/d, for d = 3.0 mm impinged onto a flat surface at different u and \( T_j \), Z/d = 2, \( \theta = 60^\circ \).
Figure 6-22: Local Nu as function of r/d, for d = 3.0 mm impinged onto a flat surface at different u and Tj, Z/d = 2, θ = 45°.

For normal impingement (θ = 90°), the profiles in Figures 6-14 and 6-15 were all symmetric, in Section 6.3.1. For oblique impingement (Figures 6-17 to 6-22), although a general 'bell-shape' was also observed for the profiles, the distribution curves were not symmetric about the stagnation point. The surface heat transfer coefficient on the uphill side was higher than that on
the down hill side. This asymmetry was the most prominent feature of oblique jet impingement heat transfer, as been reported by the previous workers in liquid jet impingement on upward-facing surfaces (see Section 3.7).

The shape of the asymmetric distribution curves of local heat transfer on up hill and down hill sides were significantly affected by the jet inclination. By comparing the local Nusselt number profiles for various jet angles the asymmetry in the profiles in the present study was seen to be accentuated at smaller jet angles. The values of heat transfer coefficient on the up hill side increased while that on the down hill side decreased as the jet inclination increased. The decrease in jet angles caused an increase in slope for the profile curves on the down hill side and decrease in slope on the up hill side. The thickness and velocity of the downstream liquid film increased on the up hill side and decreased on the down hill side as a result of reduced liquid flow from the incoming jet caused by the jet inclination. This is because on the down hill side the flow has to turn through a larger angle, and so it experiences more losses on this side, and the flow velocity reduction on the down hill side causes the reduction of local heat transfer coefficients. Therefore, significant imbalances can happen by increasing jet inclination in the heat transfer capabilities on the two sides.

Figure 6-23 shows the local Nusselt number distribution at different impingement angles for two jet velocities at an oil temperature of 100 °C. The maximum Nusselt number is found to decrease as the jet angles decreased. This was believed to be caused by the displacement of maximum heat transfer points from the geometric stagnation point, observed by previous workers (see Section 3.7). As reported in the literature, the peak heat transfer was shifted upstream.
Figure 6-23: Local Nu as function of r/d, for d = 1.5 and 3.0 mm impinged onto a flat surface at different θ and Z/d = 2

Figure 6-24 shows the average Nusselt numbers at the flat test-surface, which computed from the six r/d locations as shown in Figures 6-14 to 6-15 and Figures 6-17 to 6-22 for θ = 90° and θ = 45°, 60°, 75°, respectively. It must be noted that these average values could only be computed on x-axis centreline at the test-surface. The comparisons in Figure 6-24 show no significant changes of the average Nusselt numbers at the flat surface when the jet impinged at different inclination angles, except for a slight decrease for the oblique jet impingement at θ = 45°. This can be due to a decrease of Nusselt number in the stagnation zone, thus resulted in a decrease of average Nusselt numbers.
The following conclusions were drawn:

- The local Nusselt numbers in the stagnation zone and downstream region were found to increase as the jet Reynolds number increased.

- The local Nusselt numbers on the down hill side of the stagnation point dropped off more rapidly than those on the up hill side, and this was accentuated at larger inclinations. Thus, there are significant imbalances of cooling capability on either side of the stagnation point.

Figure 6-24: Average Nusselt numbers for jet impinged onto a flat surface at different inclination angles and Reynolds numbers.
• Comparing to the normal jet impingement results in Section 6.3.1, the local Nusselt number profiles for the oblique jet impingement showed that the local Nusselt numbers in the downstream region increased on the uphill side and decreased on the down hill side as the jet inclination angle increased. However, the average local Nusselt numbers showed no significant changes when the jet impinged at different inclination angles in the present study.

• The decrease of maximum heat transfer coefficients at the centre point was observed in the experiment with jet inclination angles ($\theta$) from 90° to 45°. The values of maximum Nusselt number decreased as jet inclination angle increased, but only moderately. The greatest decrease encountered here were in the 2% to 18% range.

6.3.3 Oil Jet Cooling on a Concave Surface

Oil jet impingement on a concave surface was studied using nozzles with diameter $d = 1.5$ and $3.0$ mm. Heat transfer coefficients were obtained for $Re_d = 263$ to $5655$, $Pr = 131$ to $705$ and $Z = 3$ to $200$ mm. In the present study, the area of concave test-surface is same as flat test-surface, as shown in Appendix B.

Figures 6-25 to 6-30 show the results of jet impingement conducted at the centre of the concave surface. Figures 6-25 and 6-26 show the results of jet impingement at small ratio of nozzle-to-surface separation to jet diameter of $Z/d = 2$. The effect of large nozzle stand-off distance was also investigated, Figures 6-27 and 6-28 shows the results for $Z = 100$ mm, and Figures 6-29 and 6-30 shows the results for $Z = 200$ mm. As before, the maximum Nusselt numbers were highest in the stagnation zone and decreased in the downstream region with radial distances ($r$) from the point of jet impact. The overall local Nusselt numbers increased as the jet Reynolds number increased. Compared to the results for jet impingement on a flat surface the
local Nusselt number profiles for concave surface declined less rapidly with radius. This is because every time when the jet impinged onto the concave surface, the downstream liquid film covered and cooled the whole area of the concave surface before it left at the edge. The curved surface structure allowed the downstream liquid film flow all over the surface area.

Figure 6-25: Local Nu as function of $r/d$, for $d = 1.5$ mm impinged onto a concave surface at different $u$ and $T_j$, $x = 0$ mm, $Z = 3$ mm.
Figure 6-26: Local Nu as function of $r/d$, for $d = 3.0$ mm impinged onto a concave surface at different $u$ and $T_j$, $x = 0$ mm, $Z = 6$ mm.
Figure 6-27: Local Nu as function of r/d, for d = 1.5 mm impinged onto a concave surface at different u and Tj, x = 0 mm, Z = 100 mm.
Figure 6-28: Local Nu as function of $r/d$, for $d = 3.0$ mm impinged onto a concave surface at different $u$ and $T_j$, $x = 0$ mm, $Z = 100$ mm.
Figure 6-29: Local Nu as function of $r/d$, for $d = 1.5$ mm impinged onto a concave surface at different $u$ and $T_j$, $x = 0$ mm, $Z = 200$ mm.
Figure 6-30: Local Nu as function of $r/d$, for $d = 3.0 \text{ mm}$ impinged onto a concave surface at different $u$ and $T_j$, $x = 0 \text{ mm}$, $Z = 200 \text{ mm}$.

In these experiments, splattering could not be observed when the jet impinged at small nozzle-to-surface separation ($Z/d = 2$) and only a very little jet splatters when impinged at large jet stand-off distances ($Z = 100$ and $200 \text{ mm}$) at high jet Reynolds numbers. This is because the splattered droplets near the stagnation zone re-impinge onto a curved surface in the
downstream region, as shown in Figure 6-31, and thus high heat transfer coefficients are found at locations near to the stagnation zone. Hence, the local Nusselt number profiles in Figures 6-27 to 6-30 have a symmetric convex-shape due to the increased downstream heat transfer coefficients, specifically for jet impinged at high Reynolds numbers.

![Diagram](image)

Figure 6-31: Splattered droplets re-impinge onto the curved surface.

Furthermore, no reduction of local Nusselt numbers in the stagnation zone and downstream region was observed when the jet impinged at large nozzle-to-surface separation, since the incoming jet is turbulent and the cooling liquid film is well retained on the concave surface until it leave at the edge, as shown in Figure 6-32.

![Images](image)

(a) $d = 1.5 \text{ mm, } Re_d = 5655, Z = 100 \text{ mm}$  
(b) $d = 3.0 \text{ mm, } Re_d = 3231, Z = 100 \text{ mm}$

Figure 6-32: Jet impingements on concave surface at high Reynolds numbers and large nozzle-to-surface separation.
Figures 6-34 to 6-39 show the results of jet impingement conducted at off-centre (in x-axis direction) locations of the concave surface. These results showed no significant changes of stagnation Nusselt number for jet impinged at off-centre locations (x) from -12.5 to -37.5 mm, and also when compared to the results for jet impinged at x = 0 mm. The downstream profiles near to the stagnation zone are similar to the results from oblique jet impingement (in Section 6.3.2) because the thickness of the downstream liquid film increased on the right side (up hill side) and decreased on the left side (down hill side) of the stagnation point. Thus, the heat transfer coefficient on the down hill side decreased more rapidly than on the up hill side. Hence, significant imbalances in the heat transfer capabilities occurred on the two sides (in Figures 6-35 and 6-38) when the jet impinged onto a steep slope (i.e. \( \theta = 60^\circ \)), as shown in Figure 6-33, and this was more accentuated at higher jet velocities. For jet impingement near the edge of the concave surface (Figures 6-36 and 6-39), the downstream profiles were decreased less rapidly because the jet impinged onto a steep slope surface at \( \theta = 41^\circ \), where it received almost all the liquid flow from the incoming jet.

![Diagram](image-url)

Figure 6-33: Jet impingement angles at different x-axis locations of concave surface.
Figure 6-34: Local Nu as function of $r/d$, for $d = 1.5$ mm impinged onto a concave surface at different $u$ and $T_j$, $x = -12.5$ mm, $Z = 3$ mm, $\theta = 76^\circ$. 
Figure 6-35: Local Nu as function of \( r/d \), for \( d = 1.5 \) mm impinged onto a concave surface at different \( u \) and \( T_j \), \( x = -25 \text{ mm} \), \( Z = 3 \text{ mm} \), \( \theta = 60^\circ \).
Figure 6-36: Local Nu as function of $r/d$, for $d = 1.5$ mm impinged onto a concave surface at different $u$ and $T_j$, $x = -37.5$ mm, $Z = 3$ mm, $\theta = 41^\circ$. 
Figure 6-37: Local Nu as function of $r/d$, for $d = 3.0$ mm impinged onto a concave surface at different $u$ and $T_j$, $x = -12.5$ mm, $Z = 6$ mm, $\theta = 76^\circ$. 
Figure 6-38: Local Nu as function of $r/d$, for $d = 3.0$ mm impinged onto a concave surface at different $u$ and $T_j$, $x = -25$ mm, $Z = 6$ mm, $\theta = 60^\circ$. 
Figure 6-39: Local Nu as function of \( r/d \), for \( d = 3.0 \) mm impinged onto a concave surface at different \( u \) and \( T_j, x = -37.5 \) mm, \( Z = 6 \) mm, \( \theta = 41^\circ \).

Figure 6-40 shows the average Nusselt numbers at the concave test-surface, which computed from the seven \( r/d \) locations as shown in Figures 6-25 to 6-26 and Figures 6-34 to 6-39 for \( x = 0 \) mm and \( x = -12.5, -25, -37.5 \) mm, respectively. It must be noted that again these average values can only be computed on \( x \)-axis centreline at the test-surface. The comparisons in Figure
show no significant changes of the average Nusselt numbers at the concave surface as the jet off-centre location increased, except for a slight decrease for jet impinging at $x = -37.5$ mm and $\theta = 41^\circ$. This can be due to a decrease of Nusselt number in the stagnation zone and thus resulted in a decrease of average Nusselt numbers, as observed in Section 6.3.2 for oblique jet impinged at $\theta = 45^\circ$. Furthermore, comparison of these results with those in Figure 6-24 shows that the average Nusselt number experienced on a concave surface is generally 20-30% higher than the corresponding value for a flat surface. This enhancement of heat transfer is attributable to re-impingement of splattered droplets.

![Diagram](Image)

Figure 6-40: Average local Nu for jet impinged onto a concave surface at different x-axis locations and Reynolds numbers.
The following conclusions were drawn:

1. The local Nusselt numbers in the stagnation zone and downstream region were found to increase as the jet Reynolds number increased.

2. The effect of curvature has been studied by comparing with flat surface results and is shown to be significant. The local heat transfer coefficients were enhanced even though the local Nusselt numbers in the stagnation zone for concave surface were almost the same as for flat surface. This is because every time when the jet impinged onto the concave surface, the downstream liquid film covered and cooled the whole area of the concave surface before it leaves at the edge.

3. The effect of curvature on downstream profiles was more prominent as the Reynolds number and nozzle-to-surface separation increased. In the test, no significant jet splatter was observed during the jet impingement because the splattered droplets near the stagnation zone re-impinge onto a curved surface in the downstream region. This causes high heat transfer coefficients at locations near the stagnation zone. For tests with central jet impingement, a symmetric convex-shape was observed for the local Nusselt number profiles due to an increase of downstream local heat transfer coefficients.

4. When the jet impinged off-centre onto a steeply sloping surface, the local Nusselt numbers on the down hill side of the stagnation point dropped off more rapidly than those on the up hill side, and this was accentuated at increased jet off-centre location. Thus, there are significant imbalances of cooling capability on either side of the stagnation point. However, the average local Nusselt numbers showed no significant changes when the jet impinged at different jet off-centre locations in the present study.
5. The decrease of maximum heat transfer coefficients in the stagnation zone was observed in the experiments with jet off-centre locations \((x)\) from 0 to -37.5 mm. The values of maximum Nusselt number decreased as jet inclination angle increased towards the edge of the concave surface, but only moderately. The greatest decrease encountered here were in the 0.5% to 16.2% range.

### 6.3.4 Oil Jet Cooling using Automotive Nozzles

Two types of automotive oil jet nozzle were used for the present research, a converging-type nozzle and a pipe-type nozzle. Both automotive jets have an orifice size of 2 mm nominal. Apart from a different design at the orifice section, the converging-type has one bend and the pipe-type has two bends, see Section 4.1.2.

The heat transfer from a flat surface was measured using different jet velocities and temperatures with varying ratios of nozzle-to-surface separation to jet diameter of \(Z/d = 25, 50, 75, 100\) and 150.

The results obtained from these tests are illustrated below and particular attention has been focused at jet temperatures of 100 and 120 °C, which are the typical range of jet operating temperatures in diesel engine.

Figure 6-41 shows the effects of Prandtl number, jet velocity and \(Z/d\) on stagnation Nusselt number for pipe-type automotive jet. The stagnation Nusselt number increased as the jet Prandtl number decreased (at high temperature jet), for each flow rate.
Figure 6-41: Stagnation Nusselt number against Pr, for pipe-type automotive jet impinging onto a flat surface at different \( u \) and \( Z/d \), \( \theta = 90^\circ \).

Unlike before, the stagnation Nusselt numbers for pipe-type automotive jet increased as \( Z/d \) increased, as shown in Figure 6-41. The increase in the present study was seen to be accentuated at high jet velocities and low Prandtl numbers, i.e. high jet Reynolds numbers. This is because at high jet Reynolds numbers, the impinging jet broke up not far from the nozzle exit, as shown in Figures 6-42 and 6-43, and it became like a spray type
impingement when discharged at large nozzle standoff distances. Hence, the spray cooling covered a large area of the test-surface with liquid film. Therefore, a liquid spray type can provide higher heat transfer coefficients than a liquid jet type with the same liquid flow rate (Cho and Wu, 1988, and Oliphant et al., 1998). It was proposed that the better cooling of sprays was due to the combined effect of evaporative cooling from the film along the impingement surface, and the unsteady thermal boundary layer expected in spray impingement (Oliphant et al., 1998).

Figure 6-42: Jet structures impinged at varying jet velocities and jet temperature of 100 °C, for pipe-type automotive jet.
Figure 6-43: Jet structures impinged at varying jet velocities and jet temperature of 120 °C, for pipe-type automotive jet.

Figures 6-42 and 6-43 show the jet structures for pipe-type automotive jet impinged at $u = 10, 15, 20$ and $25$ m.s$^{-1}$ for $T_j = 100$ and $120$ °C, respectively. The photographs were taken with the jets impinged onto a tilted test-surface, in order to lead the downstream liquid flow to other directions and reduce the splattering droplets that travel towards the optical view.

In a pipe-type nozzle, the boundary layer development along the walls of the nozzle is a primary cause of velocity nonuniformity, as discussed in Section
3.5, and for the present pipe-type automotive jet the ratio of nozzle length to
diameter of $L/D_n \approx 20$ is sufficient to achieve a fully developed turbulent
flow profile at the nozzle exit for high Reynolds numbers. With increasing jet
velocity the effect of aerodynamic forces begin to appear, and the jet
structures were distinguished which differ in that the length of breakup from
the orifice exit was shorter for jets of high Reynolds numbers, as discussed in
the literature in Section 3.6.1. Hence, jet breakup prior to impingement was
found to be more extensive for high Reynolds numbers, encouraging drop
formation.

Figures 6-44 to 6-48 show the local Nusselt number distributions as function
of $r/d$ for pipe-type automotive jet. As before, the maximum Nusselt numbers
were in the stagnation zone and decreased in the downstream region with
radial distances $(r)$ from the point of jet impact. The local Nusselt numbers
generally increased as the jet Reynolds number increased. The downstream
local Nusselt numbers for large nozzle standoff distances were found to
decrease less rapidly compared to those results obtained for small nozzle
standoff distances, and it was more accentuated for jets impinging at low $Pr$
and high velocities. This was due to spray cooling effects as discussed above,
when the jet impinged at large standoff distances and high Reynolds
numbers. Hence, the stagnation point and its nearby radial distances
received direct jet spray impingement.
Figure 6-44: Local Nu as function of $r/d$, for pipe-type automotive jet impinged onto a flat surface at different $u$ and $T_j$, $\theta = 90^\circ$, $Z/d = 25$. 
Figure 6-45: Local Nu as function of $r/d$, for pipe-type automotive jet impinged onto a flat surface at different $u$ and $T_j$, $\theta = 90^\circ$, $Z/d = 50$. 
Figure 6-46: Local Nu as function of $r/d$, for pipe-type automotive jet impinged onto a flat surface at different $u$ and $T_j$, $\theta = 90^\circ$, $Z/d = 75$. 

Pipe-Type Nozzle
Figure 6-47: Local Nu as function of $r/d$, for pipe-type automotive jet impinged onto a flat surface at different $u$ and $T_j$, $\theta = 90^\circ$, $Z/d = 100$. 
Figure 6-48: Local Nu as function of r/d, for pipe-type automotive jet impinged onto a flat surface at different u and T_j, θ = 90°, Z/d = 150.

Figure 6-49 shows the effects of Prandtl number, jet velocity and Z/d on stagnation Nusselt number for converging-type automotive jet. The Nu_d increased as the jet Prandtl number decreased (at high temperature jet), for each flow rate. At any given Prandtl number, the Nu_d increased as the jet velocity and Z/d increased. However, the Nu_d value was seen to fluctuate for
jet Prandtl numbers of 194 and 276 (at $T_j = 100$ and 85 °C), especially when the jet impinged at small nozzle-to-surface separations (i.e. $Z/d = 25, 50$ and 75). For jet impinged at $Pr = 131$ (i.e. $T_j = 120$ °C), the recorded results showed a rapid increase of heat transfer coefficient as the jet velocity and stand-off distance increased, similar to pipe-type automotive jet where the increase was seen to be accentuated at high jet Reynolds numbers (i.e. $Re_d > 4000$).

Figure 6-49: Stagnation Nusselt number against $Pr$, for converging-type automotive jet impinged onto a flat surface at different $u$ and $Z/d$, $\theta = 90^\circ$. 
This is because at high jet Reynolds numbers, the impinging jet broke up close to the nozzle exit, as shown in Figures 6-50 and 6-51, and it became like a spray type impingement when discharged at large nozzle standoff distances. The size of the spray for converging-type automotive jet (Figures 6-50 and 6-51) was found to be smaller than pipe-type automotive jet (Figures 6-42 and 6-43), because the converging section near the nozzle orifice (as seen in Section 4.1.2) tends to suppress the turbulent flow inside the nozzle. Hence, the breakup length for converging-type automotive jet would be slightly longer than pipe-type automotive jet.

Figure 6-50: Jet structures impinged at varying velocities and temperature of $T_j = 100 \, ^{\circ}C$, for converging-type automotive jet.
Figure 6-51: Jet structures impinged at varying velocities and temperature of $T_j = 120 \, ^\circ$C, for converging-type automotive jet.

Figures 6-50 and 6-51 show the jet structures for converging-type automotive jet impinged at $u = 10$, 15, 20 and 25 m.s$^{-1}$ for $T_j = 100$ and 120 $^\circ$C, respectively. With increasing jet velocity the effect of aerodynamic forces begins to appear, and the different jet structures show that the length of breakup from the orifice exit was shorter for jets of high Reynolds numbers, as discussed above for the pipe-type automotive jet. Hence, the jet breakup prior to impingement was found to be more extensive for high Reynolds numbers, and encouraging the drop formation as spray.
The fluctuations of the recorded stagnation Nusselt number values in the graphs (in Figure 6-49) occurred for jets that impinged at transitional Reynolds numbers near or in the range of 2000 to 4000, specifically at $T_j = 85$ and $100 \, ^\circ C$. The stagnation Nusselt number increased again at high Reynolds numbers ($Re_d > 4000$) as turbulent flow developing in the jet, as illustrated in Table 6-5. The instability of flow profile in the jet and strange effects in the results may be attributable to nozzle design and poor finish of converging-type automotive jet, i.e. converging section, orifice surface finishing, orifice length and bending angle, as shown in Section 4.1.2. An improper design or a poor finish of the nozzle converging section can cause unwanted cavitation inside the flow, which can seriously affect the discharge coefficient (Mian, 1997), and result in instability of flow profile in the jet. In addition, a tear-off of liquid jet surface near the orifice lip was observed in the experiments. This may be attributed to irregular or rough surfaces on the orifice lip, as shown in Appendix A, which disrupted the jet surface. However, for the case of large nozzle standoff distances, i.e. $Z/d = 100$ and 150, the problem was found to diminish as the jet would have undergone transition to turbulent flow and partial breakup before it reached the cooling surface. Figure 6-49 shows that jets impinging at low Prandtl numbers and high velocities, i.e. $Pr = 131$ and $u = 25 \, m.s^{-1}$, gave the highest heat transfer coefficient at the cooling surface as $Re_d > 4000$.

<table>
<thead>
<tr>
<th>Jet temperature (°C)</th>
<th>Jet velocity (m.s$^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10</td>
</tr>
<tr>
<td>55</td>
<td>351</td>
</tr>
<tr>
<td>70</td>
<td>610</td>
</tr>
<tr>
<td>85</td>
<td>942</td>
</tr>
<tr>
<td>100</td>
<td>1399</td>
</tr>
<tr>
<td>120</td>
<td>2154</td>
</tr>
</tbody>
</table>

Note: Bold numbers and shaded cells represent a small decrease and large decrease of stagnation Nusselt numbers, respectively.

Table 6-5: Jet Reynolds numbers defined at orifice for converging-type automotive jet.
Table 6-5 shows the calculated jet Reynolds numbers at orifice for the converging-type automotive jet. The bold values of Reynolds number inside the cells represent cases where a small decrease of stagnation Nusselt numbers was found and those inside the shaded cells represent a large decrease of stagnation Nusselt numbers, i.e. for the case of Z/d = 25 in Figure 6-49.

Figures 6-52 to 6-56 show the local Nusselt number distributions as function of r/d for converging-type automotive jet. As before, the maximum Nusselt numbers were found in the stagnation zone and decreased in the downstream region with radial distances (r) from the point of jet impact. The overall local Nusselt numbers increased as the jet Reynolds number increased.

In Figure 6-52, the results for jet impingement at Z/d = 25, Pr = 194, u = 15 and 20 m.s⁻¹ showed that not only the stagnation Nusselt number decreased (Figure 6-49) but its radial Nusselt numbers also decreased. The local Nusselt number distributions were found to decrease when the jet Reynolds numbers were in the range of 2000 to 4000, as discussed above. Jet impingement at u = 25 m.s⁻¹ (i.e. Re_d = 3497) showed the highest stagnation Nusselt number for Pr = 194, but its downstream values dropped off more rapidly than the laminar jet (Re_d = 1399) at u = 10 m.s⁻¹. However, when the jet impinged at Pr = 131, the local Nusselt number distributions for turbulent jets (Re_d > 4000) at u = 20 and 25 m.s⁻¹ were recorded higher than the transitional jets (2000 ≤ Re_d ≤ 4000) at u = 10 and 15 m.s⁻¹. A very similar trend was also observed in Figures 6-53, 6-54 and 6-55 for jets standoff distances at Z/d = 50, 75 and 100, respectively, but the fluctuations seemed to diminish as the nozzle standoff distance increased (i.e. Z/d = 150 in Figure 6-56), as discussed above.
Figure 6-52: Local Nu as function of r/d, for converging-type automotive jet impinged onto a flat surface at different u and T_j, \( \theta = 90^\circ \), Z/d = 25.
Figure 6-53: Local Nu as function of $r/d$, for converging-type automotive jet impinged onto a flat surface at different $u$ and $T_j$, $\theta = 90^\circ$, $Z/d = 50$. 
Figure 6-54: Local Nu as function of \( r/d \), for converging-type automotive jet impinged onto a flat surface at different \( u \) and \( T_j \), \( \theta = 90^\circ \), \( Z/d = 75 \).
Figure 6-55: Local Nu as function of \( r/d \), for converging-type automotive jet impinged onto a flat surface at different \( u \) and \( T_j \), \( \theta = 90^\circ \), \( Z/d = 100 \).
Figure 6-56: Local Nu as function of $r/d$, for converging-type automotive jet impinged onto a flat surface at different $u$ and $T_j$, $\theta = 90^\circ$, $Z/d = 150$.

An attempt was made to improve the quality of the converging-type automotive jet by reaming its orifice internal surface with a reamer (of 2 mm diameter), to clear the burrs and make the hole more rounded. Figure 6-57 shows the comparison of results from the reamed and the original
converging-type automotive jet for the case of $Z/d = 25$. The reamed converging-type automotive jet can be used to supply a viscous jet ($Pr = 705$) impinged $u = 25 \text{ m.s}^{-1}$, after the irregular surfaces on the orifice were cleaned. It produced a higher stagnation Nusselt number than the original converging-type automotive jet, but the recorded results showed fluctuating $Nu_d$ values, especially at $Pr = 194$ and 276. Therefore, the comparison in Figure 6-57 shows that even though the finishing of the orifice surface had been improved, the reamed converging-type automotive jet still showed the same strange results as observed earlier in Figure 6-49. Hence, such problems were attributed to jet nozzle design, as discussed above.

![Figure 6-57: Comparison of stagnation Nusselt number against Prandtl number, for reamed and original converging-type automotive jet impinged at $Z/d = 25$ and $\theta = 90^\circ$.](image)

In the oil jet cooling setup of the Perkins 1104C engine, discussed in Section 5.7, the jet was operated within the Reynolds numbers range from 1633 to 2857, for its designed jet velocities of 11.67 m.s$^{-1}$ (at low load) and 13.26 m.s$^{-1}$ (at high load) and typical operating temperatures within 100 to 120 °C. Hence, the jet structures in Perkins 1104C engine should be close to the jets in Figures 6-50 and 6-51 for $1399 < Re_d < 3231$. 

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The following conclusions were drawn:

1. The local Nusselt numbers in the stagnation zone and downstream region were found to increase as the jet Reynolds number increased.

2. It has been observed that the automotive jets exhibited breakup before impingement, and the breakup was more noticeable when the jet impinged at high Reynolds numbers. The distance from the orifice until the point where the jet started to breakup (or breakup length) became shorter as jet Reynolds number increased. The converging-type automotive jet showed a longer breakup length than the pipe-type automotive jet, because the converging section near the nozzle orifice tends to suppress the turbulent flow and boundary layer to develop inside the nozzle. The jet breakup effects on the measured heat transfer values were shown to be significant; increased Nusselt numbers are associated with spray cooling. At large jet stand-off distances the local heat transfer coefficients were found to be very high; this was due to a turbulent liquid spray covering a large area of the test-surface with liquid film.

3. The stagnation Nusselt numbers for the pipe-type automotive jet (with parabolic jet profile) were higher than the converging-type automotive jet (with uniform jet profile). This was due to the higher pressure gradient in the flow of parabolic profile than uniform profile, giving rise to heat transfer coefficients in the stagnation region, as discussed in the literature in Section 3.2.4.

4. In the experiments for the converging-type automotive jet, the recorded results showed that the local Nusselt numbers fluctuated and decreased at some points (especially at Pr = 194 and 276 for 2000 < Re<sub>d</sub> < 4000). Beside that, a tear-off of liquid jet surface near the orifice
lip was observed in the experiments. This may be attributed to irregular or rough surfaces on the orifice lip, which disrupted the jet surface. An attempt was made to improve the quality of the converging-type automotive jet by reaming its orifice surface but the recorded results only showed a slightly improvement in the local heat transfer but also showed the same $Nu_d$ values which were fluctuated and decreased at some points, as before. Therefore, the jet instability could be depending on the jet nozzle designs i.e. converging section, orifice surface finishing, orifice length and bending angle.
6.4 Cooling Oil Jet for Piston Undercrown

An experimental investigation was performed on a piston undercrown to determine the heat transfer rates for an upward free-surface impinging jet of diesel engine lubricating oil for Prandtl numbers (Pr) = 131 and 194. Heat transfer coefficients were obtained for jet Reynolds numbers (Rej) from 1049 to 5655, nozzle orifice diameters (d) of 1.5, 2.0 and 3.0 mm. The effect of nozzle stand-off distance was also investigated with Z = 100 and 200 mm, to provide clear understanding of oil jet cooling during piston at top dead centre (TDC) and bottom dead centre (BDC) inside engines.

![Diagram](image)

**Figure 6-58: Cooling oil jet position at piston undercrown.**

Figure 6-58 shows the position of the cooling oil jet impinging on the piston undercrown for the present study at radial distance of 37.5 mm, which is the same location where a thermocouple has been mounted inside the piston. See Appendix B for further details of the thermocouples arrangement inside the piston. At this radial location, the cooling oil jet impinges onto an inclined surface of approximately 45° at the piston undercrown. In the experiments, when no cooling jet impinged, the highest undercrown surface temperatures
recorded at the piston centre point were around 150 °C and reduced by 8 °C at piston radial distance of 37.5 mm. This was due to the large thickness difference of 11 mm between these two points, in conjunction with the lower thermal conductivity of piston material (aluminium-silicon alloy with \( k = 135 \text{ W.m}^{-1}.\text{K.}^{-1} \)), as well as the additional heat loss to ambient through the large piston skirt surfaces.

Figures 6-59 to 6-62 show the local Nusselt number distributions on the piston undercrown surface for jets with diameters of \( d = 1.5 \) and 3.0 mm impinging at different velocities and temperatures for \( Z = 100 \) and 200 mm. In the graphs, the stagnation point of cooling oil jet impingement throughout this study is indicated by the arrows, situated at piston radial distance of 37.5 mm (Figure 6-58). The position of the piston centre-line is indicated by the dot-dash line with label C.L.

![Figure 6-59: Local Nusselt number for \( d = 1.5 \) mm impinged at different \( u \) and \( T_j \) for \( Z = 100 \) mm. Arrows indicate \( \text{Nu}_{d}. \)](image)
Figure 6-60: Local Nusselt number for $d = 1.5 \text{ mm}$ impinged at different $u$ and $T_j$ for $Z = 200 \text{ mm}$. Arrows indicate Nu$_d$.

Figure 6-61: Local Nusselt number for $d = 3.0 \text{ mm}$ impinged at different $u$ and $T_j$ for $Z = 100 \text{ mm}$. Arrows indicate Nu$_d$. 
As before, the local Nusselt numbers generally increased as the jet Reynolds number increased. The maximum Nusselt numbers were in the stagnation zone and decreased in the downstream region with radial distances \( r \) from the point of jet impact. However, there was a sudden increase of local Nusselt number near the piston centre line. This was due the undercrown geometry that affects the liquid flow structures, which will be discussed in Figure 6-63, and hence provided extra cooling effects near to the piston centre. In the present study, the results showed an imbalance of cooling rates on either side of the piston centre point, and only a small effect of nozzle-to-surface separations (i.e. \( Z = 100 \) and 200 mm) on the local Nusselt numbers.

The results for piston undercrown cooling showed higher heat transfer coefficients than aluminium test-plates, because of large heat losses through the undercrown surfaces. The large heat losses were due to extra cooling effects from the downstream cooling oil that flows over the large piston undercrown-skirt surfaces which act as cooling fins for heat transfer enhancement. Hence, high heat transfer coefficients are found which reflect these large heat losses. This agrees with the findings of a numerical model.
reported in Appendix J. In the present study, the piston undercrown has a larger surface area than the aluminium test-plates.

Figure 6-63: Layout of undercrown geometry and liquid flow structures.

Figure 6-63 illustrates the layout of piston undercrown geometry and liquid flow structures during jet impingement. In the experiment, it has been observed that when a jet impinged onto the undercrown a portion of the incoming jet accumulated in the Region A (as shown in Figure 6-63) and flowing down via the skirt wall and gudgeon-pin boss. The liquid that flows down via the skirt wall provided extra cooling of the surface near to the jet impact point. Also, a portion of the downstream liquid film decelerated on
the crossed surface of the undercrown and fell onto the gudgeon-pin. The liquid that flows down via the gudgeon-pin boss and gudgeon pin provided extra cooling of the surface near to the undercrown centre line. Therefore, the liquid film flow structures on the undercrown surfaces correlate with the local Nusselt numbers profile in the present study.

Figure 6-64 shows images of the cooling oil jet impinging onto the piston undercrown at different temperatures and velocities for $d = 1.5$ and $3.0$ mm.

![Figure 6-64: In-house manufactured jets ($d = 1.5$ and $3.0$ mm) impinged onto a piston undercrown.](image-url)
Figures 6-65 to 6-68 show the local Nusselt number distributions on the piston undercrown surface for pipe-type and converging-type automotive jets at different jet velocities and temperatures for $Z = 100$ and 200 mm.

Figure 6-65: Local Nusselt number for pipe-type automotive jet impinged at different $u$ and $T_j$ for $Z = 100$ mm. Arrows indicate $\text{Nu}_d$.

Figure 6-66: Local Nusselt number for pipe-type automotive jet impinged at different $u$ and $T_j$ for $Z = 200$ mm. Arrows indicate $\text{Nu}_d$. 
Figure 6-67: Local Nusselt number for converging-type automotive jet impinged at different $u$ and $T_j$ for $Z = 100$ mm. Arrows indicate $\text{Nu}_{ul}$.

As before, the local Nusselt numbers increased as the jet Reynolds number increased. The maximum Nusselt numbers were in the stagnation zone and decreased in the downstream region with radial distances ($r$) from the point of jet impact. There was also a sudden increase of local Nusselt number near the piston centre line, as discussed in Figure 6-63. The results also showed an
imbalance of cooling rates on either side of the piston centre point, and only a small effect of nozzle-to-surface separations (i.e. Z = 100 and 200 mm) on the local Nusselt numbers.

In the present study, unlike the results obtained for flat test-surfaces in Section 6.3.4, the results for the pipe-type automotive jet impinging on the piston undercrown (in Figures 6-65 and 6-66) showed lower local Nusselt numbers than the converging-type automotive jet (in Figures 6-67 and 6-68) at high Reynolds numbers. This reduction is associated with extensive breakup of pipe-type automotive jets (see Figure 6-69) causing a portion of its incoming jet to be directed outside the piston undercrown, as shown in Figure 6-69(b). This in turn results in a loss of liquid mass flow at undercrown surface. The effect of liquid loss on local Nusselt numbers was larger when the jets impinged at the larger nozzle-to-surface separation (i.e. Z = 200 mm). In contrast, the converging-type automotive jet suffered less jet breakup, as shown in Figure 6-70, and consequently gave rise to higher impinging liquid mass flows and high local Nusselt numbers than pipe-type automotive jet.

Figure 6-69: Pipe-type automotive jet impinged onto a piston undercrown.
Figure 6-70: Converging-type automotive jet impinged onto a piston undercrown.

Figure 6-71: Local Nusselt number for reamed converging-type automotive jet impinged at different \( u \) and \( T_j \) for \( Z = 100 \text{ mm} \). Arrows indicate \( \text{Nu}_{ud} \).

Results from a reamed converging-type automotive jet (in Figure 6-71) did not show much difference from the results in Figure 6-67, because the effect of nozzle geometry was diminished by the crossed surface modification at piston undercrown. The extended or roughened surfaces disrupted the jet profile as it approached the modified surface (Incropera, 1999). Therefore, the
heat transfer coefficients for the piston undercrown in the present study were not significantly affected by the quality of orifice finishing.

The following conclusions were drawn:

1. The local Nusselt numbers in the stagnation zone and downstream region were found to increase as the jet Reynolds number increased.

2. The results of the tests on piston undercrown surface impingement showed higher local Nusselt numbers than aluminium test-plates because of extra cooling effects from the downstream cooling oil that flows down the large piston undercrown-skirt surfaces which act as cooling fins giving rise to heat transfer enhancement.

3. A sudden increase of local Nusselt numbers was observed near the piston centre line, which is due to the effects of undercrown geometry on the liquid flow structures. A portion of the downstream liquid film decelerates on the crossed surface at the undercrown and flows down via the gudgeon-pin boss and gudgeon pin. This provides extra cooling effects on the surface near to the undercrown centre line.

4. In the present study, the results showed an imbalance of cooling rates on either side of the piston centre line when the jet impinged on the piston at a radial distance of 37.5 mm, which resulted in less heat extracted from the piston centre and radial location far from the stagnation point. The results for \( d = 1.5 \) and \( 3.0 \) mm showed only a small effect of nozzle-to-surface separation (i.e. \( Z = 100 \) and \( 200 \) mm) on the local Nusselt numbers.

5. The results for the pipe-type automotive jet showed lower local Nusselt numbers than converging-type automotive jet when
impinging at high Reynolds numbers. This is because the large breakup of pipe-type automotive jets caused a portion of its incoming jet spray to be squirted outside the piston undercrown, causing loss of liquid mass flow impinging on the undercrown surface. The liquid loss effect on local Nusselt numbers increases with nozzle-to-surface separation (i.e. largest reduction of Nu is found at \( Z = 200 \) mm). Moreover, for the piston undercrown, the converging-type automotive jet showed reduced jet breakup and higher local Nusselt numbers than pipe-type automotive jet.

6. The results from a reamed converging-type automotive jet were not much difference from the results for the original converging-type automotive jet. This is probably attributable to the effect of the crossed surface modification on the piston undercrown (Incropera, 1999). The heat transfer coefficients for the piston undercrown in the present study were not significantly affected by the quality of orifice finishing.

6.4.1 Application to IC Engines

Experimental tests using present automotive jets have showed some poor cooling efficiency on the piston undercrown, particularly when the jet impinged at high Reynolds numbers (high jet speeds) and large nozzle-to-surface separation (i.e. during the piston upstroke). Figure 6-72 shows an example of these effects.
Chapter 6 - Experimental Results and Discussion

Jet breakup and hits out

Downstroke ($Z = 100$ mm)

Upstroke ($Z = 200$ mm)

(Liquid flow-out = Jet flow-in)  (Liquid flow-out < Jet flow-in)

Pipe-type automotive jet impinged at $T_j = 120$ °C, $u = 15$ m.s$^{-1}$, $Re_d = 3231$

Figure 6-72: Pipe-type automotive jet for piston cooling during downstroke and upstroke conditions.

It is clear from this study that the selection of appropriate oil jets for a specific piston cooling system is very important. As was observed in the present study, the pipe-type automotive jet was less effective for cooling of a piston with long stroke operation, due to jet breakup and liquid loss during the upstroke, resulting in poor piston cooling efficiency. The converging-type automotive jet would be more suitable for such piston operation but further jet designs are still needed to improve the jet quality and cooling efficiency, particularly when the jet impinges at high Reynolds numbers.

The imbalances of cooling rates on either side of the undercrown centre line can be minimised by targeting the cooling jet on the side where the piston bowl is located, as discussed in Section 2.3.

In addition, a flash evaporation or aerosol generation was also observed on the hot undercrown surface under conditions without oil jet impingement (or without oil contact), as seen in Figure 5-10 (in Chapter 5). A very thin oil film
that was left on the hot surface evaporated as aerosol. In engine cooling systems, this phenomenon can be minimised by making the oil jet speed design to match with piston maximum speed, so that a continuous flow of oil film at the surface ensuring the optimum piston undercrown cooling.

6.5 Concluding Remarks

This chapter has presented and discussed the liquid jet observations and heat transfer results obtained in the present research for downward-facing flat and concave surfaces, plus a case study on an actual IC engine piston. The effects of jet temperatures, velocities and standoff distances in piston cooling applications have also been documented in the chapter for different types of jet nozzle design.

The following summarise the main results from this chapter:

1. A bell-sheet flow pattern was observed in the present study when the upward jet impingement took place in viscous conditions and the nozzle-to-surface separation is small. But when the jet impinged at large nozzle-to-surface separation, the bell-sheet flow pattern disappeared due to jet interference: reduction of the bell sheet diameter resulting in cutting of the solid jet interrupting the flow to the heated surface, later followed by jet re-impingement onto the surface.

2. The local Nusselt numbers were found to be increased as the jet Reynolds number increased. The peak Nusselt numbers were in the stagnation zone and decreased in the downstream region with radial distances. The stagnation Nusselt number was found to increase more rapidly with Reynolds number for jet impinged at high Prandtl number (low temperature jet) because of the rapid increase of impingement area as the jet velocity increased.
3. When a jet impinged at large nozzle-to-surface separation, the aerodynamic effect created surface irregularities on the jet and led it to a turbulent flow which will tend to enhance the local heat transfer relative to a smooth jet. However, splattering due to high jet Reynolds numbers has the effect of removing liquid mass from the radial flow, hence a reduction of local Nusselt numbers in the downstream region.

4. For oblique jet impingement on a downward-facing flat surface the local Nusselt numbers in the downstream region increased on the up hill side and decreased on the down hill side as the jet inclination angle increased. Thus, there are significant imbalances of cooling capability on either side of the stagnation point. However, the average Nusselt numbers showed no significant changes when the jet impinged at different inclination angles. It must be noted that these average values could only be computed on the x-axis centreline of the test-surface.

5. The local heat transfer coefficients of a downward-facing concave test-surface were enhanced when a jet impinged onto the concave surface. The downstream liquid film covered and cooled the whole area of the concave surface before it left at the edge, and the splattered droplets near the stagnation zone re-impinged onto a curved surface in the downstream region. When the jet impinged at off-centre locations onto a sloping section of the concave surface, the local Nusselt numbers on the down hill side of the stagnation point dropped off more rapidly than those on the up hill side.

6. The automotive jets exhibited breakup before impingement, and the breakup was more noticeable when the jet impinged at high Reynolds numbers. The loss of impinging jet fluid due to jet breakup and off-centre impingement has a substantial effect on the measured heat
transfer values, which was shown to be significant in cases with spray cooling impingement. The local heat transfer coefficients were found to be very high at large jet stand-off distances because a turbulent liquid spray covered a large area of the test-surface with liquid film.

7. The converging-type automotive jet showed the local Nusselt numbers that fluctuated and decreased at some points, particularly in the transitional jet flow regime. This may be attributed to jet nozzle designs i.e. converging section, orifice surface finishing, orifice length and bending angle. However, the effect of nozzle geometry was found diminished with the crossed surface modification at piston undercrown (Incropera, 1999). Therefore, the heat transfer coefficients at piston undercrown in the present study were not significantly affected by the quality of orifice finishing.

8. The results for piston undercrown cooling showed higher heat transfer coefficients than aluminium test-plates, because of large heat losses through the undercrown surfaces. The large heat losses were due to extra cooling effects from the downstream cooling oil that flows over the large piston undercrown-skirt surfaces which act as cooling fins for heat transfer enhancement. There was also a sudden increase of local Nusselt numbers at the piston centre point and this was due to the undercrown geometry which decelerated a portion of the downstream liquid film on the crossed surface and caused it to flow downwards via the gudgeon-pin boss and gudgeon pin. This provided extra cooling effects on the surface near to the undercrown centre line.
CHAPTER 7

CORRELATED EMPIRICAL EQUATIONS

This chapter examines the validity of analytical equations and empirical correlations from previous studies on downward liquid jet impingement for the case of upward liquid jet impingement. Experimental data from the present study are also compared with those from previous workers.

New heat transfer correlations were subsequently developed based on the present cooling oil jet experimental results. These correlations are able to predict cooling oil jet heat transfer coefficient with good accuracy. They are suitable for normal and oblique liquid jet impingements on a downward-facing surface. Another correlation was developed to predict the size of upward liquid jet impingement area.

The newly generated heat transfer correlations in this chapter were compared with the piston cooling results from Chapter 6 and reasons for discrepancies between correlation and piston cooling data were explored.

Finally, this chapter is ended with concluding remarks.
7.1 Comparison of Present Lab Data to Published Data

This section compares the present lab results to other results obtained by previous workers.

Figure 7-1: Comparison of Nu_d for different nozzle diameters – graph
adapted from Leland & Pais (1999)

Figure 7-1 shows the comparison of stagnation Nusselt number (Nu_d) for present lab data with the results recorded by Leland and Pais (1999), for different nozzle and test-surface diameters. The Leland and Pais results were recorded from a test-surface of diameter 12.95 mm, jet diameters of d = 0.508 and 0.838 mm, and Z = 1 mm, whereas the present lab results were recorded using a larger test-surface of diameter 100 mm, jet diameters of d = 1.5 and 3.0 mm, and Z/d = 1. The results from Leland and Pais for each jet diameter collapse onto a single line irrespective of the jet temperature. The results from present lab data have a steeper rise when the jet impinged with low temperature and a similar slope to those of Leland and Pais when impinged with higher temperature. The data eventually collapsed onto a single line when the jet temperature continues to increase.
Figure 7-2 shows the comparison of stagnation Stanton number (St) for the present lab data with the results recorded by Metzger et al. (1974), for different nozzle and test-surface diameters, and $Z/d = 1$. The Metzger et al. results were recorded from ratios of surface-to-jet diameter ($D/d$) of 6.97 and 13.2, whereas the present lab results were recorded from larger $D/d = 33.33$ and 66.67. The Stanton number is an alternative dimensionless heat transfer coefficient defined as the ratio of the Nusselt number to the product of the Reynolds number and Prandtl number (St = Nu.Re⁻¹.Pr⁻¹).

In Figure 7-2, all the stagnation Stanton numbers from Metzger et al. (1974) and the present lab results are seen to follow the same trend: St increased when the Prandtl number and $D/d$ decreased and it decreased when Reynolds number increased. In comparison, the Stanton numbers for the present lab results were generally smaller than those in Metzger et al. (1974), but this was to be expected because the $D/d$ ratios for present study were considerably larger.
Therefore, the size of heat transfer surface has a significant effect on the measurement of local heat transfer coefficients, and most importantly, it also has implications for the relevance of empirical correlations proposed by Leland and Pais (1999) and Metzger et al. (1974) for the present experimental results. This problem is discussed in more detail in next section.

7.2 Comparison of Correlation Lines with Present Lab Data

This section compares the correlations from previous work by other authors with the present lab data. The comparison also comments on the importance of considering varying properties, ratios of cooled target diameter to jet diameter \((D/d)\), and thickness of test-surface when utilising the existing correlations from previous workers.

7.2.1 Stagnation Nusselt Number

![Graph](image)

Figure 7-3: Comparison of Leland and Pais correlation lines against data points with varying jet properties, for \(D/d = 33.33\) and 66.67.
In Figure 7-3, the present data are compared with the stagnation Nusselt number correlation generated by Leland and Pais (1999). The agreement is poor even though the other parameters (i.e. Re and Pr) were within the range of Leland and Pais (1999)'s data. Their correlations for laminar jet and turbulent jet gave negative stagnation Nusselt number values when \( D/d > 36.51 \) and \( D/d > 43.04 \), respectively. It appeared that, their correlations were only valid for ratios of surface-to-jet diameter near \( D/d = 7.61, 15.45 \) and 25.49 that were investigated by the authors, hence, not suitable for predicting the \( \text{Nu}_d \) for the present study with large \( D/d = 33.33 \) and 66.67.

![Graph showing comparison of data and correlations](image_url)

Figure 7-4: Comparison of Metzger et al. correlation lines against lab data with varying jet properties, for \( D/d = 33.33 \) and 66.67.

Figure 7-4 compares the present lab data with the stagnation Nusselt number correlation proposed by Metzger et al. (1974). At low Reynolds number, the correlation lines generated by Metzger et al. under predicted our data only for high Pr and large \( D/d \). As the Reynolds number increased for a given flow rate, the correlation lines rapidly diverged from the present data. Their correlation was derived from data for temperature differences between
surface temperature and adiabatic wall temperature \((T_w - T_{aw} = 3.3 \text{ K})\) to obtain essentially constant property behaviour. In the present study, the \(T_w - T_{aw}\) was at least 11.8 K which may have caused variation of properties in the flowing film. Apart from that, Metzger et al. only investigated ratios of surface-to-jet diameter within the range \(D/d = 1.75\) to 25.1. Thus, problems may be expected when their correlations are compared with the present data based on larger \(D/d = 33.33\) and 66.67.

In all cases, reducing the \(D/d\) values in the correlations led to an increase of stagnation Nusselt number for all jet properties and flow rates. This agrees with the investigation reported in Appendix I, where reducing the ratio of surface-to-jet diameter \((D/d)\) and increasing the ratio of cooling-to-surface diameter \((D/D)\) was found to lead to higher heat flux losses at the surface, and hence, an increase in local heat transfer coefficients.

### 7.2.2 Radial Nusselt Number

![Graph showing Radial Nusselt Number comparison](image)

Figure 7-5: Comparison of analytical predictions to present experimental data points.
Figure 7-5 shows the comparison of radial distributions of local Nusselt numbers calculated using the analytical predictions (in Tables 3-1, 3-2 and 3-3) with the present experimental data. In comparison, the stagnation Nusselt number from the experimental investigation was far below the analytical predictions, but at the large radial distances away from the stagnation point the measured local Nusselt number values were higher. The reason of this effect has been investigated by using a numerical method in Section 3.4.1, which was also discussed in Rahman et al. (2000). These results were due to the effect of different disc thickness (as shown in Appendix I), where the temperature distribution at the solid-fluid interface becomes more uniform due to thermal spreading by larger radial conduction within the disc of thicker section. This reduces the heat transfer coefficient at the centre of the thicker disc in the present study.

![Graph showing comparison of data and correlation lines]  

Figure 7-6: Comparison of Ma et al. (1997) correlation lines against lab data points.
Similar comparison was also found with Ma et al. (1997) correlation lines for the stagnation Nusselt number case, as shown in Figure 7-6, where all the heat transfer properties were taken to be those corresponding to the lab measurements. As can be seen, the present lab data were well below the correlation lines which plotted from Equation (3-26), in Section 3.4.2. Their equation was correlated using data from a small surface area (5 x 10 mm nominal) and very thin test-surface (10 µm thick constantan foil).

Concluding remarks for this section:

- The stagnation Nusselt number correlations from Leland and Pais (1999) and Metzger et al. (1974) were found to be not suitable for predicting upward impinging oil jet heat transfer coefficients for the large ratios of surface-to-jet diameter \((D/d)\) in the present study, because their empirical correlations were validated using data from tests using much smaller \(D/d\) values. By reducing \(D/d\) in their correlations all the stagnation Nusselt number increased.

- The analytical equations from Zhao and Ma (1989), Liu et al. (1991), and Webb and Ma (1995), and the empirical correlation from Ma et al. (1997) were found to be not suitable for predicting radial distributions of heat transfer coefficient for the present test-surface because these equations were validated using a heat surface made of thin sheet (less than 1 mm thickness). The larger thickness in the present test causes radial heat transfer, which redistributes the jet cooling effect.

- Although the predicted lines from previous correlations did not match the present experimental data, the trends for local Nusselt number \((\text{Nu})\) against Reynolds number \((\text{Re}_j)\), Prandtl number \((\text{Pr})\) and ratio of surface-to-jet diameter \((D/d)\) were generally similar.
7.3 Empirical Correlations

In this section new correlated empirical equations are developed for present experimental results from the downward-facing flat test-surface. Free surface jets of high Prandtl number liquid were impinged at different angles, from normal to inclined jet impingements.

All experimental results used for generating the heat transfer correlation were obtained from in-house manufactured jets (d = 1.5 and 3.0 mm) at small ratios of nozzle-to-surface separation to jet diameter (Z/d = 1 and 2) and different jet inclination angles (θ = 90°, 75°, 60° and 45°). As reported by Metzger et al. (1974), measured heat transfer rates at Z/d = 2 and 3 were in all cases, essentially identical with those measured at Z/d = 1 because the effect of surface shear stress and entrainment on the flow of the jet-gas interface was very small. In the present study, the experimental results from Z/d = 10 and 100 were not included in determining the heat transfer correlation because their Nusselt numbers did not rise consistently with jet Reynolds number, as discussed in Section 6.3.1. This inconsistency will significantly change the correlation accuracy. For determining the correlation for jet impingement size, the experimental results used in the present work were obtained with jet stand-off distances up to Z/d = 133 (for d = 1.5 mm and Z = 200 mm).

As discussed in Chapter 3, the variation of heat transfer along the test-surface also depends on geometrical features and the nature of the thermal boundary conditions. Thus, the local heat transfer may be described by a non-dimensional equation of form:

$$\text{Nu} = \Psi(\text{Re}_d, \text{Pr}, D/d, Z/d, r/d, \theta, \text{boundary condition})$$  \hspace{1cm} (7-1)
where $\Psi$ is an, as yet, unknown function to be determined, $Re_d$ is the Reynolds number of liquid flow through the nozzle orifice, $Pr$ is the coolant Prandtl number, $D$ is the diameter of heated surface or disc, $Z$ is the distance of nozzle to targeted surface, $r$ is the radial distance measured from the point of jet impact, $\theta$ is the inclination angle of jet impinging to cooling surface, and boundary condition (i.e. surface geometry, surface temperature).

The outcome of the heat transfer correlation from the present study was used to provide Nusselt numbers at the stagnation point and downstream region on a downward-facing surface. The following parameters were selected when determining the correlation:

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nusselt number (Nu)</td>
<td>Dimensionless ratio of jet convective heat transfer to diffusive heat transfer.</td>
</tr>
<tr>
<td>Jet Reynolds number ($Re_{d}$)</td>
<td>Dimensionless ratio of jet dynamic force to viscous drag force.</td>
</tr>
<tr>
<td>Prandtl number (Pr)</td>
<td>Dimensionless ratio of jet momentum diffusivity to thermal diffusivity.</td>
</tr>
<tr>
<td>Ratio of jet-to-surface diameter ($d/D$)</td>
<td>Dimensionless ratio of jet system geometry.</td>
</tr>
<tr>
<td>Ratio of radial distance from the point of jet impact to jet diameter ($r/d$)</td>
<td>Dimensionless ratio of radial distance from the point of jet impact.</td>
</tr>
<tr>
<td>Jet inclination angle ($\theta$)</td>
<td>A measure of difference relative to ‘normal’ jet impingement.</td>
</tr>
</tbody>
</table>

Table 7-1: Parameters for the present heat transfer correlation.

In the present study, for the correlated Nusselt number equation, all properties were based on the adiabatic wall temperature ($T_{\text{adw}}$), defined as in Equation (6-2). The use of the adiabatic wall temperature for the calculation of the properties was critical to obtain an excellent fit in the comparison.
between correlation lines and experimental data. This is shown in the following sub-sections.

**7.3.1 Correlation for Normal and Oblique Jet Impingements**

Based on results of the present study it was recommended that the following correlation in Equation (7-2) be used for predicting local Nusselt number for single circular jets of high Prandtl number liquid impinging normally ($\theta = 90^\circ$) and obliquely at the centre of a circular flat surface:

$$
Nu = \frac{C \text{Re}_d^m \text{Pr}^n (d/D)^a \theta^b}{1 + B(r/d)^p}
$$

(7-2)

where,

$$
B = B_0 + B_1 \sin \theta + B_2 \sin^2 \theta
$$

$$
P = P_0 + P_1 \theta + P_2 \theta^2
$$

The empirical constants and powers $C$, $m$, $n$, $a$ and $b$ were determined from the experimental data by a least-squares technique for laminar, transitional and turbulent jets respectively, and $B_0$, $B_1$, $B_2$, $P_0$, $P_1$ and $P_2$ were determined for stagnation point, up hill and down hill sides. An Excel Numerical Solver was used to iterate the best-fit values for the empirical constants and powers, and these values are presented in Tables 7-2 and 7-3. In the correlation, the radial distance from the point of jet impact ($r$) was expressed as a positive value, and the jet inclination angle ($\theta$) was expressed in radians.
Chapter 7 - Correlated Empirical Equations

<table>
<thead>
<tr>
<th>Type of flow</th>
<th>Empirical coefficients</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$C$</td>
</tr>
<tr>
<td>Laminar ($Re_d &lt; 2000$)</td>
<td>6.407</td>
</tr>
<tr>
<td>Transitional ($2000 \leq Re_d \leq 4000$)</td>
<td>6.001</td>
</tr>
<tr>
<td>Turbulent ($Re_d &gt; 4000$)</td>
<td>6.054</td>
</tr>
</tbody>
</table>

Table 7-2: Correlation coefficients for Equation (7-2) for different jet flow regimes.

<table>
<thead>
<tr>
<th>Locations</th>
<th>Coefficients</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$B_0$</td>
</tr>
<tr>
<td>Stagnation point and up hill side</td>
<td>0.332</td>
</tr>
<tr>
<td>Down hill side</td>
<td>-1.482</td>
</tr>
</tbody>
</table>

Table 7-3: Correlation coefficients for Equation (7-2) for stagnation point and up hill side, and down hill side.

In the correlation, the power $a$ for the ratio of jet-to-surface diameter ($d/D$) was constrained to a single value (see Table 7-2), because the ratio will not change for all jet conditions (i.e. laminar, transitional and turbulent).

This correlation was based on measurements with oil jets of Reynolds number and Prandtl number ranges ($267 \leq Re_d \leq 5793$, $131 \leq Pr \leq 705$), surface-to-jet diameter ratios ($D/d = 33.33$ and $66.67$), ratios of nozzle-to-surface separation to jet diameter ($Z/d = 1$ and $2$), and highly varying properties (a wall to jet temperature difference of 95 K) with jet impingement angles ($\theta = 90^\circ$, $75^\circ$, $60^\circ$ and $45^\circ$). The correlation was also tested to be valid for $0 \leq r/d \leq 1$, as shown in Appendix K.
Chapter 7 - Correlated Empirical Equations

Figure 7-7: Correlation of stagnation Stanton number for experimental results for $\theta = 90^\circ$.

Figure 7-8: Correlation of stagnation Stanton number for experimental results for $\theta = 75^\circ$. 
Figure 7-9: Correlation of stagnation Stanton number for experimental results for $\theta = 60^\circ$.

Figure 7-10: Correlation of stagnation Stanton number for experimental results for $\theta = 45^\circ$. 
Good agreement can be seen in Figures 7-7 to 7-10 between the correlations and the data in the form of Stanton number for stagnation points. The data were correlated with excellent accuracy, with mean absolute percentage error (MAPE) of 6% (i.e. average percentage of discrepancy between data and correlation).

![Graphs showing correlation between data and theoretical predictions for different angles and pipe diameters.]

Figure 7-11: Correlation of local Nu for experimental results for different $\theta$ and $d = 1.5$ mm.
Figure 7-12: Correlation of local Nu for experimental results for different $\theta$ and $d = 3.0$ mm.

Figures 7-11 and 7-12 show a comparison of the correlation of the radial distribution of local Nusselt numbers with experimental results for $d = 1.5$ and $3.0$ mm at different jet impingement angles. For all profiles, the maximum heat transfer coefficient occurred at the stagnation point and declined with radial distance. As the jet impingement angle increased, the heat transfer coefficient on the down hill side dropped off more rapidly while that on the up hill side decreased more slowly. The heat transfer rate on the up hill side was higher than that on the down hill side, and the difference between the two heat transfer rates tends to increase with
increasing jet inclination. Good agreement as found between the experimental data and the correlation curves, as shown in Figures 7-11 and 7-12. The curves for correlation lines in these graphs were plotted using smooth lines. The data were correlated with good accuracy, with mean absolute percentage error of 18%. Figure 7-13 shows the distribution of mean absolute percentage error (MAPE) at stagnation point ($r/d = 0$) and downstream region ($r/d > 0$) for Figures 7-11 and 7-12. The MAPE is small at the stagnation point and it increased at large $r/d$. The larger discrepancies between the data and the correlation at large $r/d$ are not critical because the local Nu is considerably smaller compared to Nu at stagnation point and small $r/d$.

![Diagram](image)

Figure 7-13: Distribution of mean absolute percentage error at stagnation point ($r/d = 0$) and downstream region ($r/d > 0$) for Figures 7-11 and 7-12.
7.3.2 Correlation for Jet Impingement Size

An empirical equation was developed to correlate the measured liquid jet impingement size in the present work (data given in Tables 6-1 and 6-2).

The results for impingement-to-jet diameter ratio \( \frac{D}{d} \) could be successfully correlated using Equation (7-3) for normal upward impingement of a jet of high Prandtl number liquid impinging on the surface of same temperature as the jet. The correlation equation for this study is given below:

\[
\frac{D_j}{d} = 8.18 \times 10^{-4} \text{Re}_d^{0.114} \text{Pr}^{0.62} \left( \frac{Z}{d} \right)^{0.0036}
\] (7-3)

The empirical constants in Equation (7-3) were determined from the experimental data by a least-squares technique for all flow rate conditions in the present study. The functions of jet Reynolds number \( \text{Re}_d \), Prandtl number \( \text{Pr} \) and ratio of nozzle-to-surface separation to jet diameter \( \frac{Z}{d} \) were selected because Reynolds number determines the inertia force in viscous liquid flow, Prandtl number determines the molecular momentum in liquid of different temperatures, and \( \frac{Z}{d} \) determines the effect of disturbed incoming jets due to gravitational acceleration and aerodynamic drag on the liquid film. The function of \( \text{Re}_d \) has the highest power in Equation (7-3) because the size of the jet impingement area was most influenced by inertia forces in the liquid film flow.

This correlation was based on measurements with oil jets over the ranges of \( 263 \leq \text{Re}_d \leq 3231 \), \( 131 \leq \text{Pr} \leq 705 \), surface-to-jet diameter ratios of \( \frac{D}{d} = 33.33 \) and 66.67, ratio of nozzle-to-surface separation to jet diameter of \( \frac{Z}{d} = 1, 33, 67, 133 \) (or equivalent to \( Z = 1.5, 3, 100 \) and 200 mm), and near-isothermal condition between the liquid jet and the test-surface with \( \pm 0.5 \) °C. Therefore, the effect of variable properties was minimised in the study and can be neglected from the correlation.
Chapter 7 - Correlated Empirical Equations

Figure 7-14: Correlation of $D/d$ for experimental results for $d = 1.5$ mm.

Figure 7-15: Correlation of $D/d$ for experimental results for $d = 3.0$ mm.

Good agreement is seen in Figures 7-14 and 7-15 between the correlations and the data. The data were correlated with good accuracy, with mean absolute percentage error of about 9.5%.
The largest $D/d$ that can be obtained in the present study was 66.67 and 33.33 for jet diameter of $d = 1.5$ and 3.0 mm, respectively, when impinging onto a test-surface of $D = 100$ mm. For ratios of nozzle-to-surface separation to jet diameter of $Z/d = 1$ and $Z/d > 1$, the largest Reynolds numbers used in the study were $Re_d = 3231$ and $Re_d = 1616$, respectively. This is because the test-surface was fully covered with liquid film when the jet impinged above these Reynolds numbers. At large jet standoff distances, the liquid jet received surface disturbances which cause the jet profile change and the vigorous liquid film flows more actively over the test-surface.

7.4 Comparison of Present Correlation to Concave Surface and Piston Cooling Results

This section shows the comparison of the correlations generated for a flat surface experimental result to experimental data obtained from concave surface and piston cooling results (see Sections 6.3.3 and 6.4, respectively). The generated correlation in Equation (7-2) was used to provide the heat transfer coefficient values for comparisons.

Figures 7-16 and 7-17 show the comparison of correlation of local Nusselt numbers to experimental results from concave test-surface at different $x$-axis locations (see Figure 6-33 in Chapter 6) for $d = 1.5$ and 3.0 mm, respectively.
Figure 7-16: Comparison of correlation of local Nu to experimental results from concave test-surface at different x-axis locations for $d = 1.5$ mm.
Figure 7-17: Comparison of correlation of local Nu to experimental results from concave test-surface at different x-axis locations for \( d = 3 \) mm.

Good agreement is seen in Figures 7-16 and 7-17 between the data and the predicted correlation curves. The stagnation zone data were correlated with good accuracy with mean absolute percentage error of 15.6%. As radial distance increases, the downstream data diverge from the correlation with mean absolute error of 44.9%. This is because the concave surface geometry has changed the downstream liquid film flow characteristics which enhanced the local heat transfer coefficients, as discussed in Section 6.3.3. Therefore, the correlations under-predicted the heat transfer coefficient values for experimental data for concave surface.
For all profiles, the maximum heat transfer coefficient occurred at the stagnation zone and declined with radial distance. As the jet impinged towards the edge of the concave surface (at large x-axis locations), the jet impingement angle increased. Hence, the heat transfer coefficient on the down hill side dropped off more rapidly while that on the up hill side decreased more slowly. The heat transfer rate on the up hill side was higher than that on the down hill side, and the difference between the two heat transfer rates tends to increase with increasing jet inclination.

Figures 7-18 to 7-21 show the comparison of correlation of local Nusselt numbers to experimental results from piston cooling for experimental nozzles with \(d = 1.5\) and \(3.0\) mm, pipe-type automotive jet \((d = 2\) mm\) and converging-type automotive jet \((d = 2\) mm\), respectively.

![Graphs showing heat transfer comparison](image-url)

**Figure 7-18**: Comparison of present correlations to piston cooling results for \(d = 1.5\) mm and \(Z = 100\) and 200 mm.
Figure 7-19: Comparison of present correlations to piston cooling results for $d = 3.0 \text{ mm}$ and $Z = 100$ and 200 mm.

Figure 7-20: Comparison of present correlations to piston cooling results for pipe-type automotive jet ($d = 2.0 \text{ mm}$) and $Z = 100$ and 200 mm.
Figure 7-21: Comparison of present correlations to piston cooling results for converging-type automotive jet \((d = 2.0\, \text{mm})\) and \(Z = 100\) and 200 mm.

The comparisons in Figures 7-18 to 7-21 show that the correlation curves substantially under-predicted the heat transfer coefficient values for piston undercrown cooling. The stagnation zone data deviated from the correlation by mean absolute percentage errors between 40 and 50\%. As radial distance increases the downstream data continued to diverge further from the correlation curves with mean absolute percentage errors between 40 and 70\%. The discrepancy between the data and the correlation is attributable to the effect of the large piston undercrown-skirt surfaces which gives extra cooling effects to the undercrown surfaces. The undercrown-skirt acts as cooling fins for heat transfer enhancement, as discussed in Section 6.4 and investigated in Appendix J.
Chapter 7 – Correlated Empirical Equations

7.5 Concluding Remarks

This chapter has investigated and compared the heat transfer correlations and data from the literature survey, respectively. New oil jet correlations were developed for present experimental results which were able to predict the heat transfer coefficient and jet impingement size for high Prandtl number liquid jets impinging at short nozzle standoff distance onto downward-facing surfaces.

The oil jet correlations successfully predict the concave surface results with good accuracy especially in the stagnation zone, but as radial distance increases the accuracy starts to decline due to concave surface geometry which enhanced the local heat transfer coefficients. However, the comparisons in the downstream region are not as important as in the stagnation zone because the maximum heat transfer coefficient values are always located in the stagnation zone. In comparison with piston cooling results, the correlations substantially under-predicted the heat transfer coefficient values by about 50% due to the effect of the large piston undercrown-skirt surfaces which enhanced the local heat transfer coefficients. Therefore, these comparisons have shown that the cooling surface modifications can significantly improve the local oil forced convection values.

The final chapter of this thesis discusses the conclusions from present study and suggestions for further work in oil jet piston cooling.
CHAPTER 8

CONCLUSIONS AND SUGGESTIONS FOR FURTHER WORK

The research presented in this thesis has studied oil jets for IC engine piston cooling applications. The oil jet cooling results were presented from experimental investigations.

An experimental programme was completed using various oil jet configurations and conditions, i.e. temperatures, velocities, sizes and orientations, to generate greater understanding of the parameters which were likely to affect the performance of oil forced convection. The optimum jet configuration was then selected and studied further for piston undercrown cooling. New empirical equations were correlated from this work.

This final chapter summarises the major findings and conclusions generated by this study of cooling oil jets. Recommendations for future work are also presented.
8.1 Conclusions

This research has resulted in new heat transfer results and correlations. The upward impinging jet study consisted of three different jet diameters, five different oil temperatures, six different jet velocities, five different nozzle-to-surface separations, and four different jet impinging angles to study the effects of jet cooling capability, impingement size, splatter or breakup, contraction and interference at three different heated specimens. It has also contributed new insights and understanding of oil jet cooling at downward-facing surfaces.

The important conclusions of this research are as follows:

1. Although some types of oil jet cooling, which were not studied by research, have been developed previously, there has been no published information in regards to an effective oil jet cooling impingement on downward-facing hot surfaces, or their application to IC engine cooling systems, i.e. piston undercrown. The research has demonstrated that a cooling oil jet can be effectively configured, optimally heated and targeted onto a heated metal surface.

2. During upward jet impingement on a flat surface, a 'bell-sheet' flow pattern was observed when the oil jet impinged under highly viscous conditions and at small nozzle-to-surface separations. The downstream liquid film that leaves the surface when its radial velocity is zero to form a thin liquid sheet, which shrinks in diameter as it falls and merges at the bottom. The round shape of the bell-sheet was due to surface tension forces, which pulling from all directions and a round shape has the smallest surface area per unit volume as compared to other shapes. Therefore, the shape of the bell-sheet flow pattern is dependent on the gravitational and surface tension forces.
3. Another observation was interferences between the bell-sheet flow pattern and the jet at large nozzle-to-surface separation. The reduction of the bell-sheet diameter causes it to intersect with the jet impacting and to re-impinge onto the test-surface.

Jet impingement at large Reynolds numbers and nozzle-to-surface separations also induces aerodynamic disturbances on its surface, and hence, jet instability could occur at small distances from the orifice outlet. The disturbances in the incoming jet are strongly amplified as the jet spreads into a liquid film along a wall normal to axis of the jet, causing the droplets to break free from the liquid film or known as ‘splattering’.

4. The local Nusselt numbers for upward oil jet impingement were found to increase as the jet Reynolds number increased. The peak Nusselt numbers occurred in the stagnation zone and the Nusselt number decreased in the downstream region with increasing radial distance. The stagnation Nusselt number was found to increase rapidly for jet impinged at high Prandtl number because of the rapid increasing of impingement size as the jet velocity increased, and it was observed to be more strongly dependent on Reynolds number for laminar jets (Re$_d$ ≤ 2000) than initially turbulent jets (Re$_d$ > 2000), for nozzle diameters of $d = 1.5$ and 3.0 mm.

5. The most significant increase in heat transfer associated with forced convection is believed to be through the turbulent flow of incoming jet which leads to turbulence level in the liquid film across the surface. The vigorous flow in the hydrodynamic boundary layer due to turbulence mixing in the liquid film rapidly removing the heat from the very thin thermal boundary layer.
At large nozzle-to-surface separation, the aerodynamic effect created surface irregularities on the jet leading to a turbulent flow which will tend to enhance the local heat transfer relative to a smooth jet. However, when a jet impinged at high Reynolds numbers and large nozzle-to-surface separations, the jet started to disintegrate and created liquid splattering from the free surface in the vicinity of the impingement region. Splattering has the effect of removing liquid mass from the radial flow, thus resulting in a reduction of local Nusselt numbers in the downstream region.

6. Upward oblique jet impingement showed significant difference between heat transfer coefficients on the either side of jet stagnation point as jet inclination angle increases. On a tilted downward-facing heated surface the up-hill side of the impingement receives a higher cooling effect than down-hill side because a larger proportion of liquid from incoming oblique jet flows in this direction. However, the average local Nusselt numbers at the flat surface showed no significant changes when the jet impinged at different inclination angles, except for a slight decrease for the oblique jet impingement at \( \theta = 45^\circ \).

7. The results from a concave test-surface have shown the downstream heat transfer coefficients were enhanced even though the local heat transfer coefficients in the stagnation zone for concave surface were almost the same as for flat surface. This is attributable to two effects: (i) when a jet impinged onto the concave surface, the downstream liquid film covers and cools the whole area of the concave surface before it leaves at the edge, and (ii) splattered droplets near the stagnation zone re-impinge onto a curved surface in the downstream region. Hence, the local Nusselt number profiles have a symmetric
convex-shape due to the increased downstream heat transfer coefficients.

When the jet impinged at off-centre locations onto a steep slope of concave surface, the local Nusselt numbers on the down-hill side of the stagnation point dropped off more rapidly than those on the up-hill side. Thus, there are significant imbalances of cooling capability on either side of the stagnation point, but the average local Nusselt numbers showed no significant changes.

8. It has been observed that the automotive jets exhibited breakup before impingement, and the breakup was more noticeable when the jet impinged with high Reynolds numbers. This is because the boundary layer development along the walls of the long nozzle causes the velocity nonuniformity in the jet and promotes greater turbulence at high Reynolds numbers. The distance from the orifice until the point where the jet started to breakup (or breakup length) became shorter as the jet Reynolds number increased.

The jet breakup effects on the measured heat transfer values have shown to be significant with spray cooling impingement. At large jet stand-off distances the local heat transfer coefficients were found to be very high, due to turbulent liquid spray covering a large area of the test-surface with liquid film.

9. The selected oil jet cooling set-ups were used to conduct an experimental test on a heated stationary piston undercrown. The results showed higher local Nusselt numbers than those on the aluminium test-plates. This is because of extra cooling effects from the downstream cooling oil that flows down via the large piston undercrown-skirt surfaces which act as cooling fins for heat transfer
enhancement. There was also a sudden increase of local Nusselt numbers at the piston centre point, where a portion of the downstream liquid film decelerated on the crossed surface at the undercrown and flows down via the gudgeon-pin boss and gudgeon pin, hence provided extra cooling effects at the surface near to the undercrown centre line.

The heat transfer results showed an imbalance of cooling rates on either side of the piston centre point when the jet impinged at piston radial distance of 37.5 mm. The results for $d = 1.5$ and $3.0$ mm showed only a small effect of nozzle-to-surface separation (i.e. $Z = 100$ and $200$ mm) on the local Nusselt numbers. The pipe-type automotive jet showed lower local Nusselt numbers than the converging-type automotive jet when impinging at high Reynolds numbers. This was because the large breakup of pipe-type automotive jets showed a portion of its incoming jet squirted outside the piston undercrown, hence loss of liquid mass flow at undercrown surface.

10. The recorded results for converging-type automotive jet impinged onto a flat test-surface showed that the local Nusselt numbers fluctuated and decreased at some points, particularly in the transitional jet flow regime. Beside that, a tear-off of liquid jet surface near the orifice lip was observed in the experiments. This may be attributed to the jet nozzle design i.e. converging section, orifice surface finishing, orifice length and bending angle.

For the piston undercrown, the heat transfer results from a reamed converging-type automotive jet were not much different from the results for the original converging-type automotive jet. This is because the effect of nozzle geometry was found to diminish with the crossed surface modification at piston undercrown and the extra
cooling effects from the liquid film flow structures at undercrown geometry. Therefore, the heat transfer coefficients at the piston undercrown in the present study were not significantly affected by the quality of orifice finishing.

11. The experimental data from a flat test-surface in the present work were used to generate new empirical correlations for oil jet cooling impingement on downward-facing surfaces. The heat transfer correlation was developed for normal ($\theta = 90^\circ$) and inclined ($\theta = 75^\circ$, $60^\circ$ and $45^\circ$) jet impingements. Another correlation was developed for predicting oil jet impingement sizes on a downward-facing flat surface, at isothermal condition.

12. Apart from the heat transfer and fluid dynamics observations from the present research, a flash evaporation or aerosol generation was also observed at the hot surface during jet impingement. This phenomenon appears at high temperatures when the surface is not covered by a continuous flow of liquid film or when the oil jet stops impinging. Thus, a very thin oil film that is left over on the hot surface evaporates as aerosol.

### 8.2 Recommendations for Further Work

This research has led to a number of interesting areas where further work could be directed. These include:

1. Heat transfer coefficient of cooling oil jets at downward-facing surface at static condition has been successfully measured. A further recommended step is to measure the heat transfer at dynamic surface condition, i.e. a moving downward-facing test-surface or reciprocating piston.
2. Currently, only one size of the concave surface has been used to study the free-surface liquid jet impingement. Therefore, a further study of using different sizes of concave surface would be useful to generate new empirical correlations for liquid jet impingement on downward-facing concave surfaces.

3. It is currently not possible to comment precisely on the effects of cross-hatched surfaces on the piston undercrown on the local heat transfer coefficients when the cooling oil jet impinged onto the surface. Therefore, a further study of cooling oil jet impinged onto a surface of different surface roughness or textures would be useful to design the piston undercrown surface for potential heat transfer enhancement.

4. Flash photographic technique has been successfully used to capture still images of oil jet impingement on downward-facing surfaces. A further recommended step is to use laser optical techniques for measuring liquid film thickness and velocity on downward-facing surfaces, which experience the effects of gravitational acceleration.

5. Flash evaporation or aerosol generation was also observed at the hot undercrown surface when no oil jet cooling (or no oil contact). Therefore, an aerosol generation mechanism could be further investigated to minimise the emissions from piston undercrown.
REFERENCES


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References

International des Machines a Combustion (CIMAC), April 5-9, Washington D.C., paper 34, pp. 835-869.


References

Technology-Experimental Simulation of Piston Crown, "SAE Paper 981042.


APPENDIX A

Jet Nozzles for the Present Research

Technical drawing for in-house manufactured contracting-type jets:

1.5 mm Orifice Diameter

- Orifice Diameter: 0.5 mm
- Knurl
- Diamond Knurl

3.0 mm Orifice Diameter

- Orifice Diameter: 0.75 mm
- Knurl
- Diamond Knurl

Material: Stainless Steel

Loughborough University
Digital microscope images for a contracting-type jet ($d = 1.5 \text{ mm}$):

- Chamfered orifice lip
- Round orifice with tiny kerbs at the lip
- Smooth internal surface with tiny lines texture – diamond compound finishing

Digital microscope images for a contracting-type jet ($d = 3.0 \text{ mm}$):

- Chamfered orifice lip
- Smooth round orifice
- Smooth internal surface with tiny lines texture – diamond compound finishing
Digital microscope images for a pipe-type automotive jet ($d = 2.0$ mm):

- Chamfered orifice lip
- Smooth round orifice
- Smooth internal surface

Digital microscope images for a converging-type automotive jet ($d = 2.0$ mm):

- Straight cut orifice
- Irregular round orifice
- Wavy internal surface with high-low points
Digital microscope images for a reamed converging-type automotive jet ($d = 2.0$ mm):

- Straight cut orifice
- Slightly improved round orifice
- Reamed internal surface
APPENDIX B

Surface Area of the Present Test-Surfaces

a) Flat Test-Surface

Test-surface detail:
- Flat surface diameter, $D = 100$ mm
- Surface Area $= 0.25\pi D^2 = 7.854 \times 10^{-3}$ m$^2$
b) Concave Test-Surface

Test-surface detail:
- Radius of curvature, $R = 50$ mm
- Height of concave, $R_2 = 25$ mm
- Diameter of concave base, $R_3 = 2 \sqrt{R_2 (2R - R_2)} = 86.6$ mm
- Concave surface area $= \pi \left(0.25R_3^2 + R_2^2\right) = 7.854 \times 10^{-3}$ m$^2$

**Thermocouples Setup for the Present Test Specimens**

Surface-temperature measurements with thermocouple holes inside the present test specimens.

a) Flat Surface Test-Block
b) Concave Surface Test-Block

Measuring surface temperature
Two thermocouples inside the same hole for measuring heat flux

Central axis

Test surface

12.5 mm

Central axis

12.5 mm

Measuring surface temperature
Two thermocouples inside the same hole for measuring heat flux

Aluminium Hotplate
\( (k = 240 \text{ W.m}^{-1}\text{.K}^{-1}) \)

Diesel Engine Piston
\( (\text{Al-Si alloy, } k = 135 \text{ W.m}^{-1}\text{.K}^{-1}) \)

Test-surface

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APPENDIX C

K-Type Thermocouple Data

Typical temperature gradient for type K thermocouples, as shown in the chart below - obtained from RS Components Ltd.:

![Thermocouple EMF vs Temperature Difference Chart]

Type K thermocouple accuracy, Class 1 is typically ±1.5 °C and Class 2 is typically ±2.5 °C.
APPENDIX D

Engine Oil Data

Shell Rimula X’s customer data sheet - obtained from Shell Global Solutions (UK):
Rimula X (CH-4)
Extra High Performance Diesel Engine Oil

Shell Rimula X Oils are high performance, dedicated heavy duty engine lubricants designed for use in modern high speed turbocharged diesel engines. They use exclusive additive formulations in conjunction with highly refined base oils to deliver longer life and enhanced protection relative to their predecessors.

Rimula X Oils have been reformulated for severe duty service in engines specifically designed to meet 1998/Euro 2 on-highway exhaust emission standards as well as being suitable for a wide range of heavy duty off-highway applications.

Applications
• On-Highway Heavy Duty Trucks
As an integral part of the development of Rimula X, extensive testing of the product has been carried out in road haulage operations around the world confirming the performance of Rimula X in European, American and Japanese equipment under a wide range of haulage conditions.

• Construction and Mining
Rimula X is recommended for most engine types found in construction and mining equipment. It is particularly suitable for Caterpillar, Cummins, Detroit Diesel (4-cycle) and Komatsu engines. It is formulated to provide continuous protection even where higher sulphur fuels are used.

• Agricultural Equipment
Rimula X is ideally suited for the stop-start service found in agricultural operation and protects against bearing wear and deposit formation even under high load low speed conditions when other oils can fail.

Performance Features and Benefits
• Unique New Formulation
Shell technologists have developed exclusive formulations for the new generation of Rimula X designed to deliver a step change in performance levels to improve the efficiency and life of your machinery.

• Greater Protection and Higher Temperatures
With increasing power, modern engines have to work harder for longer, resulting in increased heat and stress on the engine oil. Rimula X has been demonstrated to provide increased resistance to thermal breakdown ensuring continued protection throughout the drain interval, even under the most severe conditions.

• Longer Oil Life
With their exclusive additive formulations, the new Rimula X oils can fight the combined effects of combustion acids and increased heat for longer than their predecessors. This performance reserve ensures the oil continues to protect against wear and corrosion right up until the oil drain.

• Approved by Leading Engine Makers
The new formulation Rimula X engine oils now meets an even wider range of engine maker requirements confirming Rimula X as an ideal choice for mixed fleet diesel engine applications.

Specification and Approvals
American Petroleum Institute:
API CH-4/CG-4/ CF-4/CF2
ACEA E3
Cummins CES 200-71, 72
Cummins (B&M Series) CES 200-75
Mack EO-M
MTU Type II
MAN 271
Mercedes Benz 228.3
RVI RD
Scania E3
Volvo VDS-2
GM Allison C-4

All claims applicable to SAE 15W-40 grade.
1Claims for SAE 10W-30 grade.
2Claims for SAE 20W-50 grade.

Health and Safety
Guidance on Health and Safety are available on the appropriate Material Safety Data Sheet which can be obtained from your Shell representative.

Protect the environment
Take used oil to an authorised collection point. Do not discharge into drains, soil or water.
Continued for Shell Rimula X's customer data sheet:

# Typical Physical Characteristics

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These characteristics are typical of current production. Whilst future production will conform to Shell's specification, variations in these characteristics may occur.
Shell Rimula X (SAE 15W-40)’s thermal properties - obtained from Shell Global Solutions (UK):

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The only available data for Shell Rimula X (SAE 15W-40) obtained from Shell Global Solutions (UK), the surface tension ($\sigma$) at 25 °C is 0.035 to 0.04 N.m$^{-1}$. 

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APPENDIX E

Heat Balance for the Present Test-Block

The heat losses (W or watt) for the present test-block were measured using the three methods, i.e. the electrical power, the energy equation and the temperatures measured inside the test-block. Hence, the results obtained from these methods were used to check the accuracy of the present lab data. These three methods were discussed below:

1) The heat input into the flat surface test-block was measured using a Power & Energy Monitor unit. The Power unit measured the energy input (in kW.hr) every time when the cartridge heaters releasing heat to the test-block. Hence, the heat losses through the test-block surfaces were obtained as

- **Test 1**: The measured total heat loss of 82.95 W, from test-block to convection air and no cooling oil jet impingement.
- **Test 2**: The measured total heat loss of 224.972 W, from test-block to convection air was 79.045 W and to cooling oil jet (at \( d = 1.5 \) mm, \( T_j = 70 \) °C, \( u = 10 \) m.s\(^{-1}\)) was **145.927 W**.
• Test 3: The measured total heat loss of 400 W, from test-block to convection air was 71.359 W and to cooling oil jet (at $d = 1.5 \text{ mm}$, $T_j = 70 \, ^\circ\text{C}$, $u = 30 \, \text{m.s}^{-1}$) was **328.641 W**.

2) The heat loss to cooling oil jet was calculated using the following equation

\[
Q = \dot{m} c_p \Delta T = \rho \dot{V} c_p \Delta T
\]

where $Q$ is the heat loss, $\dot{m}$ is the mass flow rate of cooling oil jet, $\rho$ is the density of cooling oil jet, $\dot{V}$ is the volumetric rate of cooling oil jet and $\Delta T$ is the temperature difference between the incoming oil jet temperature and oil temperature leaving the test-surface ($\Delta T = T_{j-out} - T_{j-in}$), as shown in figure below.

![Diagram showing thermocouple measuring oil temperature leaving the surface, $T_{j-out}$ and incoming oil jet temperature, $T_{j-in}$](image)

Hence, the calculated heat losses were:

- Heat loss to cooling oil jet (at $d = 1.5 \, \text{mm}$, $T_j = 70 \, ^\circ\text{C}$, $u = 10 \, \text{m.s}^{-1}$) was:
  
  \[
  = 855.935 \times 2.65072 \times 10^{-5} \times 2080.3491 \times (72.6 - 70)
  \]
  
  \[
  = 108.560 \, \text{W}
  \]

- Heat loss to cooling oil jet (at $d = 1.5 \, \text{mm}$, $T_j = 70 \, ^\circ\text{C}$, $u = 30 \, \text{m.s}^{-1}$) was:
  
  \[
  = 855.935 \times 5.30144 \times 10^{-5} \times 2080.3491 \times (73.8 - 70)
  \]
  
  \[
  = 358.719 \, \text{W}
  \]
3) The heat losses inside the heater block were obtained using the lab data measured using thermocouples.

- The lab data in the table below obtained from a cooling oil jet impinged at \( d = 1.5 \text{ mm} \), \( T_j = 70 ^\circ \text{C} \), \( u = 10 \text{ m.s}^{-1} \) and the measured impingement radius on the surface was 23.6 mm:

<table>
<thead>
<tr>
<th>Radial location, ( r ) (mm)</th>
<th>0</th>
<th>12.5</th>
<th>23.6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat flux, ( q ) (W.m(^{-2}))</td>
<td>131823</td>
<td>67209</td>
<td>39500</td>
</tr>
</tbody>
</table>

The polynomial approximation from the graph above was used for \( q \) as function of \( r \), \( q(r) \), for equation below:

\[
Q = \int_{r=0}^{r=0.0236} q \, dA = 2\pi \int_{r=0}^{r=0.0236} q(r) \, r \, dr
\]

where

\[
q(r) = 113257206.28577r^2 - 6584896.11305r + 131823.81466
\]
$Q = 2\pi \int_{r=0}^{r=0.0236} [q(r)r \, dr$

$= 2\pi \int_{r=0}^{r=0.0236} (113257206.28577r^3 - 6584896.11305r + 131823.81466) \, dr$

$= 2\pi \left[ 113257206.28577 \left( \frac{r^4}{4} \right)_{0}^{0.0236} - 6584896.11305 \left( \frac{r^3}{3} \right)_{0}^{0.0236} + 131823.81466 \left( \frac{r^2}{2} \right)_{0}^{0.0236} \right]$

$= 2\pi \left[ 113257206.28577 \left( \frac{0.0236^4}{4} \right) - 6584896.11305 \left( \frac{0.0236^3}{3} \right) + 131823.81466 \left( \frac{0.0236^2}{2} \right) \right]$

$= 104.6$

Hence, from the analytical integration the measured heat loss to cooling oil with impingement radius of 23.6 mm was **104.6 W**.

- The lab data in the table below obtained from a cooling oil jet impinged at $d = 1.5 \text{ mm}$, $T_j = 70 \text{ °C}$, $u = 30 \text{ m/s}$ and the measured impingement radius on the surface was 40.7 mm:

<table>
<thead>
<tr>
<th>Radial location, $r$ (mm)</th>
<th>0</th>
<th>12.5</th>
<th>25.0</th>
<th>37.5</th>
<th>40.7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat flux, $q$ (W.m$^{-2}$)</td>
<td>184884</td>
<td>115288</td>
<td>69166</td>
<td>33889</td>
<td>27000</td>
</tr>
</tbody>
</table>

The polynomial approximation from the graph above was used for $q$ as function of $r$, $q(r)$, for equation below:
\[ Q = \int_{r=0}^{r=0.0407} q(r) \, dr \]

where

\[ q(r) = -858622639.17938r^3 + 104452750.81458r^2 - 6717306.16179r + 184832.70003 \]

\[ Q = 2\pi \int_{r=0}^{r=0.0407} q(r) \, dr \]

\[ = 2\pi \int_{r=0}^{r=0.0407} \left( -858622639.17938r^3 + 104452750.81458r^2 - 6717306.16179r + 184832.70003 \right) \, dr \]

\[ = 2\pi \left[ -858622639.17938 \left( \frac{r^4}{4} \right)_{0}^{0.0407} + 104452750.81458 \left( \frac{r^3}{3} \right)_{0}^{0.0407} - 6717306.16179 \left( \frac{r^2}{2} \right)_{0}^{0.0407} + 184832.70003 \right] \]

\[ = 2\pi \left[ -858622639.17938 \left( \frac{0.0407^4}{4} \right) + 104452750.81458 \left( \frac{0.0407^3}{3} \right) - 6717306.16179 \left( \frac{0.0407^2}{2} \right) + 184832.70003 \right] \]

\[ = 343.1 \]

Hence, from the analytical integration the measured heat loss to cooling oil with impingement radius of 40.7 mm was **343.1 W**.

Conclusions:

- For a cooling oil jet impinged at \( d = 1.5 \) mm, \( T_j = 70 \, ^\circ C \), \( u = 10 \, m.s^{-1} \), the difference of heat loss between the lab data and measured electrical power is 39.5% and calculated energy equation is 3.79%.
- For a cooling oil jet impinged at \( d = 1.5 \) mm, \( T_j = 70 \, ^\circ C \), \( u = 30 \, m.s^{-1} \), the difference of heat loss between the lab data and measured electrical power is 4.21% and calculated energy equation is 4.55%.
APPENDIX F

Bell-Sheet Temperature Measurements

- Heater off
- Thermocouple reading is 54.1 °C
- Inside air temperature is about 55 °C
- Outside air temperature near the thin sheet is about 34 °C
- Thermocouple reading is 53.3 °C
This section shows an oil jet at 55 °C impinging onto a non-heated downward-facing flat surface. The thermocouple measurements showed the thin sheet temperature decreased as it descends, due to releasing heat to the surrounding. The measurements also showed the air inside the bell-sheet is warmer than the air outside the thin sheet, due to hot air accumulation inside the bell-sheet.
APPENDIX G

Surface Tension Effects for the Present Research

Effect of capillary action (oil) in small tube and the experimental setup shown in figure below:

- Glass tubing, internal diameter is 0.6 mm
- Height of column for oil that wets the tube
- Glass bowl sitting on hotplate
- K-type thermocouple taking oil temperature
- Shell Rimula X (15W-40) engine oil
Measured oil column heights in the glass tubing, are shown in table below:

<table>
<thead>
<tr>
<th>Oil temperature °C</th>
<th>Oil column height inside the tube (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>15.98</td>
</tr>
<tr>
<td>45</td>
<td>16.34</td>
</tr>
<tr>
<td>65</td>
<td>16.95</td>
</tr>
<tr>
<td>85</td>
<td>17.32</td>
</tr>
<tr>
<td>105</td>
<td>17.67</td>
</tr>
<tr>
<td>125</td>
<td>17.99</td>
</tr>
<tr>
<td>145</td>
<td>18.05</td>
</tr>
</tbody>
</table>

Oil column height as function of temperature:

Figure below shows the oil column height in glass tubing for different temperatures. The oil column height increased as temperature increased. This is because the oil becomes more wetting at high temperatures, thus lifting up the oil level in the tube. The surface tension for the present engine oil at 25 °C is 0.035 N.m⁻¹ (see Appendix D). Hence, from the equation below, the angle of contact inside the glass tubing (0.6 mm diameter) is 53.6° and it should be decreased as surface tension decreased with temperature (Munson et al., 2006).
where $\sigma$ is the liquid surface tension, $\theta$ is the angle of contact, $\rho$ is the liquid density, $r$ is the internal radius of glass tubing, and $g$ is the gravity.
Figure below shows the effect of capillary action (water) in small tube and the water column height for different temperatures. As seen in the figure above, the water column height was fairly unchanged at 34 mm in the tests between the temperatures of 10 °C and 90 °C. The surface tension for water at 10 °C is 0.0742 N.m⁻¹ (Munson et al., 2006).

Water (997.1 kg.m⁻³ at 25 °C) has higher density than oil (883.8 kg.m⁻³ at 25 °C), and the water surface tension (0.072 N.m⁻¹ at 25 °C) is twice stronger than the oil surface tension (0.035 N.m⁻¹ at 25 °C). Therefore, the water column height is higher than the oil column height.

Compared to water, this section has shown that the oil surface tension is strongly affected by temperature with the rapid rise of oil column height in glass tubing as function of temperature.
Method for finding the surface area of three common shapes with same volume:

<table>
<thead>
<tr>
<th>Shapes</th>
<th>Volume (V) = 1 m³</th>
<th>Surface Area (A) m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sphere</td>
<td>( \frac{4\pi r^3}{3} = 1 )</td>
<td>( 4\pi \left( \frac{3}{4\pi} \right)^2 ) = 4.836</td>
</tr>
<tr>
<td>Cylindrical</td>
<td>( \frac{\pi D^3}{4} = 1 )</td>
<td>( 2 \times \frac{\pi D^2}{4} + \pi D^2 ) = 5.536</td>
</tr>
<tr>
<td>Cube</td>
<td>( L^3 = 1 )</td>
<td>( 6 \times L^2 ) = 6</td>
</tr>
</tbody>
</table>

The spherical shape in table above showed the smallest surface area per unit volume, when compared to other shapes (i.e. cylindrical and cube).
APPENDIX H

Setup for Large Jet Standoff Distances to Prevent Jet Interference

Setup of Perspex tubes for nozzle-to-surface separations (Z) of 100 and 200 mm, respectively:
Contraction of thin liquid sheet wall on outside the clear tube, to prevent jet interference during bell-sheet flow pattern:
APPENDIX I

3-D Model Study for the Present Test-Plate

Heat transfer setup for Disc1:

Materials:

\[ k_{\text{Alu}} = 202.4 \text{ W.m}^{-1}\text{K}^{-1} \]
\[ k_{\text{Copper}} = 387.5 \text{ W.m}^{-1}\text{K}^{-1} \]

Heating:
\[ T_H = 150 ^\circ \text{C} \]

Ambient:
\[ T_a = 18 ^\circ \text{C} \]
\[ h_a = 5 \text{ W.m}^{-2}.\text{K}^{-1} \]

25 mm

Cooling surface \((D_j)\):
\[ T_j = 100 ^\circ \text{C} \]
\[ h_j = 3000 \text{ W.m}^{-2}.\text{K}^{-1} \]

Test-surface \((D = 100 \text{ mm})\)

Bottom view of test-plate
Centre point heat flux loss for different ratios of cooling-to-surface diameter:

![Graph showing heat flux loss for different ratios of cooling-to-surface diameter for aluminium and copper discs.]

The distribution of heat flux prediction as function of radial location with varying cooling diameter, for aluminium disc:

![Graph showing heat flux as function of radial location for aluminium disc with different cooling diameters.]

The distribution of heat flux as function of radial location with varying cooling diameter, for copper disc:

![Graph showing heat flux as function of radial location for copper disc with different cooling diameters.]

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Discussions:
The overall heat losses inside the test-plate increased when the cooling surface area increased, and the centre point showed the highest heat flux for all cooling surface sizes. For cooling surface of the same size as test-surface, $D_i/D = 1$, the surface heat losses were quite uniform except for the radial distance after 40 mm where it decreased rapidly. This was due to the length (80 mm) of the heating at the top section of the test-plate. In the present study, the values of heat transfer coefficient were recorded at the centre point and radial direction up to 37.5 mm, hence, all studied values were measured at the surface which has receiving the same amount of heat flux.

As the size of the cooling surface ($D_i$) decreased, the surface heat fluxes were no more uniform and at distances away from the centre point the heat flux decreased gradually. The radial heat flux decreased more rapidly as the cooling surface decreased.

For different type of materials, the copper test-plate is conducting more heat than the aluminium test-plate because of its high thermal conductivity.
3D model study for test-plates of 15 mm thinner and thicker than Disc1
Centre point heat flux predictions with different cooling diameters and disc thickness (for aluminium disc):

Table below shows the predicted centre point heat flux (W.m\(^{-2}\)) for aluminium disc of different disc thickness and cooling diameters:

<table>
<thead>
<tr>
<th>Disc Type</th>
<th>Ratio of cooling surface to test-surface, (D/D)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1.0</td>
</tr>
<tr>
<td>Thinner Disc</td>
<td>116163</td>
</tr>
<tr>
<td>Disc1</td>
<td>101626</td>
</tr>
<tr>
<td>Thicker Disc</td>
<td>89341</td>
</tr>
</tbody>
</table>

In figures below, the solid lines represent results for Disc1, the dotted lines represent results for the thinner disc, and the dash-dotted lines represent results for the thicker disc.
Comparison of heat flux predictions for the thinner disc compared to Disc1:

![Graph showing heat flux predictions for the thinner disc compared to Disc1.](image)

Comparison of heat flux predictions for the thicker disc compared to Disc1:

![Graph showing heat flux predictions for the thicker disc compared to Disc1.](image)

Discussions:
In the cases of thinner and thicker discs (15 mm thinner and thicker than Disc1), the results also showed uniform heat flux when the test-surface was
fully covered by cooling, and it decreased rapidly after the radial distance of 40 mm from the centre point.

For the thinner disc, the predicted heat flux at the centre point was greater than the thicker disc. However, it showed a steeper drop of heat flux at radial direction away from the centre point and it is more noticeable when the cooling surface size decreases. The thickest disc showed more uniform heat flux throughout the test-surface, due to thermal spreading by radial conduction within the disc. Therefore, the heat flux at the centre of the disc decreased.

**Comparison of local Nusselt number for different disc thickness and** \( D/d \)**

Local Nusselt numbers were obtained from the equation below:

\[
Nu = \frac{hd}{k_{oil}} = \frac{qd}{k_{oil}(T_w - T_j)}
\]

where \(d = 1.5\) and \(3.0\) mm, \(k_{oil} = 0.132\) W.m\(^{-1}\).K\(^{-1}\), \(T_j = 100\) °C.

**Comparison of normalised local Nusselt numbers \((Nu_{local}/Nu_{r/d=0})\) for different ratios of cooling-to-surface diameter, for Disc1:**

![Graph showing comparison of normalised local Nusselt numbers for different ratios of cooling-to-surface diameter. The graph includes lines for different ratios (D/d = 0.8, 0.6, 0.4) with respect to r (m) on the x-axis and Nu/Nu-max on the y-axis. The graph is labeled D/d = 66.67.]
Predicted distributions of local $Nu$ for different disc thickness and $D/d$, for $D/d = 0.6$:

Discussion:
The maximum Nusselt number was predicted to be located at the centre point of the cooling surface for all the disc thickness, and after that, the local Nusselt numbers decreased gradually with radius. The local Nusselt number decreased more rapidly as the cooling surface decreased.

The thinner disc showed the highest Nusselt number at the centre point, but after that, the radial Nusselt numbers declined more rapidly. The thicker disc showed more uniform local Nusselt numbers throughout the cooling surface,
because the temperature distributions at the cooling interface were more uniform due to thermal spreading by radial conduction within the disc. Hence, the Nusselt number at the centre point of the disc was predicted to decrease. The local Nusselt numbers increased as the ratio of surface-to-jet diameter ($D/d$) decreased.

**Thermal contours and isothermal profiles for different disc thickness**

Predicted thermal contours and isothermal profiles for different disc (aluminium) thickness when $D/d = 0.4$:

a) Thicker disc

b) Disc1

c) Thinner disc
Comparison of predicted thermal contours and isothermal profiles for different disc materials when $D_f/D = 0.4$:

a) Aluminium Disc1

b) Copper Disc1

Discussion:

It is interesting to notice that when the disc thickness is large the temperature remains uniform over the top part of the disc. The isothermal lines were almost parallel to each other indicating one-dimensional heat conduction in region away from the cooling surface and did not significantly affect the convective heat transfer process. However, in the region near the cooling interface, the isothermal lines tend to be concentric around the centre point. The effect of non-uniform heat transfer is found only across the bottom part of the disc. When the disc is thinner, the concentric isothermal lines were more concentrated around the cooling region and it represents high temperature gradients at the test-surface. The shape of the isotherms is not significantly affected by thermal conductivity.
APPENDIX J

3-D Model Study for the Effects of Piston Skirt Cooling

Setup conditions:

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium piston</td>
<td>$k_{al} = 240 \text{ W.m}^{-1}\text{K}^{-1}$</td>
</tr>
<tr>
<td>Piston bore diameter</td>
<td>105 mm</td>
</tr>
<tr>
<td>Piston height</td>
<td>84 mm in Case 1; 34 mm in Case 2</td>
</tr>
<tr>
<td>Ambient with air flow</td>
<td>$T_a = 18 , ^\circ\text{C}; , h_a = 10 , \text{W.m}^{-2}\text{K}^{-1}$</td>
</tr>
<tr>
<td>Undercrown cooling</td>
<td>$T = 55 , ^\circ\text{C}; , k_{oil} = 0.132 , \text{W.m}^{-1}\text{K}^{-1}; , \bar{h} = 2900 , \text{W.m}^{-2}\text{K}^{-1}$</td>
</tr>
</tbody>
</table>

Case 1: Piston with *long skirt* has a height of 84 mm
As indicated by the arrow, the calculated heat flux at this point is

\[ q = \frac{k_{\text{diff}} \Delta T}{y} = \frac{240(145.14 - 120.41)}{0.0143} = 415048.95 \text{ W.m}^{-2} \]

Hence, the local Nusselt number is

\[ \text{Nu} = \frac{qd}{k_{\text{diff}} \Delta T} = \frac{415048.95(0.0015)}{0.132(65.41)} = 72.1 \]

**Case 2: Piston with short skirt has a height of 34 mm**

After trimmed away the piston skirt by 50 mm, now the piston height is 34 mm.
As indicated by the arrow, the calculated heat flux at this point is

\[
q = \frac{k_{\text{air}} \Delta T}{\gamma} = \frac{240(146.1 - 126.9)}{0.0143} = 322237.76 \text{ W.m}^{-2}
\]

Hence, the local Nusselt number is

\[
\text{Nu} = \frac{q d}{k_{\text{oil}} \Delta T} = \frac{322237.76(0.0015)}{0.132(71.9)} = 50.9
\]

Discussion:
The local Nusselt number in Case 1 is predicted to be 42% higher than Case 2, because of the large heat losses for piston undercrown with long skirt, and hence, extra cooling effects due to large undercrown surfaces.
APPENDIX K

The present heat transfer correlation in Equation (7-2) has been tested for a small range of \(0 \leq r/d \leq 1\), using the jet properties at \(r/d = 0\).

Local Nu profile for \(d = 1.5\) mm at different \(\theta\)
Discussion:

The present correlation has been shown to be valid for a small range of $0 \leq r/d \leq 1$. For all profiles, the maximum heat transfer coefficient occurred at the stagnation point ($r/d = 0$) and declined with radial distance ($r/d < 0$ or $r/d > 0$). As the jet impingement angle increased, the heat transfer coefficient on the down hill side ($r/d < 0$) dropped off more rapidly while that on the up hill side ($r/d > 0$) decreased more slowly. The heat transfer rate on the up hill side was higher than that on the down hill side, and the difference between the two heat transfer rates tends to increase with increasing jet inclination, but not significant within the small range of $r/d$. 