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Control of Nucleate Boiling with Micro-Machined Surface Features

by

Adrian Mark Holland
B.Eng (Hons) DIS

A Doctoral Thesis submitted in partial fulfilment of the requirements for the award of Doctor of Philosophy of Loughborough University

January 2004

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Abstract

This thesis discusses the production and use of laser-machined boiling grids that provide controlled nucleate boiling and enhanced heat transfer characteristics for application primarily to IC engine cooling systems.

The surface features of heated plates are known to have a significant effect on nucleate boiling heat transfer and bubble growth dynamics. Nucleate boiling starts from discrete bubbles that form on surface imperfections, such as cavities or scratches. The gas or vapours trapped in these imperfections serve as nuclei for the bubbles. After inception, the bubbles grow to a certain size and depart from the surface. The bubble departure process significantly increases heat transfer rates compared to pure convection.

In this work, special heated surfaces were manufactured by laser machining cavities into polished aluminium plates. This was accomplished with an Nd:YAG laser system, which allowed drilling of cavities of a known diameter. The size range of cavities was 25 to 300 micrometers. The resulting nucleate pool boiling was analysed using a high-speed imaging system comprising an infrared laser and high resolution CCD camera. This system was operated up to a 2 kHz frame rate and digital image processing allowed bubbles to be analysed statistically in terms of departure diameter, departure frequency, growth rate, shape and velocity. Data were obtained for heat fluxes up to 150 kW.m². Bubble measurements were obtained working with water at atmospheric pressure. The surface cavity diameters were selected to control the temperature at which vapour bubbles started to grow on the surface. The selected size and spacing of the cavities was also explored to provide optimal heat transfer. Insights into the interaction and interseeding mechanism were obtained.

The research has demonstrated that nucleate boiling can be controlled by optimally sized and spaced laser-machined cavities in heated metal surfaces.
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<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Area (m²)</td>
</tr>
<tr>
<td>C</td>
<td>Calibration value (µm.pixels⁻¹)</td>
</tr>
<tr>
<td>C₁</td>
<td>Constant (-)</td>
</tr>
<tr>
<td>C₂</td>
<td>Constant (-)</td>
</tr>
<tr>
<td>Cₚ</td>
<td>Specific heat (J.kg⁻¹.K⁻¹)</td>
</tr>
<tr>
<td>Cₛᵣᶠ</td>
<td>Rohsenow's interface coefficient (-)</td>
</tr>
<tr>
<td>Dₜ</td>
<td>Bubble departure diameter (m)</td>
</tr>
<tr>
<td>Dₜ,a</td>
<td>Bubble departure diameter, area equivalent (m)</td>
</tr>
<tr>
<td>Dₜ,m</td>
<td>Bubble departure diameter, mean (m)</td>
</tr>
<tr>
<td>Dₜ,s</td>
<td>Bubble departure diameter, Sauter mean (m)</td>
</tr>
<tr>
<td>Dₜ,v</td>
<td>Bubble departure diameter, volume mean (m)</td>
</tr>
<tr>
<td>Dₖ</td>
<td>Cavity diameter (µm)</td>
</tr>
<tr>
<td>Dₖₓ</td>
<td>Cavity diameter, x axis (µm)</td>
</tr>
<tr>
<td>Dₖᵧ</td>
<td>Cavity diameter, y axis (µm)</td>
</tr>
<tr>
<td>Dₜ</td>
<td>Hydraulic diameter (m)</td>
</tr>
<tr>
<td>E₀</td>
<td>Laser energy (W)</td>
</tr>
<tr>
<td>F</td>
<td>Friction number (-)</td>
</tr>
<tr>
<td>f</td>
<td>Laser frequency (Hz)</td>
</tr>
<tr>
<td>fₜ</td>
<td>Bubble departure frequency (Hz)</td>
</tr>
<tr>
<td>h</td>
<td>Heat transfer coefficient (W.m⁻².K⁻¹)</td>
</tr>
<tr>
<td>hₜₒᵢˡ</td>
<td>Heat transfer coefficient, boiling (W.m⁻².K⁻¹)</td>
</tr>
<tr>
<td>hₜₒᵢˡ</td>
<td>Heat transfer coefficient, convective (W.m⁻².K⁻¹)</td>
</tr>
<tr>
<td>hₜₒᵢˡ</td>
<td>Enthalpy of evaporation (J.kg⁻¹)</td>
</tr>
<tr>
<td>hₜₒᵢˡ</td>
<td>Heat transfer coefficient, mean (W.m⁻².K⁻¹)</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity (W.m⁻¹.K⁻¹)</td>
</tr>
<tr>
<td>kₙ</td>
<td>Thermal conductivity, liquid (W.m⁻¹.K⁻¹)</td>
</tr>
<tr>
<td>L</td>
<td>Heated length (m)</td>
</tr>
<tr>
<td>Lₖ</td>
<td>Cavity depth (m)</td>
</tr>
<tr>
<td>Lₖₘₐₓ</td>
<td>Drilling depth, maximum (µm)</td>
</tr>
</tbody>
</table>
\( L_{\text{crown}} \) Crown height (\( \mu m \))
\( n \) Number of cavities in a grid (-)
\( N_a \) Active site density (sites.m\(^{-2}\))
\( N_{as} \) Active site density, of cavities with small mouth angle (sites.m\(^{-2}\))
\( p \) Perimeter, section (m)
\( P \) Pressure (Pa)
\( P_0 \) Power, peak laser (W)
\( p_b \) Perimeter, bubble (m)
\( P_l \) Liquid pressure (Pa)
\( Pr \) Prandtl number (-)
\( P_{\text{sat}} \) Saturation pressure (Pa)
\( P_V \) Vapour pressure (Pa)
\( Q \) Heat flux (W)
\( q \) Specific heat flux (W.m\(^{-2}\))
\( q_{\text{boil}} \) Specific heat flux, boiling (W.m\(^{-2}\))
\( q_{\text{conv}} \) Specific heat flux, convective (W.m\(^{-2}\))
\( R_a \) Surface roughness, average (\( \mu m \))
\( r_b \) Bubble Radius (m)
\( r_c \) Cavity radius (m)
\( r_{c*} \) Active cavity radius, critical (m)
\( r_{c,\text{max}} \) Active cavity radius, maximum (m)
\( r_{c,\text{min}} \) Active cavity radius, minimum (m)
\( Re \) Reynolds number (-)
\( R_t \) Surface roughness, max peak to valley (\( \mu m \))
\( S \) Separation distance (mm)
\( S_{\text{chen}} \) Chen's suppression factor (-)
\( T \) Temperature (K)
\( t \) Time (s)
\( T_{\text{act}} \) Temperature activation (K)
\( T_{\text{amb}} \) Temperature, ambient (K)
\( T_{\text{boil}} \) Temperature, boiling point (K)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{bulk}}$</td>
<td>Temperature bulk (K)</td>
</tr>
<tr>
<td>$t_g$</td>
<td>Time, growth (s)</td>
</tr>
<tr>
<td>$T_{\text{sat}}$</td>
<td>Temperature, saturation (K)</td>
</tr>
<tr>
<td>$t_w$</td>
<td>Time, waiting (s)</td>
</tr>
<tr>
<td>$T_{\text{wall}}$</td>
<td>Temperature, wall (K)</td>
</tr>
<tr>
<td>$V_{fg}$</td>
<td>Specific volume change of vaporisation ($\text{m}^3\text{kg}^{-1}$)</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Contact angle, liquid to solid (°)</td>
</tr>
<tr>
<td>$\delta$</td>
<td>Boundary layer thickness (m)</td>
</tr>
<tr>
<td>$\psi$</td>
<td>Cavity mouth angle (°)</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>Cavity apex angle, $\varphi + \psi = 180°$ (°)</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Contact angle, vapour to solid (°)</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density ($\text{kg.m}^{-3}$)</td>
</tr>
<tr>
<td>$\nu$</td>
<td>Shape parameter (-)</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Surface tension (N.m$^{-1}$)</td>
</tr>
<tr>
<td>$\rho_l$</td>
<td>Density of liquid ($\text{kg.m}^{-3}$)</td>
</tr>
<tr>
<td>$\mu_l$</td>
<td>Viscosity, liquid ($\text{kg.m}^{-1}.\text{s}^{-1}$)</td>
</tr>
<tr>
<td>$\theta_{\text{sat}}$</td>
<td>Saturation superheat (K)</td>
</tr>
<tr>
<td>$\xi_{\text{sat}}$</td>
<td>Superheat / Saturation superheat, $\theta/\theta_{\text{sat}}$ (-)</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>Temperature difference (K)</td>
</tr>
<tr>
<td>$\Delta T_m$</td>
<td>Temperature difference, mean (K)</td>
</tr>
<tr>
<td>$\rho_v$</td>
<td>Density of vapour ($\text{kg.m}^{-3}$)</td>
</tr>
<tr>
<td>$\theta_{\text{wall}}$</td>
<td>Wall superheat (K)</td>
</tr>
<tr>
<td>$\Delta x$</td>
<td>Displacement, $x$ axis (m)</td>
</tr>
</tbody>
</table>

**Superscripts**

* critical
Subscripts

b       bubble
boil    boiling
bulk    bulk liquid
c       cavity
conv    convection
g       growth
l       liquid
max     maximum
min     minimum
sat     saturation
v       vapour
w       waiting
wall    solid surface

Abbreviations

ACR     Active Cavity Ratio
AR      Aspect Ratio
BDIA    Bubble Digital Image Analysis
CCD     Charge Coupled Device
CFD     Computational Fluid Dynamics
CNC     Computer Numerical Controlled
EG      Ethylene Glycol
fps     frames per second
He:Ne   Helium: Neon
HTC     Heat Transfer Coefficient
IC      Internal Combustion
ID      Identification Number
Nd:YAG  Neodymium Yttrium Aluminium Garnet
SI      Spark Ignition
Dedication

This thesis is dedicated to my wife Holly, for her steadfast love and support. Also to my parents, Wylda and Brian, who have supported me throughout my long educational career.

Acknowledgements

This thesis wouldn't have been possible without the help of many people. I'd like to thank Loughborough University, all at Perkins Engines Company and the EPSRC for supporting this research, both technically and financially. Particular thanks must go to my project supervisor Colin Garner, whose advice and encouragement was unfailing.

There are too many family, friends and colleagues to list individually, but their support, help and entertainment over the last three years has held me together. I would particularly like to thank Stuart Arnold and Anton Zimmermann for going beyond the call of duty and always being there for me, thank you.
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. Ever since the invention of the IC engine (Jean Lenoir, 1858) designers have been aware that in order to maximise engine durability and efficiency, cooling of engine components is essential. In 1969, French wrote an influential paper entitled “Taking 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 research reported in this thesis concerns novel laser-manufactured boiling grids for enhancing and controlling nucleate boiling heat transfer in engine cooling ducts. Nucleate boiling is shown to provide significantly enhanced engine cooling.
In this first chapter, the effect of the cooling systems on engine performance is introduced. Areas discussed include thermal fatigue, volumetric efficiency, combustion and parasitic losses of auxiliaries such as the coolant water pump. The potential benefits associated with optimising an IC engine cooling system are also reviewed. Finally, this chapter describes the structure of the thesis and the major contributions resulting from this research.

1.1 Importance of Cooling IC Engines

Some 16 - 35 % of the input fuel energy is lost as heat to the coolant circuit in a standard diesel engine (Heywood, 1988). Angus et al (1970) found that the flame face temperature could also be as high as 680 °C, which is close to the thermal limits of most cylinder head materials. It is therefore important that an engine is cooled effectively to ensure durability and efficiency. This is discussed in the following sections.

1.1.1 Durability

If the cooling system is inappropriately designed the lifetime of an engine can be significantly reduced due to the following reasons:

- **Thermal fatigue:** All materials expand and contract with temperature. Varying thermal expansion coefficients throughout the engine cause internal stresses, which increase with temperature. Thermal expansion may also affect manufacturing tolerances, for example bearing clearances or piston ring sealing. Finally thermal cycles aggravate the effect of these stress loads and can increase the fatigue on engine components. These all have a detrimental effect on component performance and can lead to premature failure.
The engine cooling system minimises the peak material temperatures and fluctuations, thereby reducing the detrimental effects of thermal cycles and component expansion.

- **Creep**: When a metal is held at an elevated temperature and is under a stress of a certain magnitude, the material will creep and may eventually rupture. Creep is a deformation dependent on stress, temperature and time. Reducing the thermal loading of a component by effective cooling minimises the onset of creep (Roy and Craig, 1996).

- **Lubrication**: The engine block transfers heat to the lubricant. If this lubricant reaches temperatures above approximately 150 °C it may begin to thin, reducing its effectiveness and life (Heywood, 1988).

### 1.1.2 Efficiency

The cooling system design will determine the temperature of the engine block and cylinder head. In turn, this will affect the engine's efficiency in several ways:

- **Volumetric Efficiency**: Air entering the cylinder must flow through the cylinder head before it enters the cylinder. If heat is transferred to the air from the hot surface, then the air density in the cylinder will be lowered, decreasing the engine's volumetric efficiency. Effective cooling of the cylinder head surrounding the inlet ports can minimise this effect.

- **Combustion**: Hotspots on the cylinder walls (specifically spark ignition engines) may cause abnormal ignition of the fuel-air mixture (known as pre-ignition or knocking). The likelihood of hotspots can be reduced by effective cooling. Knocking impairs
the engine's performance whilst also increasing component mechanical and thermal stress.

- **Emissions**: Stringent environmental legislation determines the levels of emissions such as NOx, CO, and hydrocarbons that an engine is allowed to produce. Flame temperatures and cylinder wall temperatures can have a significant effect on the engine's emissions (Heywood, 1988). Wall temperatures and in-turn gas temperatures are affected by cooling of the cylinder block and head.

- **Power Consumption and Fuel Economy**: Auxiliary engine components such as water pumps and radiator fans consume power from the engine which could be used by the engine itself. Efficient cooling system design can minimise the power consumed by these items, thereby maximising engine power output and improving fuel economy.

This section has shown how the cooling system ensures durability and efficiency of the engine, and is therefore an essential part of the modern IC engine system. As with all engines, the design of the cooling systems is an area to address for possible improved efficiency and hence the provision of associated benefits to the total system.

1.2 **Cooling System Design**

A number of authors such as French (1969) and Leshner (1983) have highlighted the potential benefits for both engine and vehicle design if the cooling system is optimised. This section discusses some of the key benefits, besides durability and efficiency, associated with effective cooling system design.
When optimising an engine cooling system, it must be remembered that it is part of the larger engine and vehicle system. Changes to small components such as the cooling system may affect engine parameters that then may affect the overall design of the vehicle. The following examples illustrate this:

1. **Design Freedom:** If an IC engine cooling system could be reduced in size it could minimise design compromises. For example, limited space in the engine cylinder head necessitates a compromised design of inlet and exhaust ports. Smaller cooling channels in the cylinder head would allow designers to use different (possibly larger) intake port geometries and configurations.

2. **Radiator Design:** In order to remove heat from the engine cylinder head and block, energy is transferred to the coolant liquid and then dissipated to the atmosphere through the radiator. Therefore, the optimisation of the cooling system is intrinsically linked to the design of the radiator. If a higher coolant temperature was possible, then the radiator could reject more heat and it could be decreased in size, leading to a reduction in the size of fan, its noise and its use of power.

3. **Package Volume:** When engines are used to power a vehicle, a reduction in the volume of the cooling system and its auxiliaries can allow additional space for other components. For example the size of the radiator will have an effect on the aerodynamic design of the vehicle because the radiator tends to determine the height of the bonnet’s leading edge.

4. **Cost:** It is important to note that the cooling system is a significant cost component of the engine system. Parts such as piping, radiators and pumps are expensive items. A decrease in their size would lead to a price reduction.
1.3 The Future of Engine Cooling

Despite the technical benefits of smaller cooling systems, outlined in Section 1.2, present market forces are encouraging manufacturers to increase the specific power of engines and therefore the requirements for cooling will grow. Cooling system design must therefore be developed further to transfer larger amounts of heat from smaller surface areas in order to keep up with increases in specific power.

Despite French (1969) highlighting the need for more research into the design of engine cooling systems, work in this area is not developing as quickly as is needed. It is suggested that a significant increase in research is required in order to optimise the design of the cooling system, the engine and ultimately the vehicle. The research reported in this thesis seeks to contribute to this.

1.4 Thesis Overview

In Chapter 1 of this thesis, the reasons for cooling IC engines have been introduced and the potential gains associated with optimising the design of a cooling system considered. These benefits, combined with a trend for higher heat fluxes, due to the increased engine specific power requirements, highlights the need for research work in this area.

Chapter 2 introduces nucleate boiling heat transfer as a potential method for significantly improving engine cooling systems and discusses previous work that has applied this idea to the cooling of IC engines. A number of approaches by different authors studying engine cooling systems and nucleate boiling heat transfer are reviewed. The objectives of this research are then detailed, highlighting the key areas of the cooling system on which this thesis is focused.

The first section of Chapter 3 investigates the fundamentals of boiling heat transfer highlighting the complexity of the topic and identifying relevant
knowledge that can be applied to engine cooling. The controlling factors affecting nucleate boiling heat transfer are identified and explained. In the second half of the chapter, boiling surface cavities are investigated. After firstly investigating naturally occurring surface cavities, this work then discusses the use of deliberately manufactured cavities of different geometries. The effect of these surface cavities on parameters, including the heat transfer coefficient and boiling initiation temperatures, is discussed. A concept is then proposed for using cavity grids in the key areas of an engine cooling system to control and enhance nucleate boiling.

Chapter 4 focuses on surface cavity manufacture, reviewing methods that past investigators have used to produce both single cavities and grids of cavities. Laser machining is chosen as the most appropriate and efficient manufacturing method. Various laser techniques are investigated, and a new methodology for the manufacture of boiling grids for use in engine cooling systems is proposed.

In Chapter 5 experimental apparatus designed to enable the laser manufactured boiling grids to be tested is described. A description is given of the optical access to the boiling chamber allowing analysis of the boiling grid performance using a laser backlight video system with digital image analysis software. The system calibration and validation tests are documented.

Chapter 6 reviews experiments that prove the effectiveness of laser machined cavities as nucleation sites. Further simple boiling experiments are described that result in optimal boiling cavity dimensions that were then evaluated and described in later chapters.

In Chapter 7 a test methodology was developed and used to assess a wide range of laser manufactured boiling grids. The boiling grids are found to
promote efficient and controlled boiling, and heat transfer coefficients were found to increase significantly when compared to pure convection. Further results of this experimental work are analysed and comparisons drawn with work by other authors. Discussions show how the novel laser manufacture boiling grids 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 controlling the nucleate boiling process.

Chapter 8 details the major conclusions of this thesis and outlines potential areas of future work.

1.5 Contributions from the Work Presented in this Thesis

The work detailed in this thesis has made four novel contributions towards the design, manufacture and analysis of laser manufactured boiling grids and their use in the cooling of an IC engine. These are:

1. A novel concept for using boiling grids to enhance cylinder head cooling in IC engines has been developed. Studies of fundamental nucleate boiling investigations were discussed and the effect of surface finish and surface cavities on nucleate boiling was found to be significant. By manufacturing a predetermined surface finish, heat transfer rates have been enhanced.

2. A fast and accurate method of manufacturing boiling grids has been established. Using laser techniques, grids up to 1000 mm$^2$ can be manufactured rapidly, typically within 10 seconds. Deep cylindrical cavities between 25 μm and 300 μm in diameter were machined.

3. A boiling chamber and optical analysis system were developed for the testing and validation of boiling grids. An automated system of digital
video and image analysis allowed statistical results to be gained for variables including bubble departure size and frequency.

4. Experimental results have proved that cavity grids enhance the heat transfer coefficient and control the nucleate boiling process.

1.5.1 Publications from this Research

One refereed paper and two patent applications have been produced from this research:


1.6 Concluding Remarks

This chapter has discussed the motivation of this research work and has given an overview of the thesis. Chapter 2 introduces heat transfer fundamentals and documents the link between nucleate boiling and engine cooling systems. The increased heat transfer coefficient associated with nucleate boiling is then identified as potentially beneficial for engine cooling.
Chapter 1 explained the importance of cooling IC engines, discussing that both the durability and the efficiency of the engine are affected by the design of the cooling system. Significantly, Chapter 1 also suggested that an engine's cooling requirements are likely to increase in the future due to a market drive for higher engine specific powers.

This chapter will focus on basic heat transfer theory and ways that this knowledge can be applied to aid engine cooling system development. Nucleate boiling is highlighted as an area for further study because it has a significantly higher heat transfer coefficient than convection alone. Some previous research in the area of nucleate boiling in engine cooling systems is also discussed in this chapter.
2.1 **Heat Transfer Theory**

If a temperature difference exists between a solid and the liquid it is in contact with, then heat energy transfers between them. The process is known as heat transfer. The simplest form is convective heat transfer, when the energy is transported without generating any phase change in the liquid. It can be described by the following equation:

\[
q_{\text{conv}} = h_{\text{conv}} \Delta T
\]

Equation 2-1

where \( q_{\text{conv}} \) is the convective heat flux, \( h_{\text{conv}} \) the heat transfer coefficient and \( \Delta T \) the temperature difference between the solid and the liquid. Equation 2-1 was formulated by Newton in 1701 to describe convection.

Convective heat transfer can be described as natural convection or forced convection, depending on whether the liquid is being deliberately moved over the heated surface by an external force or not. In a forced convection system, a pump, for example, could be used to circulate the liquid. In natural convection, the liquid movement is caused only by thermal influences.

The temperature difference between a solid and the liquid's saturation temperature (boiling point) is known as the *superheat* (or excess temperature) and is used to characterise the heat transfer regime. If the temperature of the liquid is as high as the saturation temperature then the heat transfer regime is known as *saturated*, whereas below this point it is known as *subcooled*.

IC engine cooling systems are normally designed to operate in a subcooled, forced convection regime. However, the heat transfer regime is difficult to control and tends to vary between different regions of the cooling system. In areas of flow stagnation, natural convection may occur and even nucleate boiling has been observed in areas of high heat flux. It has been known for some time that many engine cooling systems have local areas in which
boiling occurs inadvertently under certain engine load conditions. This will be discussed in more detail later in this chapter.

The energy transfer from a surface to a liquid is characterised by Figure 2-1. It shows the variation in solid wall super heat with heat flux. The graph shows positive values of superheat, meaning that the surface temperature is above the boiling point of the liquid.

![Figure 2-1: Heat flux curve for water - Bejan (1993)](image)

When the heat flux, \( q \) is low (i.e. \( q < 10^4 \) W.m\(^{-2}\)) the heat transfer from the surface to the liquid is purely convective, without any phase change. If heat flux is increased to \( 10^4 < q < 10^5 \) W.m\(^{-2}\) then the excess temperature (wall superheat) increases such that boiling occurs. Boiling is marked by the appearance of vapour bubbles in the liquid. Bubbles grow at nucleation sites on the heated surface until their buoyancy allows them to leave the surface and move into the bulk of the liquid. The number of nucleation sites increase
and boiling becomes more vigorous as the heat flux and superheat increase further.

Eventually when \( q > 10^6 \text{ W.m}^{-2} \) many nucleation sites become active and departing bubbles prevent the liquid from making contact with the surface. This is called critical heat flux and at this point, if the heat flux is maintained, the wall temperature will rise rapidly, causing film boiling. This is a heat transfer regime that can cause material failure, often termed 'burnout', of the heated surface as the layer of vapour insulates the surface, causing the surface temperature to rise further.

It was originally thought that increased heat transfer during nucleate boiling was due to the latent heat of vaporisation as bubbles were generated. However, it was later found to be caused by increased local fluid velocities close to the solid surface due to the turbulence of growing and departing bubbles. As little as 15% of the total heat flux can be attributed to the latent heat of vaporisation. The intense agitation of the liquid layer close to the heated surface is the key to excellent boiling heat transfer characteristics because the motion of the bubbles moves hot liquid away from the surface and cooler liquid towards it, ensuring a high temperature gradient and therefore high heat transfer. Factors affecting nucleate boiling heat transfer are discussed in detail in Chapter 3.

The onset of nucleate boiling occurs when the surface temperature reaches a specific point, normally 3 °C to 5 °C above saturation temperature for water. When nucleation does begin, the heat transfer coefficient increases significantly and the surface dissipates 10 times the energy with only a 4 °C temperature increase over convection.

The high heat transfer observed with nucleate boiling is effectively transferring a larger amount of energy from a smaller surface. Therefore, it
could be used in IC engine cooling ducts to transfer heat from the surface to the coolant in a more efficient manner.

2.2 Cooling Concept of Present Study

Work in this research study will consider the high heat transfer associated with a nucleate boiling regime and its application to areas of the cooling circuit which have the highest heat fluxes. Such as the engine cylinder head, where some areas have heat fluxes in excess of 1 MW.m⁻². These high heat flux areas include:

- **Valve bridge**: area in the cylinder head between inlet or exhaust ports.
- **Flame face**: immediately adjacent to the top surface of the cylinder.
- **Fuel Injectors**: normally housed near the valve bridge.

Engine designers normally seek to avoid nucleate boiling in engines (for example, by increasing the cooling pump speed) because of the excessive surface temperatures associated with film boiling, and the possibility of thermal failures such as 'burnout'. However, it is accepted that during certain engine operating conditions, some critical points of virtually all engine cooling systems will boil on some occasions. Designers allow this because it happens infrequently and in only a few areas of the cooling system.

The key to successfully implementing boiling in IC engine cooling systems is its effective control. Boiling must remain in the nucleate boiling regime so that the excessive surface temperatures associated with film boiling are avoided.
As nucleate boiling occurs in some cooling systems, there has been research into this area, as well as studies that deliberately applied nucleate boiling to IC engines. These are now discussed.

### 2.3 Boiling in IC Engine Cooling Systems

The tendency to avoid boiling is starting to change, with some engine designers realising the benefits it can offer. Some manufacturers are starting to use it as a selling point for their engines. BMW were the first manufacturer to widely publicise this in December 2002, by publishing research by Petutschnig et al. (2002), which specifically emphasised that they were exploiting nucleate boiling. However, without the ability to control boiling, there is still the risk of reaching the critical heat flux and film boiling occurring.

A number of studies have attempted to predict when and where boiling may occur in engine cooling systems. The main research papers specifically aimed at applying nucleate boiling to IC engines are listed in Table 2-1.

<table>
<thead>
<tr>
<th>Author</th>
<th>Year</th>
<th>Research</th>
</tr>
</thead>
<tbody>
<tr>
<td>Leshner</td>
<td>1983</td>
<td>Potential benefits of evaporative cooling</td>
</tr>
<tr>
<td>Watanabe et al</td>
<td>1987</td>
<td>Evaporative cooling of a CI engine</td>
</tr>
<tr>
<td>Kubazoka et al</td>
<td>1987</td>
<td>Evaporative cooling of a SI engine</td>
</tr>
<tr>
<td>Norris et al</td>
<td>1989, 1994</td>
<td>Boiling study in a convective cooling system</td>
</tr>
<tr>
<td>Ap et al</td>
<td>1999</td>
<td>Evaporative cooling of a SI and CI engine</td>
</tr>
<tr>
<td>Amelio et al</td>
<td>2001</td>
<td>Evaporative cooling of a SI engine</td>
</tr>
<tr>
<td>Lee and Cholewczynski</td>
<td>2003</td>
<td>Boiling study in a convective cooling system</td>
</tr>
</tbody>
</table>

Table 2-1: Survey of some key IC engine cooling investigations
Lee and Cholewczynski (2003) completed a thermocouple study of a standard passenger vehicle engine and Norris et al (1989 and 1994) used both temperature measurements and numerical studies to predict nucleate boiling in a diesel engine. Their key findings are now discussed.

2.3.1 Norris et al (1994)

Norris et al (1994) investigated heat transfer regimes found in an engine cooling system. Using a Cummins L10 heavy duty diesel engine, they carried out a detailed numerical analysis of the cylinder head. Some experimental work was needed in order to set the boundary conditions for the numerical model. They operated the cooling system at atmospheric pressure and with 100% water, which meant that it was difficult to apply the results to a practical production engine that is normally expected to be operated with a water/ethylene glycol coolant mixture and at elevated pressures. However, the study did create a methodology for cooling system thermal analysis.

Using a flow rig to measure the fluid velocity at key areas in the cylinder head, the results were used as boundary conditions in a meshed numerical model. Combining this velocity map with surface temperature readings from a real engine, the numerical simulation was able to calculate the effective heat transfer coefficient in different areas of the valve bridge.

Figure 2-2 shows the flow characteristics of cylinder 1 with the analysis zones marked as follows:
Zone A - An area of flow separation from the wall
Zone B - Low velocity flow region between cylinders (downstream)
Zone C - Low velocity flow region between cylinders (upstream)
Zone D - Stagnant flow area at injector bore

Figure 2-2: Schematic of the flow in cylinder head (cylinder no. 1) - Norris et al (1994)

Table 2-2 shows some of the outputs from the numerical model, compared with the expected heat transfer coefficient if the regime in that zone had been purely convective heat transfer. Where the heat transfer coefficient was greater than an order of magnitude higher than convection, the dominant mechanism was assumed to be nucleate boiling.
Table 2-2: Heat transfer coefficients in different regions of the cylinder head, see Figure 2-2 (W.m$^{-2}$.K$^{-1}$) - Norris et al (1994)

The values were not expected to be the same in each cylinder because the valves were in an alternating pattern. However, zones B and C, on cylinder 2 and 3 were found between an inlet and an exhaust valve therefore values should have been similar. As seen in Table 2-2 the values were almost identical. Finally, it should be noted that the values for cylinder 1 were high in comparison to cylinder 2 and 3. This is because the end cylinders (1 and 4) were known to run much cooler than the centre pair of cylinders because they were on the edge of the engine block.

Heat transfer coefficient values in Zone A were all significantly higher than was usual for pure convection, suggesting that the dominant mechanism is nucleate boiling. Zones B and C both had low values in comparison with the convective coefficient, possibly because this area was in a film boiling regime. Zone D had high values of heat transfer coefficient, similar to those in Zone A and was therefore assumed to be in a nucleate boiling regime.

Norris et al (1994) concluded that boiling was present in the valve bridge area of the cylinder head and that the boiling was causing local heat transfer coefficients of 30 to 40 times that of a convective regime.
2.3.2 Lee and Cholewczynski (2003)

Work conducted by Lee and Cholewczynski (2003) sought to find evidence of nucleate boiling in standard engine cooling systems. Ten thermocouples were inserted into the cylinder head of a 4 cylinder, 1.6 litres, 16 valve, spark ignition engine. In a similar manner to Norris et al (1994) they focused their investigation on the valve bridge area of the cylinder head, as it is one of the most thermally critical areas. One significant difference between this work and the previous study were the operating conditions. Unlike Norris et al (1994), Lee and Cholewczynski (2003) chose a realistic engine coolant consisting of a mixture 50% water and 50% ethylene glycol, but with a pressure of 0.985 bar gauge. The rationale behind this study was that by measuring the temperature at a steady operating condition and then reducing the system pressure whilst monitoring the surface temperature it was possible to identify areas that were originally in a nucleate boiling regime because of the observed temperature increase. Some predictive computational fluid dynamics (CFD) work was also carried out, although there were significant differences between the measured and the predicted values.

As described in the previous sections, Norris et al (1989 and 1994) and then Lee and Cholewczynski (2003) attempted to observe nucleate boiling in a standard engine cooling system. Their work highlighted the improved heat transfer associated with nucleate boiling. The next step was to design an evaporative cooling system, deliberately enhancing the heat transfer process by allowing a nucleate boiling heat transfer regime throughout the cooling system.

2.4 Evaporative Cooling System

The first nucleate boiling IC engine cooling system (evaporative system) was reported to be designed by Harrison (1926), although no significant studies developed this work further until Leshner (1983). Then, work by the Nissan

Kubozuka et al (1987) studied the effect of nucleate boiling heat transfer in different areas of a petrol SI engine (i.e. the cylinder walls and the valve bridge) before building and testing a complete evaporative cooling system, shown in Figure 2-3. Research by Watanabe et al (1987) used a passenger car light duty diesel engine and provided similar results.

Nucleate boiling was seen to control the surface temperature even after the engine had been shut down. When the engine is shut down and the cooling water pump therefore stops, the coolant temperature rises. However, in the evaporative system, even after the engine is shut down, boiling and evaporation continue and so the coolant temperature remains low, as shown in Figure 2-4.
They found that the evaporative cooling system provided acceptable heat transfer for a smaller coolant flow as well as a more uniform temperature distribution across the entire engine. A secondary benefit of continued cooling at hot shutdown was also seen.

Although several investigators have continued research in this area, including Ap et al (1999), applying this research outside of the laboratory has proven difficult. A net vapour generation cooling circuit, when mounted in a moving vehicle, caused problems such as vapour block in coolant ducts and surface dry out in key areas for prolonged periods (i.e. inclines of up to 35° need to be catered for in construction machinery and military applications). The acceleration forces experienced in everyday motoring would also cause similar problems. For these reasons, subcooled boiling, where there is no net vapour generation, would be better in commercial applications. Furthermore, problems of operating with a two-phase coolant (for example, pump cavitation) are also reduced because bubbles would condense back into the subcooled liquid.
2.5 **Concluding Remarks**

This chapter has demonstrated that nucleate boiling has the potential to increase the heat transfer coefficient significantly. However, the control of nucleate boiling is the key to its use in engine cooling systems and has so far precluded its widespread use. This present research develops methods for controlling nucleate boiling by using specially manufactured enhanced surface features.

Chapter 3 considers the fundamentals of nucleate boiling and the factors affecting nucleation.
3 Nucleate Boiling Fundamentals

This chapter investigates the fundamentals of nucleate boiling heat transfer. Focus is given to boiling parameters that control nucleation, while reducing the risk of film boiling occurring. Specifically, the effect of surface cavities is investigated, and a concept is developed for using cavity grids in the key areas of an engine cooling system to control and enhance nucleate boiling.

Firstly nucleate boiling heat transfer is discussed and surface cavities are identified as nucleation sites. Then work to categorise cavities, measure bubble growth and assess site interaction on boiling surfaces is presented, before finally, research on manufactured cavities of a predetermined size is discussed.
3.1 **Nucleate Boiling Heat Transfer**

Many correlations have been suggested that aim to predict nucleate boiling heat flux. They have mainly focused on either the nucleate boiling heat transfer coefficient or the prediction of critical heat flux.

Studies of the nucleate boiling regime have shown how the heat transfer coefficient varies with wall superheat throughout the regime. Studies of critical heat flux have been concerned with predicting the point at which the heat transfer regime moves from steady boiling towards film boiling. The critical heat flux is of interest to cooling system designers, as they wish to suppress it. However, predicting the heat transfer coefficient within the cooling channels of the cylinder head is more important, because this would be essential if nucleate boiling were to be exploited.

### 3.1.1 Chen Correlation

The Chen correlation has been considered the most important correlation since its conception in 1966, and is widely used in engine cooling system design. The empirical expression developed by Chen (1966) on a heated, up flowing, small circular tube is as follows:

\[
\begin{align*}
    h_{\text{boil}} & = 0.00122 \left( \frac{\mu_l^{0.49} \rho_l^{0.49}}{\sigma^{0.5} \mu_l^{0.29} \mu_k^{0.29} \rho_v^{0.24}} \right) (T_{\text{wall}} - T_{\text{sat}})^{0.24} \rho_v^{0.75} \text{sat}_\text{chamber} \quad \text{Equation 3-1}
\end{align*}
\]

where \( h_{\text{boil}} \) is the boiling heat transfer coefficient, calculated from \( k_l \) the conductivity of the liquid, \( C_p \) the specific heat of the liquid, \( \rho_l \) the density of the liquid, \( \sigma \) the surface tension, \( \mu_l \) the viscosity, \( h_f \) the enthalpy of evaporation, \( \rho_v \) the density of the vapour, the wall and saturation temperature, \( T_{\text{wall}} \) and \( T_{\text{sat}} \) respectively and \( \text{sat}_\text{chamber} \) the saturation pressure.
Not only were the power values in Equation 3-1 developed empirically, but importantly so was the suppression coefficient, $S_{\text{chen}}$ (described as $S$ in Figure 3-1). On his specific test rig Chen (1966) found the value of $S_{\text{chen}}$ to vary greatly as illustrated in Figure 3-1.

![Figure 3-1: Suppression factor for Chen's correlation – Chen (1966)](image)

This figure shows the graph from which suppression factors are selected. The range on the values is 0.1 - 0.8, and the error highlighted by the etched area for any particular flow conditions is large. This causes significant limitations when trying to use this as a predictive tool.

The use of this correlation for the design and prediction of an IC engine cooling duct was reviewed by Stone (1999). He correctly suggested, "The relevant geometry is not widely encountered in engines" and that "the complex geometry of the flow passage defeats correlations for heat transfer coefficient, which in general assumes some form of fully developed flow".

The Chen Correlation is inappropriate for the prediction of heat transfer in cooling ducts because of irrelevant geometry and the assumption of fully developed flow. Stone (1999) went on to discuss that the correlation would
illustrate the increase in heat transfer coefficient associated with nucleate boiling, but this was probably the limit of its use in this area.

3.1.2 Rohsenow Correlation

The Rohsenow Correlation, Rohsenow (1952) (similar to the Chen Correlation) predicts the proportion of energy that is transferred to the liquid from a hot surface due to boiling. Rohsenow's work concentrates on the effect of surface finish on the heat flux. The heat flux is described in the following expression;

\[
q_{\text{boil}} = \frac{\mu_l h_{fg}}{D_b} \left[ \frac{1}{C_{af}} \left( \frac{C_p (T_{\text{wall}} - T_{\text{sat}})}{h_{fg}} \right) \frac{1}{Pr^{1.7}} \right]
\]

Equation 3-2

where \( q_{\text{boil}} \) is the boiling heat flux, the viscosity \( \mu_l \), the enthalpy of evaporation \( h_{fg} \), the bubble departure diameter \( D_b \), the specific heat \( C_p \), the wall and saturation temperature, \( T_{\text{wall}} \) and \( T_{\text{sat}} \) respectively, \( Pr \) the Prantle Number and \( C_{af} \) Rohsenow's interface coefficient.

The surface finish is an input into Equation 3-2 as the variable \( C_{af} \) which Rohsenow describes as the interface coefficient. This was found empirically by varying surface and liquid combinations within the boiling apparatus and it can vary greatly. For example, a smooth copper surface and water would have a value of approximately 0.013, however, if the copper surface is scored, \( C_{af} \) becomes 0.007.

This correlation gives a reasonable prediction of heat transfer from a heated surface due to nucleate boiling and highlights the importance of the surface finish to nucleate boiling.
3.2 Surface Effects on Nucleate Boiling.

A photographic investigation of the boiling of Pentane and Freon 113, by Corty and Foust (1955) showed that there were marked variations in the heat transfer coefficient and the onset of nucleation with changes of surface roughness. Figure 3-2 shows variations of superheat for roughened and polished surfaces at 65 kW.m⁻² heat flux at atmospheric pressure.

![Figure 3-2: Onset of nucleate boiling - Corty and Foust (1955)](image)

Corty and Foust used a profilemeter to measure the root mean square (rms) profile of the surface, giving a measure of surface roughness. They measured values of 0.05 to 0.635 μm rms. This method of roughness classification gives a good average value of peaks and troughs on the surface. However, it has a limited ability to map out specific cavities especially if they are deeper than they are wide. Micrograph photography of the surface using yardstick balls (small spheres of a precise known diameter, scattered on the surface to give scale on microscopic photographs) showed that the values of rms were
equivalent to surface scratches and pits of approximately 0.25 to 25.4 μm wide. Later research showed that the important dimension in the nucleation site is the scratch diameter, proving that rms measurement should only be used as a comparator.

Figure 3-3: Heat transfer coefficient - Corty and Foust (1955)

The Corty and Foust findings centred around the results in Figure 3-2 and Figure 3-3 which highlight the effect of surface finish. A highly polished surface will have smaller cavities than that of a rough surface. A smaller cavity will only begin to nucleate at a higher superheat (ΔT). Figure 3-2
shows the required superheat for nucleation falling as the surface roughness increases.

Corty and Foust also noted variations in their results with time and this was put down to the ageing of the surface. During one 12-hour run, they were forced to vary the superheat by 1.7°C to hold the heat transfer coefficient constant. They postulated that there was cavity shrinkage due to oxidation of the copper surface, however, they were unable to positively conclude this.

A second observation was of bubble fronts growing across the surface (i.e. nucleation beginning on one side of the surface and spreading over the surface like wave). They attributed this to one of two effects, pressure fluctuations near to a bubble or site interseeding, when a bubble grows large enough to cover a neighbouring cavity before it departs. On departure the bubble leaves a small amount of vapour in both cavities, activating them as nucleation sites. A more likely reason for bubble fronts is the effect of uneven heating of the surface, possibly due to varying wall thickness, or varying levels of polishing on the heated surface. It was noted however, that they had proposed that interseeding was possible, but this was not conclusively proven until work by and Calka and Judd (1985).

This chapter so far, has discussed observations by Corty and Foust (1955), of nucleate boiling from surfaces of varying roughness. Recent work by Campbell et al (1997) took the next step by studying nucleation from surfaces of roughness equivalent to that found in an IC engine cooling duct. Further research by Campbell et al (1999), investigated nucleate boiling using simulated engine cooling passages.

Work by Campbell et al has studied uncontrolled nucleate boiling from various surfaces found in IC engine cooling systems. The research presented in this thesis however, aims to develop novel surface features that can be
used to control and enhance nucleate boiling in engine cooling passages. Therefore, as Corty and Foust (1955) highlighted, the surface cavities from which nucleation occurs must be considered.

3.3 Surface Cavities

A cavity containing trapped gas is more likely to become a nucleation site when heat is applied. Bankoff (1958) discussed the inability of a surface cavity to be completely filled with liquid. Using a graphical method, he predicted the width of a rectangular groove that would always entrap gas. Using the contact angle of the liquid on the solid, he then determined how deep the rectangular groove should be.

Griffith and Wallis (1960) were able to develop this research further by applying thermostatic equilibrium to a bubble as it grew in a nucleation site. The Gibbs Equation shows that at the critical radius the sum of the forces should be in equilibrium:

\[ P_l - P_v = \frac{2\sigma}{r_b} \]

where \( P_l \) and \( P_v \) and the liquid and vapour pressure respectively, \( \sigma \) is surface tension and \( r_b \) is the bubble radius. The Clausius-Clapeyron Relation describes the relationship between the temperature of a liquid and its vapour pressure, as determined by the molar enthalpy of vaporisation of the liquid. When applied to Equation 3-3 an expression was derived for the minimum cavity radius at which a bubble would start to grow:

\[ r_{c,min} = \frac{2\sigma T_{sat} V_{fg}}{h_f \Delta T} \]

where \( r_{c,min} \) is the minimum active cavity radius, \( \sigma \) is the surface tension, \( T_{sat} \) the saturation temperature, \( V_{fg} \) and \( h_f \) are the specific volume change during

30
evaporation and the enthalpy of evaporation respectively, and $\Delta T$ is the temperature difference.

Griffiths and Wallis conducted experiments which involved boiling water, methanol and ethanol from a flat plate. Using a gramophone needle to create a conical cavity of $D_c = 68 \mu m$ they produced a boiling grid in the shape of equilateral triangles, meaning all cavities were equally spaced. They assessed the boiling regime from these cavities using photographic techniques and noted that the nucleation characteristics of a cavity were set by a single dimension, namely the cavity radius.

Griffiths and Wallis also investigated re-entrant cavities. It was postulated that the large volume of the cavity would produce the most stable nucleation centres. This is due to the low initiating superheat and the predetermined activation of superheat as defined by their mouth geometries. They concluded that the liquid must actually become subcooled in order to deactivate the cavity.

Kurihara and Myers (1960) proved that surface roughness affects the heat transfer coefficient of a liquid. They began by measuring how the number of active boiling sites affected the heat transfer coefficient in a number of different liquids including water, acetone, hexane and chlorides. The heater surface was a flat copper plate submerged in the liquid. Figure 3-4 shows the results for water. Label 'I' refers to the smoothest heater surface and 'VII' to the roughest surface.
Chapter 3 - Nucleate Boiling Fundamentals

Figure 3-4: Variations of HTC with superheat for a range of surface roughnesses - Kurihara and Myers (1960)

The results show the effect of the surface finish on the heat transfer coefficient. The first notable trend is that the smoothest surface initiates nucleate boiling at the highest superheat. The roughest surface is the first to initiate boiling (at 9°C superheat) where as the smoothest surface initiates nucleation at 15°C superheat. The cavity activation temperature is directly related to the cavity size, therefore, the variation in boiling initiation temperature is due to the range of cavity sizes there are on the heater surface. Each surface has the same base material that is then polished to different levels. During polishing the larger cavities are removed leaving only smaller
ones. The more the surface is polished the smaller the cavities that remain. The size of cavity and its effect on the initiation temperature was shown in Equation 3-4.

Each test result curve plotted in Figure 3-4 has a different gradient. The curve for the roughest surface is much steeper than that for the smoothest surface. This is because the further the surface is polished, the smaller the range of cavity sizes becomes. This was summarised as a Poisson distribution of cavity sizes shown by Sadasivan et al (1995) and Wang and Dhir (1993) who confirmed the earlier work suggested by Gaertner (1963). On the roughest surface, once nucleation has been initiated and the superheat is raised, smaller cavities become active. This compares with the smooth surface which has a diminished range of cavity sizes. Therefore, as the superheat is increased above the onset of boiling, there are fewer smaller cavities to activate and so the temperature must be raised further to cause the same increases in active sites as for the rough surface.

Hsu (1962) challenged the original work by Griffiths and Wallis (1960) and was able to advance their work further. Hsu (1962) developed a limiting case equation for the radius of active cavities:

\[
r_c^* = \frac{\delta}{2C_1} \left[ (1 - \xi_{\text{stat}}) \pm \sqrt{(1 - \xi_{\text{stat}}) - \frac{4AC_3}{\delta \theta_{\text{wall}}}} \right]
\]

Equation 3-5

where \( r_c^* \) is the critical cavity radius, \( \delta \) is the boundary layer thickness, \( C_1 \) and \( C_3 \) are constants, \( \xi_{\text{stat}} \) is the ratio of superheats, \( A \) is the surface area, and \( \theta_{\text{wall}} \) is the wall superheat.

The maximum and minimum cavity radii were expressed as:

\[
r_{c,\text{max}} = \frac{\delta}{2C_1} \left[ (1 - \frac{\theta_{\text{stat}}}{\theta_{\text{wall}}}) + \sqrt{\left(1 - \frac{\theta_{\text{stat}}}{\theta_{\text{wall}}}\right) - \frac{4AC_3}{\delta \theta_{\text{wall}}}} \right]
\]

Equation 3-6
Starting with Equation 3-5, Hsu was able to include the effect of the thermal boundary layer. This is the region of liquid close to the heated surface, which is at a higher temperature than that of the bulk of the liquid and is represented by the value of $\delta$, the boundary layer thickness. This is used in Equation 3-6 and Equation 3-7. $\theta$ is the superheat value while $\theta_{sat}$ and $\theta_{wall}$ are saturation and wall superheats. $C_1$ and $C_2$ are constants and $\xi$ is equivalent to $\theta/\theta_{wall}$.

$$r_{c,\text{max}} = \frac{\delta}{2C_1}\left(1 - \frac{\theta_{sat}}{\theta_{wall}}\right) - \sqrt{\left(1 - \frac{\theta_{sat}}{\theta_{wall}}\right)^2 - \frac{4AC_3}{\delta\theta_{wall}}}$$

Equation 3-7

Figure 3-5: Active cavity range – Hsu (1962)

Hsu was able to show graphically the range of active nucleation sites on a heated surface, as illustrated in Figure 3-5. For example, reading from Figure 3-5, at a surface temperature of $15^\circ F$, active cavities radius will be between 150 - 1350 $\mu$m (i.e. the inside of horseshoe shaped curve). From Figure 3-5 it can also be seen that boiling would initiate at $5^\circ F$, which is the left hand edge.
of the curve. Hsu (1962) was the first to relate the incipience of boiling and the size of active nucleation sites to the thickness of the boundary layer.

Griffith and Wallis (1960), Kurihara and Myers (1960), Benjamin and Westwater (1961) and Hsu (1962) all investigated the relationship between the heat transfer characteristics and surface geometries. Having developed equations relating the active cavity size to thermodynamic conditions, Singh et al (1977) were the first investigators to manufacture artificial cavities of predetermined sizes. This was achieved by using laser machined cylindrical cavities. Their results were then compared to data from Tolubinsky and Ostrovsky (1966).

Singh et al (1977) were able to generate cavities of $5.4 \pm 1 \mu m$. They decided that in order to ensure that the holes could be regarded as deep, and hence ensuring that there would be some gas trapped in the cavities, the $L_e/D_c$ ratio must be greater than unity. Therefore, all the holes had $L_e/D_c$ ratios in the range of 2 to 4.

Singh et al (1977) concluded by saying that the frequency observed on an artificial laser machined cavity is much smaller than that measured by previous investigations. This is almost certainly due to the fact that a natural cavity will never be exactly cylindrical. They are never precisely round and have jagged edges, which will affect how early the bubble releases from the surface. Singh et al (1977) concluded that their sharp edged, perfectly round cavities would cause bubble growth to be more stable than that on the rough natural cavities and would cause the departure frequency to decrease.

It has been well documented that the active site criterion is based upon whether a cavity will capture gas when the heater surface is submerged. Previous work on this had been always based on conical or cylindrical cavities. Wang and Dhir (1993) proved this to be unwise as the majority of
cavities on heated surfaces are not this shape. Wang and Dhir therefore re-evaluated trapping criteria for cavities of different shapes. Work focused particularly on re-entrant type cavities. Without doing any experimentation of their own they created a mathematical model for the prediction of active cavities.

Later they tried to predict boiling characteristics of a true surface and postulated that the majority of surface cavities were based around a spherical shape. Figure 3-6 confirms their prediction that the contact angle must be greater than the mouth angle to ensure trapping of gas.

![Figure 3-6: Effect of contact angle on active cavities – Wang and Dhir (1993)](image-url)
By combining the entrapment criterion with a knowledge of the cavity diameter distribution and cavity mouth angles, the active site density was predicted quite well. When compared to Wang and Dhir’s earlier experimental results it showed some reasonable correlation as seen in Figure 3-7. However, they assumed that all cavities were some fraction of a sphere and this may be the reason for divergence away from their model. These assumptions may have only been true for cast materials; refining processes such as cutting, turning, milling and polishing may have added further, non-spherical pits and scratches to the surface.
3.3.1 Cavity Geometry

Griffiths and Wallis (1960) were the first to discuss cavity geometry, suggesting that it is important in two ways. Firstly the mouth diameter determines the superheat needed to initiate boiling, and secondly, its shape determines its stability once boiling has begun. Work on simple re-entrant cavities concluded that similar boiling characteristics from similar sites suggest that the diameter of the cavity opening is the most important parameter, rather than the shape and volume of the re-entrant cavity.

Nishikawa and Fujita (1990) studied the stability of cavities and their ability to hold a stable vapour volume. They considered a conical cavity and postulated that the wall angle and contact angle were the most important factors. A large contact angle, causing a convex meniscus, would give the best stability. They then investigated the effect of a re-entrant cavity. They found that their entrapment criterion for re-entrant cavities would always be fulfilled when the apex angle, $\phi (\phi = 180^\circ - \psi)$ was equal to $180^\circ$ as illustrated in Figure 3-8.

Figure 3-8: Contact angles of vapour trapped in a cavity – Wang and Dhir (1993)
Wang and Dhir (1993) in their most recent work, considered the gas entrapment criteria of different cavities as a method of predicting whether there were potentially active sites. They originally focused their investigation on conical cavities. However, because real surface cavities are never perfectly conical, they investigated which natural cavities were dominant. For this reason they decided to investigate spherical type re-entrant cavities. They concluded that a cavity will trap gas/vapour if the contact angle is greater than the minimum side angle.

The work of Nishikawa and Fujita (1990) and Wang and Dhir (1993) both reached similar conclusions. They postulated that a re-entrant cavity will be more effective than a conical cavity. Furthermore, they suggest the optimum would be a cavity with a horizontal ceiling and a cylindrical entrance. Wang and Dhir (1993) reasoned that such a cavity would entrap gas more effectively when the heater surface was first flooded, meaning it is more likely to become an active site. Nishikawa and Fujita (1990) suggested that it is the cavity shape which retains vapour the most consistently.

3.4 Bubble Interaction

Early nucleate boiling work was concerned with the study of bubbles departing from a surface of a known profile. Models were based on the study of active site density, wall super heat at the onset of nucleation and the heat flux from the surface. Bankoff (1958) then realised that the active nucleation sites were surface cavities. More work was then undertaken to investigate the dynamics of a single bubble, including the preparation of single nucleation sites.

Linking work on individual nucleation sites to the postulation of models for an entire surface was reported by Eddington et al (1975). Their investigation into the size range of active cavities on a brass surface was carried out using both the gas diffusion method and the nucleate boiling method. The nucleate
boiling method means measuring the activation and de-activation temperature of cavities and then using the Equation 3-4 to ascertain the cavity size for each of the nucleation sites. During the gas diffusion method the pressure is varied instead of the temperature and the active cavity sizes can therefore be calculated. There were marked differences in the results between the two test methods. This was partly attributed to the contact angle. They believed that the best correlation between the gas diffusion method and nucleate boiling method was for sharper edged cavities which might be less sensitive to the precise value of contact angle.

Eddington et al (1975) postulated that the error in the results was due to thermal interference between sites when using the nucleate boiling method, which would not have occurred during the gas diffusion method. During more recent experimentation carried out by the same authors, Kenning (1992), it was noted that on a stainless steel surface at high heat fluxes, both thermal interference and interseeding of unstable sites were important.

Work by Judd and Lavdas (1980) noted interseeding and thermal interference of nucleation sites consistent with the work by Eddington et al (1975). They described interseeding as when a nucleus is formed at the site when the dry spot had receded. If one bubble grows far enough to cover a second cavity and therefore dry it out, then the second cavity will become active.

Thermal interference occurs when, although there is no physical contact between a bubble and a nearby cavity, the area of influence of the bubble extends over this cavity. This can been seen as deactivation of the nearby cavity due to the thinning of boundary layers or the cooling of an area around an active cavity. These effects were first noted by Eddington et al (1977) and later developed by Judd and Lavdas (1980).
Judd and Lavdas used high speed photography to observe the boiling characteristics of a subcooled glass flat plate. Using heat fluxes in the range $40 - 60 \text{ kW.m}^{-2}$ and $5.3 \degree \text{C}$ subcooled they collected data from 18 naturally formed cavities in their camera's field of view. They suggested that each bubble had an area of influence around it which would enable it to alter the boiling characteristics of another cavity within that area.

Recent work by Shoji and Takagi (2001) has photographically identified the possibility of interseeding. Figure 3-9 shows the first bubble departing from a laser machined cavity. Between $t = 1.54 \text{ ms}$ and $t = 13.89 \text{ ms}$ it can be seen that the bubble is in contact with the surface across a large area. If there was a second cavity within that area then this bubble would have dried out that cavity and it could have become active again.

![Figure 3-9: First and second bubble departure (times are in milliseconds) - Shoji and Takagi (2001)](image-url)
The growth period of the second bubble can be seen between \( t = 24.69 \text{ ms} \) and \( t = 47.83 \text{ ms} \). There is no such propagation of the bubble along the surface and that suggests that only the first bubble will grow along the surface.

### 3.4.1 Separation Distance

During the 1980s, work was focused more on the measurement of the site interaction and attempts were made to apply a numerical solution to the ideas postulated by Eddington et al. (1977) and Judd and Lavdas (1980). As discussed by Calka and Judd (1985), Chekanov (1977) was the first to use gamma distributions to show the effect of cavities on one another across a varying separation difference. His experimental work was based on the use of two pin heaters pressed against a thin foil sheet, causing nucleation to occur on the top surface at points corresponding to the position of the pin heaters. By moving the heaters he was able to examine the effect of separation distance on the nucleation sites. Despite the groundbreaking nature of this work, his results were of limited use because he did not create boiling from cavities, and he made no attempt to measure the surface profile around the active site. However, the method of formulation of his results can be seen again in later work.

The use of gamma distributions and shape factors seemed the ideal method to bring such results together, see Figure 3-10. This analysis method was not used again until Calka and Judd (1985) used it to assess the boiling characteristics of a true heater surface. By heating a glass boiling surface at 40 - 90 kW.m\(^2\) at reduced pressure they were able to generate gamma distributions and shape factors as shown in Figure 3-10 and Figure 3-11.
From their observations they concluded that in the $0.5 < S/D_b < 1$ range, sites were activated by the interseeding mechanism, which has a promoting characteristic. This agrees with the observations of Eddington et al (1977) and Judd and Lavdas (1980). It can also be inferred that a shape factor of $\nu > 1$ corresponds to the promotion of activity at surrounding sites. They postulated that where $\nu < 1$, this was an inhibiting activity and that surrounding cavities would become deactivated. When the sites are far
enough apart so that shape factor $v = 1$, it is believed that the sites have no influence on each other. The trend of shape factors was found to asymptotically tend towards 1 (shown in Figure 3-11 and Figure 3-12) which might imply that there will always be some influence of adjacent cavities on one another. It was therefore suggested that the nucleation sites had negligible effect on each other when $S/D_b > 3$, as $v \approx 1$.

![Figure 3-12: Comparison of shape parameter calculations - Calka and Judd (1985)](image)

**3.4.2 Gamma Distribution and Shape Parameter**

Calka and Judd (1985), despite finding similar trends to Chekanov (1977), interpreted their findings very differently. Chekanov (1977) had postulated the complete opposite to the more recent report by suggesting that where the shape parameter $v > 1$, this was inhibiting nucleation at the nearby site. However, the work of Eddington *et al* (1977) and Judd and Lavdas (1980), shows that at small separation distances, the active sites actually allow the adjacent sites to nucleate when otherwise they would not.

The other variations, which were seen in Figure 3-12, are that Chekanov (1977) predicted a much more elongated shape parameter curve. He attributed it to the fact that he did not fit his data with a gamma distribution but used the moments method to calculate the shape parameter. In addition,
a difference in fluid and heater surface would introduce differences into the results, especially at lower separations, as the effect of the contact angle becomes increasingly important.

Judd of McMaster University has been studying the interaction of nucleate boiling for over 10 years, publishing a number of important papers. The work carried out by Calka and Judd (1985) was eventually republished with more experimental data from the same test rig. Judd and Chopra (1993) confirmed the postulations of previous investigations by Calka and Judd (1985). Collecting more data than previous investigators, Judd and Chopra were able to analyse the effect of a dominant nucleation site on a number of adjacent sites. From these results they again created gamma distributions and associated shape parameter.

With more available data they were able to show that both the separation distance, \( S \), and the bubble departure diameter, \( D_b \), are instrumental in determining what sort of effect bubble formation at the dominant site would have upon bubble formation at the adjacent site, and that a dimensionless separation distance \( S/D_b \) was not the determining factor. This is shown by Figure 3-13.

![Figure 3-13: Effect of cavity size on shape parameter - Judd and Chopra (1993)](image)
They also suggested that because the shape parameter values varied with site density, even if the separation distance and bubble departure diameter were held constant, the site density (i.e. number of nucleation sites per unit area) must also affect site interaction. This seems logical if the dominant site was not solely affecting the adjacent site but was having some effect on every site on the surface.

Judd and Chopra (1993) also suggested other possible causes for the interaction of nucleation sites. Their work appears to show that diminished availability of energy makes adjacent sites more independent. However, they believed that the activation and deactivation of adjacent sites was more likely to control nucleation than the availability of energy was. They also commented that surface temperature irregularities did not cause interaction between sites, in their opinion. They believed the scale on which temperature would have an effect would only affect $S/D_b < 1$, and would not affect the surface temperature at distances great enough to have any effect on what happens at the majority of surrounding nucleation sites.

Judd further concluded that his results had again contradicted those of Chekanov (1977), and that he was confident that $S/D_b$ effects (given in Table 3-1) were correct.

<table>
<thead>
<tr>
<th>$S/D_b$</th>
<th>Effect</th>
</tr>
</thead>
<tbody>
<tr>
<td>$&lt; 1$</td>
<td>Promotes nucleation at adjacent sites</td>
</tr>
<tr>
<td>$1 &lt; S/D_b &lt; 3$</td>
<td>Inhibits nucleation at adjacent sites</td>
</tr>
<tr>
<td>$&gt; 3$</td>
<td>Independent nucleation</td>
</tr>
</tbody>
</table>

Table 3-1: $S/D_b$ effects concluded by Judd and Chopra (1993)

Following on from some of the comments by previous authors regarding non-linear aspects of heat transfer, such as surface temperature effects and hydrodynamic effects, Sadasivan et al (1995) investigated these concepts and how they affect heat transfer from a realistic heater surface. They focused on
the high heat flux region of the nucleate boiling curve and whether the on-off behaviour of nucleation sites could introduce significant non-linear effects into the boiling system.

3.4.3 Advanced Interaction Characteristics

Sadisvan et al (1995) discussed how a model developed from the work of Pasamehmetoglu et al (1993) considered cavity size, microlayer thickness and evaporation, stem diameter and deactivation temperature. One assumption included in their model was the relationship of activation superheat, $T_{\text{act}}$ as postulated by Hsu (1962):

$$\theta_{\text{act}} = \theta_{\text{sat}} + \frac{2\sigma T_{\text{sat}}}{\rho_v h_{\text{fg}} r_c}$$

where $\theta_{\text{act}}$ is the wall superheat at which nucleation is activated, $\theta_{\text{sat}}$ is the saturation superheat, $T_{\text{sat}}$ the saturation temperature (°R), $\sigma$ is the surface tension, $\rho_v$ is the vapour density and $h_{\text{fg}}$ is the enthalpy of evaporation.

Hsu (1962) was the first to suggest a relationship between neighbouring bubbles. He believed that growing bubbles vaporised some of the boundary layer, thereby temporarily reducing the thickness, making it more difficult for the neighbouring bubbles to grow. Hsu suggested that the bubble with the shortest waiting period would have the greatest chance of development.

An assumption similar to Kenning (1992) was also made, that the deactivation temperature was 5°C below the sites activation temperature. The bubble was assumed to grow instantaneously outside the cavity, connected to the heater surface by a stem. This stem was assumed to be 10 times the size of the cavity from which it originated.

Sadisvan et al (1995) developed a model which they validated by boiling water from a horizontal copper plate. Focusing on a single section of the
plate, they were able to identify all the potential nucleation cavities and map them out. When boiling began they measured the activation of each site and its intermittent behaviour.

They considered and measured temporal variations to the surface finding that it changed with heat flux. In some cases instantaneous variations of ±3°C were noted and only 35% of sites identified as potential activation sites, were active.

Many observations were made as to the interaction of local sites. For example, a site which has been active for a long period may then become deactivated, at which point an adjacent site will become active. They suggested that this was because each active site causes local cooling, which in turn prevents or delays the activation of an adjacent site. Similarly a site which has been active for a certain duration can cool sufficiently to deactivate itself. When this happens, one or more neighbouring sites which have previously been inactive can activate.

Figure 3-14: Digital bubble interaction - Sadisvan et al (1995)

Figure 3-14 shows the activation of two sites from the experimentation by Sadisvan et al (1995). The top site is active for a long period until it cools enough to deactivate itself at which point its neighbour, only 3 mm away,
immediately becomes active. That site then nucleates until its cools itself into deactivation and the first site can reactivate. This cycle can be seen to occur twice in the time period shown in Figure 3-14.

Further observations showed that the active site density (sites per unit area, \(N_a\)) varied with time when heat flux was constant. This raises the question as to what value of \(N_a\) earlier research papers had used. Furthermore, temperature readings for different areas of the heating surface were found to vary by as much as 12°C, which over a small test piece seems excessive. However, this may be due to a combination of experimental error and local variations in active site density or heater variation.

Sadisvan et al (1995) proved intermittent site interaction and variations in surface temperature and site density. This highlighted the non-linear aspects of nucleate boiling, some of which had previously been assumed to be linear. Before nucleate boiling can be fully understood, these variables need to be investigated further.

3.5 Enhanced Surfaces

In recent years process industry tube heat exchanger manufacturers have advocated that advanced surfaces be manufactured onto the outside of pipes so they can use them in the fabrication of tube bundles. Research examining different methods of assessing the effectiveness of commercially available 'high heat flux tubing' was commissioned in association with heat exchanger manufacturers, see Marto and Lepere (1982).

Nakayama et al (1980) studied both analytically and experimentally the effect of a porous surface on the outside of the pipe and the effect it has on its heat transfer ability. They focused on a surface that instead of having a large number of unique cavities, had a matrix of tunnels just under the surface.
These tunnels were then punctured at specific points to create inter-linked cavities. An idealised drawing of the arrangement can be seen in Figure 3-15.

![Figure 3-15: Advanced heat transfer surface - Nakayama et al (1980)](image)

They found that this type of surface significantly increased the heat flux but with different cavity activation to that which has been previously found. In some cases they found that an 80 or 90% reduction in wall superheat was required to transfer the same heat flux as that of plain surfaces. They also found a significant contribution to the total heat flux from the latent heat transported in the bubbles. This is not consistent with the concept of cavity nucleation, where the most significant effect is the turbulence caused by bubble departure and not the latent heat of vaporisation.

Nakayama et al (1980) suggested that the reduction in required superheat was an effect of the tunnel matrix. In order to investigate this they built a single tunnel with cavities punched into it observe the effect within the tunnel. They found that vaporisation in the tunnel was important, as the tunnel remained almost entirely full of vapour during boiling. There seemed
also to be some pumping effect, whereby as a bubble departed, liquid was drawn in to replace it through an inactive cavity.

Attempts by Nakayama et al (1980) to model the heat flux from a porous surface of this type were relatively successful but only for low heat fluxes. Above 20 kW.m\(^{-2}\) their correlation became significantly inaccurate. Their correlations included two empirical values which they had extracted from their experimental data.

An investigation by Marto and Lepere (1982) measured the heat transfer characteristics of three commercially available high heat flux pipes and compared them with a plain tube. They found that the high flux surfaces showed significantly better heat transfer. For example at 40 kW.m\(^{2}\) the heat transfer coefficient of the high flux surface was a factor of 10 higher than for a plain surface. Much of the data in this paper gave little insight into the reasons for this, although observations into overshoot on initiation of boiling were well documented and discussed in some detail.

Figure 3-16 highlights the effect of enhanced surfaces and their ability to affect the boiling curve, Yabe et al (1999). Shifting the graph left, illustrating a higher heat flux at a lower super heat, and shifting the graph up, causing the onset of film boiling at a higher heat flux.
3.6 Concluding Remarks

This chapter has shown that bubbles nucleate from surface cavities and that their dimensions affect the boiling regime significantly. Parameters including cavity depth, diameter and spacing all contribute to bubble growth, and consequently affect the boiling regime and heat transfer coefficient.

Previous research has highlighted the potential to control nucleate boiling with surface cavities of predetermined dimensions. Commercially available enhanced boiling surfaces have been discussed and showed a significant increase in heat transfer coefficient.

The next chapter of this thesis looks specifically at the manufacture of boiling grids and enhanced surfaces. Investigating different methods used by authors who created boiling cavities, a system is developed to manufacture boiling grids using an Nd:YAG laser system.
Griffiths and Wallis (1960) postulated that active nucleation sites are at surface cavities. Since this work, much research has been undertaken in this area, as discussed in Chapter 2. Previous experimentation into boiling cavities has been carried out on either fabricated cavities of a known size, or on natural cavities whose size can be measured.

In this chapter previous manufacturing methods for both single cavities and grids of cavities are reviewed. Then a new methodology for the manufacture of boiling grids for use in engine cooling systems is proposed, using laser drilling, which was found to be the most appropriate and efficient manufacturing method.
4.1 **Cavity Manufacture**

The most popular method of cavity manufacture has been by mechanical means, such as using punches and indenters. However, the use of such methods limits the range of geometries that can be manufactured. Table 4-1 lists previous investigations that have fabricated cavities for boiling studies, some of which have focused on single cavities and others on grids of cavities. Manufacturing methods have varied from mechanical means such as indenters and drills, to more sophisticated techniques such as electro-erosion drilling.

<table>
<thead>
<tr>
<th>Author</th>
<th>Year</th>
<th>Cavity Size (μm)</th>
<th>Manufacture</th>
</tr>
</thead>
<tbody>
<tr>
<td>Griffith and Wallis</td>
<td>1960</td>
<td>50</td>
<td>Indenter</td>
</tr>
<tr>
<td>Benjamin and Westwater</td>
<td>1961</td>
<td>100</td>
<td>Drilling</td>
</tr>
<tr>
<td>Chullabodh</td>
<td>1978</td>
<td>50 - 1000</td>
<td>Indenter</td>
</tr>
<tr>
<td>Singh et al</td>
<td>1977</td>
<td>8 - 15</td>
<td>Laser</td>
</tr>
<tr>
<td>Rammig and Weiss</td>
<td>1991</td>
<td>36 - 70</td>
<td>Electro-erosion</td>
</tr>
<tr>
<td>Bhavani et al</td>
<td>2000</td>
<td>40</td>
<td>Etching</td>
</tr>
<tr>
<td>Shoji and Takagi</td>
<td>2001</td>
<td>50 - 100</td>
<td>Electrical Discharge</td>
</tr>
</tbody>
</table>

Table 4-1: Cavity manufacture

Benjamin and Westwater (1961) created a nucleation site by mechanical means. Using a plug and hole design as shown in Figure 4-1, they were able to prescribe the cavity volume and the cavity mouth diameter. This allowed more flexibility, as well as the ability to test re-entrant cavity geometries. The disadvantage of this manufacturing method was that it is a time consuming and difficult procedure.
Chullabodh (1978) used a micro indenter to produce both conical and cylindrical cavities in the size range 50 µm - 1 mm. This was a faster method than Benjamin and Westwater's, however was less accurate and re-entrant cavities are not possible.

Recent work by Shoji and Takagi (2001) used a combination of electrical discharge machining and micro-hardness machining to make the cavities shown in Figure 4-2. The diamond bit from a hardness tester was used to create conical cavities whilst the discharge machining produced cylindrical cavities. Re-entrant cavities were manufactured using a combination of both methods. This method was found to be the best, with fast, accurate results and the possibility of re-entrant cavities. The only small disadvantage was that the heat generated during discharge machining affected a large area of the surface around the cavity.

Figure 4-1: Plug and hole cavities - Benjamin and Westwater (1961)
4.1.1 Cavity Grids

In the case of the studies described in the previous section, work was undertaken on single boiling cavities, to allow measurement of bubble dynamics, particularly growth rate, shape, and departure size. Recent heat transfer studies have created a demand for the manufacture of grids of cavities which can be used to simulate the high number of boiling centres present on a real surface.

The processes discussed so far have all been limited by speed of manufacture and flexibility, making them unsuitable for production of grids of cavities. However, some studies of boiling from grids of cavities have been completed, including those listed in Table 4-2.

<table>
<thead>
<tr>
<th>Author</th>
<th>Year</th>
<th>Cavity Size (µm)</th>
<th>Grid Shape</th>
<th>Site Density (sites.m(^{-2})) (10^{-6})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Griffith and Wallis</td>
<td>1960</td>
<td>50</td>
<td>Triangular</td>
<td>0.1</td>
</tr>
<tr>
<td>Rammig and Weiss</td>
<td>1991</td>
<td>36 - 70</td>
<td>Square</td>
<td>15 - 18</td>
</tr>
<tr>
<td>Bhavani et al</td>
<td>2000</td>
<td>40</td>
<td>Square</td>
<td>0.44 - 4</td>
</tr>
</tbody>
</table>

Table 4-2: Grid manufacture
Griffith and Wallis – gramophone needle

Griffith and Wallis (1960) polished a copper surface with 2/0 emery cloth in order to remove all naturally occurring cavities and then created artificial boiling cavities. These cavities were manufactured using a specially prepared gramophone needle. The tip of the needle was ground to $18^\circ$ with a tip radius of curvature of 25 μm. It was allowed to penetrate the surface by a known distance, as shown in Figure 4-3.

![Figure 4-3: Cavity cross-section](image)

Thirty seven evenly spaced cavities of 50 μm in diameter were placed in an equilateral triangular grid. A grid spacing of 3.175 mm was used. They studied the boiling grids and the effect of pressure and temperature variations on boiling.

Rammig and Weiss – Electroerosion

Rammig and Weiss (1991) conducted experiments to study boiling from artificial cavities using liquid hydrogen and nitrogen. They manufactured square grids of cavities, with densities of between 1500 and 1800 sites.cm$^{-2}$. Electro-erosion was used to drill cavities of diameter 27 - 70 μm and 100 - 230 μm deep in copper. Figure 4-4 shows a sample of the re-entrant cavities produced.
In electro-erosion drilling, the tool and the work-piece are connected to a power supply, with the positive pole to the work-piece and the negative pole to the drill. An electric current flows through the circuit that has been created. An electro-magnet, which is part of the circuit, lifts up the drill and the current is cut off, at which point the drill is forced back to the work-piece by a spring. The cycle then starts again (Technoorg, 2003). The drilling is done by means of an electro-chemical process which takes place at the moment of breaking the electronic circuit.

This manufacturing method is slow and can only be used for high conductivity materials with the exception of aluminium. Drilling speeds in titanium for example are approximately 1 mm per minute.

**Bhavani et al – Etching**

Bhavnani *et al* (2000) etched cavities into a silicon wafer. In standard etching techniques, the silicon is coated with a light-sensitive polymer. Light is then irradiated through a stencil onto the silicon wafer and the polymer reacts with the light, causing the silicon to be removed from that region and producing cavities similar to the one shown in Figure 4-5.
Using this technique, Bhavani et al were able to produce cavities 40 μm in diameter in a precise grid 7 mm square. The cavity spacing varied between 0.5 mm, 1 mm and 1.5 mm. These were chosen to nominally represent one, two and three times the bubble departure diameter.

4.2 Laser Technology

Laser drilling of boiling cavities was first considered by Singh et al (1977). However, in their published data, it is unclear which method they used to drill their cavities. They manufactured cavities of cylindrical shape, with a mouth radius of approximately 5.4 μm, and a depth ratio of between 2 and 4.

Laser drilling works by boiling the material on which it is focused. Expanding metal vapour throws molten material away from the focus region. This reveals a new surface on which the process is repeated, drilling still further into the material. This is similar to the more common process of laser welding, although drilling has shorter pulses and a higher initial peak laser power (Steen, 1991). The laser must reach high peak powers in short pulses in order to avoid the significant transfer of heat away from the hole into the bulk of the material.
Drilling is a complex laser machining process. Crafer and Oakley (1993) suggest a number of key factors (listed below) relating to the quality of any hole or cavity, which should be considered when comparing laser drilling with other possible manufacturing processes such as mechanical drilling or electro-discharge machining.

- Diameter
- Roundness
- Cylindricity
- Speed
- Drilled surface roughness
- Recast thickness
- Recast composition and structure
- Delaminating
- Micro cracks

When assessing these factors, laser drilling of boiling cavities is seen as an ideal manufacturing method, especially because boiling cavities are required to be of various diameters, manufactured at high speed and to be machined into a polished surface.

Methods of laser hole drilling can be put into two general groups: direct drilling and rastered drilling. Direct drilling uses a laser beam focused on to the material surface with a beam diameter equivalent to the required hole size. The laser can operate with either a single pulse (single shot drilling) or multiple shots or passes (percussion drilling). Rastered drilling involves focusing the laser beam at a small diameter onto the material surface. The laser is then scanned over the cross section of the required hole, removing a small amount of material at a time.
4.2.1 Laser Systems

Steen (1991) suggested a great variety of applications for laser drilling including aerosol valves, baby’s teats, spray nozzles, optical apertures and compact discs. Commonly laser drilling systems are now being used for any process involving large numbers of holes, holes with small diameters, the machining of hard materials or for drilling at a low angle to a surface (Crafer and Oakley, 1993).

Laser drilling equipment can utilise any laser source capable of producing a high enough peak power to vaporise the material that requires machining. Industrial laser drilling systems currently tend to be limited to those shown in Table 4-3.

<table>
<thead>
<tr>
<th>Laser Type</th>
<th>Wavelength (μm)</th>
<th>Output Type</th>
<th>Power</th>
</tr>
</thead>
<tbody>
<tr>
<td>Excimer</td>
<td>0.15 – 0.3</td>
<td>Pulsed</td>
<td>Tens of Watts</td>
</tr>
<tr>
<td>Nd:YAG</td>
<td>1.064</td>
<td>Pulsed or Continuous</td>
<td>Kilo Watts</td>
</tr>
<tr>
<td>CO₂</td>
<td>9 - 11</td>
<td>Pulsed or Continuous</td>
<td>Tens of Kilo Watts</td>
</tr>
</tbody>
</table>

Table 4-3: Guide to laser variations - Hecht (1992)

Each of these are now discussed.

Excimer Lasers

Excimer lasers contain a mixture of gases in which energy changes its form to become laser light. Comprising of 90-99% buffer gas, most commonly nitrogen, the remaining fraction of the gas mixture consists of the Excimer molecules. Excimer gases consist of two identical atoms, which only exist in the excited states, whilst the buffer gas is there to aid the energy transfer. Commonly used Excimer gasses include argon fluoride, krypton fluoride, xenon fluoride and xenon chloride. Energy is deposited in the gas by a pulse
of electrical discharge, which is then converted into laser energy. This is controlled using a number of mirrors before lenses focus the energy onto the work piece.

An Excimer laser produces a beam of ultraviolet light of approximately 650 nm wavelength. The Excimer does not remove material by burning or vaporising, but by breaking down the atomic bonds of the material. Unlike other types of laser, this means that adjacent material is less affected by the heat that is generated. The laser is pulsed, removing material with each pulse. The amount of material removed is dependant on the material itself, the length of the pulse and the intensity (fluence) of the laser light. Fluence below the threshold of the material will have no effect. As the fluence is increased above the threshold, the depth of cut is increased.

Figure 4-6: Possible shapes machined by Excimer laser - Banks (2000)

Figure 4-6 from Banks (2000) shows some of the shapes that could be manufactured using an Excimer laser. These have the potential to be used as cylindrical, conical and re-entrant boiling cavities.

**CO₂ Lasers**

The active gas in a CO₂ laser is a mixture of carbon dioxide, nitrogen and, in most cases, helium. Carbon dioxide is the light-emitting gas. The nitrogen assists excitation and the helium acts as a buffer gas to aid transfer of energy. Excitation can be initiated by several different methods including electrical discharge or radio frequency. During the process CO₂ is broken down into carbon monoxide and oxygen. This re-forms to become carbon dioxide. Hydrogen or water is sometimes added to aid the regeneration.
CO₂ lasers are large, containing expensive gasses, which means they are expensive to operate. They are normally used for sheet metal cutting and have simple optical arrangements, meaning that they are hard to calibrate if needed to machine relatively small holes such as boiling cavities.

**Nd:YAG lasers**

Neodymium: Yttrium Aluminium Garnet (Nd:YAG) is part of the solid state laser family. This type of laser operates by exciting a crystalline material with light from an external source. The crystalline material is mounted in an optical cavity, allowing the generation of feedback.

In the case of an Nd:YAG laser, Yttrium Aluminium Garnet crystalline solid is impregnated with approximately 1% of Neodymium. Although not a highly efficient laser material, it has the advantage of being able to produce a high quality continuous beam at room temperature.

### 4.3 Laser Parameters

Each system discussed so far has its particular advantages and disadvantages and when combined with a computer numerically controlled (CNC) moving table top, can produce grids of cavities. The choice of which system to use depends on a number of parameters such as the material to be drilled, hole depth, hole shape, speed, flexibility and recast layer.

#### 4.3.1 Material Choice

Ready (1997) suggested that the maximum depth, \( L_{c,\text{max}} \), of material that could be vaporised can be calculated based on the conservation of energy:

\[
L_{c,\text{max}} = \frac{E_0}{A \rho \left[ C_p \left( T_{\text{boil}} - T_{\text{amb}} \right) + h_{\text{fg}} \right]}
\]

Equation 4-1
where $C_p$ is the specific heat capacity of the material, $T_{\text{boil}}$ the boiling temperature, $T_{\text{amb}}$ the ambient temperature, $h_{fg}$ the latent heat of vaporisation, $\rho$ the density of the material, $E_0$ the laser energy and $A$ the area of the beam. This model assumes that all the energy of the laser pulse is employed in vaporising the material.

The laser power can be assumed to be the peak average power, $P_0$, per pulse, where the pulse frequency is $f$;

$$E_0 = \frac{P_0}{f} \quad \text{Equation 4-2}$$

Rearranging Equation 4-2 and substituting into Equation 4-1 gives:

$$L_{c,\text{max}} = \frac{P_0}{fA\rho[C_p(T_B - T_0) + h_{fg}]} \quad \text{Equation 4-3}$$

As can be seen from Equation 4-3, the choice of parameters such as laser power and frequency are directly related to the material that is being drilled. This is especially important when considering material properties such as density and boiling temperature which can vary greatly from metal to metal. Table 4-4 shows approximate values for aluminium and cast iron, highlighting these variations.

<table>
<thead>
<tr>
<th></th>
<th>Aluminium</th>
<th>Cast Iron</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, $\rho$ (kg.m$^{-3}$)</td>
<td>2700</td>
<td>7900</td>
</tr>
<tr>
<td>Boiling Temp, $T_B$ (°C)</td>
<td>2470</td>
<td>2750</td>
</tr>
<tr>
<td>Specific Heat, $c$ (kJ.kg$^{-1}.$°C$^{-1}$)</td>
<td>0.97</td>
<td>0.46</td>
</tr>
<tr>
<td>Latent heat of vaporisation, $h_{fg}$ (kJ.kg$^{-1}$)</td>
<td>10900</td>
<td>5800</td>
</tr>
</tbody>
</table>

Table 4-4: Material properties - Ready (1997)
4.3.2 Drilling Depth

Using Equation 4-3, the depth of vaporisation can be calculated. This theoretical value will be higher than that found in practice, due to absorption, recast layers and the type of gas nozzle.

These calculations assume the entire laser power is used to vaporise the metal, whereas in practice different materials will absorb different amounts of laser power. Ready (1997) suggested Figure 4-7 as a rough guide to the expected level of absorption for different materials at different laser wavelengths. This level may vary throughout the duration of the laser pulse, as the preliminary energy burst may disturb the surface enough to destroy its reflectiveness.

Figure 4-7: Material absorption at varying wavelengths of light - Ready (1997)

Equation 4-3 calculates the depth of material that would be vaporised and not the depth of hole that will be drilled. Only if all the vaporised material is removed from the hole will they have the same value. The recast layer,
material which is not removed, causes the observed hole depth to be less. Gas nozzles are frequently used with laser drilling to reduce the recast layer.

Gas nozzles apply a close coaxial gas flow to the laser beam which increases the force on the molten metal. If oxygen is used then a further reduction in the recast layer will be seen on materials whose oxide acts as a flux, as oxidation can also contribute to the energy in the process. Gas nozzles also protect the focusing lens from material that is being thrown away from the focus region. However, they are not always used on laser drilling as they do have the disadvantage of cooling the focus area.

So far, this work has only discussed the depth limit of a single pulse. Percussion drilling and rastered drilling both use multiple laser pulses to drill holes of greater depth than can be drilled with a single pulse. However, there are still limits to the depth of hole that can be drilled, for reasons such as the ability to eject molten material from the bottom of a deep blind hole.

![Figure 4-8: Effect of numerous passes on hole depth - Hecht (1992)](image)

Figure 4-8 shows an asymptotic trend towards a limited depth of hole as the number of passes is increased. Two main factors, the focal depth and the vaporised metal removal, are the cause of this upper limit. The laser is focused onto the initial surface of the material. As each pulse removes a layer
of metal, the surface moves further below the focal plane. This means the effective power of the laser becomes less with each pulse. Secondly, as the depth of the hole increases, the removal of the vaporised material becomes more difficult. Material that has not been removed solidifies in the hole as a recast layer. This is a particular problem with blind holes, where the drilling does not break through the bottom surface.

4.3.3 Hole Shape

During single shot drilling, variations in hole depth are achieved with changes in pulse power. In general, a laser machined hole, especially with single shot drilling, will take the shape of a cone, although if the laser power is high enough the hole will continue straight through the sheet. Figure 4-9 shows the variation in hole shape which will occur with variation of the power of laser drilling on a metal sheet. Crafer and Oakley (1993) suggest it is easier to drill a complete hole than blind holes, especially if a gas nozzle is being used.

![Hole shape changes with laser power - Hecht (1992)](image-url)

Figure 4-9: Hole shape changes with laser power – Hecht (1992)
Although laser hole drilling can be very accurate, there are a number of defects which can occur regularly with this manufacturing process. Figure 4-10 shows the cross-section of a typical laser drilled hole. Defects illustrated by this hole include rough sides, lack of roundness, taper of the hole and the offset of the centreline of the hole from normal to the surface (Ready, 1997). Some of these defects are difficult to eliminate from laser-drilled cavities.

![Figure 4-10: Possible laser drilling cross-section - Ready (1997)](image)

Furthermore, it is acknowledged that very small holes are only achieved with some sacrifice in uniformity. Small variations in material properties can effect individual holes when laser drilling at a diameter of less than 200 µm.

### 4.4 Laser Manufacturing of Boiling Cavities

In previous investigations boiling cavities with diameters 8 - 1000 µm have been studied. The majority of work has used cavity diameters of around 100 µm. Various cavity profiles have been used, including conical, cylindrical and re-entrant. It is accepted that re-entrant cavities produce stable boiling sites and are essential when the boiling medium has a small contact angle, for example paraffin. Water has a relatively high contact angle and re-entrant cavities are not necessarily required.

The work described here aimed to produce deep cylindrical cavities of 25 - 300 µm diameter and depth to diameter ratio of not less than two. This was to ensure that they would be stable boiling centres, as highlighted by
Singh et al (1977). The cavities need not be cylindrical, as deep conical cavities can be assumed to be cylindrical. The differences between conical and cylindrical cavities are only noticeable when shallow cavities are being studied. Finally, it should be noted that the shape of the cavity mouth is not believed to have a significant effect on boiling. In this present study, cavities will have circular openings.

4.4.1 Excimer Laser

Preliminary testing was undertaken with an Excimer laser, normally used for surface profiling and micro engraving. The system, which is specified below in Table 4-5, was chosen for its high accuracy and flexibility.

<table>
<thead>
<tr>
<th>Lambda Physik EMG 203 MSC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wavelength</td>
</tr>
<tr>
<td>Pulse Length</td>
</tr>
<tr>
<td>Average power</td>
</tr>
<tr>
<td>Pulse Energy</td>
</tr>
</tbody>
</table>

Table 4-5: Specification of Excimer laser facility

The laser spot size, at the work piece, is adjusted by blocking off the area of light not required, with a mask. A lens then focuses the beam before it strikes the work piece. Focusing the beam reduces the diameter by a factor of 15 and the hole in the mask is therefore manufactured 15 times bigger than the cavity being drilled. A 100 µm cavity is produced using a 1.5 mm mask which could be easily drilled using a conventional pillar drill, or similar. If a smaller cavity is required then the laser could be used to manufacture the mask, for example a 1.5 mm mask machines a 100 µm hole, which can then in turn be used as the mask to machine a 6.6µm cavity.
In this present work, the laser was operated at 100 Hz and the shutter held open for 2 - 5 minutes at a time. Masks were selected to machine cavities of diameter 10 – 400 µm. Despite the long exposure time, manufacturing deep cavities was a problem. Cavities of diameter 100 – 400 µm had a depth ratio of approximately 1. Cavities with diameters 10 – 100 µm had a depth ratio of approximately 0.2.

The laser was rated at a 100 W, however, this power was spread across the diameter of the beam. A mask was then placed in the beam blocking a significant amount of the power. A mask with a 1.5 mm hole in it (used to manufacture 100 µm cavities) reduced the beam cross-sectional area by a factor of 150. With such a low power, machining metal was not found to be feasible because:

- Molten metal was not removed from the cavity and re-cast itself to create a shallow cavity.
- Power loss at a distance from the focal plane was too great.
- The high thermal conductivity of metal rapidly transferred energy away from the cavity.

4.4.2 Nd:YAG

A system that focuses the entire power of the beam to the required spot size, rather than blocking it, would avoid the problems encountered with the Excimer laser. An Nd:YAG laser system with an adjustable focusing multiplier was used to overcome problems. Using an adjustable lens to vary the spot size ensured that the full power of the beam was utilised.
Table 4-6: Specification of Nd:YAG laser facility

Table 4-6 describes the Nd:YAG laser system. A gas nozzle was mounted on the laser head, focusing a jet of oxygen on to the work piece, therefore increasing the effective laser power. By adjusting the multiplier lens, the laser spot size was changed. This was more difficult than changing masks in the Excimer laser but did have some flexibility. The laser was operated at 20 Hz, and the shutter held open for 2 - 5 seconds. This system also had a computer controlled X - Y table. This allowed grids to be machined automatically and accurately positioned to a spatial tolerance of ± 0.001 mm.

Preliminary testing was done with the multiplier lens in a position that produced cavities of approximately 120 μm. Figure 4-11 shows the first cavity manufactured with this system.
The cavity in Figure 4-11 has a number of key features including:

- Eccentric mouth diameter \( D_{c,x} (P_2-P_{2R}) = 107.52 \, \mu m \) and \( D_{c,y} (P_1-P_{1R}) = 152.02 \, \mu m \).
- \( L_c/D_c > 3 \)
- Crowning/splatter attached to the outside of the cavity mouth.

The eccentricity of the cavity mouth was caused by a small misalignment of the laser focusing mirrors and lens. The beam striking the work piece was of similar shape to the cavity because the beam had not been aligned to the centre of the lens. The eccentricity was exaggerated further by the fact that poorly set mirrors caused the laser power to spread unevenly across the beam diameter.
The time the shutter was open was limited to 2 - 5 seconds, however, if left open longer this system could have machined deeper cavities. This laser system has been able to produce deep cavities, unlike the Excimer laser, because the beam power density was created by focusing the beam instead of blocking it. The oxygen gas nozzle also increased the effectiveness of the laser.

Finally, the most significant problem highlighted in preliminary testing was the recast layer around the top of the cavity, known as the crown. This was caused by molten metal cooling as it met the cold air and attached itself to the rim of the cavity. This problem would normally be solved by increasing the power and decreasing the shutter opening time (exposure time), so that molten metal would be thrown clear of the cavity. Because the laser was already at peak power, this solution could not be used in this case.

Figure 4-12 shows a comparison of two cavities, one machined with a long exposure time and the other with a short exposure time, measured with a Zygo 3D surface profiler. It is clear that if the exposure time is shorter then the crowning effect around the cavity is significantly reduced. The laser system was already operating at peak power so both the cavities shown in Figure 4-12 are at the same laser power, therefore reducing exposure time also reduces the depth of the cavity. The cavity on the right has an $L_e/D_e \approx$
1.5. If the exposure time were reduced further in an attempt to remove the crowning effect, the cavity would become too shallow.

Surface preparations such as PTFE-based pastes were used in an attempt to protect the cavity rim from the recast material. This removed the largest crowns, whilst simultaneously causing difficulty when the gas nozzle was turned on as the paste was blown away from the required area. Attempts to harden the paste by allowing it to dry before laser machining improved the process. No solution that completely removed the crowning effect was found when the continuous Nd:YAG laser system was used.

### 4.4.3 Q-switched Nd:YAG

In an effort to reduce the crowning effect further, an alternative configuration and set-up of Nd:YAG laser was used, as specified in Table 4-7. With a Q-switch laser chamber and a scanning head instead of X-Y tables, it was anticipated that crowning would not occur.

<table>
<thead>
<tr>
<th>Coherent Inc. Q-switched Nd:YAG</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Wavelength</strong></td>
</tr>
<tr>
<td>1.064 µm</td>
</tr>
<tr>
<td><strong>Pulse Length</strong></td>
</tr>
<tr>
<td>120 - 150 ns</td>
</tr>
<tr>
<td><strong>Average power</strong></td>
</tr>
<tr>
<td>20 W</td>
</tr>
<tr>
<td><strong>Pulse Energy</strong></td>
</tr>
<tr>
<td>4mJ per pulse @ 500 Hz</td>
</tr>
</tbody>
</table>

Table 4-7: Specification of the Q-switched Nd:YAG laser system

The Coherent Q-switched Nd:YAG laser is similar to the standard model, although it has a different method of releasing the energy from the laser rod. In a standard Nd:YAG laser, the rod is excited by pulses from a diode, causing light to reflect back and forth, magnifying the power within the rod. At one end a window allows a proportion of the energy to trickle out,
creating a pulsed output beam. In the Q-switched system, the window is replaced by a fast shutter that gathers up energy and then releases it in one fast, high powered pulse. Figure 4-13 shows the difference in pulse energy between the Q-switched and the standard Nd:YAG laser. The main differences are high peak power and shorter pulse length. It should be noted that the output frequency of the laser remains the same.

Figure 4-13: Schematic comparison of pulse waves in standard Nd:YAG and Q-switched Nd:YAG

The second significant difference in the system set-up is that Coherent Q-switched laser systems use a pair of scanning mirrors to move the laser beam to any point on the work piece, instead of an X-Y table that moves the work piece. A schematic of which is shown in Figure 4-14.
This is not only faster than the X-Y tables but it also allows the system to be used in Raster mode. Focusing the beam to a small spot, this beam is then scanned over the area that requires machining, removing small areas at a time. Because the material is removed gradually, the re-cast layer does not attach to the cavity rim and no crown appears. Figure 4-15 shows a 70 μm cavity machined using a Q-switched Nd:YAG laser in Raster mode.

Figure 4-15: Sample of cavity machined with Q-switched laser
Accuracy of Laser Manufacture

Laser machined cavities between 25 and 250 μm in diameter (discussed further in Chapter 5) were measured using a microscope and reticle. The size of the cavities was found to be slightly larger than prescribed. This is shown in Figure 4-16. The discrepancy was a result of the high number of passes required to machine deep cavities, resulting in cavities larger than programmed because of increased heat input to the surface.

![Figure 4-16: Programmed diameter vs. actual diameter](image)

To cancel out the discrepancy, the computer was programmed to machine undersized cavities in order that the machined cavities would then be correctly sized. This is one of the most significant advantages of using a computer-programmed manufacturing technique.
4.5 Concluding Remarks

As discussed in the previous chapters, the objective of this study is to enhance engine cooling systems. The aim is to focus on the high heat flux areas such as the valve bridge and the use of the high heat transfer coefficient that is associated with a nucleate boiling regime. A literature survey has highlighted that surface features, specifically surface cavities, are a key contributing factor to the enhancement and control of nucleate boiling. Lasers have been found to be a suitable method of manufacturing a boiling surface grid with the aim of enhancing the heat transfer.

A Nd:YAG Q-switched laser system has been proved to be a successful and an ideal method of producing good quality boiling cavities. Manufacturing cavities in this way means a grid of up to 1000 mm² (i.e. a similar surface area to high heat flux areas of an engine cooling system) can be machined typically within 10 seconds. The size of the cavities can be varied from 10 μm to 500 μm and it is possible to create cavities of varying sizes within the same boiling grid. The key to a cavity’s effectiveness as a nucleation site is its depth ratio, \( \frac{L_c}{D_c} \). This system is capable of machining cavities to a ratio in excess of 2. Finally, the crown around the cavity edge was identified as a potential hindrance to nucleation, and with the Nd:YAG Q-switched laser system the crown height \( L_c \) is negligible.

Laser manufacturing offers a number of other advantages as well as those already discussed. How to manufacture enhanced surfaces for use in engine cooling ducts must also be considered. One option would be to produce an insert that could be manufactured and then mounted in the cooling system. A second option could be that this type of enhanced surface could be machined directly into the cooling duct wall. This would be possible using a fibre optical cable to laser machine cavities directly into the cylinder head. These are both possible with using an Nd:YAG Q-switched laser system.
5 Experimental Apparatus and Equipment

Studies of nucleate boiling fundamentals in Chapter 3 discussed the reasons for the increased heat transfer coefficient during the nucleate boiling process. Surface cavities were identified as predominantly controlling boiling as bubbles initiate from them.

Cavities machined into grids control boiling by specifying where on the surface vapour bubbles will be generated. In Chapter 4 a fast, efficient and accurate method for cavity grid production using a Nd:YAG laser was developed.

This chapter discusses the parameters controlling the design of boiling grids for use in IC engine cooling systems. Criteria, including pressure and temperature, are used to select the cavity grid dimensions.
The design of the experimental equipment for testing the effectiveness of the boiling grids is also explained. An optical boiling chamber to house boiling grids was designed, and a bubble digital image analysis (BDIA) system for observation of the boiling regime was incorporated.

The accuracy of the experimental equipment is discussed and a validation experiment is reported. The experimental study of boiling from laser machined boiling grids will be discussed in Chapter 6.

5.1 Boiling Chamber

In order that laser-manufactured boiling grids can be used in practical applications it was necessary to first validate the concept experimentally. An experimental rig was designed and manufactured. This rig was comprised of a boiling chamber and a BDIA system.

Investigation of previous studies and their experimental equipment was valuable during the design stage of the new apparatus in order to compare results. Table 5-1 summarises the equipment used by other engine heat transfer investigators prior to this study. A number of these studies used boiling chambers with dimensions similar to that found in real engine cooling systems, which enabled the authors to make valid comparisons with working systems.
<table>
<thead>
<tr>
<th>Duct Shape</th>
<th>Duct Size (mm)</th>
<th>AR</th>
<th>Hydraulic Diameter $D_h$ (mm)</th>
<th>L/D</th>
<th>Liquid Type</th>
<th>Liquid Velocity (m s$^{-1}$)</th>
<th>Liquid Temp (°C)</th>
<th>Liquid Pressure (bar)</th>
<th>Heat Flux (kW m$^{-2}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>French (1969)</td>
<td>Rectangular</td>
<td>50.8 x 25.4</td>
<td>2</td>
<td>33.9</td>
<td>-</td>
<td>Demineralised Water</td>
<td>0.09 - 4.6</td>
<td>50 - 70</td>
<td>-</td>
</tr>
<tr>
<td>Smith et al (1970)</td>
<td>Rectangular</td>
<td>38.1 x 38.1</td>
<td>1</td>
<td>38.1</td>
<td>5.3</td>
<td>Demineralised Water</td>
<td>0.03 - 0.09</td>
<td>80</td>
<td>-</td>
</tr>
<tr>
<td>Finlay et al (1988)</td>
<td>Circular</td>
<td>-</td>
<td>1</td>
<td>6.35</td>
<td>9.4</td>
<td>Water &amp; EG</td>
<td>0.1 - 5.5</td>
<td>85</td>
<td>1 - 2</td>
</tr>
<tr>
<td>Perry et al (1985)</td>
<td>Circular</td>
<td>-</td>
<td>1</td>
<td>2.0 - 8.0</td>
<td>20</td>
<td>Water &amp; EG</td>
<td>1.0 - 7.0</td>
<td>90</td>
<td>2</td>
</tr>
<tr>
<td>Boyle et al (1991)</td>
<td>Rectangular</td>
<td>50.0 x 5.0</td>
<td>10</td>
<td>9.09</td>
<td>11.0</td>
<td>Water &amp; EG</td>
<td>0.1 - 3.0</td>
<td>90</td>
<td>2</td>
</tr>
<tr>
<td>Cipolla (1989)</td>
<td>Valve Bridge</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>Water &amp; EG</td>
<td>0.1 - 1.5</td>
<td>55 - 107</td>
<td>2, 3</td>
</tr>
<tr>
<td>Norris et al (1994)</td>
<td>Valve Bridge</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>Water &amp; EG</td>
<td>0.52 - 1.52</td>
<td>90</td>
<td>1.3</td>
</tr>
<tr>
<td>Campbell et al (2000)</td>
<td>Rectangular</td>
<td>10.0 x 10.0</td>
<td>1</td>
<td>-</td>
<td>-</td>
<td>Water &amp; EG</td>
<td>0.25 - 2.0</td>
<td>40 - 90</td>
<td>1 - 2</td>
</tr>
<tr>
<td>Gollin et al (1995)</td>
<td>Circular</td>
<td>-</td>
<td>1</td>
<td>15.9</td>
<td>10.4</td>
<td>Water &amp; EG</td>
<td>0.1 - 0.5</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 5-1: Comparison of heat transfer apparatus - adapted from Campbell et al (2000)

Note: '-' indicates that the authors did not quote this dimension.
Many of the authors' experimental rigs given in Table 5-1 used the length ratio \((L/D_h)\) and aspect ratio \((AR)\) to describe their experimental arrangement. \(L/D_h\) is a ratio of heated length against hydraulic diameter and \(AR\) is defined by Equation 5-1:

\[
AR = \frac{4 \times A}{p}
\]

Equation 5-1

where \(A\) is the cross sectional area and \(p\) is the wetted perimeter.

The geometry of actual engine cooling ducts has also been studied by Cipolla (1989) and Norris (1994), despite the difficulty of measuring and classifying them. Difficulties arise because the cooling ducts are inside the cylinder head and are not visible; therefore, 3D CAD models have been used to visualise the size and shape of the cooling ducts.

![Figure 5-1: Cylinder head sand cores](image)

In this study, sand cores shown in Figure 5-1 were also used to visualise the cylinder head cooling ducts. Sand cores are used to manufacture the casting and are an invert of the final object, therefore the solid sand is where the cooling ducts are in the cast head. Ducts that were hard to measure or visualise in the actual cylinder head were easily accessible.
It is also difficult to classify the water gallery dimensions because of the large variety of sizes and shapes found within the cylinder head. Lee and Cholewczynski (2003) commented that in thermally critical areas of the engine cooling system, the geometry was extremely complex and difficult to classify.

Using Table 5-1 for reference, as well as studying details of real life engine geometries, it was decided to manufacture a rectangular test section 40mm by 20mm, which is of similar size to the cooling ducts found in current automotive diesel engines. It is accepted that no cooling ducts found in engine cylinder heads are exactly rectangular in cross section, however, it allowed for ease of manufacture and flexibility.

With a heated length of 30 mm, the boiling chamber parameters (shown in Table 5-2) are similar to previous studies and therefore allow comparisons to be drawn.

<table>
<thead>
<tr>
<th>Aspect Ratio</th>
<th>2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydraulic Diameter (mm)</td>
<td>26.6</td>
</tr>
<tr>
<td>L/D_h</td>
<td>11</td>
</tr>
</tbody>
</table>

Table 5-2: Observation chamber parameters

The heat was applied to the base of the test section. This allowed windows to be installed on the other three sides for observation. A pair of windows opposite each other allowed the boiling regime to be recorded and measured. Optical measurement equipment is ideal for the study of nucleate boiling as it is a non-contact method and has no affect on the process. The third window on the top of the section gave further flexibility in test setup, including the possibility of using laser light sheets to illuminate the system, or the measurement of active site densities.
The heater block has air gap insulation (shown in Figure 5-2 and Figure 5-3), which ensured that > 97% of the energy was transferred into the water. One-dimensional heat transfer analysis and finite difference modelling of the block was carried out to confirm this.

Figure 5-2: Cross sectional schematic of the boiling chamber

Figure 5-3 shows the final boiling chamber whose key features are:

- Realistic flow cross-section area comparable to IC engine cooling passages.
- Optical viewing area of the nucleation sites from 3 sides.
- Short length in order to avoid fully developed flow.
- Pressure limit in excess of 3 bar gauge (300 kPa).
- Flow and stagnation capabilities.
- Cartridge heater which can be changed according to heat flux required.
5.2 Bubble/Boiling Analysis

The benefits and disadvantages of different experimental approaches to recording and analysing nucleate boiling were reviewed. Consideration was given to studies such as Clarke et al (1959) and Griffith and Wallis (1960) who used simple photographic techniques. Authors including Calka and Judd (1985), Shoji and Takagi (2001) and Golobic and Gjerkes (2001) developed more complex automated methods for boiling analysis.

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

5.2.1 Previous Photographic Studies

Studies such as those by Clarke et al (1959) and Griffiths and Wallis (1960) used simple photographic techniques to observe and measure boiling characteristics. Singh et al (1977) advanced these techniques by combining the use of a stroboscope with photographic techniques.
In each case, boiling was photographed at set time intervals and the authors were therefore able to observe key features such as bubble size, growth, boundary layer and departure frequency. An obvious limit to this method of boiling analysis was the speed at which images could be taken. Bubbles may depart the surface at a frequency as high as 200 Hz and therefore in order to study the departure characteristics of a single bubble, the camera needs to operate at a minimum speed of 1000 frames per second. Advances in camera technology over the last few years means that equipment capable of operating at this speed is now much more affordable.

The analysis and measurement of the results undertaken by Clarke et al (1959) and Griffiths and Wallis (1960) was carried out manually. Some of the pictures they recorded were projected on to a wall. Measurements of the bubbles were then taken using a ruler and then scaled down to allow for the magnification of the projector. This was a time consuming method of data analysis and open to systematic and human error. Even recent work by Bhavnani et al (2000) chose high-speed photography because of its simplicity, and despite the obvious difficulty of the manual analysis which would then need to be undertaken.

The use of digital charge coupled device (CCD) cameras was avoided in the past because of low frame rates. However, equipment is now capable of frame rates in excess 2000 frames per second (fps), a speed previously only reached with a 35 mm drum cameras. Therefore, it is now possible to gather high speed, high quality images which can then be reviewed and analysed using computer software. Having the images in a digital format instead of film allows for automated analysis techniques.
5.2.2 Automated Data Collection/Analysis

The shortcomings of previous photographic studies were the time required to analyse large quantities of data. Although CCD technology has improved, this problem still remains an issue. Calka and Judd (1985) were the first to develop a truly automated system that could generate large quantities of data in a relatively short period of time, with minimal human interaction.

They investigated the effect of cavity separation distance on bubble departure frequency and produced an experimental system which automatically collected relevant boiling data from a desired surface. Using electrical resistance heating of stannic oxide coated glass, Calka and Judd (1985) designed an optical system to observe nucleation from the underside of the heated surface. Using a 632.8 nm wavelength He-Ne laser they were able to partly reflect the beam from the heating surface and observe changes in the intensity of the reflection, which signalled a departing bubble. They then used a phototransistor to record these variations and calculated the bubble departure frequency and position. This enabled them to generate data quickly, accurately and with good repeatability.

The largest single disadvantage to this system was that it required a transparent heated surface such as glass. This meant it was hard to create high heat fluxes without damaging the surface. Additionally, as glass is very smooth, it produced different results to the metallic surfaces which other authors have used. However, their system was ideally suited to the precise experiments that they were performing.

There would be difficulties in applying their technology to the measurements required in this study. However, it can be seen that accuracy, repeatability and speed are all key benefits of developing an analysis system which is fully automated.
5.2.3 VisiSizer System

Previous studies have shown the significant benefits of having a fully automated system for data analysis. The development of high speed CCD cameras now enables photographic studies to become significantly less reliant on manual analysis, as digital image processes can be used to automate the analysis of measurements.

The benefits of optical non-contact measurements, combined with the speed, accuracy and reliability of an automated system is an ideal combination. In this study a novel system including high speed video, digital image processing and automated data analysis was chosen. The system comprised infrared laser backlighting to illuminate the region of interest and a high resolution CCD camera (see Figure 5-4).

Figure 5-4: Laser and camera system

Because the bubbles refracted the laser light, shadow images of the bubbles were created which were then recorded with the CCD camera. The camera and laser were triggered so that a single laser pulse arrested the motion of the bubble image during each frame captured. Figure 5-5 is a sample image
from the CCD camera. This system makes use of the best features of the systems used in previous research.

Figure 5-5: Sample shadow image of bubbles

Images were read from the camera by software which had an analysis rate of 30 fps. When a higher frame rate was required, the camera recorded a sequence of images and played them back into the software at the lower frame rate for analysis. This gave the system the capability to operate up to 2000 fps.

The images were thresholded in order to distinguish between the bubble and the illuminated background. The pixel area of the bubble was then measured. Prior calibration of the system allowed the equivalent bubble diameter to be reported. The system was capable of measuring bubbles that were out of focus by measuring the edge gradient of the bubble to determine the correct size. Bubbles that were too far out of focus to be measured accurately were rejected.
Bubble Sizing

Specifying the appropriate calibration value (C) for the chosen lens, the program converted the measured pixel values into actual diameters. Bubbles are sometimes not spherical, therefore the BDIA system calculated the equivalent diameter \( D_{b,a} \) of a bubble from its pixel area (A) as shown:

\[
A = \pi \frac{D_{b,a}^2}{4C^2}
\]

Equation 5-2

\[
D_{b,a} = C\sqrt{\frac{4A}{\pi}}
\]

Equation 5-3

It also determined the bubble diameter (\( D_b \)) from its perimeter (\( P_b \)) as seen in Equation 5-4 and Equation 5-5. The maximum and minimum diameters of a non-spherical bubble were calculated (Oxford Lasers, 2000):

\[
p_b = \pi \frac{D_p^2}{C}
\]

Equation 5-4

\[
D_b = C\sqrt{\frac{P_b}{\pi}}
\]

Equation 5-5

The sphericity of the bubbles was also calculated for values between 0 and 1, using Equation 5-6:

\[
0 < \left( \frac{D_{b,a}}{D_b} \right)^2 < 1
\]

Equation 5-6

Up to 250 bubbles per frame were calculated and the dimensions of each individual bubble were reordered and output to a data file. Statistical analysis, including bubble size distribution, was reported both graphically and in tabular form. Percentile values for 10%, 50%, and 90%, were obtained. The software also calculated the mean number (\( D_{b,m} \)), mean volume (\( D_{b,v} \)) and Sauter mean (\( D_{b,s} \)) diameters for the bubbles. The software then reported the number of empty frames, number of bubbles measured, percentage focus
rejections, background average and elapsed time to provide information on the validity and accuracy of the results.

**Velocity measurement**

The system was able to measure the velocity of bubbles passing through the frame. Velocity data was obtained using a technique known as bubble tracking. During each picture frame the laser was pulsed twice with a known pulse separation. The resulting image consisted of a pair of images of the bubbles. The image analysis procedure then matched the pairs of images and calculated the bubble velocity from its relative position in consecutive frames.

### 5.3 Temperature Measurements and Data Logging

So that parameters such as heat flux and heat transfer coefficient can be measured accurately, it is essential that the temperature be measured throughout the apparatus. Most importantly surface temperature $T_{\text{wall}}$ and bulk temperature $T_{\text{bulk}}$ must be measured as these are then used to calculate the heat transfer coefficient (Equation 5-7) which is of key importance in this study:

$$h = \frac{q}{(T_{\text{wall}} - T_{\text{bulk}})}$$

Equation 5-7

Previous studies have used various temperature measurement techniques such as thermocouples, thermistors, infrared detectors, and even thermally sensitive lacquers. Each method has its own particularly advantages and would be suited to different measurement scenarios.

In this study, K-type thermocouples were used to measure temperature throughout the apparatus, including the wall temperature ($T_{\text{wall}}$), bulk fluid temperature ($T_{\text{bulk}}$) and ambient temperature ($T_{\text{amb}}$). Thermocouples measure
the temperature change at their tip, where the two dissimilar wires are in contact, normally soldered together. The simplicity of their assembly means they offer a reliable source of temperature measurement. Importantly their simple assembly also means they can installed in various configurations as they are both flexible and very small (less than 1.5 mm in diameter). Their flexibility also enables them to be used for all the temperature measurements throughout the rig which helps to maintain the simplicity of the experiment.

An 8 channel data logger was used to record temperature values up to 56 kHz. Using only the data acquisition equipment and computerised spreadsheets eliminated any uncertainties due to human error. Temperature data was saved directly to text files and was entered into MATLAB and Excel. The MATLAB program was constructed to show the data in a graphical format, which enabled immediate assessment of the data before a more thorough validation was undertaken in MS Excel.

Using a finite difference form of the Fourier equation;

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

Equation 5-8

where \( q \) is the specific heat flux, \( \Delta T \) the temperature difference and \( \Delta x \) the distance between points, it was possible to calculate the temperature difference between the thermocouple reading and the actual surface temperature. The worst case scenario of the thermocouple being 1 mm (i.e. \( \Delta x = 10^{-3} \) m) from the surface returned a temperature difference of 0.3 °C at 100 °C, equivalent to 0.4 %. This is within the expected error of ± 0.1 °C associated with thermocouple measurements.

For measurement of the surface temperature, thermocouples were inserted into the aluminium block 600 - 1000 μm from the heater surface. As discussed by Baker et al (1953) a single thermocouple measurement could be used to measure the surface temperature if the thermal gradient in the block was
small. The heater blocks were manufactured from aluminium with a thermal conductivity, $k$, of approximately 200 W.m$^{-1}$K$^{-1}$. This suggested that the thermal gradient would be low and measurements were therefore assumed to be equivalent to surface temperature.

### 5.4 Experimental Arrangement

This Chapter has discussed the individual parts from which the apparatus used in this study is made up. The key components that have already been introduced were the boiling chamber, bubble analysis system and thermocouples used for temperature measurement. In this section the apparatus as a whole is now discussed.

Figure 5-6: Schematic of test equipment

Figure 5-6 shows a schematic of the experimental arrangement. All the equipment is controlled centrally from a single computer, with the exception of the heater. Measurements were taken from thermocouples mounted throughout the chamber. The resulting signals from up to eight thermocouples were fed back to a PC-based software program through the
data loggers screw terminal box. The laser and camera parameters were set on the computer, before using a synchroniser, to ensure that the camera shutter was co-ordinated with the laser pulse. The cartridge heaters were controlled manually by a variable transistor box.

All the equipment was arranged in a way which ensured accuracy, repeatability and safety:

- **Accuracy**: During setup the lens was focused onto a known surface before the vernier slide, on which the camera was mounted, was adjusted to place the camera’s focal plane, where required. This ensured that images were taken of nucleation from the correct cavity.

- **Repeatability**: The CCD camera lens had a seven-position zoom, each position set by a locking ball bearing. Therefore, the camera zoom was always set in one of the seven calibrated positions.

- **Safety**: The camera and laser were mounted on a rail to ensure that the camera was facing the laser at all times. For safe use of the laser, an enclosure was built using the guidance of BSEN 60825-1: 1994 “Safety of laser products, equipment classifications, requirements and user guide”. The enclosure included interlocked Perspex doors that guarded against accidental exposure of the operator.

### 5.5 Accuracy and Calibration

Every effort was made to ensure accuracy and to minimise errors. Particular attention was paid to how the boiling images were gathered and analysed. Other areas where inherent errors are likely to occur include the temperature readings and the heat flux measurement. Both these issues were discussed earlier in this chapter.
5.5.1 Image Acquisition

The camera lens had a calibrated set focal distance of 86 mm, with a field of view (FOV) of between 541 µm and 6532 µm depending on the magnification setting. The magnification settings were adjusted on a locking ball bearing, which allowed repeatable adjustments to be made. The lens was also calibrated separately at each magnification setting and this was checked regularly using a reticle.

The camera was mounted on a motorised Vernier slide to enhance fine positioning of the focal plane. When combined with the 2D Vernier platform on which the boiling chamber was mounted, it offered a fully 3D positioning of the camera’s field of view. Each of the Vernier slides was adjustable to ± 0.01 mm.

5.5.2 Analysis and bubble measurement

Once the images had been recorded by the CCD camera and downloaded to the computer, a digital image analysis software package processed each frame. The operation of this software is discussed in greater detail in section 6.4 of this thesis.

Images were thresholded in order to distinguish between the bubbles (which are black) and the background (which is light grey). The threshold level used was adjusted by the user in order to allow for the level of laser power used in each test.

The pixel area of the bubble was measured using a calibrated lens. This was then converted to the actual area by a calibration value, C (µm/pixel) i.e. normally 1 - 10 µm/pixel. The lens was calibrated at each magnification setting and these figures entered into the software. Calibration was achieved by photographing a reticle at each magnification setting.
To ensure accurate results, bubbles that were not in focus were disregarded. Focus of each bubble was defined by edge sharpness, i.e. the gradient of the transition from the light background to the black interior at a bubble boundary. When a bubble was in focus, the border between it and the background was a step change from black to white. As a bubble became out of focus this border became more poorly defined until the edge was a gradual greyscale transition from the light grey background to the dark centre.

It was essential that out of focus bubbles were rejected to avoid the system analysing them as undersized bubbles. The BDIA software allowed a depth of focus (normally in the range 100 - 1000 μm) to be specified with a corresponding gradient. Larger depths of field corresponded to shallower gradients from black to white.

Finally the system measured its own parameters in order to give the operator an idea of the validity and accuracy of the results. These parameters included the number of empty frames, number of bubbles measured, percentage focus rejections, background average and elapsed time.

5.6 Validation Experiment

An experiment was performed to assess the effectiveness and accuracy of the apparatus described in this chapter. Attention was focused on the operation of the BDIA system and the accuracy of the optical bubble measurement technique developed.

5.6.1 Experimental Set-up

The apparatus was arranged similarly to Figure 5-6. However, the optical boiling chamber was replaced with a Perspex container. No temperature measurements were taken as the aim was to test the accuracy of the BDIA system. The container was approximately 40 mm × 40 mm × 20 mm and did
not have a lid. Two sides of the container were made from flat, clear Perspex which allowed excellent optical image quality, viewing directly through it.

The container was filled with water which had been filtered to remove any solid contaminates. Several thousand polymer spheres known as Dynospheres were then added to the water.

Dynospheres are manufactured to be precisely spherical and approximately 20 μm diameter (± 3 %). They have a density similar to water and therefore remain approximately neutrally buoyant i.e. they have no overall tendency to either float nor sink.

The flat sided Perspex vessel containing these Dynospheres suspended in water was placed between the laser and the CCD camera. A number of single frames was then recorded and downloaded to the computer.

The CCD camera was focused on to the front window of the container and then, using the vernier slide, the camera was moved 20 mm towards it. This ensured that the focal plane of the camera was in the bulk of the suspended particles. A number of images including Figure 5-7 were recorded. Some of the Dynospheres can be clearly seen suspended in the middle of the image.

Figure 5-7: Digital photograph of 20 μm Dynospheres
This image was then viewed in the digital image analysis software, VisiSize. It is suggested by Oxford Lasers (2000) that an image must have a background average of 130 – 170 (where 240 is pure white) for accurate discrimination between the background and the particles. Figure 5-8 shows the background average graph for Figure 5-7.

![Graph of background average](image)

**Figure 5-8: Graph of background average**

The value of the background average was found to be 146, ideal for high accuracy measurement of particle size. The background average could be adjusted before the final image was recorded by adjusting the laser power. The laser was directed through a simple diffuser before it reached the test area and this ensured that the background average was consistent across the entire field of view. The laser power could also be adjusted from the laser control panel attached to the central computer.

As discussed earlier in this chapter, the VisiSize software converted the image from greyscale to black and white with a standard adaptive threshold or percentage threshold, selected by the user according to the background...
average and the level of zoom used on the camera. In this case Figure 5-9 shows a standard adaptive threshold of 70% being used to convert the greyscale image into a pure black and white image.

![Image](image.png)

Figure 5-9: Original image (Left), binorized image (Right)

The image was taken using a CCD camera at a known calibrated zoom setting, in this case a nominal value of two, for which \( C \), the magnification setting is \( 1.92 \ \mu m \cdot pixels^{-1} \). The VisiSize used the magnification setting \( C \) to calculate the diameter of all the particles. Particles out of focus, not spherical or clearly the wrong size were automatically ignored. Particles that were measured were highlighted with an identification (ID) number.

### 5.6.2 Results

The digital image analysis software calculated the diameter, perimeter and shape factor for each particle in the frame. Table 5-3 shows these results and the ID number is that given to each particle in Figure 5-10.
Figure 5-10: Image with measured ID numbers

<table>
<thead>
<tr>
<th>ID</th>
<th>Diameter (µm)</th>
<th>Perimeter (µm)</th>
<th>Shape Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>18.76</td>
<td>73.94</td>
<td>0.92</td>
</tr>
<tr>
<td>2</td>
<td>18.95</td>
<td>75.71</td>
<td>0.90</td>
</tr>
<tr>
<td>3</td>
<td>19.22</td>
<td>79.18</td>
<td>0.84</td>
</tr>
<tr>
<td>4</td>
<td>20.81</td>
<td>80.22</td>
<td>0.95</td>
</tr>
<tr>
<td>5</td>
<td>20.29</td>
<td>79.18</td>
<td>0.93</td>
</tr>
<tr>
<td>6</td>
<td>18.01</td>
<td>71.87</td>
<td>0.91</td>
</tr>
<tr>
<td>7</td>
<td>19.76</td>
<td>79.18</td>
<td>0.89</td>
</tr>
<tr>
<td>8</td>
<td>18.95</td>
<td>75.34</td>
<td>0.91</td>
</tr>
<tr>
<td>9</td>
<td>19.40</td>
<td>75.34</td>
<td>0.95</td>
</tr>
<tr>
<td>10</td>
<td>20.72</td>
<td>90.70</td>
<td>0.74</td>
</tr>
<tr>
<td>11</td>
<td>19.22</td>
<td>77.78</td>
<td>0.87</td>
</tr>
<tr>
<td>12</td>
<td>20.64</td>
<td>84.06</td>
<td>0.85</td>
</tr>
</tbody>
</table>

Table 5-3: VisiSize output

5.6.3 Analysis and Conclusion

The resulting particle size data can then be analysed statistically in either a simple MATLAB program used to quickly visualise the results, in Excel or within the VisiSize software which had simple graph-plotting capabilities. Figure 5-11 shows the distribution graph of particle diameters.
Dynospheres are manufactured to be 20μm ± 3%. The results showed that the VisiSize measured only 25% of the particles within the manufacturing tolerance. However, it did measure over 90% of the particles to be 20μm ± 6%.

There is a tendency for the analysis software to measure particles as smaller than their true size. This is particularly apparent when analysing images where there are many particles in the foreground and background. These are out of focus particles and cause the background colour to be a darker shade of grey than in a scenario where there are no particles in the background or foreground.

When the image is thresholded it is difficult for the software to distinguish between the dark grey background and the black particle. For this reason it is essential that the laser power is increased for densely populated samples. This validation experiment is a prime example of this. Although care was taken to avoid it, the results still show a slight trend of undersizing particle diameters.

Figure 5-11: Cumulative distribution of bubble size.
5.7 Concluding Remarks

This chapter has described the design and development of test apparatus for an automated photographic study of nucleation from boiling grids. Past investigations were discussed and the lessons learned were used to select the individual components of the apparatus used in this study.

The focus of the design was on the generation of accurate, fast and repeatable results. Human interaction was minimised by choosing a CCD camera and data logging system. A single central computer system was used to control all the components of the apparatus as well as recording and saving the results.

A validation experiment was carried out in a controlled environment to test the effectiveness of the system. Results were taken and analysed and were shown to give adequate accuracy and repeatability. The test apparatus was therefore shown to be appropriate for this research and boiling grids could therefore be manufactured and tested.
This chapter describes the development of test boiling grids and the experiments performed to prove the concept of using boiling grids to control and enhance nucleate boiling heat transfer. Fundamental laws of boiling are applied to the experimental results to produce a selection of boiling grids, which are to be tested in Chapter 7.

6.1 Boiling Grid Design Criteria

The two key dimensions affecting the design of boiling grids are the cavity diameter and spacing. These are known to have a significant affect on both the nucleation and heat transfer characteristics and therefore must be considered before the experimental boiling grids are designed.
The optimising of these cavity dimensions is determined by the thermodynamic conditions in which the boiling grid will operate. In this study the aim was to use the grid in an engine cooling system that could be exposed to varying system parameters. Table 6-1 shows approximate design values for heavy-duty diesel engine cooling systems such as those used to power trucks or off-highway vehicles. One of two cooling systems (A or B) would be used, depending on the size, application and type of engine.

<table>
<thead>
<tr>
<th></th>
<th>System A</th>
<th>System B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Cap Setting (kPa)</td>
<td>50</td>
<td>100</td>
</tr>
<tr>
<td>Max Coolant Temperature (°C)</td>
<td>103</td>
<td>110</td>
</tr>
<tr>
<td>Radiator ΔT (°C)</td>
<td>6 (min 3 - max 10)</td>
<td></td>
</tr>
<tr>
<td>Coolant Saturation Temperature (°C)</td>
<td>111.37</td>
<td>120.23</td>
</tr>
<tr>
<td>Thermostat fully open at (°C)</td>
<td>93</td>
<td>93</td>
</tr>
<tr>
<td>Max Duct Wall Temperature (°C)</td>
<td>160 - 200</td>
<td></td>
</tr>
<tr>
<td>Coolant velocity (m.s⁻¹)</td>
<td>2 (max 5)</td>
<td></td>
</tr>
</tbody>
</table>

Table 6-1: System operating parameters - Perkins Engine Company (2002)

Pressure caps determine the system pressure and therefore the saturation temperature. The radiator temperature difference is assumed to be 6 °C during the design of the cooling system. However, during operation this figure may vary from 3 - 10 °C, depending on ambient conditions. The duct wall temperature also varies as a result of the engine operating conditions.

Finally, coolant velocity varies throughout the cooling system depending on the cross-sectional area and position of the duct being studied. In general it can be assumed to be 2 m.s⁻¹, whilst in small ducts or areas of highest heat flux it could be found to be as high as 5 m.s⁻¹.
During the design stages of an engine cooling system, the velocity of the coolant is a parameter that the designer may use in an attempt to suppress areas of nucleate boiling. Upgrading or speeding up the water pump has been accepted as a simple and effective method for increasing coolant velocity to areas of high heat flux. Suppression of nucleate boiling has been studied by Thornecroft et al (1998) who suggested that the ratio of theoretical values, \( r_{c,\text{max}} \) and \( r_{c,\text{min}} \) (discussed in Section 3.3) was the controlling parameter and a low value of \( r_{c,\text{max}} / r_{c,\text{min}} \) meant boiling was suppressed. The calculation includes Reynolds number, which allows for the effect of fluid velocity.

The data contained in Table 6-1 was used to calculate a range of boiling grids that would adequately control boiling in a heavy-duty diesel engine cooling system. The following sections describe the design calculations used to develop a range of experimental boiling grids. These grids could be tested in laboratory conditions before being applied to a real engine.

### 6.1.1 Cavity Size

Griffith and Wallis (1960) developed an expression to describe the minimum active cavity size:

\[
r_{c,\text{min}} = \frac{2\sigma T_{\text{sat}} V_{f}}{h_{f} \Delta T}
\]

where \( r_{c,\text{min}} \) is the minimum active cavity radius, \( \sigma \) is the surface tension, \( T_{\text{sat}} \) the saturation temperature, \( V_{f} \) and \( h_{f} \) are the specific volume change during evaporation and the enthalpy of evaporation respectively, and \( \Delta T \) is the temperature difference.

This was later developed by Hsu (1962) to describe the range of active cavities:

\[
r_{c,\text{max}} = \frac{\delta}{2C} \left[ 1 - \delta \frac{\theta_{\text{sat}}}{\theta_{\text{wall}}} \right] + \sqrt{\left[ 1 - \frac{\theta_{\text{sat}}}{\theta_{\text{wall}}} \right] - \frac{4AC}{\delta \theta_{\text{wall}}}}
\]

Equation 6-2
where \( r_c^* \) is the critical cavity radius, \( \delta \) is the boundary layer thickness, \( C_1 \) and \( C_3 \) are constants, \( \xi_{\text{sat}} \) is the ratio of superheats, \( A \) is the surface area, and \( \theta_{\text{wall}} \) is the wall superheat. These equations were introduced in Chapter 3.

Using the conditions described in Table 6-1 a range of active cavities expected to boil under typical engine cooling system conditions were calculated. These results are shown in Table 6-2.

<table>
<thead>
<tr>
<th>( r_{c,\text{min}} ) (( \mu \text{m} ))</th>
<th>50</th>
</tr>
</thead>
<tbody>
<tr>
<td>( r_{c,\text{max}} ) (( \mu \text{m} ))</td>
<td>250</td>
</tr>
</tbody>
</table>

Table 6-2: Range of active cavities to study

As this study aims to focus on boiling grids that can be used to control boiling in an engine cooling system, it was decided to limit the experimental investigation to cavities of this size.

### 6.1.2 Cavity Separation

Theoretical studies concerning boiling cavity separation were discussed in Chapter 3 and were found to have been limited to a few authors including Eddington et al (1977), Chekanov (1977), Judd and Lavdas (1980) and Calka and Judd (1985). The results of these studies suggested that the separation ratio, \( S/D_b \) (\( S \) is separation distance between cavities and \( D_b \) is the diameter of the bubble at departure) is the predominant dimension that controls cavity interaction. Although Chekanov (1977) disagreed, it has been agreed by all the other authors that the \( S/D_b \) ratio has the following effect on nucleation:
S/Db < 1 promotes nucleation
1 < S/Db < 3 inhibits nucleation
S/Db > 3 has no effect on nucleation

This information can be used to calculate a range of cavity separation distances, S, for boiling grids which may be used to control boiling in IC engines cooling systems.

Following a similar approach to that taken by Bhavani et al (2000), the diameter of the bubble at departure can be calculated from thermodynamic conditions and therefore the separation distance can be calculated for various S/Db ratios.

Using the engine cooling system parameters highlighted in Table 6-1 it is possible to calculate Db using Equation 6-4, suggested by Ruckstien (1963) and Equation 6-5 suggested by Cole (1967):

\[
D_b = \left[ \frac{\rho \alpha^2}{g(\rho_l - \rho_v)} \right]^{\frac{1}{3}} \left[ \frac{\rho L C_p \Delta T}{\rho_v h_{fg}} \right]^{\frac{1}{2}} \quad \text{Equation 6-4}
\]

\[
D_b = \phi \left[ \frac{\sigma}{g(\rho_l - \rho_v)} \right]^{\frac{1}{2}} \quad \text{Equation 6-5}
\]

where \( \rho_l \) and \( \rho_v \) are the densities of the liquid and the vapour respectively, \( \sigma \) is surface tension, \( h_{fg} \) is the latent heat, \( C_p \) the specific heat and \( \phi \) the contact angle. Averaging the results from Equation 6-4 and Equation 6-5 an estimate of bubble departure diameter can be made, as shown in Table 6-3.
Using the average value of $D_b$, the separation distance ($S$) between cavities can be predicted for boiling grids with various separation ratios.

### 6.1.3 Surface Preparation

The final point to consider during the manufacture of boiling grids is the preparation of the surface before the cavities are laser drilled. To control nucleation from a surface, cavities of a known size and spacing were precisely positioned to initiate boiling. This would not be possible if the surface already had a number of naturally occurring cavities.

To remove all naturally occurring cavities capable of initiating boiling, the entire surface area was polished on a rotary table polisher, to a mirror finish. This made any natural cavities on the surface too shallow to trap vapour, i.e. less than $0.1 \mu m$ in depth, which is a magnitude smaller than the limit suggested by Corty and Foust (1955).

Figure 6-1 and Figure 6-2 are surface plots of the near mirror finish produced by the rotary table polisher. These images were taken by a Zygo white light interferometer 3D surface profiler which measures variations in the reflected light intensity from the surface.
Figure 6-1 gives a cross sectional profile of the surface in a number of different planes and shows that surface cavities are no deeper than 0.4 μm. Imperfections of this size are not large enough to trap vapour and therefore would not act as nucleation sites. The system calculated the surface to $R_a = 0.119 \, \mu m$ and $R_t = 0.61 \, \mu m$. These values agree with the results taken on a traditional surface profiler shown in Figure 6-3.
Laser machined cavities with these predetermined characteristics were then manufactured on the prepared polished surface.

6.2 Proof of Concept

Initial work was carried out on a line of 12 laser machined cavities, as shown in Figure 6-4, varying in diameter from 25 to 300 μm.

Figure 6-4: Line of cavities, diameters 25 to 300 μm, depth 300 μm
Operating with demineralised water at atmospheric pressure, the boiling process was recorded using the CCD camera. To make results more applicable to practical solutions, the water was not degassed. McAdams et al (1949) used degassed water in an experimental investigations of nucleate boiling. The dissolved gasses increasing, due to nucleation, was found to increase the heat transfer coefficient in sub cooled boiling. It was also noted that the contact angle reduced with time as dissolved gasses left the liquid. It is therefore recognised here, that degassing liquids can have a significant affect on nucleation. However, to keep the results applicable to IC engine cooling systems, filtered water was used.

Figure 6-5 shows that bubbles formed in the mouth of each cavity and that boiling did not occur on the polished areas of the heated surface. This proved the effectiveness of laser machined cavities to become boiling centres and showed that polishing the surface to a near-mirror finish removed all naturally occurring cavities capable of initiating boiling.
6.3 Development of Cavity Design

The previous section looked at the basic concept of boiling grids. Boiling was proved to initiate from the cavities whilst the polished surface remained free from bubbles. The study therefore investigated boiling from laser-machined cavities of a known shape and size. Boiling grids were then designed and tested in the boiling observation chamber.

It has previously been suggested that cavity size is the key factor which affects nucleation sites. A range of cavity diameters was calculated, created and tested and the results were used as the start of this experimental study. The following sections discuss the development of boiling cavities and their effectiveness.

6.3.1 Rate of Nucleation at Active Cavities

Preliminary testing of laser machined cavities of various sizes showed variations in boiling activity at different sized cavities. A single heater block with four varying grids of cavities was used as a simple test to evaluate the effect of cavity diameter on bubble departure data.

Figure 6-6 shows four grids machined into a single polished heater surface. Each grid has a different cavity size, varying between 80 and 270 μm in diameter, in grids of six by seven lines.
Figure 6-6: Four boiling grids of cavity diameters 80 to 270 µm

Using the CCD camera and digital image analysis software, the departure frequencies of bubbles from the laser machined cavities were measured, the results of which are shown in Figure 6-7.

Figure 6-7: Effect of cavity diameter on departure frequency

Figure 6-7 shows that departure frequency decreased in larger cavities, but that the departure frequency increased with heat flux. A decrease in
superheat, caused by the reduced heat flux, causes the bubble departure frequency to decrease. It is suggested that this occurs because the waiting time \( t_w \) and the bubble growth time \( t_g \) both increase and as Equation 6-6 shows that the departure frequency \( f_b \) will therefore decrease:

\[
f_b = \frac{1}{(t_w + t_g)}
\]

Equation 6-6

Singh et al (1977) found departure frequencies from laser machined cavities to be lower than departure frequencies from the naturally occurring cavities investigated by Tolubinsky and Ostrovsky (1966). This work confirms the findings of Singh et al (1977). The departure frequency was found to be in the region of 80-120 Hz at a superheat of approximately 2°C. This compares to Tolubinsky and Ostrovsky (1966) who found natural cavities to boil at about 3 kHz. Variations in water quality may be a partial cause of these variations, whilst the cavity density and cavity manufacturing method may also have had some effect.

Singh et al's study of nucleate boiling of water from laser-drilled cavities at atmospheric pressure predicted frequencies increasing from zero to 2.5 kHz at 10°C superheat. Their experimental results showed lower values of approximately 1 kHz. The results of this study at 2°C superheat fit with these results and are significantly lower than those from naturally occurring cavities. Singh et al suggested that natural cavities are smaller than laser machined cavities. They also suggested that the sharp mouth radii of the laser machined cavities may contribute to this effect because the rim would affect the bubble release mechanism.

6.4 Bubble Departure Size and Cavity Separation

In chapter 5, equations by Ruckstien (1963) and Cole (1967) were used to estimate bubble departure diameters, \( D_b \) (See Table 6-3). This value could then be used to assume \( S/D_b \) values for grids. Preliminary boiling results
were used to verify these calculations and then an appropriate spacing (S) of cavities in the boiling grids was selected.

Using the CCD camera, images like those shown in Figure 6-8 were generated. This video was reordered at 500 frames per second with the calibrated camera lens set to be 13.3 μm.pixel$^{-1}$. The images show a bubble growing and departing over a period of 20 msec.

![Figure 6-8: Sequence of boiling from a laser machined cavity](image)

The final frame, at $t = 0.020$ seconds, was downloaded to the BDIA system. Digital image analysis was then used to measure the diameter of the bubble at departure.

![Figure 6-9: Calculation of standard bubble diameter](image)
Figure 6-9 shows the diameter, $D_b$, to be 1.443 mm. This compared favourably to the calculated value in Table 6-3 which has been found to be 1.5 mm.

An experimental study by Bhavani et al (2000) calculated the expected bubble departure diameter, $D_b$, and therefore boiling grids were manufactured that had separation ratios of 1, 2 and 3. Using the $D_b$ value, the actual spacing for separation ratios 0.5, 1, 2 and 3 was calculated.

<table>
<thead>
<tr>
<th>Separation Ratio, $S/D_b$</th>
<th>1</th>
<th>2</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Separation (mm)</td>
<td>1.5</td>
<td>3</td>
<td>4.5</td>
</tr>
</tbody>
</table>

Table 6-4: Spacing equivalents

6.5 **Testing of Laser Machined Boiling Grid**

This chapter discussed the development and testing of laser machined boiling cavities including manufacturing tolerances, active cavity diameters, bubble departure diameters and separation distance. The results of these tests were used to design various boiling grids which could be mounted into the optical boiling chamber for evaluation.

In the following section, the knowledge gained throughout the study is used to design, manufacture and evaluate laser machined boiling grids.

6.5.1 **Boiling Grid Layout**

Each boiling grid was machined into individual polished aluminium blocks (as described in Chapter 5). In this study, the cavities were machined in an equilateral triangular grid formation as shown in Figure 6-10. This pattern was selected as it ensured that each cavity was equidistant from its six neighbours. The separation distance $S$ is shown in Figure 6-10.
The boiling grids were set out in a diamond shape as shown in Figure 6-11. This shape was chosen to allow the BDIA system to focus on a single cavity if required. This was achieved by focusing on a cavity at the point of the diamond shape, as there would be no cavities behind or in front of it. Having too many nucleating cavities in the background of an image was known to cause problems during thresholding. Secondly, using the diamond grid meant that at the periphery of the grid there were cavities with varying numbers of neighbours.

Figure 6-11: Example diamond grid form, 225 cavities 125 μm diameter and 300 μm depth
Each diamond shaped grid, covered the same surface area of the aluminium block, 440 mm². However, each grid had a different number of cavities on it because of the varying cavity spacing.

### 6.6 Grid Selection

Table 6-5 shows the four grids that were manufactured for this study.

<table>
<thead>
<tr>
<th>Grid Name</th>
<th>Cavity Size (μm)</th>
<th>Cavity Depth (μm)</th>
<th>Spacing (mm)</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>125</td>
<td>300</td>
<td>1.5</td>
<td>Diamond shaped grid</td>
</tr>
<tr>
<td>B</td>
<td>125</td>
<td>300</td>
<td>3</td>
<td>Diamond shaped grid</td>
</tr>
<tr>
<td>C</td>
<td>125</td>
<td>300</td>
<td>4.5</td>
<td>Diamond shaped grid</td>
</tr>
</tbody>
</table>

Table 6-5: Experimental boiling grid specification

### 6.7 Concluding Remarks

This chapter has described the design processes used in the development of boiling grids to be tested. Some departure frequency and bubble size measurements were taken in this chapter and compared to previously known results. These results have been used to design a number of boiling grids with varying dimensions. Separation ratios ranging from 1 to 3 was chosen. The next chapter tests each grid to find the optimum design.
7 Evaluation of Boiling Grid Technology

This chapter reports on the results of experiments on boiling grids designed to control and enhance nucleate boiling heat transfer. A number of grids were developed in Chapter 6 using both theoretical calculations and preliminary experimental results. Their effectiveness is evaluated and the effect of cavity crowning, grid dimensions and the heat transfer coefficient is investigated.

The automated photographic apparatus, comprising an optical boiling chamber in which the grids are mounted and a BDIA system to video and analyse the boiling regime, as described in Chapter 5, was used throughout this study.
7.1 Experimental Investigation

The experimental investigation into the control and enhancement of nucleate boiling heat transfer by the use of boiling grids can be broken down into a number of stages:

- Proof of concept (Reported in Chapter 6).
- Effectiveness of laser machined cavities (Reported in Chapter 6).
- Variations in grid dimensions.
- Optimum grid design.

Preliminary tests reported in Chapter 6 investigated the feasibility of the basic boiling grid concept by proving that nucleation occurs at cavities and that a mirror finish inhibits boiling. The manufacturing tolerances of laser drilling were also tested to prove the effectiveness of laser-drilled holes as nucleation sites.

The following sub-sections describe the test methods used to evaluate boiling grids.

7.1.1 Test Procedure

A test procedure was developed to evaluate the effectiveness of laser machined boiling grids. Each grid was therefore tested in an identical manner to allow the results to be compared.

To allow rapid heating of the liquid (water) the boiling chamber was connected to a pump and heated liquid bath. The system circulated water through the chamber until its temperature had been raised from ambient conditions to near saturation (this took approximately 30 minutes). The circulator pump and heated bath was not used during testing, at which point the cartridge heater was the only source of heat input.
Figure 7-1 shows a schematic of the testing profile used throughout this experimental study.

![Figure 7-1: Profile of testing time schedule](image)

The boiling chamber water was raised to a temperature of 98.5 °C by the circulating pump and heater system before the data logger was started. The circulator then held the system at 98.5 °C for 200 seconds (i.e. t = 0 seconds to t = 200 seconds) to allow the entire system to reach a stable temperature.

The circulator was then stopped, the boiling chamber vent valve was opened and the cartridge heater turned on. The temperature was seen to rise quickly before levelling at its equilibrium point. The system remained at its equilibrium point for a further 200 seconds (i.e. t = 250 seconds to t = 450 seconds) within which time all measurements were taken. This ensured that all measurements were taken prior to t = 500 seconds, at which point the heater was turned off.

**7.1.2 Benchmark Case**

All boiling grids were tested in identical conditions in order that comparisons could be made between results, including surface temperature
and heat transfer coefficient. A benchmarking study was also used during this investigation.

One test piece was polished to the same specification as all the other aluminium blocks \((R_a = 0.119 \, \mu m \text{ and } R_t = 0.61 \, \mu m)\), although in this case no cavities were machined into the surface. This was then used as a benchmark in order to gauge the effect of the laser machined grid cavities on the heat transfer regime.

![Figure 7-2: Plain polished surface without cavities](image)

Figure 7-2 is a photograph of the polished surface operating at 85 kW.m\(^{-2}\), similar to the tests on the boiling grids. A single nucleation site is seen at a surface imperfection created at what was found to be a small occlusion in the original aluminium billet.

### 7.2 Observation of Boiling Regime

The boiling grids to be used in this experimental study were designed in Chapter 6 and were machined into a polished aluminium surface using the Nd:YAG Q-switched laser system. This surface was the top face of the block housing the cartridge heater and the block was inserted into the base of the boiling chamber.
The chamber was flooded with demineralised water and the heater was turned on. Observation of the boiling regime from grids A, B and C (S/D₀ = 1, 2 and 3) are shown below in Figure 7-3, Figure 7-4 and Figure 7-5. These illustrative photographs were taken through one of the side windows of the chamber using a standard 35 mm camera.

Preliminary tests were designed to look at whether the concept of using boiling grids was effective. Tests similar to those described in Chapter 6 showed that boiling occurred at the laser machined cavities whilst the polished surface remained free of nucleation sites.

During these experiments, few measurements were taken as the aim was simply to prove that boiling grids nucleated effectively. However, even from these simple 35 mm photographs it is noticeable that the boiling regime was distinctly different on each grid.

Figure 7-3 shows the grid with the largest separation distance, S/D₀ = 3, on which there is very little nucleation. However, the vapour which is being
produced is being generated only from the laser machined cavities and not the polished surface.

Figure 7-4: Boiling from grid B: $S/D_b = 2$

Figure 7-4 shows the boiling grids with separation ratio $S/D_b = 2$. The boiling regime is similar to the first grid although there are more active nucleation sites.

Figure 7-5: Boiling from grid A: $S/D_b = 1$
The boiling regime seen in Figure 7-5, which shows the grid with separation ratio $S/D_{b_0} = 1$, is different from both the previous results. Large quantities of vapour are being produced and nucleation seems to be occurring across the entire grid.

The preliminary test of boiling grid effectiveness proved successful in validating the concept. However, it also showed that the separation ratio, $S/D_{b_0}$, has a significant affect on the number of cavities that become active. The results were all taken at an identical heater power of 112 Watts (i.e. $q = 85\ \text{KW.m}^{-2}$), yet results showed distinctly different boiling regimes.

### 7.2.1 Active Cavity Ratio

During preliminary observations it was noted than the number of active sites increased as the separation ratio, $S/D_{b_0}$, of the grid decreased. The boiling grids were all made up of cavities of 125 $\mu$m in diameter and each grid covered the same surface area. Therefore, the number of cavities in a boiling grid will increase significantly as the separation ratio decreases.

Although the grids had varying numbers of cavities, in order to keep the surface area constant, there is a notable variation in active cavity ratio, as seen in Figure 7-6.
The boiling grid which had an $S/D_b = 1$ had nucleation at all of its 225 cavities, equivalent to an active cavity ratio (ACR) of 100%, when:

$$ACR = \frac{N_a A}{n}$$  \hspace{1cm} \text{Equation 7-1}

where $N_a$ is the active site density, $A$ the surface area and $n$ the number of cavities.

This compared to the grid $S/D_b = 3$ which had only 5 active cavities from 25 sites, equivalent to an ACR of only 20%.

The reason why a grid with a small separation ratio has a high active cavity density is discussed at length later in this chapter since it is an important finding from this thesis. From this preliminary observation of boiling on laser machined boiling grids, it was noted that nucleation seemed to occur at some cavities more than others. This latter point is now discussed.
7.2.2 Physical Dimensions of Active Cavities

During preliminary studies, boiling grids were repeatedly tested. Results showed reasonable consistency of measurements of active site density. However, it was noted that during the repeated experiments that although the active site density remained constant, the cavities which were active changed.

Within the grids, some particular cavities consistently nucleated and others consistently remained inactive. The cavities that were highlighted as consistently active or inactive were therefore studied. *Active* cavities were those that nucleated in 100% of the repeated tests whilst *inactive* cavities did not nucleate at any point during testing. As mentioned earlier, grid A had an ACR of 100%, which means no cavities were inactive and therefore the cavities on this grid were not used in this analysis.

Measurements were taken of all active and inactive cavities from boiling grids B and C (S/Dc = 2 and 3). Using a microscope, it was possible to focus on one plane, before adjusting the focus to a second plane. The focus adjustment was then recorded. The crown height and cavity depth were both measured in this way. Using a reticle, the diameter of the cavities was also measured to ensure that they were all 125 μm.

All cavities measured were found to have 125 μm diameter ± 8%, the cavity depth, Lc was found to be 220 μm ± 20 μm (± 9 %) both of which are acceptable tolerances for laser drillings of blind holes.
Figure 7-7: Comparison of active and inactive cavities.

Taken through the microscope, Figure 7-7 shows two cavities, one which is active (left) and one which is inactive (right). The large rim of material which is seen to be out of focus on the inactive cavity is the crown and was measured as 30 µm in height. This compares to the cavity shown on the left, which had a small crown, measured to be 12 µm in height.

In Chapter 4 it was shown that one reason the Nd:YAG, Q-switched laser was chosen for this work was that it produced the smallest crowns. These measurements showed that the largest crown height, $L_{\text{crown}}$ was measured to be 30 µm. All the cavities measured had some residual crowning, the average $L_{\text{crown}}$ being 19 µm and the smallest being 12 µm.

The measurements taken with the microscope were confirmed using a Zygo surface profiler. Figure 7-8 and Figure 7-9 show the 3D surface profile and the cross-section respectively.
Figure 7-8: Zygo scan of the surface showing the crowning

Figure 7-9: Cross section of the cavity showing the crown to be 30 μm high

Although the microscopic measurements were known to be accurate, the Zygo surface profiler offered the opportunity to both visualise the size and shape of the crown and to confirm the measurements taken on the microscope. Checks were not carried out on all the microscope measurements as the Zygo was a slow technique, where as the microscope is significantly quicker and still provided the required accuracy. The cavity depth L_c was not checked on the Zygo as the equipment is not capable of
measuring cavities deeper than 58 \( \mu m \). As seen in Figure 7-9 the cavity wall cannot be traced all the way to the bottom of the cavity.

The effect of diameter on whether a cavity was active or inactive offered no trend, although it was noticed that the measurements of cavity depth and crown height did. Figure 7-10 shows the results of the microscope measurements of cavity depth and crown height. It was noted that the size of the cavity crown appears to affect its ability to act consistently as a nucleation site.

![Diagram showing crown height and cavity depth effects](image)

**Figure 7-10: Crown height and cavity depth effects**

Cavities with a large crown seemed to inhibit nucleation as they predominantly remained inactive, whilst cavities with smaller crowns tended to be active. Figure 7-10, however, suggested that a combination of both dimensions control a cavity’s ability to become active. Those that have
large crowns in proportion to their depth tend not to act as nucleation sites whilst deep cavities with smaller crowns do.

The linear equation attributed to the dividing line, shown in Figure 7-10, was found to be:

\[ L_{\text{crown}} = 0.288L_c - 50 \]  
Equation 7-2

where \( L_{\text{crown}} \) is the crown height (\( \mu \text{m} \)) and \( L_c \) is the cavity depth (\( \mu \text{m} \)).

Therefore it is suggested that if \( L_{\text{crown}} > 0.288L_c - 50 \), then the cavity is likely to remain inactive, whilst \( L_{\text{crown}} < 0.288L_c - 50 \) then the cavity is likely to be active.

### 7.3 Control of Nucleation

One of the main aims of this study was to consider the control of nucleation from a surface, particularly with a view to avoiding the film boiling which is known to insulate the surface and cause a significant increase in surface temperature. As described earlier, and shown in Figure 7-5, boiling grid A (i.e. \( S/D_b = 1 \)) nucleated from all of the laser manufactured cavities.

It is possible to suggest that a boiling grid that has 100% ACR could be mistaken for being a film boiling regime, as it would look similar. Film boiling is characterised by a layer of vapour covering the heated surface. This is caused by high quantities of vapour being generated from nucleation sites. Neighbouring cavities begin to merge to the extent that a layer of vapour is produced across the heated surface. This causes the surface to become insulated from the bulk liquid and causes the surface temperature to rise excessively.

However, grid C (\( S/D_b = 1 \)) shown in Figure 7-5, was studied closely and it became clear that the boiling remained in a discrete nucleation regime.
Figure 7-11 shows a magnification of the boiling from the grid and highlights the positions of the laser machined cavities which make up this boiling grid. It is seen that boiling occurs purely from the cavities and that the vapour columns do not merge until they are a significant distance from the surface. It therefore, can be concluded unequivocally that the boiling regime is nucleate boiling, as opposed to film boiling.

Figure 7-11 clearly shows that nucleation remained at discrete locations rather than developing into film boiling. This is important for cooling of surfaces such as engine cooling duct walls, because the generation of the vapour layer in film boiling insulates the surface and can cause the surface temperature to rise excessively. High wall temperatures cause physical deterioration or, in extreme cases, failure of the wall material and must therefore be avoided.

Figure 7-11 shows boiling generated by a heater power of 112 W, which is equivalent to 85 kW.m⁻². Further studies of this grid were carried out in stages up to 150 kW.m⁻² in order to evaluate the ability of the cavities to remain in control of the nucleation. Results showed that even at higher heat fluxes film boiling did not occur and that the boiling grid controlled the
boiling regime and ensured that it remained as nucleate boiling. This conclusively proved the potential of boiling grids to control nucleate boiling, and again highlighted the benefits this concept would have for IC engine cooling ducts.

7.4 Heat Transfer Coefficient Measurements

The essence of this work was aimed at improving the heat transfer coefficient (HTC) in IC engine cooling ducts. As HTC is measured in kW.m\(^{-2}\).K\(^{-1}\), any improvement offers engine designers the possibility of either increasing the amount of energy rejected from a known surface or rejecting a known quantity of energy from a smaller surface, both of which would be advantageous in engine design.

As discussed throughout this study, nucleate boiling is known to enhance the heat transfer coefficient of a surface. In testing the effectiveness of boiling grids this study focused in two areas, firstly a comparison between convective and boiling grid HTC, and secondly the variations of HTC in relation to cavity separation ratios. These two points are now discussed.

7.4.1 Convective HTC versus Nucleate Boiling HTC

Convective heat transfer is the movement of heat from a solid to a fluid without a phase change i.e. without boiling. During nucleate boiling heat transfer, vapour bubbles grow and depart from surface cavities, causing more energy to be transferred into the liquid and therefore creating a higher heat transfer coefficient.

A test surface was used to evaluate the difference in heat transfer between convection and boiling regimes. Boiling grid C which had a separation ratio \(S/D_b = 3\) was operated at a range of heat fluxes between 20 and 85 kW.m\(^{-2}\). Measurements of surface temperature and bulk liquid temperature were
recorded using the data logging equipment, the results of which are shown in Figure 7-12.

![Graph](image_url)

Figure 7-12: Temperature measurements

Below 35 kW.m\(^{-2}\) there was no phase change in the liquid, suggesting that it was in a convective heat transfer regime. This was confirmed by the bulk temperature measurement that was in the range 78 to 90 °C, below the saturation temperature, \(T_{\text{sat}}\). With an ambient barometer reading of 735.06 mmHg and therefore a bulk liquid pressure of 98 kPa the saturation temperature \(T_{\text{sat}}\) was known to be 99.03 °C. If the bulk temperature measurement remains below saturation temperature, it is unlikely that there will be any phase change.

Above 35 kW.m\(^{-2}\) bubbles were seen forming at, and departing from, a number of the laser machined cavities, suggesting the presence of a nucleate boiling heat transfer regime. The bulk temperature measurement was approximately equal to the saturation temperature, \(T_{\text{sat}} = 99.03 \, ^\circ\text{C}\), and therefore conditions were ideal for phase change in the liquid.
Finally it can be seen in Figure 7-12 that above 35 kW.m\(^{-2}\) the surface temperature rose steadily reaching 105.22 °C at 85 kW.m\(^{-2}\). The fact that the temperature rose steadily suggests that film boiling did not occur at the highest heat flux. Film boiling would have been accompanied by a rapid increase in wall temperature as the layer of vapour insulated the heated surface. This did not occur and therefore it can be assumed to have remained in a nucleate boiling regime.

Observations of the heater surface and boiling grid during the experiment allowed measurements to be allocated to either convective heat transfer or nucleate boiling regime. From the temperature measurements a heat flux curve can be plotted as seen in Figure 7-13.

![Graph showing heat transfer curve](image)

**Figure 7-13: Heat transfer curve**

This graph can be compared to the theoretical heat flux curve presented by Bejan (1993), show in Figure 7-14 and discussed in Chapter 1. It show similar results to Bejan (1993), placing the transition point to the nucleate boiling trend in the region or 10 - 100 kW.m\(^{-2}\) and 1 - 10 °C superheat. Other similarities to the theoretical heat flux curve are:
Convection is a straight line trend.

Boiling trend is a straight line.

The gradient is significantly steeper during the boiling than the convective regime.

The transition point occurred at positive superheat, not at 0°C.

Using the bulk temperature and the wall temperature, as shown in Figure 7-12, the heat transfer coefficient was calculated using Equation 7-3:

\[ Q = hA\Delta T \]  

Equation 7-3

where Q is the power input (W), h is the heat transfer coefficient (kW.m⁻².K⁻¹), A is the surface area (m²) and \( \Delta T \) is the temperature difference between the wall temperature and the bulk liquid temperature (°C). Figure 7-15 shows the results obtained.
The trend of heat transfer coefficient versus superheat shown in Figure 7-15 is similar to the heat flux curve. As superheat becomes positive, the heat transfer coefficient rises rapidly. Figure 7-13 shows that in this region nucleation begins and the heat transfer regime becomes that of nucleate boiling. The onset of nucleate boiling, as expected, causes a significant rise in the heat transfer coefficient from less than 5 kW.m\(^{-2}.K^{-1}\) to approximately 15 kW.m\(^{-2}.K^{-1}\) in the nucleate boiling regime.

The aim of this section of the study was to show the potential increase in heat transfer coefficient to be gained by nucleate boiling. In this sample case the boiling heat transfer coefficient was more than three times the convective heat transfer coefficient. This was as expected as previous studies have suggested that boiling could cause an increase in the heat transfer coefficient of between twice and ten times that of convection.

### 7.4.2 HTC Comparison Between Various Boiling Grids

The study of heat transfer coefficients discussed in this work, has to-date concentrated on the increase in HTC due to the transition from convection
into nucleate boiling. It has been shown that convective heat transfer has a magnitude of less than a third that of nucleate boiling, and this led to focusing on the effect that different boiling grid dimensions had on the enhancement of the heat transfer coefficient.

Various boiling grids were manufactured on their own individual, but identical heater blocks which were mounted in the base of the optical boiling chamber. K-type thermocouples, installed in the heater block, were used to measure the surface temperature of the boiling grid. The temperature of the surface and the bulk liquid were measured and recorded by the data logging system.

Each boiling grid was brought up to temperature by the circulator pump and water heater, before being heated solely by the cartridge heater. This was operated at 112 W, equivalent to 85 kW.m⁻². The benchmark aluminium block was polished to a near-mirror finish and had no laser-machined cavities. It was tested in an identical manner to that of the boiling grids. Therefore, results could be compared with a known benchmark.

The three boiling grids developed in chapter 5 each had identical surface areas and cavity diameter. The significant difference between them was the separation ratio of the cavities which was chosen to be $S/D_0 = 1, 2, \text{ and } 3$, following the technique used by Bhavani et al (2000). The equivalent separation distances were therefore 1.5 mm, 3.0 mm and 4 mm.

Each grid was tested at constant heat flux. Figure 7-16 shows that as the separation of the cavities increased, so did the surface superheat of the grid. The dotted line shows the superheat value of the benchmark surface to be 7.31 °C. As there are no cavities machined onto the benchmark surface, this ratio is undefined and could theoretically be described as having an infinite separation ratio (or possibly no separation ratio). It is therefore to be
expected that, as shown in Figure 7-16, the results tend towards the benchmark surface as separation value increases.

![Figure 7-16: Surface superheat vs. separation distance](image)

At a separation distance of 3 mm the superheat value is 6.17 °C, whilst at a separation distance of 1.5 mm, the superheat is 4.59 °C, a reduction of over 1.5 °C. This is a significant reduction in superheat for what is only a small change in the design of the boiling grid.

Using the results shown in Figure 7-16 and Equation 7-3 it was possible to calculate the heat transfer coefficient for the various boiling grids. The superheat value can be used as an indication of the cooling effectiveness, because a low superheat value suggests that the surface is being efficiently cooled and must therefore have a high heat transfer coefficient. The resulting heat transfer coefficients for each boiling grids are shown in Figure 7-17.
Chapter 7 - Evaluation of Boiling Grid Technology

Figure 7-17: HTC versus separation distance

The HTC at a separation distance of 1.5 mm was calculated to be 18.78 kW.m\(^{-2}\).K\(^{-1}\), which is an increase of 160% on that of the benchmark surface, which had a heat transfer coefficient of only 11.78 kW.m\(^{-2}\).K\(^{-1}\). Figure 7-17 clearly highlights the improvement in the heat transfer coefficient between the benchmark plain surface and the surfaces with cavities laser machined in them. This suggests that the grid with a cavity separation ratio \(S/D_b = 1\) has the lowest surface temperature and therefore the highest heat transfer coefficient.

7.5 Optimum Boiling Grid Design

The boiling grids were developed with the aim of both enhancing and controlling nucleate boiling heat transfer. The work has focused on the application of this in IC engine cooling ducts.

A number of different boiling grids were designed and tested, and have been shown to be an effective method of controlling and enhancing nucleate boiling. Tests have also shown that grids with various dimensions offer different results. This research work has centred on the effect of the cavity
spacing on the boiling regime, although early tests on various other parameters were used to design the grids used in this study.

Results of this experimental study into boiling grids conclusively showed that a spacing of $S/D_b = 1$, offered three key features that other grids did not:

- Surface temperature, $T_{wall} = 103.2 \, ^\circ C$
- Heat transfer coefficient, $h = 18.7 \, kW\cdot m^{-2}\cdot K^{-1}$
- ACR = 100 %

Each result is fundamentally linked to the next and together they offer significant benefit to IC engine cooling system designers by proposing a more efficient and controlled cooling mechanism and showing a greater understanding of nucleate boiling heat transfer. These three key factors are now discussed in more detail.

### 7.5.1 Surface Superheat

Figure 7-18 shows a comparison of the surface temperature of the different boiling grids used in this study. The dotted line shows the value of wall temperature for the benchmark study of heat transfer from a polished aluminium block without any laser-machined cavities.
These results show similar results to Figure 7-16 which shows a trend of superheat increasing with cavity spacing. When cavity spacing equalled 1.5 mm (S/Db = 1) the surface temperature was at its lowest, 103.2 °C. As the experiments were done at atmospheric pressure, the superheat during this experiment is similar, 4.6 °C (bulk temperature of 98.6 °C). This graph can be used as a gauge of the effectiveness of cooling. If the surface is poorly cooled i.e. has a low heat rejection, then the surface temperature will be high, as energy is unable to transfer into the liquid.

The advantage of a low surface temperature in an IC engine cooling system is that at high temperatures metal surfaces can degrade and even fail. Therefore, reducing the surface temperature is of benefit in terms of engine durability.

There is a difference between the surface temperature of grids A and C (S/D₀ = 1 and 3) of approximately 3 °C. Whilst this is a considerable variation in surface temperature, it is to be expected because of the significant difference in the number of active cavities on the different boiling grids.
7.5.2 Active Cavity Ratio

Figure 7-6 showed that there was a significant variation in the number of active sites on different boiling grids. Grid C with $S/D_b = 1$ had bubbles forming and departing from every cavity and is therefore shown in Figure 7-6 as having an ACR of 100%. This result was seen to be consistent, as every cavity on the grid nucleated in all tests performed with this grid. This differed from the other grids which not only had significantly fewer active cavities, but also had an active site density that varied between tests, resulting in a grid design that was not reliable.

Past work in the area of cavity interaction has offered numerous separation distances at which a cavity is believed to affect its neighbours in a positive manner. This value has been suggested by most investigators, including Judd and Chopra (1993) as a separation ratio of approximately unity or below ($S/D_b \leq 1$). The exact mechanics as to why such short separation ratios enhance boiling at neighbouring cavities yet to be proven. Judd and Chopra (1993) suggested that interseeding may be the cause.

In the following, interseeding is investigated and a new mechanism is proposed.

Figure 7-19 describes the mechanism that previous authors have suggested occurs during the interseeding of neighbouring cavities. A bubble grows out over its neighbour, causing vapour to become trapped in the other cavity, and therefore making it become active.
Figure 7-19: Classical/Previous understanding of the interseeding mechanism

One essential requirement for this to occur is that the bubble must grow along the surface in a dome shape. Figure 7-20 shows two different shaped bubbles. The first is a spherical bubble and the second is a dome shaped bubble. They are similar in volume and when a stem begins to grow as the bubble departs they will be have the same diameter. However, at this stage of bubble growth, when the cavity is in firm contact with the surface, they look and act differently.
The mechanism shown in Figure 7-19 relies on the bubble growing in domes, rather than spheres, as the contact patch of the bubble is significantly larger in the dome bubble than the spherical bubble. In Figure 7-20, the contact angle is the controlling factor of bubble growth type. A system using demineralised water and polished aluminium, as in this study, has a surface wetability and contact angle (the angles between vapour, liquid and solid) which tend to produce spherical bubbles.

Figure 7-21 shows the measurement of a droplet of water on the surface of the polished aluminium heater block. This image can be used to measure the actual contact angles in the system discussed here.
Figure 7-21 also shows the needle used to steadily increase the size of the droplet by injecting a known quantity of liquid into its centre. At each step, the contact angle between the solid and the liquid is calculated. Table 7-1 shows the results.

<table>
<thead>
<tr>
<th>Run Number</th>
<th>Droplet Volume</th>
<th>Contact Angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.99654</td>
<td>95.5</td>
</tr>
<tr>
<td>2</td>
<td>5.23952</td>
<td>96.4</td>
</tr>
<tr>
<td>3</td>
<td>6.64744</td>
<td>96.3</td>
</tr>
<tr>
<td>4</td>
<td>8.23355</td>
<td>96.9</td>
</tr>
<tr>
<td>5</td>
<td>26.5618</td>
<td>102.7</td>
</tr>
<tr>
<td>6</td>
<td>39.9558</td>
<td>101.4</td>
</tr>
<tr>
<td>7</td>
<td>52.7631</td>
<td>102.8</td>
</tr>
<tr>
<td>8</td>
<td>65.7549</td>
<td>103.6</td>
</tr>
<tr>
<td>Average</td>
<td></td>
<td>99.45</td>
</tr>
</tbody>
</table>

Table 7-1: Contact angle measurements

It is accepted that these results do not precisely model those that occur during nucleation from a heated surface, the significant difference being that
during nucleation the liquid is at a raised temperature and is not ambient. However, Sasges and Ward (1998) showed that the liquid-vapour-solid contact angles of water on a smooth, homogeneous, non-dissolving solid was negligibly affected by temperature change in the range with which this work is concerned.

The average contact angle measurement between liquid and solid was found to be approximately $100^\circ$. It should be noted that the droplet in Figure 7-21 is water surrounded by air, whereas in nucleation the droplet is a bubble containing air and surrounded by water. The solid to liquid contact angle is $99.45^\circ$ and therefore the solid to gas contact angle is $180^\circ - 99.45^\circ = 80.55^\circ$. As this value is less than $90^\circ$ the bubbles form at the heated surface in a sphere instead of a dome, as highlighted in Figure 7-20 and are therefore unlikely to stretch over to neighbouring bubbles because they grow up away from the surface, instead of out across the surface. This is confirmed by images taken by the CCD camera (refer to Figure 6-8.)

However, results from boiling grid C still suggested that there was some positive interaction between neighbouring cavities, although dome bubble interseeding is thought to be unlikely due to the contact angle. Reviewing images of boiling from grid C, it was noticed that bubbles interacted between cavities by sliding across the surface to a neighbouring cavity before departing.

Figure 7-22 shows pictures of the development and departure of bubbles during nucleate boiling from two adjacent laser machined cavities. By using a CCD camera, it is possible to observe that some bubbles departed from the surface by sliding along the surface to the next cavity. When this occurred, a trace of vapour was left in the cavity, enabling it to start boiling.
It was seen to occur across the whole of the boiling grid at very high speed. Figure 7-22 covers a time span of only 0.012 seconds and accounts for the boiling grid nucleation from 100% of its cavities. The advantage offered by this mechanism is that it causes a reliable repetition of the heat transfer effect, as all the cavities boil each time. This mechanism is highlighted in Figure 7-23.
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Bubble grows in Cavity 1, then slides across the surface to Cavity 2

Bubble departs Cavity 2, leaving some vapour

Cavity 1 & 2 are Active

Figure 7-23: Proposed new interseeding mechanism
Chapter 7 - Evaluation of Boiling Grid Technology

7.5.3 Heat Transfer Coefficient

Figure 7-15 clearly shows the benefits of using nucleate boiling as a method of increasing heat transfer from a surface. Work has been undertaken to evaluate the difference in performance between various boiling grids and Figure 7-17 shows the heat transfer coefficient on grid S/Db = 1, to be the highest. The benchmark surface was calculated to be 11.7 kW.m⁻².K⁻¹. This compared to 18.7 kW.m⁻².K⁻¹ for boiling grid C which has a separation ratio, S/Db = 1.

As discussed in the previous chapters of this study there are many reasons why the heat transferred during nucleate boiling is greater than that transferred during convection. The most significant reason is that the departing bubbles cause turbulence in the water which increases the convective proportion of the heat transfer coefficient significantly.

The most significant past work in the area was by Chen (1966) who proposed, that the overall heat transfer coefficient, h was made up of the heat transfer coefficient for both convection, \( h_{\text{conv}} \) and nucleate boiling, \( h_{\text{boil}} \). i.e.;

\[
h = h_{\text{conv}} + h_{\text{boil}} \quad \text{Equation 7-4}
\]

The turbulence due to bubbles departing causes the value of \( h_C \) to increase significantly. As the boiling grid with S/Db = 1 has a considerably higher number of active cavities it is not surprising that it also has a significantly higher heat transfer coefficient. The larger separation distance, S, creates less active nucleation sites. Figure 7-24 shows that at smaller separation distances the heat transfer coefficient is higher.
7.6 Application to IC Engines

In this work, the design of boiling grids has been aimed at IC engine cooling ducts although the results showed that they could offer benefits in many other applications. Experimental results reported so far have been for water, although this is liquid would rarely be used in a real-life engine cooling system.

As a preliminary experiment, a single test was performed using a mixture of water and Ethylene Glycol (EG), commonly used as a coolant. (i.e. Halfords Anti Freeze and Summer Coolant).

A standard engine coolant was chosen as a source of EG, the content of which is shown in Table 7-2.
<table>
<thead>
<tr>
<th>Component</th>
<th>% wt</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ethylene Glycol</td>
<td>90-97</td>
</tr>
<tr>
<td>Additives</td>
<td>3-10</td>
</tr>
</tbody>
</table>

Table 7-2: Table of coolant content - Halfords (2003)

As shown, the majority of the coolant is EG, the remaining fraction is comprised of additives, including corrosion inhibitors and bittering repellent agent (designed to avoid accidental swallowing). In this brand of coolant, OAT (Organic Acid Technology) is used as the corrosion inhibitors.

![Figure 7-25: Effect of BG on heat transfer - Benjamin and Westwater (1961)](image)

Previous work by Benjamin and Westwater (1961) studying the effect of water and EG mixtures on heat transfer is summarised in Figure 7-25. They found that the heat transfer coefficient was found to decrease as the quantity
of EG was increased. A small anomaly showed that, rather than pure EG having the lowest heat transfer coefficient, it actually occurred with a mixture including 5.5% water. Although an interesting result, this is not applicable to IC engines, which typically operate in the range of 50% to 100% water.

A 50:50 mixture of water and coolant was used in the boiling apparatus, as shown in Figure 7-26. This was then tested using the same methodology to previous experiments reported in this thesis. Thermocouple readings were used to measure surface temperature, $T_{\text{wall}}$ and the bulk fluid temperature $T_{\text{bulk}}$ and heat flux was recorded.
Figure 7-27: Temperature readings for water and EG mixture

EG is known to have a boiling point of 135 °C and so (using standard boiling point evaluation formulas) the boiling point of water and pure EG in an 50:50 mixture (without additives) was calculated to be 108.3°C, Redpulsar (2003). Using this, and the reading taken from the apparatus, the heat flux curve in Figure 7-28 can be drawn.

Figure 7-28: Heat flux curves for pure water and for 50% water and 50% Ethylene Glycol (EG)
Figure 7-28 shows a comparison between heat flux curves for water and a 50:50 mixture of water and EG. At a heat flux of 85 kW.m\(^{-2}\) the \(h\) value was found to be 14.34 kW.m\(^{-2}\).K\(^{-1}\) for water and 11.45 kW.m\(^{-2}\).K\(^{-1}\) for water and EG. As expected, the EG inhibited the heat transfer coefficient.

Despite the overall reduction in heat transfer coefficient when the EG was added to the water, the laser-machined cavities of the boiling grid operated as they had done for pure water, and the heat transfer coefficient was still significantly higher than for pure convection. For example, using the grid A \((S/D_b = 1)\), nucleation occurred at 100% of the laser-machined cavities, and despite a further increased heat flux, film boiling did not occur. These results showed the potential of laser-machined boiling grids to operate as desired in an engine cooling system.

7.7 Concluding Remarks

This research has established that surfaces with cavities act as nucleation sites whereas highly polished surfaces without any cavities do not initiate nucleate boiling.

It has also been shown that the spacing between cavities and their relationship to each is an important factor to consider. The design of a grid of laser-machined cavities on a highly polished surface influences the efficiency of the cavity grid in creating and controlling nucleate boiling. A new interseeding mechanism has been observed which explains why optimally spaced cavities can be found to provide fully active boiling grids.

Observations and testing of the grids, which were subjected to detailed analysis with sophisticated computer software, has highlighted the potential offered by choosing a specific design of boiling grid.
Conclusions and Suggestions for Further Work

The research presented in this thesis has studied the novel use of laser-machined grids of cavities for the control of nucleate boiling for application to IC engines. The grids are surface configurations comprised of laser-machined cavities in a polished metal surface. The research has considered their contribution to the control of boiling and the enhancement of heat transfer in combustion engines cooling ducts.

Boiling grids were shown to enhance the heat transfer coefficient by up to three times between a hot surface and a liquid coolant because nucleation occurred at each laser machined cavity site, creating local turbulence. At the same time, the position, pattern and relationship of the cavities to each other controlled the boiling regime, particularly by deterring it from changing from nucleate boiling into film boiling.
A comprehensive experimental programme was completed using various boiling grid and cavity designs to generate greater understanding of the parameters which were likely to effect the performance of boiling grids. The optimum grid design was then selected and studied further.

This final chapter summarises the major findings and conclusions generated by this study of the control of nucleate boiling by the use of micro-machined surface features. Recommendations for future work are also presented.

8.1 Conclusions

This research has resulted in new techniques for the control of nucleate boiling. It has also contributed new insights and understanding of nucleate boiling at a fundamental level.

The important conclusions of this research are as follows:

1. The research has demonstrated that nucleate boiling can be controlled by optimally sized and spaced laser-machined cavities in a heated metal surface.

   Although some types of boiling grids, which were not laser machined, have been developed previously, there has been no published information in regards to laser-machined boiling grids, or their application to IC engine cooling systems. This research, therefore, has resulted in a new technology in this area.

2. Previous investigations have suggested that nucleation sites on metal surfaces are found at surface cavities or imperfections. Therefore, in order to eliminate nucleation from these random cavities and imperfections on the boiling surface, the aluminium surface was polished to a near-mirror finish of Ra ≈ 0.1 μm before the laser-
machined cavities were manufactured. Throughout the testing and the evaluation of the various boiling grid designs, the polished surface always remained free of nucleation sites, justifying the effectiveness of polishing.

3. During the laser-machining of cavities, the molten aluminium, which had been removed during the process, was found to solidify around the rim of the hole, creating a cavity crown. This crown affected the ability of the cavity to act as a bubble nucleation site. When the height of the crown, compared to the depth of the cavity, was disproportionately large, it was less likely that the cavity would act as a nucleation site. Through experimentation, it is possible to establish a new relationship between the dimensions of the crown and the cavity can be defined in the form of an equation. Adopting the use of an Nd:YAG Q-switched laser system, minimised the crown height when machining the cavities.

4. A boiling grid that had cavities positioned at a separation ratio of greater than 1, ensured that the nucleation sites remained discreet. Increasing the heat flux to a significant level had no effect on the boiling regime which retained its nucleate boiling characteristics and avoided the transition to film boiling. This is an important finding, because when the transition to film boiling occurs there is an accompanying large increase in surface temperature, which is capable of damaging the heated surfaces of an engine.

5. Previous investigations have shown that a nucleate boiling heat transfer regime has a higher heat transfer coefficient than that of a convective heat transfer regime. This study has therefore suggested that nucleate boiling could be used to improve heat transfer in engine cooling ducts if boiling grids are used on the surfaces of these ducts to
control the nucleation process. Experimental results confirmed that the heat transfer associated with the nucleate boiling regime generated by boiling grids was significantly greater than that generated by pure convection.

6. The most significant increase in heat transfer associated with nucleate boiling is believed to be through turbulence in the liquid caused by bubbles departing from the heated surface. Therefore, the number of active cavities in a boiling grid was assumed to affect the heat transfer coefficient because more active cavities create more turbulence. This study has shown for the first time that there is a significant increase in the heat transfer coefficient of boiling grids with small separation ratios.

7. Tests were carried out using various boiling grid and cavity designs. Observation and measurements of nucleate boiling on each grid enabled the optimum design to be selected. A boiling grid that had cavities of 125 μm in diameter positioned at a separation ratio of 1 had an active cavity ratio (ACR) of 100% and had the highest heat transfer coefficient of all the grids evaluated. Boiling remained at discreet nucleation sites and ensured film boiling did not occur. This finding was found to be an optimum design for application to engine cooling ducts.

8. At a separation ratio of 1, the interaction between cavities was found to be optimum and favourable. At this particular spacing of the cavities, interseeding was likely to occur. Turbulence causes some bubbles to slide between cavities before the buoyancy of the bubble causes it to depart. When a sliding bubble reaches an adjacent cavity, it deposits some residual vapour into that cavity before departing from its new cavity in the normal fashion. As a result of this
interseeding, the new cavity is also activated. The sliding of bubbles across the heated surface from one cavity to an adjacent one and thereby activating it has not previously been observed and this is believed to be a new mechanism to explain site interaction.

A separation ratio of greater than 1 may also generate interseeding. This research has shown that a positive interaction takes place between adjacent cavities with a separation ratios of 1 but increasing this ratio to 2 resulted in the loss of nucleation of a significant number of the cavities. Therefore, a ratio of 2 is too large to allow interseeding to occur.

9. Most of the experiments were carried out using water as the boiling liquid. However, in an engine cooling system, the engine coolant is rarely made up of 100% water. Some experiments were therefore performed using a mixture of 50% water and 50% ethylene glycol. These results showed that the laser-machined boiling grid technique activates nucleate boiling in the same way as in the 100% water-based experiments.

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. The understanding of nucleate boiling fundamentals has been improved by this study, however, it has also highlighted a number of areas in boiling grid knowledge that could be further investigated. These include an extended study of the effect of cavity shape, variations in cavity pattern and cavity depth.
2. This new technology has been developed with IC engine cooling systems in mind. Consideration of potential cavity erosion over prolonged periods of operations should therefore be assessed. Before boiling grids can be introduced to engine cooling systems, further research is needed into the effect of positioning these grids on the upper and side surfaces of ducts.

3. Further investigation of the new interseeding mechanism identified in this work is suggested. It is recommended that the boundaries of its application be researched as this study found it to occur at a particular separation ratio. The parameters that affect this mechanism, which may include liquid contact, surface polishing angle and cavity size, should be investigated.
References


French C.C.J. (1969) Taking the Heat off the Highly Boosted Diesel, *SAE 690463*


Halfords (2003) Technical Data Sheet - Summer and Winter Coolant


Perkins Engines Company (2002) *Private communication with: Middlemiss I.D.*


This appendix is a guide to the design and manufacturing techniques that can be used to develop a boiling grid suitable for operation in a particular system.

1) Identify the system parameters:
   a. System Pressure, $P$
   b. Bulk Fluid Temperature, $T_{\text{bulk}}$
   c. Expected Wall Temperature, $T_{\text{wall}}$

2) Identify the coolant and its properties:
   a. Saturation Temperature (Boiling Point), $T_{\text{sat}}$, at system pressure.
      - If using a water/ethylene glycol mixture, use standard boiling point evaluation formulas to gain $T_{\text{sat}}$. (Logan, 1996)
b. Coolant Surface Tension, $\sigma$.
c. Liquid Density, $\rho_l$.
d. Vapour Density, $\rho_v$.
e. Latent Heat of Vaporisation, $h_{fg}$.

3) Select the material of the heated surface.

**Calculation of contact angle**

4) Use a Video-Based Contact Angle Meter (DataPhysics OCA-20) to measure the solid-liquid contact angle, $\beta$ (through the vapour) of the chosen liquid and surface material chosen.

5) Calculate the solid-vapour contact angle, $\phi$ (through the liquid) using the equation, $\phi + \beta = 180^\circ$ (assuming an approximately flat heater surface).

**Choosing the Cavity Dimensions**

6) Use Hsu’s (1962) Equation 6-2 and Equation 6-3 to calculate values of $r_{c,\text{max}}$ and $r_{c,\text{min}}$.

7) Equation 3-4 can be re-arranged to give the activation temperature, $T_{\text{act}}$, of cavities $r_{c,\text{max}}$ and $r_{c,\text{min}}$ in size.

8) Select the cavity diameter that has an appropriate activation temperature, $T_{\text{act}}$ for just below the system wall temperature, $T_{\text{wall}}$.
   - i.e. Approximately $2^\circ\text{C}$ below expected surface temperature.
**Grid Dimensions**

9) Use Ruckstien’s (1963) Equation 6-4 and Cole’s (1967) Equation 6-5 to calculate the predicted bubble departure diameter. Average the results to calculate the predicted $D_b$.

10) The expected optimum spacing should be separation ratio of one. Therefore, assume the separation distance, $S = D_b$.

11) The grid pattern can be chosen to suit, however, an equilateral triangular grid, as shown in Figure 6-10, offers the benefit of all a cavity’s neighbours being the same distance away.

**Grid Manufacture**

12) Polish a sample of the material using a rotary table polisher (or similar) to polish to a near mirror finish (i.e. approximately $Ra < 0.3 \mu m$)

13) Using a Q-switched Nd:YAG laser, program required cavity dimensions and pattern into the rotary mirror control system.

14) Equation 4-1 or Equation 4-3 should be used to calculate the depth per pass capable by the laser. Therefore, the number of passes required can be calculate, noting:
   a. The cavity should normally be machined to have a depth greater than twice its width, (i.e. $L_c > 2D_c$)
   b. If passes the number of passes required is high (i.e. greater than 5) then reduce programmed cavity size by approximately 30 $\mu m$ (to allow for cavity over sizing, due to overheating)
**Laser Machining Validation**

15) Use a reticle and microscope to measure the cavity diameter \((D_c)\), cavity depth \((L_c)\) and crown height \((L_{crown})\). To confirm the laser parameters were correctly set.

- If found to be incorrect then adjust laser parameters and re-machine the test sample.

**Optical Validation**

16) Once machining in sample materials have been finalised then machine grid optical boiling chamber base plate.

17) Operating at expected conditions, use the VisiSizer (BDIA system) to measure \(D_b\). Use the CCD camera to check the calculated contact angle is approximately correct.

18) Replace the predicted \(D_b\) with the new measured value then re-do all the previous stages.

19) Using the temperature measurements taken during testing, calculate the heat transfer coefficient of the boiling grid.

**Final Stage**

20) Now that the design has been finalised and tested, manufacture the boiling grid into the required piece of material. (i.e. engine cylinder head cooling duct) and evaluate.