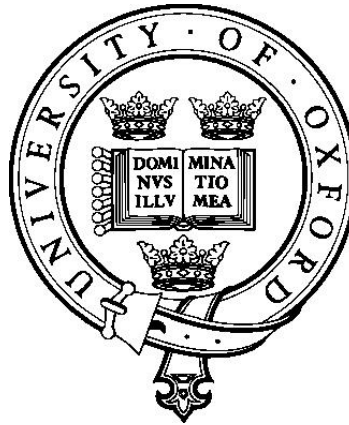




# Pool Boiling in Electrical Systems

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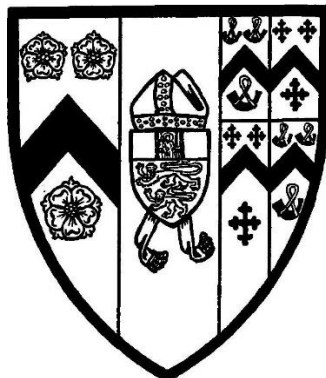
Supervised by

Professor Edmond J. Walsh

A thesis submitted in partial fulfilment of the requirements for the degree of Doctor of Philosophy in Engineering Science at the University of Oxford.

Trinity Term

MMXIX





*Pool Boiling in Electrical Systems*

Roy Sowdagar Bartle, Brasenose College

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## **Abstract**

This thesis presents the results of research into pool boiling in electrical systems, with particular application to stator end windings cooling. Pool boiling heat transfer characterisation of wire-wound motor windings geometry (MWG) surfaces was characterised, with MWG surfaces returning higher heat transfer even with area-compensation. Extant correlations were evaluated against both surfaces; the Rohsenow correlation with curve-fitting parameters was the best descriptor for both surfaces, whilst the Gorenflo-Kenning and unfitted Rohsenow provided good *a priori* predictive accuracy. Pool boiling of a heated vertical surface parallel to an unheated surface was shown to considerably enhance heat transfer coefficient at low heat flux, and considerably degrade heat transfer coefficient at high heat flux, when  $Bo < 0.58$ . Reducing liquid level below a similar level produced a marked deterioration in heat transfer from a horizontal surface. A semi-empirical model was developed to evaluate bulk void fraction for confined and unconfined mini-tube bundles. Heat transfer coefficient was shown to decline where void fraction exceeded a certain threshold. Exceeding the same bulk void fraction threshold anywhere in the column was shown to predict that dryout would occur first at the top tube in the column rather than at the bottom tube. The maximum power density of a mini-tube bundle was obtained around  $30 \text{ MW m}^{-3}$ . Bubble nucleator devices were shown to be effective at mitigating against bundle effect by activating dormant nucleation sites. These novel discoveries enable stator end windings geometry and packaging to be optimised to maximise pool boiling heat transfer, in turn allowing the design of machines with higher power density and efficiency and cooled by a system that has the potential for entirely passive operation.

## **Publications**

Four publications have resulted from this research hitherto:

1. Bartle, R. S., Menon, K. and Walsh, E. J. (2017) ‘Effect of Spatial Confinement and Liquid Level on Nucleate Boiling’, in *Proceedings of the 15th UK Heat Transfer Conference (UKHTC2017)*. Uxbridge, pp. 166–167.
2. Bartle, R. S., Menon, K. and Walsh, E. J. (2018) ‘Pool Boiling of Resin-Impregnated Motor Windings Geometry’, *Applied Thermal Engineering*, 130, pp. 854–864. <https://doi.org/10.1016/j.applthermaleng.2017.11.053>
3. Bartle, R. S. and Walsh, E. J. (2019) ‘Bubble Nucleators to Enhance Pool Boiling in Tube Bundles’, in *Proceedings of the 14th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics (HEFAT2019)*. Wicklow, Ireland.
4. Bartle, R. S. and Walsh, E. J. (2019) ‘Pool Boiling of Horizontal Mini-Tubes in Unconfined and Confined Columns’, *International Journal of Heat and Mass Transfer*, 145, 118733. <https://doi.org/10.1016/j.ijheatmasstransfer.2019.118733>

## Preface

I started surveying boiling literature from October 2015 whilst working in Zimbabwe. Work on this thesis began apace in January 2016, continued to September 2019, and was conducted mainly at the Oxford Thermofluids Institute (OTI), University of Oxford. Funding was provided by the Engineering and Physical Sciences Research Council (Grant N<sup>o</sup>. 1658609) and supplemented by Rolls-Royce plc (RR), to whom we are grateful.

Special thanks must be given to my supervisor, Prof. Ed Walsh, whose exacting standards and penchant for thorough scrutiny and refinement has emphasised to me the importance of rigorous scientific method in research techniques and outputs; basically, Ed's supervision has shown me how to conduct good research. Although I continue archaically to use the indefinite article "an" on more h-words than he would like, Ed's help as a mentor during the invariable struggles of doctoral life, and life more generally, has been much appreciated and enjoyed.

Various Rolls-Royce colleagues: firstly, Dr. Rory Stieger, my main industrial contact, and alongside whom I worked on secondment aided by the occasional packet of *droëwors*. I am thankful for Rory's encouragement and sense of perspective through my DPhil. Also to Dr. Kalyani Menon for developing the RR-OTI collaboration, and to Njål Rotevatn and Kolbeinn Kristjansson for stimulating applied conversations at the early stages of the research.

At OTI, Drs. Alex Murray and Nick Holgate, and Messers. Jack Lavender and Daniel Fahy, brought welcome spontaneity to the CDT Area. In Ed's Microfluidics Lab, the arrival in mid-2018 of Ensign Nick Stovall-Kurtz, United States Navy, to pursue a Masters' with us made the final months of my experimental programme an altogether more sociable pursuit.

I am thankful to my parents, Paul and Kulvant, for encouraging me to ask questions and pursue answers, and helping me immeasurably along the way. In 2017 I was joined in Oxford by Raffles, a blue British Shorthair (cat), whose soft sentience provides sure cheer after a long day. The London Congregation of the Free Presbyterian Church of Scotland was a spiritual home during my DPhil and I will miss our Lord's Day discussions over lunch.

Finally, I expect I shall have everlasting gratitude to the Triune God for His works of creation in all their complexity; and for the electing love of the Father, the redeeming love of the Son, and the sanctifying love of the Spirit. "This is a faithful saying, and worthy of all acceptation, that Christ Jesus came into the world to save sinners; of whom I am chief" (1Ti. 1:15).

Roy Bartle  
*Portvoller, Isle of Lewis*  
*September 2019*



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## Nomenclature

|           |                                       |                                  |
|-----------|---------------------------------------|----------------------------------|
| $A$       | area                                  | $\text{m}^2$                     |
| $BE$      | bundle effect                         |                                  |
| $BF$      | bundle factor                         |                                  |
| $Bo$      | Bond number                           |                                  |
| $C$       | spatial confinement                   | $\text{m}$                       |
| $CHF$     | critical heat flux                    | $\text{W m}^{-2}$                |
| $C_{s,l}$ | surface-liquid curve fitting constant |                                  |
| $c_p$     | specific heat capacity                | $\text{J kg}^{-1} \text{K}^{-1}$ |
| $d$       | diameter                              | $\text{m}$                       |
| $G$       | mass flow rate                        | $\text{kg s}^{-1}$               |
| $F$       | force                                 | $\text{N}$                       |
| $f$       | frequency                             | $\text{s}^{-1}$                  |
| $f_A$     | true-to-planar area ratio             |                                  |
| $g$       | acceleration due to gravity           | $\text{m s}^{-2}$                |
| $H$       | height                                | $\text{m}$                       |
| $h$       | heat transfer coefficient             | $\text{W m}^{-2} \text{K}^{-1}$  |
| $h_{fg}$  | latent heat of vaporisation           | $\text{J kg}^{-1}$               |
| $I$       | current                               | $\text{A}$                       |
| $i$       | counter, stator windings turn number  |                                  |
| $J$       | current density                       | $\text{A m}^{-2}$                |
| $j$       | counter, tube number                  |                                  |
| $K$       | coefficient                           |                                  |
| $K_f$     | fill factor                           |                                  |

|           |                                                            |                                   |
|-----------|------------------------------------------------------------|-----------------------------------|
| $k$       | thermal conductivity                                       | $\text{W m}^{-1} \text{K}^{-1}$   |
| $k_b$     | Boltzmann constant                                         | $\text{J K}^{-1}$                 |
| $L$       | length                                                     | m                                 |
| $M$       | molar mass                                                 | $\text{mol g}^{-1}$               |
| $m$       | mass                                                       | kg                                |
| $m$       | curve fitting index                                        |                                   |
| $MD$      | mean absolute difference                                   |                                   |
| $N$       | total number – of bubbles, datapoints, tubes, turns        |                                   |
| $Nu$      | Nusselt number                                             |                                   |
| $n$       | curve fitting index                                        |                                   |
| $n$       | index                                                      |                                   |
| $P$       | perimeter                                                  | m                                 |
| $Pr$      | Prandtl number                                             |                                   |
| $p$       | pressure                                                   | $\text{N m}^{-2}$                 |
| $Q$       | heat or power                                              | W                                 |
| $Q'''$    | heat or power density                                      | $\text{W m}^{-3}$                 |
| $\dot{q}$ | internal heat generation                                   | $\text{W m}^{-3}$                 |
| $q''$     | heat flux                                                  | $\text{W m}^{-2}$                 |
| $R$       | resistance                                                 | $\Omega$                          |
| $R$       | universal gas constant                                     | $\text{J K}^{-1} \text{mol}^{-1}$ |
| $R'$      | Bakhr-Lienhard characteristic dimensionless number         |                                   |
| $R_a$     | mean surface roughness                                     | m                                 |
| $R_c$     | mean height of profile irregularities of roughness profile | m                                 |

|           |                                                                             |                   |
|-----------|-----------------------------------------------------------------------------|-------------------|
| $R_{max}$ | maximum peak to valley height of roughness profile within a sampling length | m                 |
| $R_p$     | Glättungstiefe surface roughness by DIN 4172                                | m                 |
| $R_q$     | root-mean-square surface roughness                                          | m                 |
| $R_t$     | maximum peak to valley height of roughness profile                          | m                 |
| $R_v$     | maximum valley height of roughness profile                                  | m                 |
| $R_z$     | mean peak-to-valley height                                                  | m                 |
| Re        | Reynolds number                                                             |                   |
| $S$       | slip ratio                                                                  |                   |
| $SD$      | standard deviation                                                          |                   |
| $SE$      | sample error                                                                |                   |
| $S_m$     | mean spacing between peaks                                                  | m                 |
| $s$       | tube spacing                                                                | m                 |
| $T$       | temperature                                                                 | K                 |
| $t$       | time                                                                        | s                 |
| $u$       | velocity                                                                    | $\text{m s}^{-1}$ |
| $V$       | volume                                                                      | $\text{m}^3$      |
| $w$       | width                                                                       | m                 |
| $w$       | uncertainty                                                                 |                   |
| $x$       | axis dimension                                                              | m                 |
| $y$       | axis dimension                                                              | m                 |
| $z$       | axis dimension                                                              | m                 |

**Special Letters**

|               |                           |                                 |
|---------------|---------------------------|---------------------------------|
| $\Re$         | specific gas constant     | $\text{J g}^{-1} \text{K}^{-1}$ |
| $\beta$       | contact angle             | $^{\circ}$                      |
| $\varepsilon$ | void fraction             |                                 |
| $\zeta$       | dryness fraction          |                                 |
| $\mu$         | dynamic viscosity         | $\text{N s m}^{-2}$             |
| $\nu$         | kinematic viscosity       | $\text{m}^2 \text{s}^{-1}$      |
| $\rho$        | resistivity               | $\Omega \text{ m}$              |
| $\rho$        | density                   | $\text{kg m}^{-3}$              |
| $\sigma$      | surface tension           | $\text{N m}^{-1}$               |
| $\tau$        | temperature coefficient   | $\text{K}^{-1}$                 |
| $\varphi$     | surface inclination angle | $^{\circ}$                      |
| $\Psi$        | nucleation site desity    | $\text{m}^{-2}$                 |

**Subscripts**

|      |                      |
|------|----------------------|
| $A$  | area                 |
| $b$  | bubble               |
| $C$  | critical             |
| $c$  | characteristic       |
| $d$  | departure            |
| col  | column               |
| comp | comparative          |
| CSA  | cross-sectional area |
| Cu   | copper               |
| case | casing               |
| con  | conductor            |

|            |                         |
|------------|-------------------------|
| end        | end                     |
| flat       | flat                    |
| ins        | insulation              |
| <i>l</i>   | liquid                  |
| max        | maximum                 |
| MWG        | motor windings geometry |
| <i>NB</i>  | nucleate boiling        |
| <i>r</i>   | reduced                 |
| ref        | reference               |
| <i>s</i>   | surface                 |
| <i>sat</i> | saturation              |
| slot       | slot                    |
| <i>VPC</i> | vapour curve            |
| <i>v</i>   | vapour                  |
| <i>w</i>   | wall                    |
| $\infty$   | bulk                    |

# 1. Introduction

## 1.1. Background

Perhaps the overarching motive behind engineering research in the early 21<sup>st</sup> century is energy efficiency. This motive has many components including the traditional drivers of cost, performance, and reliability. Coupled to this, factors including pollution, climate and ecosystem change, resource scarcity, and environmental activism, are increasingly relevant. In 2018, the transportation sector was responsible for 28% of global energy use; within transportation, diesel fuels accounted for 27% and jet fuels 11% (U.S. Energy Information Administration, 2019).

Rolls-Royce plc operate in the transportation sector with core expertise in aviation propulsion; they also produce power systems for marine and rail applications. Throughout the transportation industry, a shift away from fossil fuel consumption exists due primarily to environmental and sustainability concerns and has redoubled interest into the electrification of propulsion systems. The ideal behind such systems is the production of electrical energy from renewable sources (Department of Energy and Climate Change, 2011), which can then be converted to kinetic energy by means of electrical motors.

Aircraft efficiency has increased considerably since the advent of commercial jets. Between 1960 and 2014, fuel consumption per passenger per unit distance decreased by around 50% (Kharina and Rutherford, 2015). Nonetheless, consistent with the trend throughout the transportation industry and driven by the same motivators, work on aircraft electrification continues apace. Figure 1 shows the history and projected growth of electrical propulsion programmes in aircraft. Mass and volume reduction are especially critical in aeronautical applications, resulting in electrical machines and circuitry with high power densities. By reviewing existing installed electrical capacity

and research programmes, the National Academies of Science, Engineering, and Medicine (U.S.) (2016) project that, by 2036, the power density of an electrical motor installed in an aircraft will be around  $9 \text{ kW kg}^{-1}$ .

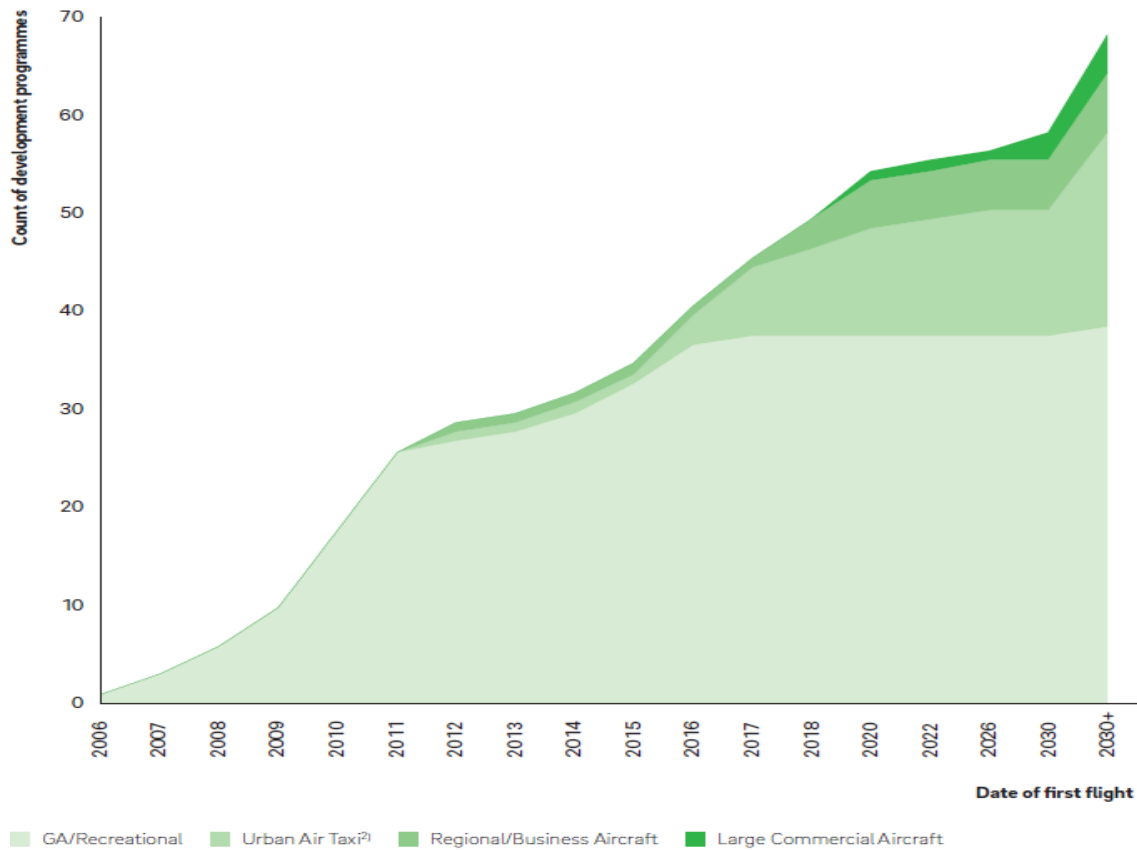


*Figure 1: History and development of electrical propulsion programmes in aircraft, by date of first flight (Thomson et al., 2017).*

Although advanced electrical machines are extremely efficient, invariably waste heat is generated at high heat fluxes from power dense electrical machines and their associated power electronics. Thermal management of waste heat is important for the following principal reasons:

- Some materials used in aircraft components have upper temperature limits that, when exceeded, cause immediate material failure (Mishra, 2004);

- Elevated temperatures and thermal cycling reduce the reliability of electrical components (Khazaka *et al.*, 2015); and,
- As the resistivity of most electrical conductors increases with temperature, increased temperatures result in decreased electrical machine efficiency (Aarniovuori *et al.*, 2018).

Thermal management solutions deployed in aircraft must accommodate the high heat fluxes generated and be highly reliable, whilst minimising system mass, volume, and power consumption.

## 1.2. Electrical Motors

Permanent magnet (PM) synchronous machines are presently being researched for propulsive (Donea and Gerling, 2016; Yoon *et al.*, 2016; Ismagilov *et al.*, 2019) and flight control (Villani *et al.*, 2012; Madonna *et al.*, 2018) applications. Electric propulsion systems for aviation were recently reviewed by Henke *et al.* (2018) and include fully-electric systems and jet-electric hybrid systems. The main features of a PM motor are diagrammed in Figure 2 and generally consist of an internal rotor to which are attached permanent magnets, and an external stator with slots (or teeth) through which wire or other conductors are run. Synchronous PM machines offer high efficiencies ( $> 95\%$ ) but the energy losses can still present considerable thermal management challenges, especially when the machine power density is high. Losses can be categorised as:

- Copper losses from Joule heating of the conductors in the stator slots;
- Iron losses, composed of hysteresis losses from changing the direction of the magnetic domains, and eddy currents from Lenz' opposing flux density;
- Windage losses from the drag of the rotor against the air in the air gap; and,
- Mechanical losses from bearing friction.

Typically, copper losses dominate, accounting for between 50-70% of total losses (Jianfeng, Ting and Jianming, 2011; Madonna *et al.*, 2018; Giangrande *et al.*, 2019; Ismagilov *et al.*, 2019), and are the particular focus for the electrical motor cooling aspect of this thesis.

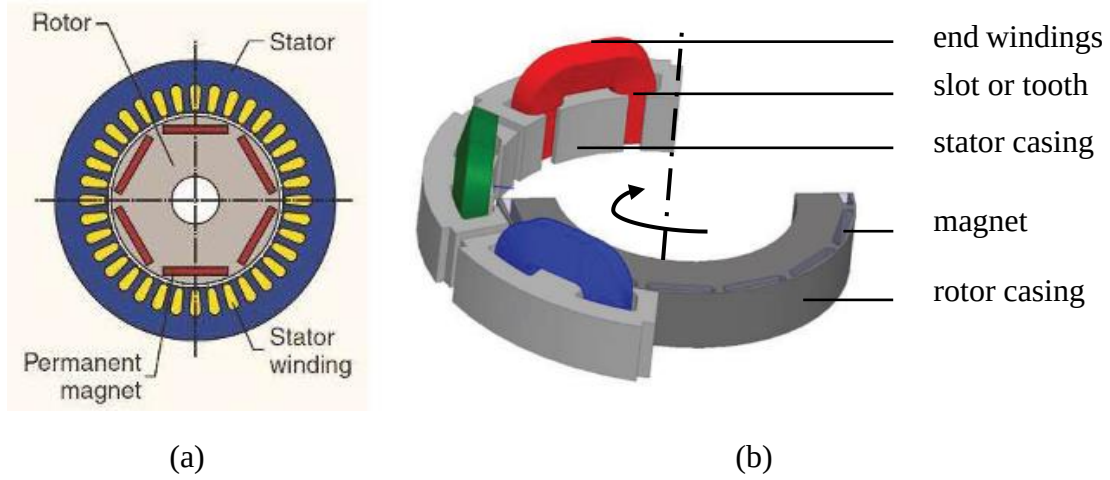


Figure 2: (a) Section diagram of a synchronous PM motor with magnets embedded in rotor (Jaszczolt, 2017); (b) 3D cutaway diagram of the end of a synchronous PM motor with magnets embedded in rotor (adapted from Donea and Girling (2016)).

Copper losses exist due to Joule heating of the stator windings, which must be dissipated away from the machine and ultimately to the environment outside the aircraft.

The heat generated due to Joule heating is

$$Q = I^2 R, \text{ where resistance} \quad (1.1)$$

$$R = \frac{\rho_{\text{ref}} [1 + \tau(T - T_{\text{ref}})] L}{A}. \quad (1.2)$$

Where the temperature coefficient is negligible, the expression reduces to

$$R = \frac{\rho_{\text{ref}} L}{A}. \quad (1.3)$$

Conductors are placed within the stator slots, with insulation preventing each conductor contacting another. The most common conductor design is to use circular profile wire windings in resin-impregnated stator slots, as shown in Figure 3(a), because

the manufacture and assembly involved is relatively straightforward and inexpensive. However, because of the desire to increase efficiency without reducing power density, research is turning to conductor geometries that increase copper fill factor and thus current density, as shown in Figure 3(b). The complex relationship between efficiency (in the form of copper loss reduction) and power density (in the form of slot area) is shown in the equation for copper losses, obtained by combining eqns. (1.1) and (1.2):

$$Q_{Cu} = \frac{I^2 \rho_{ref} \left[ 1 + \tau (T - T_{ref}) \right] L}{A_{slot} K_f} . \quad (1.4)$$

Decreasing the slot area provides a smaller machine, increasing power density for the same power output. However, assuming unchanged fill factor, the copper area also decreases, increasing copper losses and decreasing machine efficiency. A dichotomy therefore exists between power density and efficiency. Increasing fill factor without increasing slot area is therefore desirable to reduce copper losses without increasing the machine volume, hence the move from wire windings to edge wound conductors. However, an increased fill factor with the same copper area also means that the heat is transferred across a smaller area, increasing conductor temperatures. Cooling the machine is important because eqn. (1.4) shows that when the thermal coefficient is positive (as it is with copper), increasing material temperature increases copper losses and so decreases machine efficiency.



(Axalta Coating Systems, 2019)



(Fraunhofer IFAM, 2019)

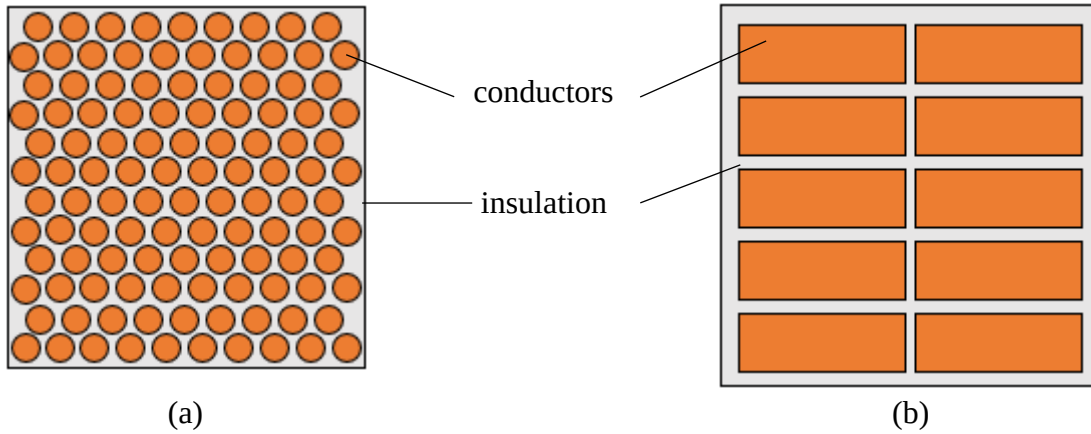


Figure 3: Photographs and cross-sectional diagrams of stator slots filled with (a) circular wire; and (b) high fill factor, edge wound conductors.

Existing efforts to cool the stator windings include cooling on the end windings by blown air (e.g. Ismagilov *et al.*, 2019) and liquid jet impingement (e.g. Davin *et al.*, 2015). Liquid jackets surrounding the machine have been developed (e.g. Volpe *et al.*, 2018). Liquid cooling within the stator has also been proposed: with channels around the stator slots (Schiefer and Doppelbauer, 2016), and in each winding turn (Wohlers *et al.*, 2018). Each method has certain merits. Blown air cooling simply involves channelling bleed air through a pump and/or fan onto the end windings, but extremely high pumping power is required to obtain high heat transfer coefficients. Liquid pools are similarly straightforward but, in natural convection conditions, heat transfer coefficient is modest; this increases somewhat in forced convection. Cooling on the end windings also limits machine length, as shown later. Liquid jackets outside the stator casing result in a more uniform axial temperature distribution but, as heat transfer coefficients are modest and

the thermal resistance through the stator windings and casing is typically high, the usefulness of liquid jackets is limited to relatively thin machines, as shall also be demonstrated. Popescu and Cavagnino (2015) and Henke *et al.* (2018) reviewed some of these methods.

Except for cooling channels in the stator, thermal management approaches generally use cooling methods on the outer surfaces of the machine to avoid affecting the electromagnetic properties of the motor. These approaches usually take advantage of one of two thermal paths: radially from the stator windings, through the casing, to the outer shell; or axially through the stator to the end windings. Next, Chapters 1.3 and 1.4 analyse these thermal paths.

### 1.3. Thermal Analysis of Radial Conduction Path

Considering first the radial thermal path through the stator slot and casing, as shown in Figure 4. Conduction takes place only in the direction of the  $x$  axis through the area  $A = wL$ . Internal heat generation  $\dot{q}$  ( $\text{W m}^{-3}$ ) occurs due to Joule heating and, for a known heat output  $Q$  (W) per turn and dimensions as diagrammed in Figure 5, is

$$\dot{q} = \frac{Q}{A\Delta x}. \quad (1.5)$$

Alternatively, where the electrical properties are known:

$$\dot{q} = \frac{I^2 R}{V}, \quad (1.6)$$

where  $V$  is volume. The heat equation with internal heat generation for the  $x$  axis only is

$$\frac{\partial^2 T}{\partial x^2} + \frac{\dot{q}}{k} = 0. \quad (1.7)$$

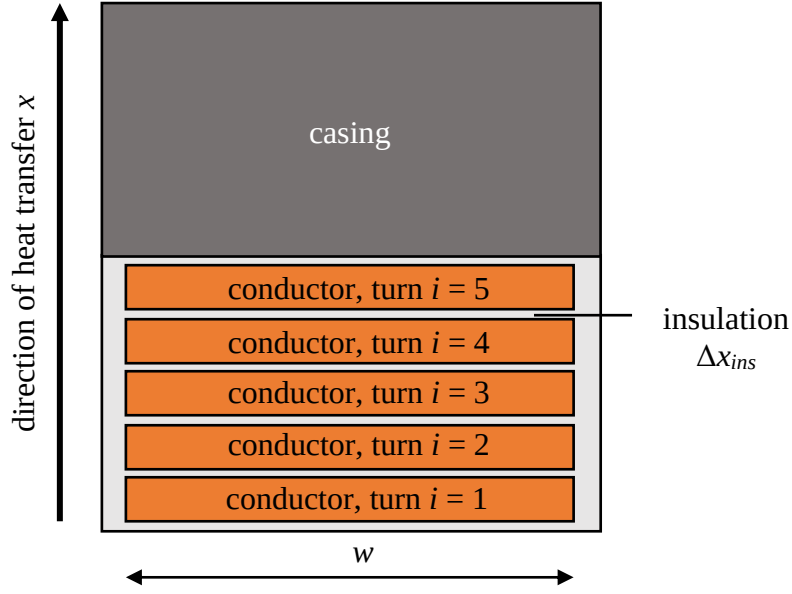


Figure 4: Diagram showing the radial thermal path of Joule heat generated by the windings in the stator slot.

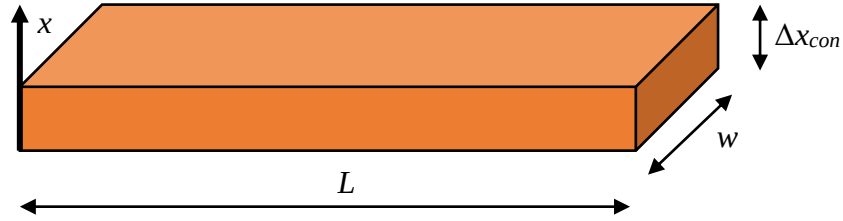


Figure 5: Nomenclature used in stator winding dimensions.

With full working in Appendix A, it can be shown that the temperature profile through a stator slot is

$$\Delta T_{1 \rightarrow i} = \frac{Q}{A} \left( i^2 \frac{\Delta x_{con}}{2k_{con}} + i \left( \frac{i+1}{2} \right) \frac{\Delta x_{ins}}{k_{ins}} \right). \quad (1.8)$$

Depending on the design of the stator windings, the top turn can comprise a half thickness of insulation rather than the full thickness between the other turns. Then, the temperature difference across the windings to the top turn  $N$  is

$$\Delta T_{1 \rightarrow N} = \frac{N^2}{2} \cdot \frac{Q}{A} \left[ \frac{\Delta x_{con}}{k_{con}} + \frac{\Delta x_{ins}}{k_{ins}} \right]. \quad (1.9)$$

In such a setup, eqn. (1.8) should be used to obtain the temperature profile through the stator turns where  $i < N$ , and eqn. (1.9) should be used when  $i = N$ . Where the top turn has a full thickness of insulation, eqn. (1.8) is used throughout  $i=1:N$ .

Finally, the temperature difference through the casing is simply

$$\Delta T_{case} = N \frac{Q}{A} \left( \frac{\Delta x_{case}}{k_{case}} \right). \quad (1.10)$$

From eqns. (1.8) and (1.9), it is observed that the temperature difference in the radial direction of the stator slot squares with the number of turns. Additionally, the thermal conductivity of insulation  $k_{ins}$  is typically around  $0.1 \text{ W m}^{-1} \text{ K}^{-1}$  and the thermal conductivity of the conductor  $k_{Cu}$ , usually copper, is  $390 \text{ W m}^{-1} \text{ K}^{-1}$ . The thickness of the insulation layer is typically around 10-20% of the copper thickness; the thermal conductivity of a silicone steel casing  $k_{case}$  may be in the order of  $20 \text{ W m}^{-1} \text{ K}^{-1}$  and thickness of the casing is typically 2-3x the stator slot height. Therefore, the thermal resistance of the insulation and casing layers dominate the radial thermal path, being each around 200-400x larger than the thermal resistance of the copper layers. Because of the effect of stator casing geometry on eddy currents (and so iron losses), simply reducing the casing thickness is seldom feasible. Additionally, whilst it is possible to optimise the number of turns and layout of the stator windings, selection of these parameters is usually performed at the initial stage of machine design, and in conjunction with the selection of magnet material and their locations. Hence a more integrated approach to the design process, simultaneously considering machine efficiency, power density, electromagnetic effects, and thermal management is required for this; as the author of this thesis is not an electrical engineer, this avenue has not been pursued further. Irrespectively, any slot (even an optimised arrangement) will contain several winding turns, separated by resin layers with high thermal resistance, outside of which is a casing with high thermal

resistance. Therefore, without significant multi-objective optimisation, thermal management solutions relying on radial heat conduction to the outer casing do not appear feasible for high power density machines.

#### 1.4. Thermal Analysis of Axial Conduction Path

The thermal path along the axial length of the stator windings is next considered. Neglecting the thermal coefficient of the material, substitution of eqns. (1.1) and (1.3) into (1.5) gives the internal heat generation  $\dot{q}$  due to Joule heating as

$$\dot{q} = J^2 \rho, \quad (1.11)$$

where current density

$$J = I / A. \quad (1.12)$$

Assuming heat transfer only in the machine axial axis  $z$ , the heat equation is identical in form to eqn. (1.7):

$$\frac{\partial^2 T}{\partial z^2} + \frac{\dot{q}}{k} = 0. \quad (1.13)$$

A full derivation of the succeeding eqns. (1.14) to (1.17) is given in Appendix B. If  $L$  is here defined as the half-length of the stator as shown in Figure 6, the maximum temperature in a stator is

$$T_{\max} = \frac{\dot{q}}{k} + \frac{L^2 \dot{q}}{k} - \frac{2L^2 \dot{q}^3}{k^3 (T_{w,1} - T_{w,2})^2} + \frac{T_{w,1} + T_{w,2}}{2}. \quad (1.14)$$

The effect of the heat transfer coefficient associated with the end windings cooling method can be incorporated into the analysis. Convective heat flux is

$$q'' = h(T_w - T_\infty), \quad (1.15)$$

where  $T_\infty$  is the bulk temperature of the cooling medium. However, the conduction and convection areas are not identical because end windings connect from one slot to the next,

and so the additional heat transfer area of the end windings is significantly larger than the conduction area. Ergo the conductor cross-sectional area  $A$  and the end windings heat transfer area  $A_{\text{end}}$  must be included into the equations. For temperature variation along a symmetrically cooled stator turn with end windings heat transfer coefficient  $h$ :

$$T(z) = \frac{\dot{q}}{k} \left( L^2 - z^2 \right) + T_{\infty} + \frac{\dot{q}LA}{hA_{\text{end}}}, \quad (1.16)$$

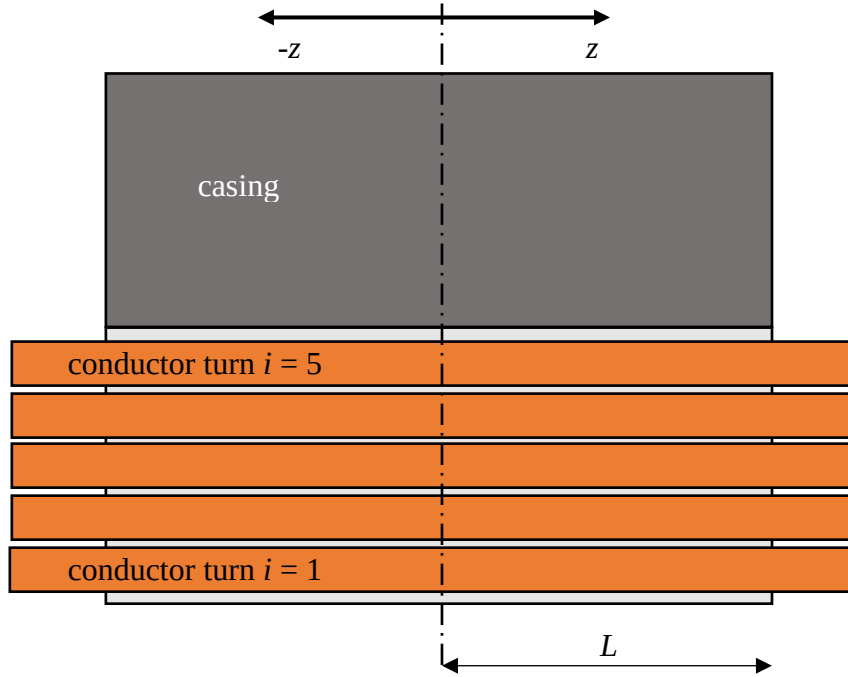


Figure 6: Diagram showing the  $z$ -axis conduction path along the stator conductors.

The maximum temperature expression similarly becomes

$$T_{\text{max}} = \dot{q}L \left( \frac{L}{k} + \frac{A}{hA_{\text{end}}} \right) + T_{\infty}. \quad (1.17)$$

Typically, the maximum permissible temperature in the stator windings is known and the relationship between machine length and maximum current is desired. Substitution of (1.11) into (1.17), and rearranging, provides the relationship:

$$I = \sqrt{\frac{A(T_{\max} - T_{\infty})}{\rho L \left( \frac{L}{k} + \frac{A}{hA_{\text{end}}} \right)}}. \quad (1.18)$$

The approach assumed in eqn. (1.11) is that internal heat generation  $\dot{q}$  is constant with axial position, which is a simplification; eqn. (1.2) shows that resistance is a function of temperature, which changes with axial position, and so internal heat generation is slightly dependent on axial position. When the thermal coefficient is included, the internal heat generation term becomes

$$\dot{q} = J \rho_{\text{ref}} [1 + \tau (T_{\max} - T_{\text{ref}})], \quad (1.19)$$

and eqn. (1.18) becomes

$$I = \sqrt{\frac{A^2 (T_{\max} - T_{\infty})}{\rho_{\text{ref}} L [1 + \tau (T_{\max} - T_{\text{ref}})] \left( \frac{L}{k} + \frac{A}{hA_{\text{end}}} \right)}}. \quad (1.20)$$

This provides an analytical solution for maximum current for a given machine design and stator cooling method, where the thermal coefficient of the material is important. However, a closed analytical solution for the temperature variation along the stator windings with axially-varying thermal conductivity is not possible. A somewhat unwieldy analytical expression can be obtained but must be solved iteratively. Ergo, where the temperature profile is desired, it is recommended to use eqn. (1.16) for an initial estimate, then employ an iterative 2D numerical algorithm to obtain the final temperature profile with axially-varying thermal conductivity included. Nonetheless, when real-world values are substituted, these equations show that axial conduction, utilising the high thermal conductivity of the stator windings material, is an attractive option for stator windings cooling.

Of the parameters that can be readily altered without electromagnetic side-effects, to achieve maximal cooling the bulk coolant temperature  $T_\infty$  should be minimised, and heat transfer coefficient  $h$  and end windings area  $A_{\text{end}}$  should be maximised. The bulk coolant temperature is selected by considering suitable fluids in the context of the whole cooling system, which is not considered here. Existing attempts to enhance the heat transfer coefficient with end windings cooling were outlined in Chapter 1.2. Natural convection into a liquid pool can be used as part of a passive cooling system but provides low heat fluxes, in the order of  $10^2 \text{ W m}^{-2} \text{ K}^{-1}$ . Liquid jet impingement provides acceptable heat transfer coefficients but, as the heat transfer enhancement is only over a limited surface area, an array of jets is required which increases the pressure drop and so required pumping power. Nucleate pool boiling is an heat transfer mechanism that combines the potential for passive, or low pumping power, operation with high heat transfer coefficients. As the coolant is changing phase through evaporation, the dominant thermofluidic property is the latent heat of vaporisation  $h_{fg}$  which is, for typical dielectric coolants suitable for use in electrical systems, around 100 times larger than the specific heat capacity  $c_p$  for the same fluid. A detailed literature review of pool boiling constitutes Chapter 2.

Maximising end windings area is the other method identified to enhance heat transfer. Figure 7(a) diagrams a typical end windings arrangement with edge wound conductors, separated by conductor layers, similar to Figure 6. In this geometry, noting that the end windings heat transfer area is shared between two slots (i.e. half the total area end windings area transfers heat from a single slot), the heat transfer area exposed to the fluid and pertaining to a single slot is

$$A_{\text{end}} = N \left[ \Delta x_{\text{con}} \left( \frac{w_{\text{end}}}{2} + L_{\text{end}} \right) \right] + w_{\text{end}} L_{\text{end}} . \quad (1.21)$$

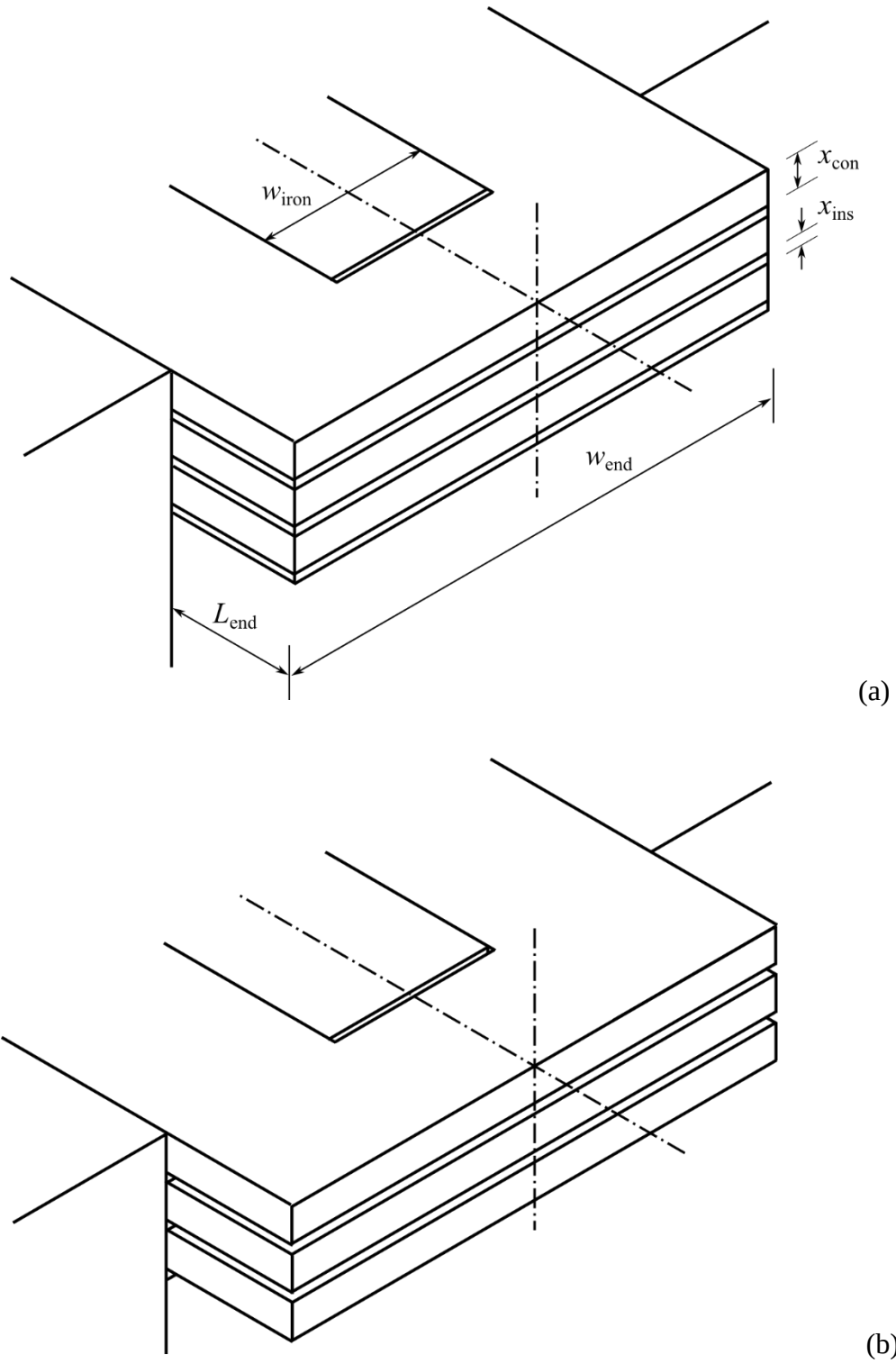


Figure 7: Cross-sectional diagrams of end windings area with (a) insulation layers, and (b) insulation layers removed.

However, if the insulation layers can be removed at the end windings, as shown in Figure 7(b), the heat transfer area potentially increases to

$$A_{\text{end}} = N \left[ \Delta x_{\text{con}} \left( \frac{w_{\text{end}}}{2} + \frac{w_{\text{iron}}}{2} + L_{\text{end}} \right) + w_{\text{end}} L_{\text{end}} \right]. \quad (1.22)$$

Removing the resin impregnation at the end windings of traditional round wire windings similarly increases the surface area potentially contacting the coolant.

By considering a simple case study, the effect of heat transfer coefficient  $h$  and end windings heat transfer area  $A_{\text{end}}$  on machine design is elucidated. The variables used in the case study are shown in Table 1. The effects of increasing heat transfer coefficient from a typical liquid natural convection ( $200 \text{ W m}^{-2} \text{ K}^{-1}$ ) to a typical liquid forced convection ( $1 \text{ kW m}^{-2} \text{ K}^{-1}$ ) and a typical mid-range pool boiling heat transfer coefficient ( $4 \text{ kW m}^{-2} \text{ K}^{-1}$ ), and the effect of increasing heat transfer area from the regular case to the bare end windings case, is graphed in Figure 8. For a machine half-length of 0.1 m, increasing heat transfer coefficient from 0.2 to  $4 \text{ kW m}^{-2} \text{ K}^{-1}$  allows more than 2.5x the current to be generated in the machine; removing the insulation between the end windings turns increases this to just under 3x. Or, for a 500 A current, the maximum allowable machine length increases 6.5x from the base case to the  $4 \text{ kW m}^{-2} \text{ K}^{-1}$ , insulation removed case.

*Table 1: Case study variables used to illustrate the effect of enhancing heat transfer coefficient and area.*

|                         |                       |                                         |
|-------------------------|-----------------------|-----------------------------------------|
| $k_{\text{con}}$        | 390                   | $\text{W m}^{-1} \text{ K}^{-1}$        |
| $w_{\text{con}}$        | 20                    | mm                                      |
| $\Delta x_{\text{con}}$ | 2                     | mm                                      |
| $L_{\text{end}}$        | 20                    | mm                                      |
| $w_{\text{end}}$        | 60                    | mm                                      |
| $w_{\text{iron}}$       | 20                    | mm                                      |
| $\tau$                  | 0.00386               | $\Omega \text{ m } ^\circ\text{C}^{-1}$ |
| $\rho_{\text{ref}}$     | $1.68 \times 10^{-8}$ | $\Omega \text{ m}$                      |
| $T_{\text{ref}}$        | 20                    | $^\circ\text{C}$                        |

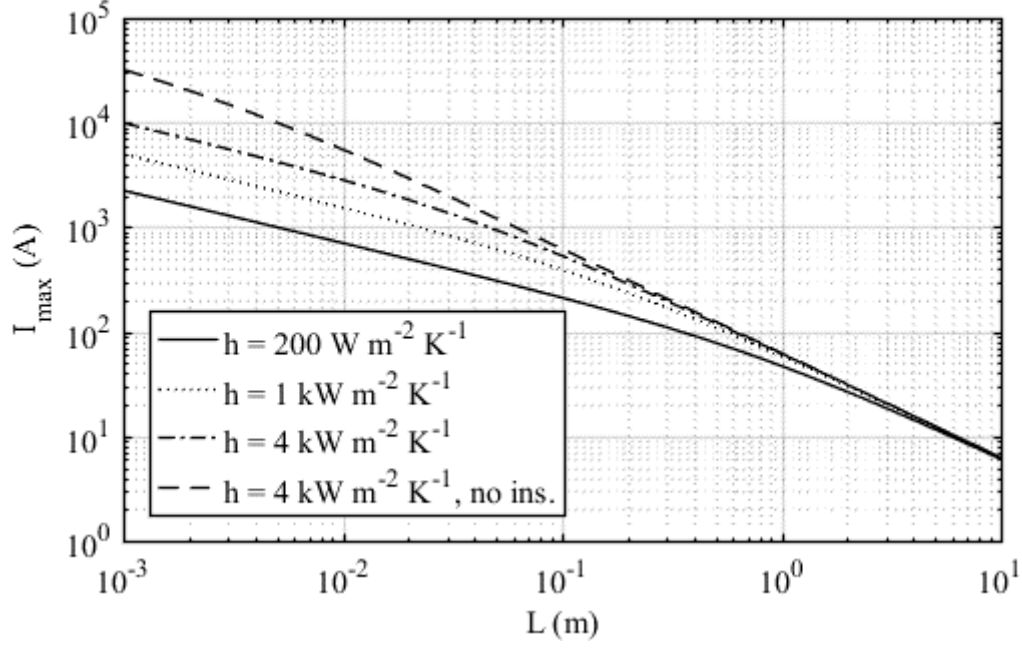


Figure 8: The effect of increasing end windings heat transfer coefficient and heat transfer area on the relationship between machine length and maximum current.

These advantages translate to considerable improvements in machine power density and efficiency, allowing for the design of smaller, lower mass machines that are easier to integrate into an airframe and with a lower fuel consumption penalty. In turn, this promises lower aircraft operational costs and environmental advantages. To evaluate whether these envisaged gains are achievable, research is required in several areas. The formulation of these specific research questions is therefore considered in the next section.

### 1.5. Research Questions

In the preceding section, it was demonstrated that the high conductivity thermal path renders end windings cooling a good thermal management strategy, and that increasing end windings heat transfer coefficient and heat transfer area considerably increases the current that can be produced in a machine without exceeding the maximum stator temperature. However, research is required into several aspects of the problem in order to more accurately quantify the gains and map the design envelope in which they

occur. A summary of the research questions and locations is provided in Table 2, followed by a more detailed discussion in Section 1.6 of their formulation in the narrative of this thesis.

*Table 2: Research questions and the Chapter in which they are addressed.*

| <b>No.</b> | <b>Research Question</b>                                                                                                                                                                  | <b>Chapter</b> |
|------------|-------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|----------------|
| 1(a)       | Can the pool boiling performance of a circular-wire motor windings geometry (MWG) surface be acceptably predicted by existing correlations?                                               | 3              |
| 1(b)       | How does the pool boiling heat transfer of an MWG surface compare with heat transfer from a flat plate for various surface orientations?                                                  | 3              |
| 2          | How does removal of the insulative coating on a conductor affect heat transfer in pool boiling conditions?                                                                                | 4              |
| 3          | How do spatial confinement and liquid level affect pool boiling heat transfer from a flat surface?                                                                                        | 5              |
| 4(a)       | How may bulk void fraction be modelled for a bubble plume around a column of conductors in spatially unconfined and confined pool boiling?                                                | 6              |
| 4(b)       | How does the bulk void fraction in the vicinity of a single conductor in a column affect its pool boiling heat transfer coefficient and critical heat flux?                               | 6              |
| 5          | Can the use of a low power nucleator device mitigate “bundle effect”, the reduced heat transfer coefficient of the bottom conductor in a boiling tube bundle at low heat flux conditions? | 7              |

## 1.6. Summary Narrative of Thesis

In this introductory Chapter 1, the scene has been set for the presentation of the research that composes this integrated thesis. Firstly, market trends towards electric propulsion in aircraft were noted, and then the challenges and potential opportunities associated with the thermal management of electrical machine stators were sketched. From these, nucleate boiling of stator end windings was particularly selected for further research as offering a favourable thermal path along axially-opposed stator winding conductors, high nucleate boiling heat transfer coefficients, and the potential for buoyancy-driven passive operation.

The Literature Review in Chapter 2 briefly outlines the fundamentals of pool boiling and then evaluates the extant research on specific aspects of pool boiling on planar and tubular surfaces. Tubular surfaces feature prominently in this thesis since they provide a good representation both of circular profile wire windings and also elucidate many of the same physical phenomenon that may be observed in stacked edge-wound rectangular profile end windings. On planar surfaces: size, orientation, microgeometry, confinement, and coolant level literature are analysed. Publications on single tubes and tube bundles are surveyed and competing phenomenological interpretations discussed.

Chapters 3 to 7 present the relevant research conducted during the doctoral studies of the author. Nucleate boiling heat transfer coefficients are relatively well documented for flat surfaces and common fluids, and correlations exist to predict these. However, the wavy geometry generated by resin-impregnated circular motor windings has never been tested in nucleate boiling. Can existing correlations be used to predict nucleate boiling heat transfer coefficient on the resin-impregnated end windings geometry? As end windings run at all orientations from horizontal to vertical and can vary in size, the effects of surface orientation and scale should also be tested. These subjects are considered in

Chapter 3: Pool Boiling of Resin Impregnated Motor Windings Geometry, which was published in *Applied Thermal Engineering* (Bartle, Menon and Walsh, 2018).

Resin impregnation of windings is often used to ensure insulation of each conductor. However, the thermal conductivity of resin is very low, and the development of phase-changing dielectric coolants makes it possible to have bare surface end windings without an insulation coating. The potential thermal gains of a bare surface are quantified by comparing the nucleate boiling of a bare copper surface with a resin-coated copper surface in Chapter 4: Pool Boiling of a Copper Surface with Epoxy Resin Microlayer.

For the heat transfer area gains described in eqn. (1.22), depicted in Figure 7(b), and quantified in Figure 8, nucleate boiling must be able to occur in vertically and horizontally confined spaces: between each windings layer, and between the stator iron and the windings stack. Additionally, the relationship between boiling heat transfer coefficient and liquid coolant level must be known in order to design suitable coolant reservoirs. Therefore, the effect of spacing between a heated and an unheated surface in vertical and horizontal configurations, and the effect of coolant level, on heat transfer coefficient are explored in Chapter 5: Effect of Spatial Confinement and Liquid Level on Nucleate Boiling, which is a paper read at the 15th UK Heat Transfer Conference 2017 (Bartle, Menon and Walsh, 2017).

When nucleate pool boiling is generated on conductor end windings, as depicted in Figure 7(b), a bubbly fluid flow is generated around the conductors. If the confinement is significant, for example in a channel or a narrow fluid reservoir, for a buoyancy-driven flow the void fraction of the fluid must increase in the direction opposite to gravity. In flow boiling, increasing void fraction can result in a significant heat transfer deterioration depending on flow conditions, so meriting an investigation whether similar phenomena

are observed with stacked conductors in pool boiling. Hence, the fluid field around a tube column in pool boiling is experimentally investigated for unconfined and confined conditions and a range of inter-tube spacings. A semi-empirical model is developed to evaluate bulk void fraction, and the relationship between void fraction and heat transfer coefficient is shown. This work constitutes Chapter 6: Pool Boiling of Horizontal Mini-Tubes in Unconfined and Confined Columns, which was published in the *International Journal of Heat and Mass Transfer* (Bartle and Walsh, 2019b).

Resulting from the work in Chapter 6 was the observation that the heat transfer coefficient on the bottom conductor is lower than on the conductors above at low heat flux conditions. Therefore, as Joule losses are a function of material temperature due to the positive temperature coefficient of most conductors, the bottom turn of an end-windings pool boiling cooled stator will produce slightly higher losses than the turns above. To attenuate this, Chapter 7: Bubble Nucleators to Enhance Pool Boiling, evaluates whether a low power electrical nucleator device positioned under Joule heated conductors can increase the heat transfer coefficient of the lower conductor. This Chapter is a paper read at the 14<sup>th</sup> International Conference of Heat Transfer, Fluid Mechanics and Thermodynamics 2019 (Bartle and Walsh, 2019a).

In Chapter 8: Discussion, the research undertaken in Chapters 3 to 7 is applied more particularly to the pool boiling of stator end windings, and then briefly extended to other aircraft power electronics. Limitations of the research are evaluated. Potential directions for future work in the areas of pool boiling fundamentals, and boiling in electrical machines and power electronics, are then suggested.

Finally, Chapter 9: Conclusions, summaries the general and problem-specific conclusions resulting from this research.

## 2. Literature Review

In Chapter 1, a review of phase-change cooling technologies was presented. Chapter 2 begins with a history of the development of research into boiling. It then considers the effects of fluid, and surface size, orientation, and microgeometry on planar surfaces. Finally, attention is given to boiling on the outside of single tubes or cylinders and boiling in tube bundles.

### 2.1. Pool Boiling Fundamentals

#### 2.1.1. A Description and History of Pool Boiling

The vast majority of studies on boiling have been undertaken with the immersion of an heated surface in a liquid coolant. The boiling curve of water is shown in Figure 9. The curve shape, and the phenomena it illustrates, are applicable to fluids in general, but the values of excess temperature  $\Delta T$  and heat flux  $q''$  are material- and coolant- specific. Natural convection-dominated boiling occurs in Region I and transitions with gradually-increasing bubble density into Region II: the nucleate boiling regime. Small imperfections in the hot surface (e.g. scratches, troughs) provide a greater local wetted heat transfer area and so a higher local temperature in the fluid immediately adjacent, which causes a localised pocket of vapour (i.e. a bubble) to develop. As excess temperature continues to increase, the vapour pockets produced at the nucleation sites change from being individual bubbles to long slugs and, eventually, vapour streams running up to the free surface. For most engineering purposes, boiling within Region II is most desirable. Region II concludes at critical heat flux at  $q''_{max}$ . Region III is unstable and can result in extremely high and/or fluctuating surface temperatures, rendering it undesirable in most engineering applications. The Leidenfrost point  $q''_{min}$  marks the beginning of Region IV, in which there are no distinct nucleation sites as the entire hot

surface is covered by a vapour layer. Where the material of the hot surface has a service temperature higher than the temperature at point E, Region IV can be used to provide an heat transfer rate in excess of those obtainable in Region II.

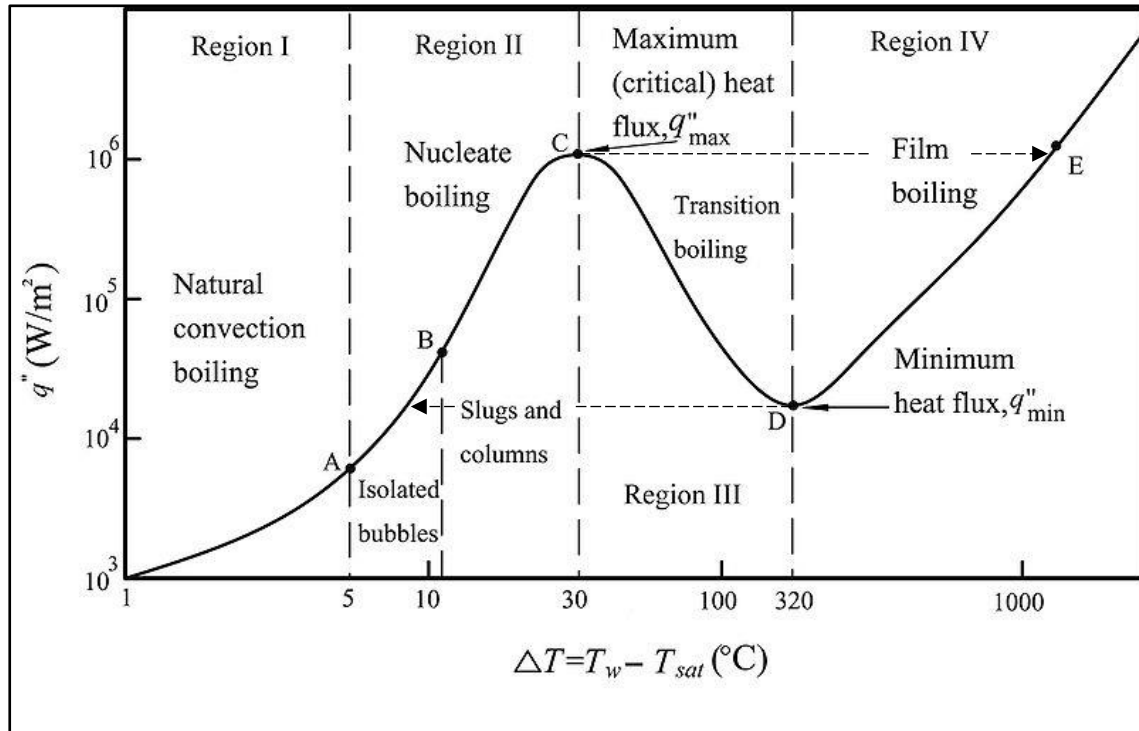


Figure 9: A graph to show the boiling curve of water at atmospheric pressure (Zhang, 2010).

Perhaps the earliest study of droplet form in literature is by the Dutch chemist Boerhaave (1732) but it is Leidenfrost (1756), who references Boerhaave, who is usually credited as the father of boiling research. Leidenfrost dropped water into a bowl of a skillet. He noticed that, with a skillet heated to just above the water saturation temperature  $T_{sat}$ , the dropped water would bubble away quickly into steam but, when the spoon was heated to much above  $T_{sat}$ , the water would coalesce into spherical drops, agitate in the bowl, and take much longer to change phase. He concluded that this was the result of a thin layer of vapour being developed on the hot surface; as the thermal conductivity of steam is much poorer than that of water, much less heat flux results and the suspended water droplet takes longer to boil. This vapour layer, and its associated phenomena,

subsequently became known as the Leidenfrost effect. Leidenfrost may therefore also be credited with observing what is today termed critical heat flux *CHF*, the point at which the boiling regime departs from stable nucleate boiling with significant surface dry out.

Nukiyama (1934) obtained the most part of the curve by varying the electrical current passed through a bar immersed in a water pool. The heat flux generated by the element was increased by increasing the current flow through the element, which provided the curve portion origin to C and then E onwards. The heat flux was then decreased from the top of the curve, which provided the return curve to D, and then from B down to origin. In heat flux controlled experiments of this type, thermal hysteresis is observed as the boiling regime rapidly transitions from point C to E as heat flux is increased, and rapidly transitions from D to B as heat flux is decreased. Most engineering problems are of this type, where heat flux is known and surrounding temperature increase unknown. However, from a research perspective, heat flux controlled experiments have the disadvantage that the transition boiling regime cannot be accessed for observation; Nukiyama could only conjecture of the existence of the curve between C and D.

This curve portion was supplied by Drew and Müller (1937). By placing an hollow tube into a pool and flowing preheated steam through the tube, the surface temperature of the tube  $T_w$  could be varied; as the saturation temperature of water at atmospheric pressure is known, the effect of surface temperature on heat flux could be investigated.

Building on Zuber and Berenson's own work, Moissis and Berenson (1963) postulated that transition from bubble boiling to jet or column boiling (in Figure 9, regions A-B and B-C respectively) would take place when the rate of bubble generation causes the bubbles to touch one another, so forming a vapour column. They argued that the

diameter of the bubbles depended on the contact angle  $\beta$  with the flat plate, and postulated a transitional heat flux. This seldom appears in modern literature as subsequent research suggests that the features described as associating with transition boiling are not stable and are better considered as occurring fully-developed nucleate boiling

### 2.1.2. Pool Boiling Heat Transfer

Pirotto (2004b) identified two main approaches to determine the heat transfer coefficient in nucleate pool boiling. The first approach, followed by Kruzhilin (1947), Borishanskii (1969) and Mostinski (1963), Labuntsov (1972), Stephan and Abdelsalam (1980), and Kutateladze “new” (1990), is to set constants for coefficients and indices, which provides general applicability at the expense of surface-liquid combination optimisation. The second, followed by Rohsenow (1952), Leiner (1994), and Pirotto, leave a variable surface coefficient and/or variable indices of dimensionless numbers. Pirotto (2004b) and Forrest *et al.* (2013) published excellent reviews of nucleate boiling heat transfer correlations.

Kruzhilin (1947) published the first correlation predicting heat transfer coefficient in the nucleate boiling regime  $h_{NB}$ :

$$h_{NB} = 0.082 \frac{k_l}{L_c} \left( \frac{q'' h_{fg}}{g T_{sat} k_l} \cdot \frac{\rho_v}{\rho_l - \rho_v} \right)^{0.7} \left( \frac{T_{sat} c_{p,l} \rho_l \sigma}{h_{fg}^2 \rho_v^2 L_c} \right)^{1/3} \text{Pr}^{-0.45} \quad (2.1)$$

where  $g$  is acceleration due to gravity,  $k$  is thermal conductivity,  $\rho$  is density,  $h_{fg}$  is the latent heat of vaporisation,  $\sigma$  is surface tension,  $c_p$  is the specific heat capacity of the coolant,  $T_{sat}$  is the coolant saturation temperature,  $\text{Pr}$  is the Prandtl number, and  $L_c$  is a characteristic length; the subscripts  $s$ ,  $l$  and  $v$  denote surface, liquid, and vapour, respectively. Sarma *et al.* (2010) suggest  $L_c$  may be the bubble diameter

$$d_b = C \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \quad (2.2)$$

with the constant  $C$  obtained by regression of experimental data. Alternatively, a bubble departure diameter correlation, such as Fritz (1935) or Fazel and Hosseyni (2016) may be used to provide  $L_c$ .

Borishanskii (1969) used thermodynamic similarity to correlate the physical properties of fluids and proposed a nucleate boiling correlation as a function of a number of reference points. These were supplied several years earlier by Mostinski (Mostinski, 1963), and the combined Borishanskii-Mostinski nucleate boiling correlation (Forrest *et al.*, 2013) is

$$h_{NB} = 0.1 p_C^{0.69} q^{0.67} \left( 1.8 p_r^{0.17} + 4 p_r^{1.2} + 10 p_r^{10} \right) \quad (2.3)$$

where  $p_c$  is critical pressure (in bar, for this equation only), and reduced pressure  $p_r = p/p_c$  where  $p$  is the actual pressure.

Labuntsov (1972) conceptualised a thin liquid layer under the vapour region on the boiling surface, and proposed the following correlation for the nucleate boiling of non-metallic liquids under moderate pressures:

$$h_{NB} = 0.075 \left[ 1 + 10 \left( \frac{\rho_v}{\rho_l - \rho_v} \right)^{2/3} \right] \left( \frac{k_l^2}{\nu \sigma T_{sat}} \right)^{1/3} q^{2/3} \quad (2.4)$$

where  $\nu$  is kinematic viscosity.

Stephan and Abdelsalam (1980) proposed 13 dimensionless groups and from this obtained the general equation, presented by Forrest *et al.* (2013) in terms of  $h_{NB}$ :

$$h_{NB} = 0.207 R_p^{\frac{2}{15}} \left( \frac{q''}{T_{sat}} \right)^{0.674} \left( \frac{\beta^{0.066} c_{p,l}^{0.042} h_{fg}^{0.371} k_l^{0.284}}{\sigma^{0.317}} \right) \times \left( \frac{\rho_l^{1.825} \rho_v^{0.297}}{g^{0.033} (\rho_l - \rho_v)^{1.763}} \right) \quad (2.5)$$

where  $R_p$  is mean surface roughness by DIN 4172 (in  $\mu\text{m}$ ) and may be assumed as 1  $\mu\text{m}$ ; Stephan and Abdelsalam use  $\beta = 35^\circ$  even in cases where the contact angle obtained by experiment is lower.

Prior to Borishanskii's own correlation (1969), he published collaborative work with Kutateladze (1966) that may be referred to as the Kutateladze "old" correlation. This early work was refined by Kutateladze some years later to create the Kutateladze "new" correlation (1990):

$$h_{NB} = \sqrt[3]{3.37 \times 10^{-9} \frac{k_l}{L_c} \left( \frac{q'' c_{p,l}}{h_{fg}} \right)^2 \left( \frac{(\rho_l / \rho_v)^2}{(\sigma g) / (\rho_l - \rho_v)} \right)} \quad (2.6)$$

The second group of correlations are those that explicitly define constants or indices to reflect the specific surface-fluid interaction. Rohsenow (1952) published the most popular correlation predicting heat flux  $q''$  in the nucleate boiling regime. Rohsenow suggested that the mechanism of nucleate boiling is dominated by turbulent convection caused by dispersing vapour pockets, and postulated the following correlation linking the Nusselt Nu, Reynolds Re and Prandtl Pr numbers:

$$\text{Nu} = C_{s,l} \cdot \text{Pr}^m \text{Re}^n, \quad (2.7)$$

where  $C_{s,l}$  is a curve-fitting constant dependent on surface effects such as geometry and roughness, and  $m$  and  $n$  are curve-fitting indices. Rohsenow formed Nu and Re hence:

$$\text{Nu} = \frac{h_{NB}}{k_l} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}, \text{ and} \quad (2.8)$$

$$\text{Re} = \frac{q''}{h_{fg} \rho_l} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \cdot \frac{\rho_l}{\mu_l}. \text{ Rearranging gives} \quad (2.9)$$

$$q'' = \mu_l h_{fg} \left( \frac{g(\rho_l - \rho_v)}{\sigma} \right)^{1/2} \left( \frac{c_{p,l} \Delta T}{C_{s,l} \text{Pr}^{1-m}} \right)^{1/n} \quad (2.10)$$

where  $m$  is a fluid-dependent parameter with value 0 for water and -0.7 for all other fluids,  $n = 2/3$  as default,  $\mu$  is viscosity, excess temperature  $\Delta T = T_w - T_{sat}$  where  $T_w$  is surface temperature, and  $h_{NB}$  is the nucleate boiling heat transfer coefficient defined

$$q'' = h_{NB} \Delta T. \quad (2.11)$$

The Rohsenow equation can be rearranged for  $h_{NB}$  thus:

$$h_{NB} = \left( \frac{h_{fg}}{q''} \right)^{n-1} \left( \frac{g(\rho_l - \rho_v)}{\sigma} \right)^{n/2} \left( \frac{c_{p,l} \mu_l^n}{C_{s,l} \text{Pr}^{1-m}} \right). \quad (2.12)$$

Nishikawa *et al.* (1982) continued the approach of Stephan and Abdelsalam (1980) by broadening the range of reduced pressures tested and suggested the correlation

$$h_{NB} = 31.4 \frac{p_C^{0.2}}{M^{0.1} T_C^{0.9}} \cdot 8 R_p^{(1-p_r)/5} \cdot \frac{p_r^{0.23}}{(1-0.99 p_r)^{0.9}} \cdot q''^{0.8}, \quad (2.13)$$

where  $M$  is the molar mass, reduced pressure  $p_r$  is in SI units, and  $R_p = R_a/0.4$  where  $R_a$  is the mean surface roughness (in  $\mu\text{m}$ ).

The University of Oxford's own Cooper (1984) developed a simplified correlation to predict heat transfer based on reduced pressures, thus:

$$h_{NB} = 55 p_r^{0.12 - \log_{10} R_p} (-\log_{10} p_r)^{-0.55} M^{-1/2} q''^{2/3}, \quad (2.14)$$

with molar mass  $M$  in units  $\text{mol g}^{-1}$  rather than the SI  $\text{mol kg}^{-1}$ .

The Gorenflo (1991) correlation continued in the same vein of reduced pressure similarity, but requires the knowledge reference values  $h_{NB,0}$  and  $q''_0$  to be used in the correlation

$$h_{NB} = h_{NB,0} F(p_r) \left( \frac{q''}{q''_0} \right)^{0.9-0.3p_r^{0.3}} \left( \frac{R_a}{R_{a,0}} \right)^{0.133}, \text{ where} \quad (2.15)$$

$$F(p_r) = 1.2 p_r^{0.27} + p_r \left( 2.5 + \frac{1}{1-p_r} \right), \quad (2.16)$$

$R_{a,0} = 0.4 \mu\text{m}$ , and  $q''_0 = 20 \text{ kW m}^{-2}$ . A development of the correlation was published in 2010 (Gorenflo and Kenning, 2010), but has yet to enter popular use in literature. Reference value  $h_{NB,0}$  can be obtained from literature for 55 fluids (Gorenflo *et al.*, 2014), but the Gorenflo family of correlations is somewhat less useful if one's fluid of interest is not listed.

Leiner (1994) continued on this avenue, and developed Borishanskii's use of thermodynamic similarity by noting that certain fluid parameters do not fit the principle of thermodynamic similarity, but can instead be characterised by fluid-specific nondimensional values. The use of fluid-specific values makes the Leiner correlation readily applicable to a much wider range of fluids than the Gorenflo correlations. Forrest *et al.* (2013) replaced Leiner's nondimensional values to give

$$\begin{aligned} h_{NB} = & 0.06955 p_C^{0.1277} R_a^{0.133} \left( \frac{\Re^{1.257} T_C^{0.4244} \rho_C^{0.9166} c_{p,l}^{0.2113}}{h_{fg}^{0.0521} k_b^{0.0443}} \right) \\ & \times \left( 43000^{0.15-0.3p_r^{0.3}} \right) \left( 1.27 p_r^{0.27} + p_r \left( 2.5 + \frac{1}{1-p_r} \right) \right) \\ & \times \left( \frac{q''}{p_C \sqrt{\Re T_C}} \right)^{0.9-0.3p_r^{0.3}} \end{aligned} \quad (2.17)$$

Subscript  $c$  denotes the thermodynamic critical point for the parameter; the specific gas constant  $\mathfrak{R} = R/M$  ( $\text{J g}^{-1} \text{K}^{-1}$ ) where  $R$  is the universal gas constant  $8.314 \text{ J K}^{-1} \text{ mol}^{-1}$ ; and the Boltzmann constant  $k_b = 1.38 \times 10^{-23} \text{ J K}^{-1}$ .

Pirotto (2004b) presented a modified version of the Rohsenow correlation:

$$h_{NB} = \frac{C_{s,l} k}{L_c} \left( \frac{q''}{h_{fg} \rho_v^{1/2} (\sigma g (\rho_l - \rho_v))^{1/4}} \right)^{2/3} \text{Pr}^m \quad (2.18)$$

where  $C_{s,l}$  and  $m$  are listed for various surface-fluid combinations or may be obtained experimentally.

In summarising heat transfer correlations, Rohsenow (1952) is by far the most recognised correlation and, in contemporary reviews, performs well against newer correlations that aim to account for a greater number of boiling parameters. As a relatively simple correlation, it lends itself to benchmarking new correlations and applied industrial use and, whilst it does require the look-up or experimental deduction of the surface-liquid curve fitting parameter, most common surface-liquid combinations have been tabulated over the past half century.

### 2.1.3. Pool Boiling Critical Heat Flux

Critical heat flux  $CHF$  is the heat flux at which the boiling regime transitions from nucleate boiling to film boiling. At this point, a vapour layer develops uniformly over the heated surface. This separation of the liquid from the heated surface causes a significantly lower heat transfer coefficient subsequent to this point. In the context of electrical and electronics cooling, operating a system above  $CHF$  causes rapid surface temperature increase and can lead to system failure. Therefore, the ability to predict  $CHF$  for a particular surface-fluid combination is also important from a component reliability perspective. Helpful reviews exist in Sadasivan *et al.* (1995), Kandlikar (2002), Arik *et*

*al.* (2011), Fang and Dong (2017), and Liang and Mudawar (2018a, 2018b). The latter work delineates five competing models explaining the critical heat flux condition: hydrodynamic instability, hot/dry spot development, bubble interference, macrolayer dryout, and interfacial lift-off. As this thesis does not research the mechanism behind CHF in detail, the first two models – with the widest acceptance today – are selected for fuller review below. The reader is encouraged to consult Liang and Mudawar (2018a) for review of the remaining three models.

A summary of *CHF* correlations based on the hydrodynamic theory are presented in Table 3. Zuber, in his 1958 thesis, is credited with first developing the hydrodynamic theory predicting critical heat flux. This argues that, during boiling, vapour jets are formed approximately uniformly on a surface due to the Rayleigh-Taylor instability caused by the generation of less dense vapour underneath denser liquid. Because the velocities are different in the liquid and vapour phases, Kelvin-Helmholtz instability occurs and, it is suggested, CHF occurs when the magnitude of the Kelvin-Helmholtz oscillation is such that the vapour plumes interact and merge with each other, causing a vapour mushroom. The published form (Zuber, 1958, 1963) is still widely used today:

$$CHF = K \rho_v h_{fg} \left( \frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right)^{1/4} \quad (2.19)$$

where the constant  $K = 0.131$  where the critical Helmholtz wavelength is used,  $\rho$  is density,  $h_{fg}$  is the latent heat of vaporisation,  $\sigma$  is surface tension, and the subscripts  $l$  and  $v$  denote properties for the liquid and vapour phases, respectively. There was some critical reception to this paper at the time (Bernath, 1963; Chang, 1963; Costello and Frea, 1963) because, amongst other matters, although Zuber noted the formulation of Taylor waves in the resulting vapour streams, he did not incorporate surface characteristics into the equation. Lienhard *et al.* (Lienhard and Dhir, 1973; Lienhard, Dhir and Rihard, 1973)

modified the formulation of the Helmholtz wavelength in the derivation of (2.19), slightly increasing predicted  $CHF$  as per eqn. (2.23). Both are found in literature today, but Zuber (1958) remains the more common formulation. Lu *et al.* (2015) modified the hydrodynamic model to include nucleation site density. The negative relationship between contact angle  $\beta$  and  $CHF$  has been modelled by Moissis and Berenson (1963) and Kirichenko and Chernyakov (1971), to which Ramilson *et al.* (1992) and Kim *et al.* (2016) added mean surface roughness  $R_a$  and mean spacing between peaks  $S_m$ . Generally decreased contact angle and increased surface roughness increase  $CHF$  but the empirical relationships presented would benefit from testing against an extended metric of surface-fluid combinations.

Table 3:  $CHF$  correlations based on the hydrodynamic theory (table adapted from Liang and Mudawar (2018a)).

| Publication                      | $K$ Coefficient Formulation                                                                                                                                                                              |        |
|----------------------------------|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|--------|
| Zuber (1958)                     | $K = 0.131$                                                                                                                                                                                              | (2.20) |
| Moissis and Berenson (1963)      | $K = 0.11\beta^{1/2}$                                                                                                                                                                                    | (2.21) |
| Kirichenko and Chernyakov (1971) | $K = 0.171 \frac{(1 + 0.324 \times 10^{-3} \beta^2)^{1/4}}{(0.018\phi)^{1/2}}$                                                                                                                           | (2.22) |
| Lienhard and Dhir (1973)         | $K = 0.149$                                                                                                                                                                                              | (2.23) |
| Ramilson <i>et al.</i> (1992)    | $K = 0.0044(\pi - \beta)^3 R_a^{1/8}$                                                                                                                                                                    | (2.24) |
| Kim <i>et al.</i> (2016)         | $K = 0.811 \left( \frac{1 + \cos \beta}{16} \right) \times \left( \frac{2}{\pi} + \frac{\pi}{4} (1 + \cos \beta) + \frac{351.2 \cos \beta}{1 + \cos \beta} \left( \frac{R_a}{S_m} \right) \right)^{1/2}$ | (2.25) |

Before passing from the hydrodynamic theory, it should be noted that Zuber built on Kutateladze's foundational work. Kutateladze (1961) argued that liquid-vapour

structures are destroyed near *CHF* as the vapour momentum reaches a critical point, but did not link this to either Rayleigh-Taylor or Kelvin-Helmholtz instabilities. From dimensional analysis, he proposed the form of equation (2.19) with  $K$  empirically obtained as 0.16. Borishanskii (1969) provided a formula to refine Kutateladze's constant  $K$ :

$$K = 0.13 + 4 \left( \frac{\rho_l \sigma^{3/2}}{\mu^2 (g(\rho_l - \rho_v))^{1/2}} \right)^{-0.4} \quad (2.26)$$

Passing now to the hot- or dry- spot theory, this argues that *CHF* occurs when an irreversible dry spot develops on a surface during boiling. It was first presented in Yagov (1988). The refined Yagov (2014) model offers a selection of three correlations depending on reduced pressure  $p_r$ . For  $p_r > 0.03$ :

$$CHF_{\text{high}} = h_{fg} \rho_v^{3/5} \sigma^{2/5} \left( \frac{g(\rho_l - \rho_v)}{\mu_l} \right)^{1/5} \quad (2.27)$$

For  $p_r < 0.001$ :

$$CHF_{\text{low}} = 0.5 \frac{h_{fg}^{81/55} \sigma^{9/11} \rho_v^{13/110} k_l^{7/110} g^{21/55} f(\text{Pr}_l)}{\nu_l^{1/2} c_{p,l}^{3/10} R_i^{79/110} T_{\text{sat}}^{21/22}}, \text{ where} \quad (2.28)$$

$$f(\text{Pr}_l) = \left( \frac{\text{Pr}_l^{9/8}}{1 + 2 \text{Pr}_l^{1/4} + 0.6 \text{Pr}_l^{19/24}} \right)^{4/11} \quad (2.29)$$

except for liquid metals, where  $f(\text{Pr}_l) = 0.5$ . Interpolation is suggested for  $0.001 < p_r < 0.03$ :

$$CHF = \sqrt[3]{CHF_{\text{high}}^3 + CHF_{\text{low}}^3} . \quad (2.30)$$

An alternative model based on the dry spot theory is presented by Theofanous and Dinh (Theofanous, Dinh, *et al.*, 2002; Theofanous, Tu, *et al.*, 2002; Theofanous and Dinh, 2006), which argues for a form similar to the hydrodynamic formulation in eqn. (2.19):

$$CHF = K^{-1/2} h_{fg} \rho_v \left( \frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right)^{1/4}. \quad (2.31)$$

Definition of the coefficient  $K$  was left to Kim *et al.* (2007):

$$K = \left( 1 - \frac{\sin \beta}{2} - \frac{(\pi/2) - \beta}{2 \cos \beta} \right)^{-1/2}. \quad (2.32)$$

Choi and No (Choi, No and Kim, 2016) developed a model that suggests two conditions are required to make a dry spot unquenchable: (1) the coalescence of a dry spot with neighbouring dry spots, and (2) the peripheral temperature of the dry spot arriving at Leidenfrost temperature. This model has subsequently been applied to forced convective and transitional boiling with, respectively, good and fair correspondence with experimental data (No *et al.*, 2019).

Before concluding, it is worth introducing a model that belongs to neither of the above families, but which has demonstrated good general applicability. Kandlikar (2001) performed a force-balance on a vapour bubble attached to an heated surface. He suggests that  $CHF$  occurs when the force drawing the bubble interface into the liquid (i.e. expanding the bubble) exceeds the sum of the forces holding the bubble interface in place; this condition causes the bubble to expand across the heated surface to form a vapour layer. The conclusion of his force-balance is a prediction for  $CHF$ :

$$CHF = h_{fg} \rho_v^{1/2} \left( \frac{1 + \cos \beta}{16} \right) \left( \frac{2}{\pi} + \frac{\pi}{4} (1 + \cos \beta) \cos \varphi \right)^{1/2} (\sigma g (\rho_l - \rho_v))^{1/4} \quad (2.33)$$

where  $\varphi$  is the heater surface angle ( $^\circ$ ) with respect to horizontal.

To conclude, from the most contemporary reviews (Fang and Dong, 2017; Liang and Mudawar, 2018b) it is difficult to make firm recommendations on which correlations best predict *CHF*. It is fair to suggest that the hydrodynamic theory remains the most common explanation of *CHF* and perhaps, taken as a family, the modern hydrodynamic *CHF* correlations are more consistently accurate. However, it is also fair to suggest that the hydrodynamic theory has had more attention, and therefore that family of correlations are more refined, than the other theories. By 1985, having undertaken much work on the hydrodynamic theory, Lienhard came to a position of “mature scepticism” (Lienhard and Witte, 1985) on it. Liang and Mudawar (2018b) show that that hot spot Yagov (2014) correlation is comparable to the hydrodynamic correlations in predictive accuracy. The Kandlikar (2001) correlation is shown in both Fang and Dong (2017) and Liang and Mudawar (2018b) as providing relatively good accuracy across a wide range of parameters. Ergo, whilst any of the *CHF* correlations recommended in these reviews is likely to fulfil the needs of the practicing industrial engineer for the present, the fundamental question of what mechanisms precipitate *CHF* is not altogether resolved.

## **2.2. Pool Boiling on Planar Surfaces**

### **2.2.1. Surface Size**

The effect of boiling surface size on nucleate pool boiling has been considered in a number of studies. Bakhu and Lienhard (1972) studied the pool boiling of a platinum wire with diameter 1 mil ( $10^{-3}$  inches) in water, acetone, benzene, isopropanol, and methanol. From the Zuber (1958) *CHF* criterion of Helmholtz instability, for spheres and cylinders it was noted that

$$CHF/CHF_{\text{Zuber}} = f(R') \quad (2.34)$$

where the characteristic dimension  $R'$  considers the ratio of buoyancy forces to surface tension forces and is defined as

$$R' = R(g(\rho_l - \rho_v)/\sigma)^{1/2}. \quad (2.35)$$

Bakhu and Lienhard's  $R'$  and the Bond number  $Bo$  have the relation  $R' = Bo^{1/2}$ . Experimental data showed that the model in eqns. (2.34) and (2.35) ceased to apply where  $R' < 0.15$ . Photographs indicated that bubbles grew on the wires until sufficiently large to detach from buoyancy forces, whilst no evidence of the waves present with larger  $R'$  values was found. At  $R < 0.03$ , the boiling curve showed a monotonic increase in excess temperature with respect to heat flux, with no unstable transition boiling region (Region III in Figure 9). Therefore, when the characteristic dimension of an immersed solid in pool boiling is very small, there is no critical heat flux point.

Gogonin and Kutateladze (1977) boiled ethanol on stainless steel flat plates with widths from 5 mm to 50 mm and concluded no correlation between surface size and critical heat flux. Park and Bergles (1988), Rainey and You (2001), and Kwark *et al.* (2010) found no effect on the boiling curve, except that critical heat flux arrived sooner with increasing heater size. Lu *et al.* (2015) observed the same effect and suggested that small heaters may be more influenced by edge effects than large heaters – nucleate boiling on a polished flat surface typically begins at the edges.

### 2.2.2. Surface Orientation

Jakob and Linke (1935) were the first to study the effects of orientation on boiling, by experimenting with horizontal upward-facing boiling surfaces and vertical boiling surfaces; however, the author has not been able to source a copy of this publication and so cannot comment on the conclusions at which they may have arrived.

Marcus and Dropkin (1963) built on this with an experiment to investigate the effects of inclination angle  $\theta$  from  $0^\circ$  (horizontal upwards) to  $90^\circ$  (vertical) on boiling. A cuboid copper block with one surface exposed to distilled water was enclosed in a micarta case, which was in turn suspended by pins that enabled the boiling surface to be rotated. They found that, for boiling where convection effects dominates, the heat transfer coefficient  $h$  was higher for horizontal surfaces than vertical surfaces; however, in nucleate boiling, the reverse was the case – heat transfer  $h$  was higher for vertical surfaces than horizontal surfaces. Noting the hydrodynamic theory that the superheated boundary region is agitated by the departing vapour bubbles, the theory rendered for the observed orientation effect was that an increased surface angle causes a greater bubble length through the boundary region and so increased agitation per departing bubble.

Chen (1978) extended the range of inclination angles  $\varphi$  investigated from  $0^\circ$  to  $180^\circ$  (horizontally downward-facing). A rectangular copper surface was epoxy glued into an insulating block, with the boiling surface exposed to Freon 11, and the whole test section mounted to a copper rod rotatable to adjust inclination angle. Heat transfer at a given wall superheat increased from  $0^\circ$  to around  $150^\circ$ , but then dropped swiftly in the definitively downward-facing orientations. Chen noted that, for inclinations above  $90^\circ$ , the heated surface impeded bubble departure; instead, the bubbles slide along the inclined surface. As bubbles from lower down the inclined surface are driven by buoyancy into bubbles at nucleation sites higher up the surface, they collide and coalesce; this agitation increases the departure frequency of the coalesced bubbles. In the unique case where the heated surface is horizontally downward ( $\varphi = 180^\circ$ ), the net buoyancy is zero and the bubbles are entirely impeded, with generated convection currents unable to displace them. Therefore, the bubbles coalesce at relatively low heat fluxes.

Nishikawa *et al.* (1984) considered inclination angle and surface finish in the nucleate boiling of water. A rectangular copper surface was exposed to the coolant, Fourier's heat flux calculated with thermocouple measurements, and photographs of the boiling surface taken. The heat transfer behaviour observed by Marcus and Dropkin from  $0^\circ$  to  $90^\circ$  was confirmed, and the coalescing bubble behaviour at high inclinations - discussed by Chen - considered in more detail. Nishikawa divided the boiling behaviour with respect to orientation into three categories: (a) low heat flux, less than  $70 \text{ kW m}^{-2}$ ; (b) medium heat flux,  $70\text{-}170 \text{ kW m}^{-2}$ , and (c) high heat flux, above  $170 \text{ kW m}^{-2}$ . (a) In the low heat flux category, as inclination angle increases, bubble diameter increases and nucleation site density decreases, up to an inclination angle of  $120^\circ$ . Beyond  $150^\circ$ , some bubbles generated grow rapidly along the heated surface against the direction of gravity, coalescing with other bubbles to form large coalesced bubbles over the heated surface. (b) In the medium heat flux category, individual coalesced bubbles begin to form even at inclinations less than  $120^\circ$ , and small bubbles exist alongside coalesced bubbles at inclinations above  $150^\circ$ . (c) In the high heat flux category, coalesced bubbles are generated across the heated surface for all inclinations, the top surfaces of which pulsate irregularly. Heat transfer at high heat flux was suggested to be independent of inclination angle. The effects of surface roughness coupled to orientation were also investigated. For an inclination angle less than  $120^\circ$ , the rough surface provided greater heat transfer than the smooth surface at all heat fluxes, whilst this was only true at higher heat fluxes for inclination angles above  $150^\circ$ .

Jung *et al.* (1987) made further progress on enhanced surfaces and orientation in boiling heat transfer although, as it claimed to be the first paper on the subject, it does not appear that the authors were aware of the embryonic work on orientation and surface roughness in Nishikawa *et al.* A polished copper surface was compared to two enhanced

surfaces, made by metal powder deposition, in nucleate pool boiling of R-11 in orientations from 0° to 180°. A 15-25% increase in heat transfer was reported as inclination angle was varied from 0° to 165° at the low heat fluxes of 10 to 40 kW m<sup>-2</sup>. No orientation effect on heat transfer was reported, with the exception of the 180° inclination, where a significant decrease in heat transfer was noted consistently with previous studies. The paper suggests that two mechanisms are at play in nucleate boiling – a bubble agitation mechanism, and an evaporative mechanism. At high heat fluxes, it is suggested, the evaporative mechanism dominates and so the effect of the bubble agitation mechanism described by Marcus and Dropkin (1963), and Chen (1978) – and which is a function of surface inclination – is minimal. Chen's reasoning of trapped bubbles for the low heat transfer in the 180° orientation was reiterated and concurred with.

Brusstar *et al.* (1997) conducted an unusual boiling experiment in which a low velocity (0.04 m s<sup>-1</sup>) flow boiling arrangement was said to approximate pool boiling. It found that critical heat flux *CHF* is negatively correlated with bubble residence time for all surface inclinations. Brusstar and Merte (1997) developed this theory into an analytical model for *CHF* with upward- and downward- facing surfaces and suggests that, for surfaces at inclinations greater than 90°,

$$CHF = CHF_{\text{flat}} |\sin \phi|^{1/2} \quad (2.36)$$

is a fair modification for critical heat flux predictions that do not already include orientation effects.

Rainey and You (2001) compared orientation effects on plain and a microporous enhanced copper surfaces in saturated FC-72. For the plain surface, a significant decrease in heat transfer is noted when increasing inclination from 90° to 180°. Attention was

drawn to the superheat hysteresis of fluorocarbons, in which a higher heat flux is initially required to activate nucleation sites; researchers using fluorocarbons are advised not to neglect decreasing  $q''$  runs in their experimentation programme. Unlike Marcus and Dropkin (1963), Nishikawa (1984), and Jung (1987), no significant orientation effect was observed at low heat fluxes; Rainey and You attribute this absence to the thermal hysteresis associated with fluorocarbons.

El-Genk and Bostanci (2003) investigated inclination angle on pool boiling of Novec 7100 on a smooth 10 x 10 mm copper block, sized to approximate an electronic microchip. They noted that, for low heat fluxes, heat transfer increased as inclination was increased from 0° to 90° but, at high heat fluxes, heat transfer decreased for the same inclination change. At inclinations greater than 90°, heat transfer continued to decrease for high heat fluxes, but no significant orientation effect was observed at flow heat fluxes.; this is consistent with Rainey and You (2001), but disagrees with Nishikawa *et al* (1984).

Priarone (2005) boiled fluorocarbons FC-72 and Novec 7100 on a smooth copper surface at a range of inclinations. At low heat fluxes, heat transfer increased with orientation angle; a finding that contradicts Rainey and You (2001). At high heat fluxes, orientations from 0° to 90° did not appear to affect heat transfer, but orientations above 90° adversely affected heat transfer, consistent with previous studies.

Liao *et al.* (2008) investigated water pool boiling on a standard copper surface and two surfaces with increasing degrees of hydrophilicity ( $\beta = 30^\circ$  and  $\beta = 0^\circ$ ), varied from orientations  $\phi$  between 0° and 180°. At orientations above 90°, lower critical heat fluxes were achieved on all surfaces. The most hydrophilic surface exhibited a small adverse effect of increasing orientation between 0° and 90° on critical heat flux, whilst the other two surfaces did not exhibit any effect on critical heat flux for the same

orientation range. Kwark *et al.* (2010) tested a plain and a nanocoated surface for orientation effects and the results confirmed Liao *et al.*, that surface coating does not appear to change orientation effects.

Jung and Kim (2016) noted the very small spaces between bubble nucleation sites in the nucleate boiling regime and suggested accordingly that the agitation caused by bubbles sliding across the heated surface cannot be the main mechanism governing orientation effects in non-horizontal orientations, but rather the effects of the coalescence that occurs when bubbles slide locally into each other to form larger bubbles.

In summary, heat transfer coefficient is usually reported to increase modestly from horizontally upwards ( $0^\circ$ ) up to at least vertical ( $90^\circ$ ) and possibly somewhat further. The physical mechanism enhancing heat transfer in these orientations is debated. A heated surface facing horizontally-downwards returns an early boiling crisis as vapour generated on the surface is trapped.

### **2.2.3. Surface Microgeometry**

Investigation of the effect of surface roughness on nucleate boiling began with Jakob and Fritz (1931) who boiled water, at atmospheric pressure, on a horizontal copper plate with varying roughnesses obtained by etching regular geometrical grooves onto the boiling surface. An initial improvement in heat transfer coefficient was observed for roughened surfaces, but this improvement decreased with boiling time. Collier and Thome (1994) comments that this indicates the cavities generated by the abraded roughness were unstable. A similar phenomenon was observed by a 500 h aging study by Chaudri and McDougall (1969); Roy Chowdhury and Winterton (1985) note this also, which they describe as a surface “aging” effect, but do not attempt to explain it physically.

However, Kravchenko and Ostrovsky (1979) conducted what seems to have been a long-duration boiling study on abraded surfaces without observing an aging effect.

Corty and Foust (1955) used sandpaper roughened surfaces with mean surface roughnesses  $R_a$  between 0.01 and 0.6  $\mu\text{m}$  and found that the roughest surfaces produced a 60% greater  $h_{NB}$  than the smoothest surface. Kurihara and Myers (1960) boiled water and a range of other liquids on a flat copper plate. The number of nucleation sites  $n_n$  was observed to increase with surface roughness, as was  $h_{NB}$ , and so the link established that the greater heat transfer coefficient observed with rougher surfaces is the result of a greater number of nucleation sites. When these additional nucleation sites are stable with time, a greater heat transfer coefficient results. Berenson (1962) considered the effects of surface roughness in the nucleate and transition boiling regimes of *n*-petane on nickel and found that the roughest surface provided a nucleate boiling heat transfer coefficient 600% greater than the smoothest surface. Nishikawa *et al.* (1982) observed that as the system pressure approached the critical pressure, the effect of surface roughness on heat transfer decreased.

Griffith and Wallis (1958) investigated the microgeometry of nucleation sites and concluded that the mouth diameter of the cavity determines the excess temperature needed to begin nucleation, whilst the shape of the cavity determines the stability of the nucleation site. Pioro (2004a) reviewed this by noting that a stable nucleation site exists when the cavity is not filled with liquid during the departure of a vapour bubble and that the three main variables to this are the surface wettability and the cavity shape and size. Webb (1981) provided a comprehensive literature review of the first thirty years of serious study on surface roughness effects.

Kandlikar and Steinke (2002) tested surfaces with roughness  $R_a$  between 0.02 and 0.62  $\mu\text{m}$  in water and made an interesting observation. Up to  $R_a \sim 0.15 \mu\text{m}$ , contact angle appeared to decrease with increasing surface roughness, indicating improved wettability, which is consistent with higher heat transfer coefficients for rougher surfaces. However, they also observed that, beyond 0.15  $\mu\text{m}$  surface roughness, the contact angle increased again. The investigators suggested that higher surface roughness interfered with the spreading of the liquid but recommended more research to better understand the trend.

Hibiki and Ishii (2003) devised a model to link nucleation site density (noted as being positively correlated with heat transfer coefficient), to critical cavity size (a function of the fluid) and the contact angle (a function of the fluid and boiling surface). McHale and Garimella (2013a) found this correlation to agree within their experimentally-observed nucleation sites within 50%; the same study also noted that nucleation site densities are significantly greater for rough surfaces, whilst average bubble departure diameters were lower for rough surfaces, and bubble departure frequencies did not vary significantly between rough and smooth surfaces.

A number of nucleate boiling correlations have expressly included defined surface roughness parameters (Stephan and Abdelsalam, 1980; Nishikawa *et al.*, 1982; Cooper, 1984; Gorenflo, 1991; Leiner, 1994; Gorenflo and Kenning, 2010). Questions have been raised over the extent to which surface roughness parameters adequately capture the microgeometry features relevant to nucleate boiling (Pioro, Rohsenow and Doerffer, 2004a; McHale and Garimella, 2013b). For example, two surfaces with somewhat different surface profiles and cavity shapes can have the same mean surface roughness  $R_a$ . However, hitherto no alternative approach to surface characterisation has been widely adopted and it may simply be prudent to acknowledge surface roughness as one useful characterisation variable amongst others.

Alongside mechanically-roughened and etched surfaces, many authors have studied the greater surface roughness properties that can be obtained by applying a coating to the boiling surface. Greater long-term nucleation site stability is achieved when the higher surface roughness value is achieved by an enhanced coating rather than mechanical abrasion of the surface (Webb, 1981). A detailed study of heat transfer enhancing coatings is not within the scope of this research programme; the author suggests Leong *et al.* (2017) for a recent review of surface enhancements with dielectric fluids.

#### 2.2.4. Spatial Confinement

Chernobyl'skii and Tananaiko (1966) produced data on boiling water in a narrow annulus, but Ishibashi and Nishikawa (1969) was the first to conduct a comprehensive study on the effect of spatial confinement on nucleate boiling performance. By adjusting the space between two vertical heating surfaces in distilled water, three regimes were identified visually. An isolated bubble regime was found to give similar heat transfer coefficients to unconfined pool boiling. When the space was decreased below the bubble departure diameter, the isolated bubbles coalesced to leave a few large, coalesced bubbles on the heating surfaces; in this coalesced bubble regime, decreasing space was found to significantly increase heat transfer. Further decreasing space gave rise to a liquid deficient regime, in which changing heat flux had little impact on the heat transfer coefficient; continuing to decrease space eventuates in a boiling crisis.

Yao and Chang (1983) boiled Freon 113, acetone, and water, and similarly found three boiling regimes. They associated them with the Bond number  $Bo$ , which measures the ratio of the gap size and nominal bubble departure diameter.

$$Bo = s / \left( \frac{\sigma}{g(\rho_l - \rho_v)} \right)^{1/2}, \quad (2.37)$$

where spacing  $s$  is the distance between the heated surface and the confinement. With low Bond number and low heat flux, heat transfer was in the isolated bubble regime. At low Bond number with higher heat flux, the coalesced bubble regime was accessed. With high heat flux and the Bond number slightly higher than unity, bubble nucleation was observed on the heated surface below the coalesced bubble layer. With high heat flux and the Bond number slightly lower than unity, violent expansion and coalescence of bubbles was observed. By using the Bond and boiling numbers, a map was produced to predict the confined boiling regime for given fluid and confinement conditions. The boiling number was defined as

$$B_1 = \frac{q''L}{h_{fg}\rho_v s U}, \text{ where the characteristic bubble rise velocity} \quad (2.38)$$

$$U = (gD(\rho_l - \rho_v) / \rho_l)^{1/2}, \quad (2.39)$$

and  $L$  is the channel or annulus characteristic length.

Fujita *et al.* (1988) developed the field by studying the relationships between boiling confinement, critical heat flux, and surface orientation. They confirmed the previous findings of Ishibashi and Nishikawa concerning the relationship between spacing and heat transfer coefficient, noting that a spacing much lower than the nominal bubble departure diameter ( $s = 0.15$  mm in water) caused significant vapour contact with the heated surface and an heat transfer coefficient independent of heat flux. With regards to surface orientation in confined boiling, it was found that a  $150^\circ$  orientation with  $s = 5$  mm and  $0.6$  mm did not significantly alter the boiling curve from the vertical  $90^\circ$  configuration because, they suggest, the decrease in bubble rise velocity is insignificant so long as the supply of liquid is adequate to wet the heated surface. The boiling curves for the  $175^\circ$  confined configurations were significantly different to the  $90^\circ$  confined boiling curves. At  $s = 5$  mm, the heat transfer coefficient was significantly larger than for

the vertical surface, explained by the flattened vapour bubbles being pushed up against the heated surface and causing thin liquid film evaporation to take place between the vapour and the heated surface. However, at  $s = 0.6$  mm, vapour bubbles accumulated and coalesced on the heating surface and lead to a lower heat transfer coefficient than the equivalent  $90^\circ$  curve.

Bonjour and Lallemand (1998) noted that the three boiling regimes, and their transitions, identified in extant literature were not precisely defined because they were based on subjective measures obtained from photograph, film, or live observations. Hot wire anaemometry was used to measure the flow vapour fraction, bubble passage time, and bubble frequency in R-113 boiled from a confined vertical heated surface. The boiling crisis occurred at around 50% vapour fraction for large or unconfined spacing (similar to pool boiling), but at 80 to 100% for narrow channels (similar to forced flow boiling). Bubble passage time increased with increasing heat flux; for a constant heat flux, passage time increases with increased confinement because the bubble is squeezed between the heated and unheated surfaces. For large channels, bubble frequency increased with increasing critical heat flux whilst, for small channels, bubble frequency increased until the heated surface was covered almost entirely by a vapour layer and individual bubbles ceased to be. Bonjour and Lallemand also redefined a flow map to predict flow regime based on the Bond number and the ratio of actual heat flux to unconfined critical heat flux.

Passos *et al.* (2005) boiled FC-72 at relatively low heat fluxes from horizontally downward facing heated surface in confined spacings ranging from 0.2 to 13 mm, and found the heat transfer coefficient to be enhanced at spacings up to 0.5 mm. Photographic visualisation of the boiling surface was also provided and showed large surface area, flattened bubbles as previously reported.

Misale *et al.* (2009) investigated the effect of surface orientation, and spacing between 0.5 mm and 20 mm, on boiling heat transfer in Novec 7100. They found spacing to affect heat transfer at  $Bo < 4$  and initially just at high heat fluxes. For  $1 < Bo < 4$ , confinement did not appreciably affect heat transfer at low and medium heat fluxes except for the  $0^\circ$  orientation (horizontally-upwards), where the heat transfer coefficients were slightly lower than in the unconfined case. For  $Bo \leq 1$ , the heat transfer coefficient was higher at low heat fluxes but much lower at high heat fluxes. Misale *et al.* (2011) probed further by investigating the effect of both confinement spacing and confinement area on Novec 7100 boiling from a horizontally-upwards heated surface. Two different unheated confinement discs were used, one with a diameter of 30 mm (equating to the boiling surface diameter) and the second with a diameter of 60 mm (equating to the overall test section diameter). The reduction in heat transfer coefficient with decreasing spacing at high heat fluxes was found to be more pronounced with the larger diameter confinement disc than with the smaller disc.

Rops *et al.* (2009) investigated a range of circular boiling surfaces from 4.5 to 15 mm diameter and with liquid levels of 2 to 6 mm above the heated surface. They found a reduction in diameter to increase heat transfer coefficient but no significant liquid head effect.

Geisler and Bar-Cohen (2008) researched boiling heat transfer in rectangular plate finned heat sinks in FC-72 and found heat transfer to be optimised when the channel width was around twice the fin thickness and the Bond number over unity. The same authors (2009) found the relative decrease of critical heat flux with respect to decreasing spacing to be independent of surface material or finish, but strongly dependent on channel aspect ratio. Geisler and Natarajan (2015) performed a similar study in FKE-774, focusing particularly on the effect of aspect ratio on critical heat flux. Only mediocre agreement

was obtained between the experimental data and the Bonjour and Lallemand (1998) correlation, and the error bars on the experimental data in this paper are not insignificant.

Finally, Souza *et al.* (2014) investigated the effect of enhancing a boiling surface with maghemite nanoparticles with diameters of 10 and 80 nm. For both free and confined ( $s = 0.1$  mm) conditions, the 10 nm coated surfaces provided higher heat transfer coefficients than the uncoated copper reference surface. However, whilst the 80 nm coated surface displayed the same trend for the free case, the heat transfer coefficient decreased relative to the uncoated surface in the confined case.

In conclusion, the literature suggests that, with decreasing spacing, heat transfer coefficient first increases as convective effects are pronounced, then decreases sharply with localised dryout. The mechanisms and respective contributions of these convective effects (flow acceleration, bubble sliding, coalescence) is not agreed and warrants further exploration.

#### **2.2.5. Coolant Level Effects**

The effect of liquid level above the boiling surface on nucleate boiling characteristics has not been the subject of much attention. By testing water, ethanol, R-113, and R-11, Pioro (1998) found that the coolant level beyond which heat transfer ceased to be impacted was 2.4 mm on an aluminium surface. Pioro *et al.* (2004a) observed that increasing coolant head above 2 mm has no significant effect on heat transfer, and cites Kutepov *et al.* (1977) and Tolubinskiy (1980); however, these Russian language works cited are not readily accessible and the present author has hitherto been unable to corroborate the references.

Relatively few papers on nucleate boiling record the level of the liquid coolant above the boiling surface. Chen (1978) maintained the R-11 liquid level at 100 mm above

the bottom of the chamber but dimensions relative to the boiling surface were not given. Guo and El-Genk (1992) established the water liquid level at 57 mm above the boiling surface and noted that this head decreased during the course of the experiment. With R-11, R-123, and R-134a, Kim and Choi (2001) used a liquid level of 50 mm above the boiling tube. For FC-72 and HFE 7100, El-Genk (2012) maintained a liquid head of 80 mm above the boiling surface.

Zhang *et al.* (Zhang, Liu and Wang, 2013, 2014) produced experimental studies on a confined boiling setup in which condensation took place within the confined chamber. The distance between the heated surface and condensation surface was 26 mm and water layer thicknesses of between 10 and 16 mm were used. They found that, with increasing water depth, the heat transfer coefficient initially increased and then decreased, suggesting an optimal fluid quantity at which both boiling and condensation heat transfer coefficients are maximised.

## **2.3. Pool Boiling on Tubes**

### **2.3.1. Single Tubes**

Aside from flat surfaces, cylindrical or tubular surfaces are the next most explored geometry in boiling. The word “tube” used widely in boiling publications to describe a cylindrical geometry and does not necessarily imply an internal flow. Nukiyama (1934) and Drew and Müller (1937) both experimented with tubes in foundational contributions to pool boiling literature. Sun and Lienhard (1970) demonstrated the applicability of Zuber’s (1958) hydrodynamic vapour jet theory to CHF prediction. Bakhru and Lienhard (1972) showed experimentally, on Joule heated platinum wires of 25.4  $\mu\text{m}$  diameter, that nucleate boiling does not occur on very small cylinders. Kew and Houston (2002) suggest that heat transfer coefficient increases with decreasing tube diameter.

The local heat transfer coefficient variation with angular position on a tube has been investigated. Lance and Myers (1958) found that maximum heat transfer coefficients existed at the bottom and the sides of the tube for methanol and n-hexane respectively. Using R-113, Cornwell and Einarsson (1990) reported that the heat transfer coefficient in pool boiling increased from the bottom to the sides, then decreased towards the top of the tube, and that the trend was independent of heat flux in the 15 to 40 kW m<sup>-2</sup> range. With the same fluid, Louahlia-Gualous *et al.* (2003) disagree, suggesting instead that the heat transfer coefficient was maximal at the top and minimal at the sides. Hence, whilst each publication suggests that there is a statistically-significant variation in local heat transfer coefficient with angular position, there is no agreement what that relationship is, and further work is needed.

### 2.3.2. Tube Bundles

Tube bundles with external boiling on horizontal tubes are found in a range of engineering applications including flooded refrigerant (Robinson and Thome, 2004), falling film (Fernández-Seara and Pardiñas, 2014), and desalination (Liu and Ishibashi, 2001) evaporators; and kettle reboilers in process distillation systems (McNeil *et al.*, 2010). Useful contemporary reviews are available in Browne and Bansal (1999), Ribatski and Thome (2007), and Swain and Das (2014). Most published research is on flooded boiling heat exchangers with pipes around ¾" (19 mm), and the trends discovered by these are valid when the experimental coolant flow rates are sufficiently low that forced convective effects are negligible.

An early work was published by Akin and McAdams (1939) on a 10 row bundle reported significant temperature variation across the tube rows but no trend. Cornwell and Schuller (1982) reported that heat transfer coefficient increased towards the top of a tube bundle. On finned tubes, Hahne and Müller (1983) noted that heat transfer

coefficient was enhanced on upper tubes in a bundle at low heat fluxes but that the effect disappeared at high heat fluxes. Subsequent studies on plain tubes confirm this. Hsu *et al.* (1993) reported a bundle heat transfer enhancement that decreased with increasing heat flux and pressure but increased with dryness fraction. Memory *et al.* (1995) contributed helpful nomenclature: defining bundle effect as the ratio between heat transfer from a given tube row in the bundle over heat transfer from the same tube heated in isolation, and bundle factor as the average heat transfer coefficient of the bundle over that of a single tube. Tube bundles with smooth and enhanced surfaces were tested and all except the porous surfaces confirmed that bundle effect declined with increasing heat flux; the porous surfaces showed no bundle effect. Fujita and Hidaka (1998) showed that a heated column of tubes surrounded by unheated tubes returns a lower tube heat transfer coefficient at low heat flux, and higher heat transfer coefficient at high heat flux, than when surrounded by heated tubes. For a 5 row, 3 column tube bundle, Gupta (2005) confirmed bundle effect and that the heat transfer coefficient on the central column was slightly higher than on the two side columns. Adom *et al.* (2010) reported that bundle effect continues to exist on small 3 mm diameter tubes; however, the experimental setup used thermocouples inside a Joule heated tube, which is cautioned against because the accuracy of thermocouples is adversely affected by electromagnetic interference (OMEGA Engineering, 2019).

Liu and Ishibashi (2001) investigated compact tube bundles; they reported that low tube spacing (i.e. highly compact) enhanced heat transfer at low and moderate heat fluxes and that heat transfer was maximum from the middle rows of the bundle. Liu and Qiu (2002, 2006) refined the findings of Liu and Ishibashi to suggest instead that, with decreasing spacing, heat transfer coefficient first increases and then decreases. This is partly corroborated by Gorgy and Eckels (2016), who observed enhanced heat transfer

coefficient for intermediate spacing at medium heat flux. Swain *et al.* (2017) reported bundle effect and increased heat transfer coefficient with decreased spacing but did not report a subsequent drop in heat transfer; test of a closer spacing may have been desirable from this paper.

Attention has been given to modelling dryness and void fraction in a tube column in flow boiling: experimentally (Mercier and Tadrist, 2007; Rooyen *et al.*, 2012), analytically (Robinson and Thome, 2004; Aprin, Mercier and Tadrist, 2011), and computationally (Tian *et al.*, 2016). To the best of the knowledge of the author of this thesis, the only void fraction modelling published for pool boiling in tube bundles is the numerical method in Ahmadpour *et al.* (2018).

Two popular theories have been advanced to explain the enhanced heat transfer with increasing row number in tube bundles at low heat flux. Usually, bundle effect is explained by the promotion of convective mixing caused by departing bubbles (Hahne, Chen and Windisch, 1991; Gupta, 2005; Kang, 2015). Additionally, Cornwell and Schuller (1982) developed the theory of a vapour microlayer (with the high heat transfer coefficient associated with evaporative cooling) caused by sliding bubbles. Later, Cornwell (1990) expanded this to suggest that nucleation does not actually occur on upper tubes in a bundle and the heat transfer there is entirely a combination of liquid convection and evaporation through the sliding bubble microlayer. Both theories are commonly used without discrimination to explain bundle effect (Hsu, Lin and Jensen, 1993; Memory, Sugiyama and Marto, 1995; Aprin, Mercier and Tadrist, 2011).

In summary, the vast majority of publications indicate that bundle effect exists in pool boiling and is most pronounced at low heat fluxes. The mechanism behind this, and

their relative contributions, are worthy of further exploration. Very little work has hitherto been undertaken on evaluating dryness and void fraction in pool boiling.

### 3. Pool Boiling of Resin Impregnated Motor Windings Geometry

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#### Abstract

More effective cooling strategies enable lighter and more power dense electrical machines to be developed. Pool boiling using the fluorocarbon Novec 7000 was generated on copper surfaces, both flat and modified to reflect the geometry of resin-impregnated windings of an electrical machine. Four experimental surfaces were exposed to the fluorocarbon in the nucleate pool boiling regime. The motor windings geometry (MWG) surfaces displayed an improved heat transfer coefficient compared to the flat surfaces due to a longer contact line between the heated surface and the vapour during bubble development. An MWG surface was also tested at orientations from 0° (horizontally-upward) to 90°, which did not significantly affect heat transfer; while the 128° orientation gave higher heat transfer coefficients at low heat fluxes, and lower heat transfer coefficients at high heat fluxes. The 180° orientation produced an immediate boiling crisis. The flat plate experimental data was evaluated and compared with the predictions of the Borishanskii and Mostinski, Gorenflo-Kenning, Kruzhilin, Kutateladze, Labuntsov, Leiner, Rohsenow, and Stephan-Abdelsalam nucleate boiling correlations, with the Rohsenow and Labuntsov models showing best agreement with data.

### 3.1. Introduction

The power densities of modern electrical and electronic machines are increasing as technology becomes more compact and requirements for higher power densities emerge. These trends result in the need for heat removal technologies that can maintain temperatures within acceptable operating conditions to ensure reliable operation. The heat flux density of a modern microchip is several orders-of-magnitude larger than that of an equivalent wired circuit. In most scenarios, the heat produced by a component must be dissipated to another location, which can be complicated by restrictive geometries (Green *et al.*, 2015; Garimella *et al.*, 2016; Hoffman, 2016).

Nucleate pool boiling provides amongst the highest heat transfer coefficients of any cooling mechanism. Accordingly, alongside the significant body of nucleate boiling literature on cooling individual electronic components (Anderson and Mudawar, 1989), multi-chip printed circuit boards (PCBs) (Ramakrishnan *et al.*, 2014), and computing hardware in server rooms (Tuma, 2010); nucleate pool boiling is also relevant to nuclear reactor cooling (Park *et al.*, 2012), hot-pressing of paper (Krook and Stenstrom, 1998), and even the manufacture of dairy confectionary (Kumar, Prakash and Kasana, 2011). Understanding and predicting nucleate boiling heat transfer characteristics is therefore important across a range of industrial sectors.

The present research investigates the use of nucleate boiling heat transfer to cool electrical motors, which consist of a magnetic rotor outside of which is a stator, consisting of windings. These windings are made from a material with good properties of electrical and thermal conduction, typically copper or steel, with resin impregnated through the windings to provide electrical insulation between each windings turn, mechanical fastening, and a thermal conduction path to the winding surface. In a high power density electrical motor, forced convection is typically used to cool the outer surface of windings

(Staton and Cavagnino, 2008). The effective cooling of windings is believed to contribute significantly to the reliability of the machine, where increased winding temperature has a significant adverse effect on the longevity of an electrical motor. Montsinger's law (1930) suggests that a 10°C increase in winding temperature causes a halving of the lifespan of the stator winding insulation, which in-turn may cause a short-circuit and machine failure (Cowern, 2000). Therefore, more effective cooling of windings may result in longer lifespan for a specified power, or increased power output for a given lifespan, leading to a more compact machine; all of which provide key competitive advantages to the manufacturer. A more compact machine is especially important where power density is critical, such as in aircraft applications.

When implementing boiling as a thermal management strategy in applications where the coolant makes contact with an electrically-charged surface, a dielectric is required. The materials company 3M produce a range of dielectric fluorocarbon based coolants with varying engineering properties, many of which offer low values on the global warming potential (GWP) scale. The GWP scale rates fluids on the quantity of heat a particular gas traps in the atmosphere, relative to CO<sub>2</sub>. Also, 3M fluorocarbons have been the subject of widespread research and use in the electronics industry and hence are the fluids investigated in this paper for potential cooling of motor windings.

This paper investigated nucleate pool boiling on flat and motor windings geometry (MWG) surfaces of varying sizes and at different orientations. Section 3.2 outlines the development of nucleate boiling theory and discusses competing heat transfer correlations; work on surface size effect, and the sometimes contradictory results reported by studies on surface orientation effect, are also presented. Section 3.3 describes the experimental methods used to generate data to investigate these topics. Section 3.4 analyses the extent to which the nucleate boiling correlations fit the experimental results

for flat and MWG surfaces of geometrically similar geometries with different length scales. Furthermore the surface orientation effects are compared against published results for flat surfaces. These results provide quantification of pool boiling heat transfer on a resin-impregnated windings geometry relevant to the use of nucleate boiling within an electrical machine.

## 3.2. Theory

### 3.2.1. Nucleate Boiling

Building on Boerhaave (1732) and Leidenfrost (1756), Nukiyama (1934) obtained most of the boiling curve by varying the electrical current passed through a bar immersed in a water pool. The boiling curve was completed by Drew and Müller (1937); by placing an hollow tube into a pool with preheated steam flowing through the tube, the surface temperature of the tube  $T_w$  could be varied by varying the steam temperature and pressure.

Pioro *et al.* (2004b) identified two main approaches to determine heat transfer coefficient. The first approach, followed by Kruzhilin (1947), Borishanskii (1969) and Mostinski (1963), Labuntsov (1972), , and Kutateladze (1990), is to set initial constants for coefficients and indices, which provides *a priori* predictions. Cooper (1984) also followed this approach, but the molar mass applicability bounds exclude the fluid used in this paper, and so the correlation is not included here. The second, followed by Rohsenow (1952), Stephan and Abdelsalam (1980), and Leiner (1994) leave a variable surface coefficient and/or variable indices of dimensionless numbers to be obtained from measured data. Mudawar (2001), Pioro (2004b), and Forrest *et al.* (2013) published excellent reviews of nucleate boiling heat transfer correlations relevant to this work.

Table 4 provides a summary of prominent correlations from literature. Kruzhilin (1947) published the first correlation predicting heat transfer coefficient in the nucleate

boiling regime, shown as eqn. (3.7). Borishanskii (1969) used the law of corresponding states, or “thermodynamic similarity” (Novikov, 1987), to correlate the physical properties of fluids from their properties at the critical point. These were supplied several years earlier by Mostinski (1963), and the combined Borishanskii-Mostinski nucleate boiling correlation (2013) is in eqn. (3.11), where  $P_c$  is critical pressure (in bar, for this equation only), and reduced pressure  $P_r = P/P_c$  where  $P$  is the actual pressure.

Labuntsov (1972) conceptualised a thin liquid layer under the vapour region on the boiling surface, and proposed the correlation shown in eqn. (3.11) for the nucleate boiling of non-metallic liquids under moderate pressures.

Prior to Borishanskii’s own correlation (1969), he published collaborative work with Kutateladze (Kutateladze and Borishanskii, 1966). This early work was refined by Kutateladze some years later to create the Kutateladze “new” correlation (1990), presented in eqn. (3.11).

The second group of correlations are those that explicitly define constants or indices to reflect the specific surface-fluid interaction. Rohsenow (1952) published the most popular correlation predicting heat flux  $q''$  in the nucleate boiling regime. Rohsenow suggested that the mechanism of nucleate boiling is dominated by turbulent convection caused by dispersing vapour pockets, and postulated the following correlation linking the Nusselt Nu, Reynolds Re and Prandtl Pr numbers:

$$\frac{Re \cdot Pr}{Nu} = \frac{c_p \Delta T}{h_{fg}} \quad (3.1)$$

Rohsenow provided indirect scaling for the variables in Nu and Re to give more physical relevance to the nucleate boiling regime as:

$$\frac{c_p \Delta T}{h_{fg}} = C_{s,l} \text{Pr}^m \left( \frac{q''}{\mu_l h_{fg}} \left( \frac{\sigma}{g(\rho_l - \rho_v)} \right)^{\frac{1}{2}} \right)^n, \quad (3.2)$$

where  $m$  is a fluid-dependent parameter with value 1 for water and 1.7 for all other fluids,  $n = 1/3$  as default,  $\mu_l$  is viscosity, and excess temperature

$$\Delta T = T_w - T_{sat}, \quad (3.3)$$

where  $T_w$  is surface temperature, and the nucleate boiling heat transfer coefficient is

$$h_{NB} = q'' / \Delta T. \quad (3.4)$$

Rohsenow remains the most widely used nucleate boiling correlation. Substituting eqn. (3.4) in eqn. (3.2) gives the expression in terms of  $h_{NB}$  as in eqn. (3.11).

In 1969 Mikic and Rohsenow (1969) published another correlation for nucleate boiling heat transfer. This has not been widely used, requiring *a priori* knowledge of the nucleation site density at a particular excess temperature for a given surface-fluid combination; the correlation also uses dimensional fitting parameters that require inputs to be in US Imperial units.

Some correlations require surface roughness parameters, including Stephan and Abdelsalam (1980), who proposed 13 dimensionless groups and from this obtained a general equation, where  $R_p$  is mean surface roughness (in  $\mu\text{m}$ ) and is suggested to equal 1 for most engineering surfaces in the absence of information to the contrary. This model constitutes eqn. (3.12). Although the variable  $\beta$  appears in the equation, instead of using an experimentally-derived value of this, Stephan and Abdelsalam and subsequent studies (Sathyabhama and Hegde, 2010; Forrest *et al.*, 2013; Zajackowski, Halon and Krolicki, 2016) recommended the use of  $35^\circ$  for refrigerants and fluorocarbons, and so is used in this paper also. For an heat flux of  $160 \text{ kW m}^{-2}$ , a 12% decrease in predicted heat transfer

coefficient results when a more realistic  $5^\circ$  contact angle is used instead of the recommended  $35^\circ$ .

Leiner (1994) developed Borishanskii's use of thermodynamic similarity by noting that certain fluid parameters do not fit the principle of thermodynamic similarity, but can instead be characterised by fluid-specific nondimensional values. Forrest *et al.* (2013) replaced Leiner's nondimensional values to give eqn. (3.10). Gorenflo and Kenning (2010) similarly identified variable groups impacting strongly on heat transfer and defined associated reduction factors to produce the semi-empirical correlation shown in eqn. (3.6).

Table 4: Correlations of heat transfer coefficient for nucleate boiling on flat surfaces.

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Borishanskii-

Mostinski

(Mostinski, 1963;

Borishanskii, 1969)

$$h_{NB} = 0.1 p_C^{0.69} q''^{2/3} (1.8 p_r^{0.17} + 4 p_r^{1.2} + 10 p_r^{10}) \quad (3.5)$$

$$h_{NB} = h_0 F_q F_{p_r} F_f F_w,$$

$$\text{where } F_q = (q''/q_0'')^n, \quad n = 0.95 - 0.3 p_r^{0.3},$$

Gorenflo-Kenning

(2010)

$$F_w = \left( \frac{R_a}{R_{a0}} \right)^{2/15} \left( \frac{(k\rho c)_w}{(k\rho c)_{Cu}} \right)^{1/4}, \quad F_f = \left( \frac{p_{f0}}{p_{f,ref}} \right)^{0.6} \quad (3.6)$$

$$p_{f0} = \frac{(dp/dT)_{VPC}}{\sigma}$$

$$h_0 = 3.58 \text{ kW m}^{-2} \text{ K}^{-1}, \quad p_{f,ref} = 1 \text{ (K } \mu\text{m)}^{-1},$$

$$q_0'' = 20 \text{ kW m}^{-2}, p_{r,0} = 0.1, R_{a0} = 0.4 \text{ } \mu\text{m}$$

$$\begin{aligned} \text{Kruzhilin (1947)} \quad h_{NB} &= 0.082 \frac{k_l}{L_c} \left( \frac{q'' h_{fg}}{g T_{sat} k_l} \cdot \frac{\rho_v}{\rho_l - \rho_v} \right)^{0.7} \left( \frac{T_{sat} c_{p,l} \rho_l \sigma}{h_{fg}^2 \rho_v^2 L_c} \right)^{1/3} \\ &\times \text{Pr}^{-0.45} \end{aligned} \quad (3.7)$$

$$\text{Kutateladze (1990)} \quad h_{NB} = \sqrt[3]{3.37 \times 10^{-9} \frac{k_l}{L_c} \left( \frac{q'' c_{p,l}}{h_{fg}} \right)^2 \left( \frac{(\rho_l / \rho_v)^2}{(\sigma g) / (\rho_l - \rho_v)} \right)} \quad (3.8)$$

$$\text{Labuntsov (1972)} \quad h_{NB} = 0.075 \left[ 1 + 10 \left( \frac{\rho_v}{\rho_l - \rho_v} \right)^{2/3} \right] \left( \frac{k_l^2}{\nu \sigma T_{sat}} \right)^{1/3} q''^{2/3} \quad (3.9)$$

$$\begin{aligned} \text{Leiner (1994)} \quad h_{NB} &= 0.06955 p_C^{0.1277} R_a^{0.133} \left( 43000^{0.15 - 0.3 p_r^{0.3}} \right) \\ &\times \left( \frac{\Re^{1.257} T_C^{0.4244} \rho_C^{0.9166} c_{p,l}^{0.2113}}{h_{fg}^{0.0521} k_b^{0.0443}} \right) \\ &\times \left( 1.27 p_r^{0.27} + p_r \left( 2.5 + \frac{1}{1 - p_r} \right) \right) \left( \frac{q''}{p_C \sqrt{\Re T_C}} \right)^{0.9 - 0.3 p_r^{0.3}} \end{aligned} \quad (3.10)$$

$$\text{Rohsenow (1952)} \quad h_{NB} = \left( \frac{h_{fg}}{q''} \right)^{n-1} \left( \frac{g (\rho_l - \rho_v)}{\sigma} \right)^{n/2} \left( \frac{c_{p,l} \mu_l^n}{C_{s,l} \text{Pr}^m} \right) \quad (3.11)$$

$$\begin{aligned} \text{Stephan-} \quad h_{NB} &= 0.207 R_p^{15} \left( \frac{q''}{T_{sat}} \right)^{0.674} \left( \frac{\beta^{0.066} c_{p,l}^{0.042} h_{fg}^{0.371} k_l^{0.284}}{\sigma^{0.317}} \right) \\ \text{Abdelsalam (1980)} \quad &\times \left( \frac{\rho_l^{1.825} \rho_v^{0.297}}{g^{0.033} (\rho_l - \rho_v)^{1.763}} \right) \end{aligned} \quad (3.12)$$

### 3.2.2. Surface Size Effects

The effect of boiling surface size on nucleate pool boiling has been considered in a number of studies. Gogonin and Kutateladze (1977) boiled ethanol on stainless steel flat plates with 150 mm length and widths from 5 mm to 50 mm, and concluded that there

is no relationship between surface size and critical heat flux. Park and Bergles (1988), Rainey and You (2001), and Kwark *et al.* (2010) found the trajectory of the boiling curve unchanged but that critical heat flux occurred at a lower excess temperature with increasing heater size. Lu *et al.* (2015) observed the same effect and noted that small heaters are more influenced by edge effects than large heaters – nucleate boiling on a polished flat surface typically begins at the edges.

### 3.2.3. Orientation Effects

Jakob and Linke (1935) were the first to study the effects of orientation on boiling, by experimenting with horizontal upward-facing boiling surfaces and vertical boiling surfaces. Marcus and Dropkin (1963) built on this with an experiment to investigate the effects of inclination angle  $\phi$  from  $0^\circ$  (horizontal upwards) to  $90^\circ$  (vertical) on boiling. They found that, for low heat flux boiling where convection effects dominate, the heat transfer coefficient was higher for horizontal surfaces than vertical surfaces; however, in nucleate boiling, the reverse was the case. Jung *et al.* (1987) concurred with this, reporting a 15% to 25% increase in heat transfer for increasing orientations from  $0^\circ$  to  $165^\circ$  at low heat fluxes of 10 to 40 kW m<sup>-2</sup>.

Chen (1978) extended the range of inclination angles  $\phi$  investigated from  $0^\circ$  to  $180^\circ$  (horizontally downward-facing). Heat transfer at a given wall superheat increased from  $0^\circ$  to around  $150^\circ$ , but then dropped swiftly in the definitively downward-facing orientations. Chen noted that, for inclinations above  $90^\circ$ , the heated surface impeded bubble departure; instead, the bubbles slide along the inclined surface. As bubbles from lower down the inclined surface are driven by buoyancy into bubbles at nucleation sites higher up the surface, they collide and coalesce; this agitation increases the departure frequency of the coalesced bubbles. Jung and Kim (2016) instead suggest that, because of the small spaces between nucleation sites, bubble coalescence, and not agitation due to

bubble movement, is the main mechanism governing orientation effects. In the unique case where the heated surface is horizontally downward ( $\varphi = 180^\circ$ ), the motion of the bubbles due to buoyancy is entirely impeded, with generated convection currents unable to displace them from underneath the surface. Therefore, the bubbles coalesce at relatively low heat fluxes. Nishikawa *et al.* (1984) found comparable experimental results but with a turning point from increasing to decreasing heat transfer at  $120^\circ$ .

Rainey and You (2001) and El-Genk and Bostanci (2003) used 3M fluorocarbons similar to that used in this present orientation work: FC-72 and Novec 7100 respectively. Rainey and You observed no significant orientation effect on heat transfer coefficient at low heat fluxes, and attributed this to a thermal hysteresis peculiarly associated with fluorocarbons. Priarone (2005) tested both FC-72 and Novec 7100 on a smooth copper surface and found heat transfer increased with orientation at low heat fluxes but decreased with orientation at high heat fluxes. Jun *et al.* (2016) echoed these findings in comparing a plain copper surface with a microporous surface at various orientations in deionized water.

### 3.3. Method and Materials

Two 12.5 x 12.5 mm and two 25 x 25 mm copper blocks were manufactured with dimensions shown in Figure 10 and Table 5, where  $y_1$  and  $y_2$  are the through-plane dimensions of  $x_1$  and  $x_2$  respectively. Two surfaces were manufactured consisting of a waveform (shown in Figure 10 and Figure 11) on the top surface to approximate the geometry of the surface of resin-impregnated motor windings, the other blocks retained flat boiling surfaces. A mirror image tool of the desired surface was machined onto a

copper electrode using a bullnose cutter. An electrical discharge machine was used to remove material from the workpiece, and so produce the desired surface geometry.

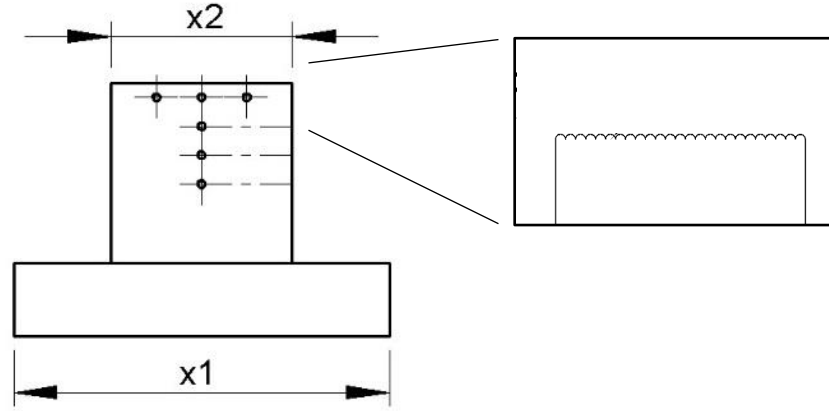


Figure 10: Main drawing showing the copper block flat surface, and inset showing the MWG surfaces with surface profile shown in Figure 11.

Table 5: Boiling surface area with reference to Figure 10.

|                          | $A_w$ (mm <sup>2</sup> ) |
|--------------------------|--------------------------|
| small flat (12.5 x 12.5) | 156                      |
| small MWG (12.5 x 12.5)  | 192                      |
| large flat (25 x 25)     | 625                      |
| large MWG (25 x 25)      | 703                      |

Onto the base of each 12.5 mm block was adhered a 28V Kapton flexible silicone adhesive heater. The 25 mm blocks had 12 V silicone adhesive heater mats affixed. Type K thermocouples, calibrated to  $\pm 0.1^\circ\text{C}$  against a thermostatic bath rated to  $\pm 0.02^\circ\text{C}$ , were placed in the bores of each heated element, as shown in Figure 10. A thermocouple was affixed to the underside of the heater to ensure that the maximum temperature of the heater was not exceeded during operation. Each heated element was placed into an aluminium container, prepared with a layer of epoxy resin to provide thermal insulation between the heated element and the container. Epoxy resin was then poured into the container in stages until the resin was flush with the boiling surface, so only the upper surface was exposed to the coolant. Once the top layer of resin was set, the modules were

cured at 85°C for three hours according to the manufacturer's specifications. Photographs of two larger test surfaces embedded in resin are shown in Figure 12.

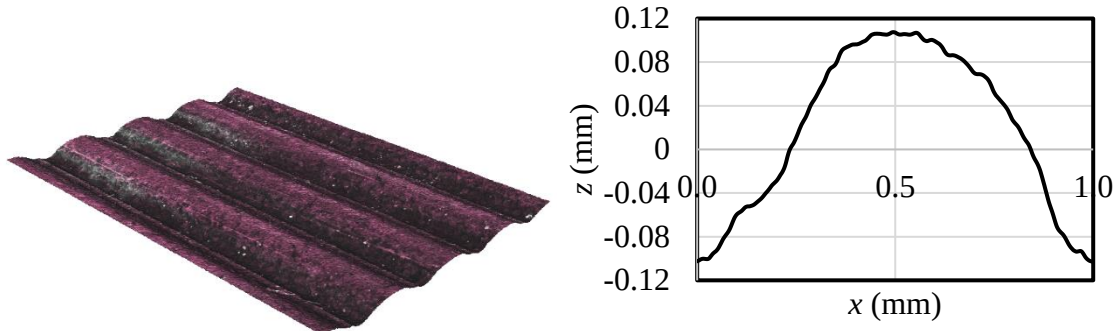


*Figure 11: Left image shows a section of the large MWG surface as scanned by the Alicona microscope. Graph shows the profile of a single wave period of the large MWG surface.*

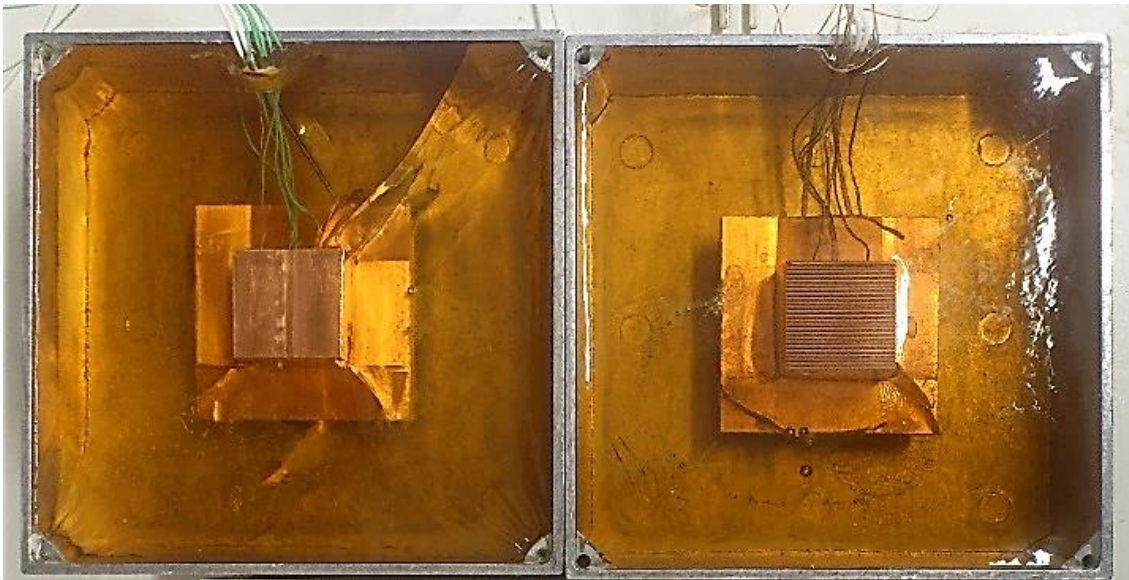


*Figure 12: Photograph of 25 mm flat and 25 mm MWG primary modules.*

Prior to each experiment, the selected primary module was placed into a transparent acrylic box to provide a boiling chamber that enabled the boiling surface to be observed by the naked eye. Type K thermocouples read vapour and pool temperature. The acrylic box was filled with coolant to 25 mm above the boiling surface. A relay-controlled heating element was immersed into the coolant to maintain saturation temperature. This assembly is shown in Figure 13.

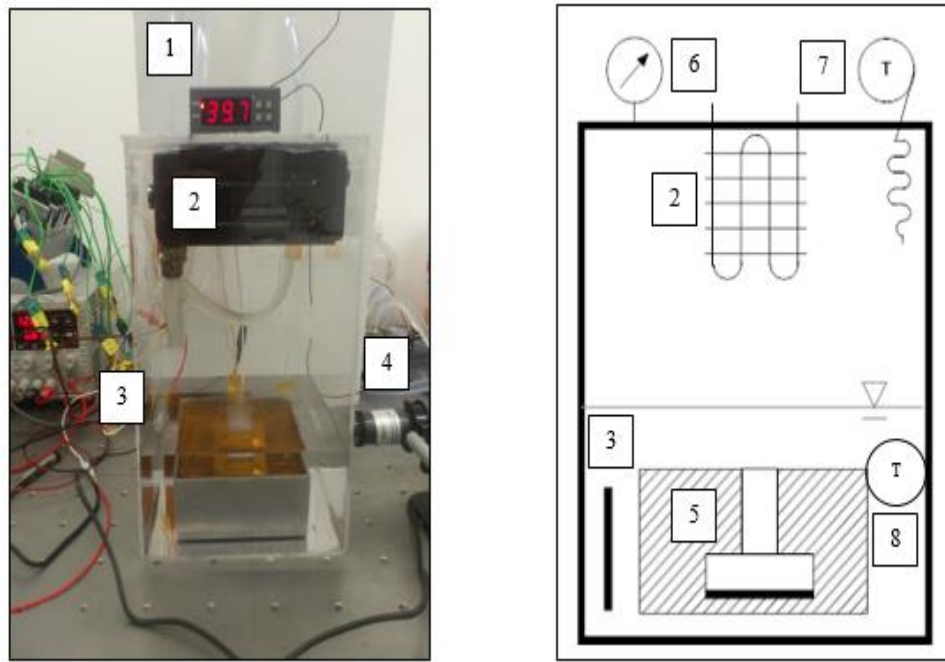


Figure 13: Boiling chamber apparatus: (1) Thermistor controlled relay to control auxiliary pool heater. (2) Condenser. (3) Auxiliary pool heater. (4) Camera. (5) Primary module consisting of heated block encased in resin. (6) Pressure transducer. (7) Vapour thermocouple. (8) Pool thermocouple. The external acrylic box is square-profile with 175 mm side.

Adjustment of the orientation was made by mechanically supporting the primary module. For the  $45^\circ$  orientation, the base of the primary module set on the hypotenuse of a  $45^\circ$  aluminium prism. The  $90^\circ$  orientations were obtained by setting the primary module on its side, and the  $180^\circ$  orientation (heater facing down) was achieved with four aluminium cuboid pillars placed to elevate the heated surface 50 mm from the base of the boiling vessel. The  $128^\circ$  orientation was achieved by using both the prism and the pillars to suspend the primary module at an angle.

Condensation of the coolant was achieved with a heat exchanger, mounted underneath the top surface of the boiling vessel, and fed by water. This water was maintained below the saturation temperature of the coolant by being pumped through a water-air heat exchanger. The forced convection of air was provided by two 12 V fans mounted on the surface of the heat exchanger, which dissipated the heat to the room.

Pressure and temperature data was captured electronically. A relay was used to automatically cut power at a predetermined temperature to avoid burnout (120 °C and 180 °C for the 12.5 and 25 mm surfaces respectively). All experiments were undertaken at atmospheric pressure.

High concentrations of non-condensable gases significantly accelerate the onset of boiling in both water (Torikai, Shimamune and Fujishiro, 1970; Chao *et al.*, 1994) and fluorocarbons (You *et al.*, 1995; O'Connor, You and Chang, 1996). Bar-Cohen and Simon (1988) suggest that the presence of noncondensables, at a partial pressure, in a developing bubble reduces the superheat required to initiate detachment. Accordingly, to ensure repeatability, before each run of experiments the discharge of non-condensable gases was performed by vented boiling under an extractor for one hour (Forrest *et al.*, 2013; Fazel and Hosseyni, 2016; Sahu *et al.*, 2016).

Each test began at the highest power setting possible without overheating the primary module; thereafter, the power to the heated block was decreased incrementally, with a time-step of 15 mins at each power increment, to ensure steady-state conditions. When the lowest power setting was reached, the power would then be increased incrementally, until the initial power setting. Thus, a single decreasing heat flux run followed by a single increasing heat flux run constituted one repeat. Three repeats were made of each test, at the same power supply settings. Although the existence of an initial thermal hysteresis is a subject of debate (Jung, Venart and Sousa, 1987; El-Genk and Bostanci, 2003; Priarone, 2005), data from the initial run of each heated surface was excluded from the presented results in order to mitigate against initial thermal hysteresis and initial surface aging effects and so improve repeatability.

Heat flux  $q''$  in this experiment is obtained from Fourier's law

$$q'' = k \nabla T, \quad (3.13)$$

where copper thermal conductivity  $k = 390 \text{ W m}^{-1} \text{ K}^{-1}$  and the gradient is measured in the  $z$ -direction by an equi-spaced thermocouple stack within the copper block (the bore holes observable in Figure 10. This is compared with the input power, obtained from the power supply display, to calculate the heat loss through surfaces other than the boiling surface and was found to be up to 20%. The experimental nucleate boiling heat transfer coefficient  $h_{NB}$  is

$$h_{NB} = q'' / (T_w - T_{sat}). \quad (3.14)$$

Uncertainty analysis was performed according to the Kline and McClintock (1953) method, where  $w_T$  is  $\pm 0.1 \text{ }^\circ\text{C}$  and  $w_z$  is  $\pm 0.1 \text{ mm}$ , and shown in Table 6.

*Table 6: Kline and McClintock uncertainty for three representative heat fluxes.*

|             |                       |                       |                        |
|-------------|-----------------------|-----------------------|------------------------|
| $q''$       | 39 $\text{kW m}^{-2}$ | 94 $\text{kW m}^{-2}$ | 134 $\text{kW m}^{-2}$ |
| Uncertainty | 17.68%                | 7.33%                 | 5.14%                  |

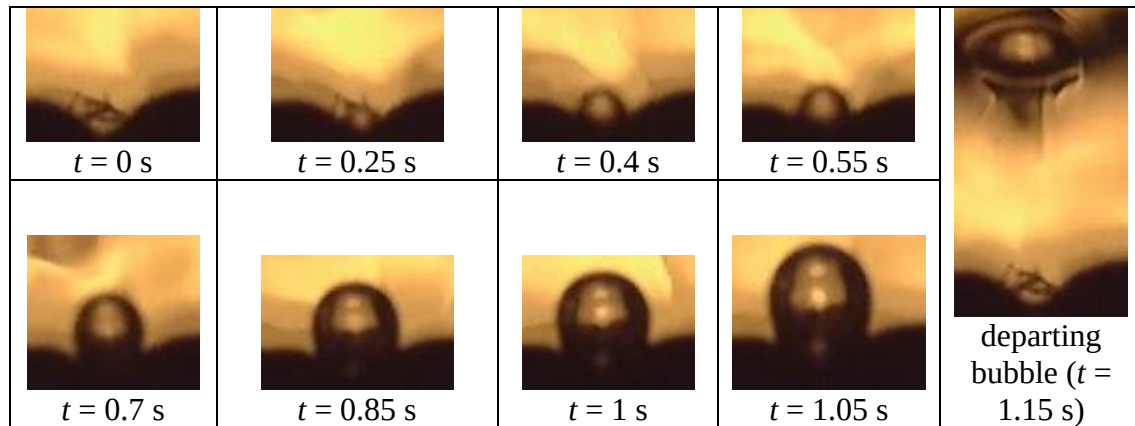
Surface roughness data was obtained for the flat and MWG surfaces by profilometry, as shown in Table 7. A calibrated Alicona InfiniteFocus 3D optical microscope was used to scan a section of the 25 mm flat and both 12.5 mm and 25 mm MWG surfaces. With the flat surface, the surface roughness variables are isotropic. The arithmetic mean surface roughness is  $R_a$ , the root-mean-square roughness is  $R_q$ , and the mean peak-to-valley height is  $R_z$ . Bubble departure diameter was obtained with a high speed camera and software, as shown in Figure 14 and Figure 15.

Table 7: Surface roughness characteristics.

|                     | [ $\mu\text{m}$ ] | $R_a$ | $R_q$ | $R_z$ |
|---------------------|-------------------|-------|-------|-------|
| <b>flat</b>         | mean              | 0.832 | 0.470 | 3.66  |
|                     | $SD$              | 0.092 | 0.435 | 0.543 |
| <b>along ridge</b>  | mean              | 0.493 | 0.628 | 2.71  |
|                     | $SD$              | 0.138 | 0.181 | 0.848 |
| <b>along trough</b> | mean              | 1.01  | 1.50  | 5.60  |
|                     | $SD$              | 0.338 | 0.707 | 2.22  |



Figure 14: Bubble nucleation process and departure diameter at the onset of nucleate boiling on flat surface, showing attached and growing, departing, and departed bubbles.

Figure 15: Time lapse stills showing the growth and departure of a single bubble on the large MWG surface. Note the stable vapour pocket remaining in the trough at  $t = 1.15$  s.

### 3.4. Results and Discussion

#### 3.4.1. Data Statistics

As three repeats were made of each test, the repeat data was averaged to provide a sample mean heat flux, excess temperature, and heat transfer coefficient at each power setting. The sample error was calculated for each power setting as

$$SE = \frac{SD}{\sqrt{N}}, \quad (3.15)$$

where  $SD$  is sample standard deviation, and  $N$  is the number of data points. As the sample errors are too small to be plotted graphically as error bars on each data point, for each test, the mean sample error for each variable was calculated as 0.7% for  $q''$ , 0.78% for  $\Delta T$ , and 1.07% for  $h_{NB}$ . The error values are very low and indicate good repeatability. No statistically-significant difference in excess temperature was observed between decreasing-flux and increasing-flux conditions.

### 3.4.2. Surface Profile and Size Effects

Figure 16 indicates that, for a given heat flux, the excess temperature  $\Delta T$  is lower for the MWG than for the flat surface for both surface sizes, except that the 25 mm surfaces appear to converge at low heat fluxes. The excess temperature difference between the 12.5 mm surfaces is more pronounced than between the 25 mm surfaces.

Considering size effects, the 12.5 mm flat surface exhibited a greater excess temperature at all heat fluxes compared to the 25 mm flat surface. The 12.5 mm MWG surface gave a consistently lower excess temperature throughout the range of heat fluxes tested, compared to the 25 mm MWG surface. This is partly explained by the larger true-to-planar area ratio of the smaller MWG surface. The increased influence of edge effects on small surfaces may also contribute; the ratio of perimeter to area is two times higher for the smaller surfaces compared to the larger surfaces.

The areas corresponding to the heat fluxes and heat transfer coefficients expressed in this paper are planar (i.e. the area observed when looking perpendicularly onto the surface from above). However, because the MWG surfaces consist of a series of ridges, the surface area wetted by the coolant  $A_w$  is greater than the flat case. Considering the wetted area to obtain the heat flux, using

$$f_A = A_{\text{MWG}} / A_{\text{flat}}, \text{ and} \quad (3.16)$$

$$q''_{\text{comp}} = q''_{\text{MWG}} / f_A, \quad (3.17)$$

where the true-to-planar area ratio  $f_A$  is obtained from profilometry data as 1.23 for the smaller surface and 1.13 for the larger surface. The resultant surface area compensated flat plate boiling curves are compared with the equivalent sized MWG plate boiling curves in Figure 17. Both MWG surfaces continued to display improved heat transfer coefficients, even with area compensation, which suggests a geometrical advantage independent of surface area.

Bubble nucleation on the MWG surface first takes place in the surface troughs. Figure 15 shows the development and departure of a single bubble from a trough, whilst the nucleation of several bubbles in troughs is shown in Figure 18(a). The onset of nucleate boiling occurring in the troughs appears to be the result of advantageous microgeometry in the troughs, due to the EDM fabrication process, giving stable nucleation sites. This is evidenced by the greater surface roughness in the troughs than on the ridges. Nucleation sites are stable when a local vapour pocket is maintained in a microfracture in the surface, so enabling regular bubble ebullition. By comparison, the roughness of the flat surface is lower than the trough roughness and so a higher heat flux is required to generate the same number of stable nucleation sites on the flat surface. Modelling the surface augmentation as a triangular fin of 0.2 mm height, and assuming a constant heat transfer coefficient of  $2 \text{ kW m}^{-2} \text{ K}^{-1}$  results in a  $0.002 \text{ }^\circ\text{C}$  temperature difference between ridge and trough, indicating that ridge-to-trough temperature difference is not the cause of trough bubble nucleation at the onset of nucleate boiling.

Bubble departure diameter helps explain the enhanced heat transfer coefficients demonstrated by the MWG surface. Average bubble departure diameters were obtained

by analysing 20 bubbles each from the flat and MWG surfaces, at the point of departure and at the onset of nucleate boiling, and were found to be 270  $\mu\text{m}$  (SD = 43.1  $\mu\text{m}$ ) and 586  $\mu\text{m}$  (SD = 39.0  $\mu\text{m}$ ) respectively. Suszko and El-Genk (Suszko and El-Genk, 2016) observed that bubble departure diameter on a dimpled copper surface was larger than on a flat surface, that the bubble departure diameter increased with increasing dimple diameter, and that very little nucleation took place on the flat surfaces between the dimples. This phenomenon can be explained in terms of surface tension force  $F_\sigma$  causing the vapour to adhere to the solid surface:

$$F_\sigma = \sigma L, \quad (3.18)$$

where  $L$  is the length of the contact line between the vapour and the surface. Comparing the bubble detachment from the flat surface in Figure 14 and the MWG surface in Figure 18 reveals that the MWG surface produces a significantly longer contact line, and therefore a significantly higher surface tension force. For a bubble to detach, the buoyancy force must exceed the surface tension force. At the moment of detachment, in eqn. (3.18) length  $L = \pi d_b$ . Hence, the mean buoyancy force required to cause bubble detachment from the MWG surface is around twice as large as from the flat surface. Therefore, a larger bubble is required to produce sufficient buoyancy force to detach the bubble from the MWG surface compared to the flat surface. As the heat removed by a single bubble is given by

$$Q = \frac{\pi}{6} d_b^3 f_d \rho_v h_{fg}, \quad (3.19)$$

a doubling in bubble diameter corresponds to an eightfold increase in heat removal, all other parameters remaining constant. However, for the same planar heat flux, a larger bubble will take longer to develop than a smaller bubble. This was observed experimentally, explaining why the actual heat transfer coefficient enhancement of the

MWG surface is much more modest than predicted solely by the change in bubble departure diameter.

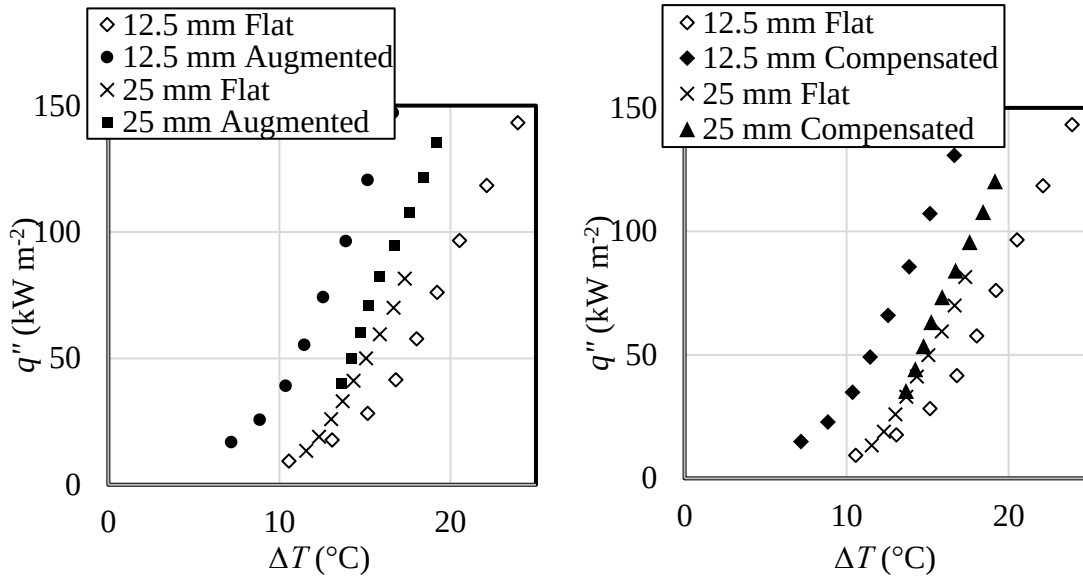


Figure 16 (left): A comparison of the flat and MWG surface boiling curves for both surface sizes. Figure 17 (right): A comparison of the boiling curves for the flat surfaces with the area-compensated boiling curves for the MWG surfaces.

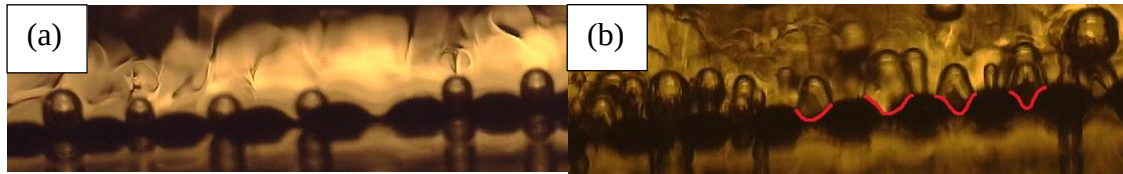


Figure 18: Photo showing (a) the onset of nucleate boiling with nucleation exclusively in troughs, and (b) fully developed nucleate boiling with nucleation predominantly in troughs and the contact line highlighted in red for selected bubbles.

### 3.4.3. Nucleate Boiling Correlations

The Rohsenow correlation was “fitted” by performing logarithmic regression to obtain the index  $n$  and the surface-fluid constant  $C_{s,l}$  for each surface, as shown in Table 9. Unsurprisingly, the fitted Rohsenow correlation gave the lowest MD for every surface tested. The fitting parameters in Table 9 are consistent with the report of previous studies (Chen, 1978; Fazel and Hosseini, 2016). For the MWG surface, the roughness parameter used in the Leiner and Stephan-Abdelsalam correlations is the mean of the direction-specific MWG roughness values presented in Table 7.

Figure 19 shows the experimental data against the nucleate boiling correlations discussed in Section 3.2.1. The mean absolute difference  $MD$  for each correlation was calculated as

$$MD = \frac{1}{N} \sum_{i=1}^N \frac{|h_{NB,correlation,i} - h_{NB,experiment,i}|}{h_{NB,experiment,i}}, \quad (3.20)$$

where  $N$  is the number of data points. Means of the  $MD$  values were then taken to provide a single  $MD$  for each correlation against each surface type irrespective of fluid or surface size. Finally, an average  $MD$  for each correlation without respect to fluid, surface size, or surface type, was provided, as shown in Table 8. The  $MD$  for correlations with respect to both flat and MWG surfaces is tabulated.

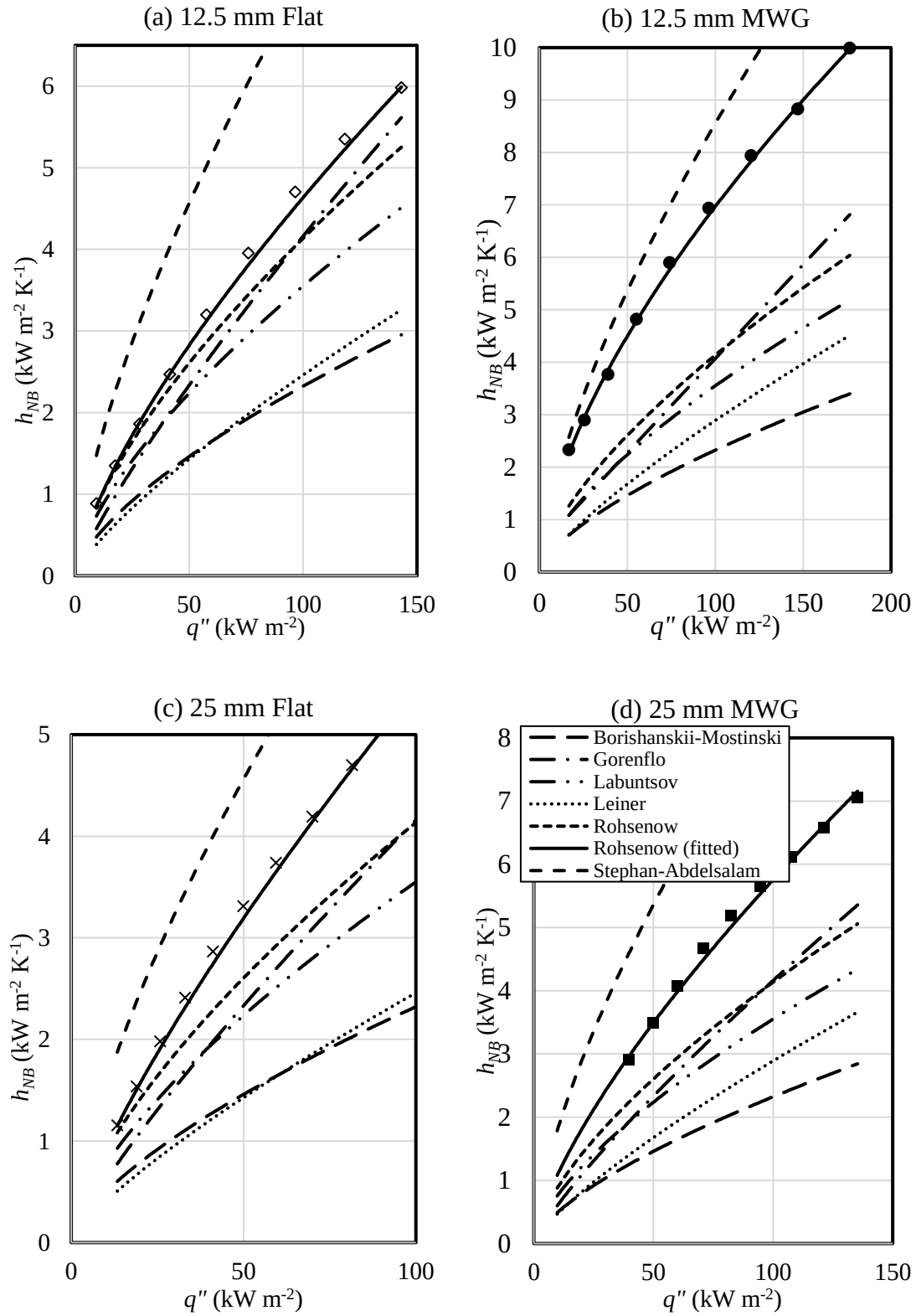


Figure 19: A comparison of the experimental data with nucleate boiling correlations; legend inset to graph (d).

Table 8: Mean absolute difference  $MD$  between correlations and measured data.

|                        | Flat  | MWG   | Mean  |
|------------------------|-------|-------|-------|
| Borishanskii-Mostinski | 51.5% | 61.4% | 56.5% |
| Gorenflo-Kenning       | 24.4% | 37.6% | 31.0% |
| Kutateladze            | 80.9% | 85.6% | 83.3% |
| Kruzhilin              | 81.4% | 39.1% | 60.3% |
| Labuntsov              | 25.7% | 44.1% | 34.9% |
| Leiner                 | 53.5% | 56.1% | 54.8% |
| Mikic-Rohsenow         | 33.3% | 12.3% | 22.8% |
| Rohsenow               | 13.5% | 34.9% | 24.2% |
| Rohsenow (fitted)      | 5.77% | 4.93% | 5.35% |
| Stephan-Abdelsalam     | 51.4% | 34.4% | 42.9% |

Table 9: Curve-fitting parameters for the Rohsenow correlation.

|           | 12.5   | 12.5 MWG | 25     | 25 MWG |
|-----------|--------|----------|--------|--------|
| $C_{s,l}$ | 0.0053 | 0.0034   | 0.0046 | 0.0029 |
| $n$       | 0.2845 | 0.3687   | 0.2248 | 0.2797 |

The Leiner and Stephan-Abdelsalam correlations' modeling of roughness were examined. The roughness variable in both correlations is index to the power of  $2/15$ . Increasing the surface roughness from  $1.98 \mu\text{m}$  to  $2.78 \mu\text{m}$  increased the predicted heat transfer coefficient by 4.67% in both correlations, which is lower than the experimental uncertainty. They are not especially good predictors of heat transfer coefficient overall, returning  $MD$  of 53.5% and 51.4%, respectively, for flat surfaces. The Gorenflo correlation gave better experimental agreement than either but, at  $MD = 24.4\%$  for flat surfaces, was not as good as the Rohsenow correlation and only marginally better than the *a priori* Labuntsov correlation.

The “unfitted” Rohsenow correlation, using the default fitting values suggested for copper on refrigerant of  $n = 1/3$  and  $C_{s,l} = 0.0059$  fitted the flat surface data slightly better than the Labuntsov correlation, at 13.5% and 25.7%  $MD$  respectively. This raises the question as to whether a correlation that requires fitting to data is useful for engineering design purposes. A logarithmic regression of a raw dataset almost invariably

provides a best-fit curve with less error than an *a priori* correlation, but only applies immediately to the particular dataset from which it originates. In reviewing the Rohsenow correlation, it was found that this argument was raised in 1977 by Fand and Ho (1977). Rohsenow's 1979 response (Rohsenow, 1979) criticised a surface aging process used by Fand and Ho in their experimental setup, and suggested use of the later Mikic-Rohsenow correlation, but failed to address the fundamental argument that a correlation requiring data fitting is not especially useful as a predictor.

The Borishanskii-Mostinski correlation appeared to have the correct curve trajectory but under-predicted each surface. The Kutateladze correlation under-predicted each surface by some margin and is not shown on Figure 19.

#### **3.4.4. Augmented Surface Orientation Effects**

The effect of surface orientation  $\phi$  on heat transfer with nucleate boiling of Novec 7000 was examined and shown in Figure 20, where  $0^\circ$  is the default orientation (heated horizontal surface facing upwards) and  $90^\circ$  is a vertically oriented surface. As the surface profile is anisotropic in the x-y plane (see Figure 10), the orientation of the ridges can be expected to affect heat transfer where the surface orientation is not horizontal, because the ridge orientation may aid or impede bubble flow. To study the effect of ridge orientation, the  $90^\circ$  surface orientation was repeated with the ridges aligned parallel to the x-axis (horizontal) and the ridges parallel to the y-axis (vertical). All other surface orientations were tested with the ridges parallel to the x-axis, as shown in Figure 21.

The  $180^\circ$  orientation saw a boiling crisis occur almost instantaneously, as the vapour bubbles formed were held against the heated surface by buoyancy, confirming Chen's observations (1978). As nucleate boiling did not occur at  $180^\circ$ , the orientation is not represented in Figure 20.

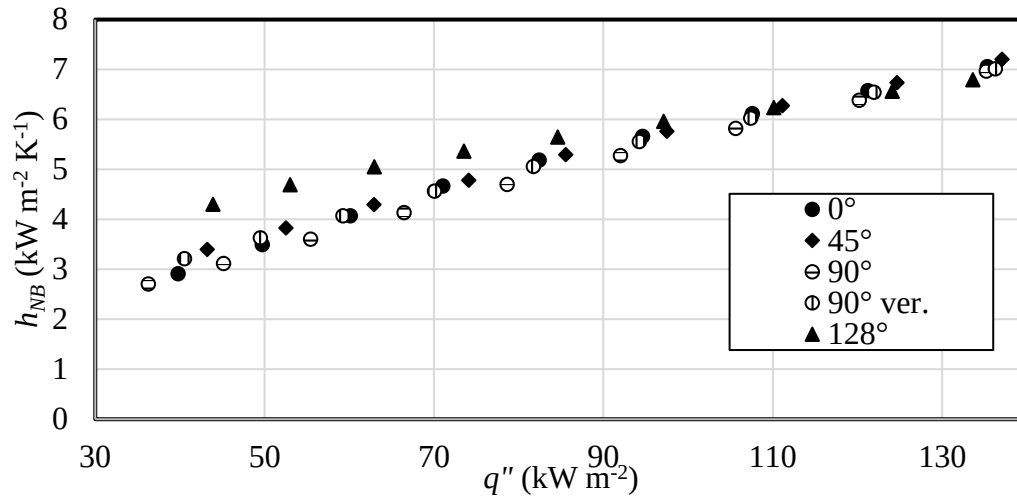


Figure 20: The effect of MWG surface orientation on heat transfer coefficient.

Changing the orientation from 0° to 90° did not greatly affect the heat transfer coefficient across the range of heat fluxes tested. Compared to the default 0°, the 128° orientation exhibited greater heat transfer coefficients at low heat fluxes (<110 kW m<sup>-2</sup>) and lower coefficients above this. This data is consistent with previous orientation experiments with fluorocarbons on flat surfaces (Rainey and You, 2001; El-Genk and Bostanci, 2003; Priarone, 2005).

The effect of the augmentation alignment at 90° orientation was also examined. The vertically-aligned ridges displayed a higher heat transfer coefficient than the horizontally-aligned ridges across all heat fluxes. In the case of the vertically-aligned ridges, the bubbles slid unimpeded up the valleys to activate other nucleation sites and form large, coalesced bubbles as observed by Jung and Kim (2016). With the horizontally-aligned ridges, a bubble must negotiate out of the valley in which it nucleated in order to rise freely; therefore the vapour, which has a lower thermal conductivity than the liquid, is in contact with the surface for longer and results in a lower heat transfer coefficient.

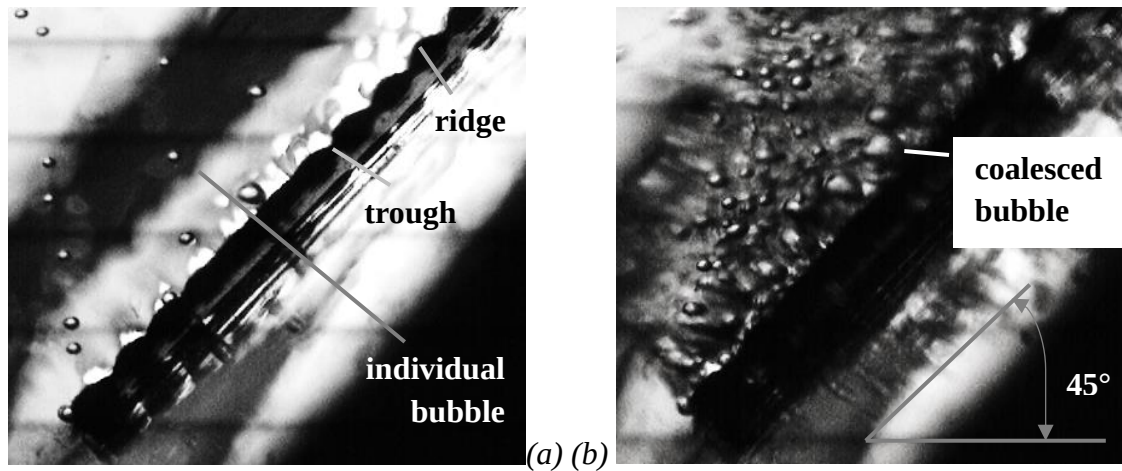


Figure 21: (a) Initiation of nucleate boiling with uniform bubble departure diameter, and (b) vigorous nucleate boiling with coalesced bubbles, on  $\phi=45^\circ$  MWG surface.

### 3.5. Conclusions

Nucleate boiling on 12.5 x 12.5 and 25 x 25 mm flat and motor windings geometry (MWG) surfaces in Novec 7000 was performed with the following conclusions:

- The MWG surfaces resulted in higher heat transfer coefficients at all heat fluxes, with a mean 8.1% increase between 49 and 82 kW m<sup>-2</sup>.
- Where accounting for increased wetted area, the MWG surfaces continued to show superior heat transfer to the flat surfaces, at a mean 4.2% increase between 35 and 73 kW m<sup>-2</sup>. The mean bubble departure diameter from the MWG surfaces was significantly larger than from the flat surfaces.
- The Labuntsov correlation provided the best entirely unfitted *a priori* prediction of the heat transfer coefficient for flat surfaces.
- The 1952 Rohsenow model could be made to fit the data well using the experimentally-determined constants as recommended.

The 25 mm MWG surface was tested in Novec 7000 at a range of surface orientations from 0° to 128°. For orientations from 0° to 90°, heat transfer coefficient was not significantly affected by orientation. The effect of ridge alignment at the 90° surface orientation was considered, with the vertically-aligned ridges displaying lower

heat transfer coefficients than the horizontally-aligned ridges throughout the heat flux range tested. The 128° orientation displayed manifestly higher heat transfer coefficients at low heat fluxes, and lower coefficients at high heat fluxes.

Characterisation of motor windings geometry surfaces in pool boiling has been obtained, which enables more effective cooling strategies to be designed and, in turn, more efficient and environmentally-friendly electrical machines. Further work into size effects on heat transfer with the surface profiles tested may yield greater insight in the future.

### 3.6. Statement of Authorship

#### Statement of Authorship for joint/multi-authored papers for PGR thesis

To appear at the end of each thesis chapter submitted as an article/paper

The statement shall describe the candidate's and co-authors' independent research contributions in the thesis publications. For each publication there should exist a complete statement that is to be filled out and signed by the candidate and supervisor.

|                     |                                                                                                                                                                                                                                                                                                                                            |
|---------------------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| Title of Paper      | Pool Boiling of Resin-Impregnated Motor Windings Geometry                                                                                                                                                                                                                                                                                  |
| Publication Status  | <input checked="" type="checkbox"/> Published<br><input type="checkbox"/> Accepted for Publication<br><input type="checkbox"/> Submitted for Publication<br><input type="checkbox"/> Unpublished and unsubmitted work written in a manuscript style                                                                                        |
| Publication Details | Bartle, R. S., Menon, K. and Walsh, E. (2018) 'Pool Boiling of Resin-Impregnated Motor Windings Geometry', <i>Applied Thermal Engineering</i> , 130, pp. 854–864. Available at:<br><a href="https://www.sciencedirect.com/science/article/pii/S1359431117342400">https://www.sciencedirect.com/science/article/pii/S1359431117342400</a> . |

#### Student Confirmation

|                                          |                                                                                                                                 |            |  |
|------------------------------------------|---------------------------------------------------------------------------------------------------------------------------------|------------|--|
| Student Name:                            | Roy Sowdagar Bartle                                                                                                             |            |  |
| Contribution to the Paper                | Co-designed experimental rig and method with supervisor;<br>performed test programme;<br>analysed data;<br>prepared manuscript. |            |  |
| Signature [redacted from online version] | Date                                                                                                                            | 21/05/2019 |  |

### Supervisor Confirmation

By signing the Statement of Authorship, you are certifying that the candidate made a substantial contribution to the publication, and that the description described above is accurate.

|                                               |      |         |
|-----------------------------------------------|------|---------|
| Supervisor name and title: Prof. Edmond Walsh |      |         |
| Supervisor comments<br>None                   |      |         |
| Signature [redacted from online version]      | Date | 11/9/19 |

This completed form should be included in the thesis, at the end of the relevant chapter.

## **4. Pool Boiling of a Copper Surface with Epoxy Resin Microlayer**

Unpublished thesis chapter.

### **4.1. Introduction**

This experiment exists to gain insight into the effect of a thin layer of resin above a copper surface in pool boiling. The setup replicates the surface of a turn of copper windings with a thin layer of electrical insulation on the surface, as can exist in the stator of an electrical machine. This introduces an additional thermal resistance between the copper and the coolant. Additionally, the effect of a resin microlayered surface on pool boiling performance is explored.

#### **4.1.1. Method and Materials**

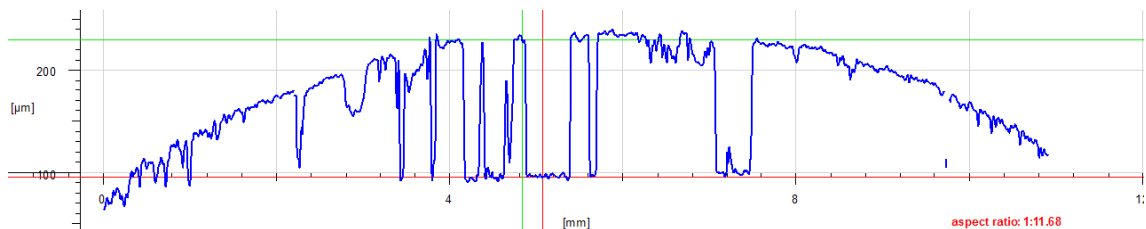
The basic experimental rig and method is as per Chapter 3.3. Once the module had set, a paintbrush was dipped into resin, excess resin was removed, and the brush was run three times over the surface and left to cure; this covered the copper surface in a layer of resin with thickness in the order of microns. The cured module was then placed into the boiling chamber. The coolant used was Novec 7000.

To provide specimens for destructive surface profilometry, three specimens with identical dimensions were fabricated from the same copper stock and prepared with a resin coating in the same manner as the boiling surface. Profilometry served two roles in this experiment. Firstly, it provided experimental values for surface roughness values, as used in pool boiling correlations, of the resin-coated surface. The experimental surface roughness values were obtained as described in Chapter 3.3 and are detailed in Table 10. Secondly, the thickness of the resin layer was quantified. To obtain this, scratches into

the resin were made with a stylus to remove the resin layer without eroding into the copper. Two samples each received two scratches, oriented in perpendicular directions, running from the centre of the sample to the edge. Using the Alicona software, the resin thickness was measured at six points on each scratch. The mean thickness and standard deviation was calculated for each scratch; the mean thickness was calculated as 60.0  $\mu\text{m}$  (SD = 27.1  $\mu\text{m}$ ). The raw profilometry plots of the scratched resin surfaces are shown in Figure 22.

*Table 10: Surface roughness metrics for the plain copper and resin microlayered surfaces.*

|           | Plain Copper |       | Resin<br>Microlayered |       |               | Description                                                                 |
|-----------|--------------|-------|-----------------------|-------|---------------|-----------------------------------------------------------------------------|
|           | mean         | SD    | mean                  | SD    |               |                                                                             |
| $R_a$     | 833          | 93.0  | 556                   | 75.8  | nm            | Average roughness of profile                                                |
| $R_q$     | 470          | 435   | 724                   | 88.0  | nm            | Root-Mean-Square roughness of profile                                       |
| $R_t$     | 5.24         | 0.921 | 4.21                  | 0.921 | $\mu\text{m}$ | Maximum peak to valley height of roughness profile                          |
| $R_z$     | 3.66         | 0.543 | 3.04                  | 0.489 | $\mu\text{m}$ | Mean peak to valley height of roughness profile                             |
| $R_{max}$ | 5.17         | 1.00  | 4.04                  | 0.960 | $\mu\text{m}$ | Maximum peak to valley height of roughness profile within a sampling length |
| $R_p$     | 2.43         | 0.435 | 2.12                  | 0.448 | $\mu\text{m}$ | Maximum peak height of roughness profile                                    |
| $R_v$     | 2.81         | 0.597 | 2.08                  | 0.707 | $\mu\text{m}$ | Maximum valley height of roughness profile                                  |
| $R_c$     | 3.19         | 0.498 | 2.08                  | 0.328 | $\mu\text{m}$ | Mean height of profile irregularities of roughness profile                  |



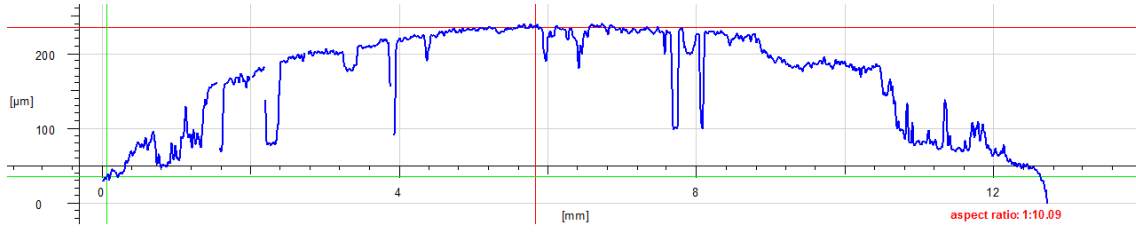


Figure 22: Raw profilometry plots of the scratched resin microlayered surfaces.

The nomenclature used in the experiment is diagrammed in Figure 23. For the resin microlayered surface, the temperature  $T_w$  was calculated from Fourier's law,

$$T_w = T_{Cu} - \frac{q''}{k} \Delta z, \quad (4.1)$$

where  $T_{Cu}$  is the temperature at the copper-resin interface and  $\Delta z$  is the mean thickness of the resin microlayer. Heat flux  $q''$  and temperature  $T_{Cu}$  were calculated with eqn. (3.13).

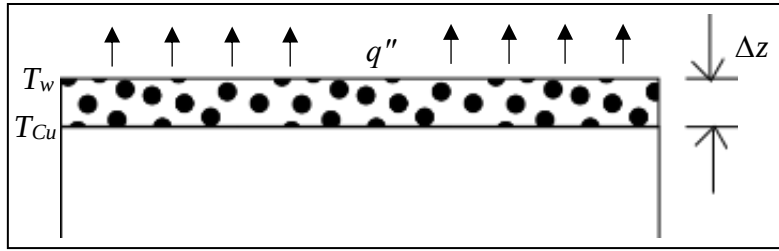


Figure 23: Epoxy-resin microlayer with thickness  $\Delta z$  on copper block.

#### 4.1.2. Results and Discussion

To illustrate the effect of the resin microlayer, boiling curves for the plain copper surface and the resin microlayered surface are plotted in terms of heat transfer coefficient (Figure 24) and excess temperature (Figure 25).

The contribution of the reduced surface roughness to the decreased heat transfer coefficient of the resin microlayer may be estimated using correlations that include surface roughness parameters: the Cooper (1984) correlation with  $R_a$  (eqn. (2.14)), and the Gorenflo-Kenning (2010) correlation with  $R_p$  (eqn. (3.6)), are used here. Both correlations predict a heat flux -independent percentage decrease in heat transfer coefficient due to decreased surface roughness: 3.83% and 5.25% for Cooper and

Gorenflo-Kenning respectively. However, Figure 24 shows that the heat transfer coefficient ratio between the resin microlayered and plain surfaces is not independent, but decreases, with heat flux. Furthermore, the experimental heat transfer coefficient decrease between the plain and the resin surface was larger than predicted. Therefore the predictive correlations, utilising single surface roughness metrics, only partly capture surface microgeometry effects on heat transfer coefficient.

The small discrepancy between the predicted and experimental heat transfer coefficient ratios of the plain and resin microlayered surfaces emphasises the inadequacy of a single surface roughness metric to quantify the effect of surface microgeometry on heat transfer coefficient. This has been emphasised in literature (Pioro, Rohsenow and Doerffer, 2004a; McHale and Garimella, 2013a) but hitherto no pool boiling correlation has been introduced that describes surface microgeometry effects in detail. A higher number of active nucleation sites gives a lower wall temperature for a given heat flux (Hibiki and Ishii, 2003). As the activation and stability of nucleation sites is linked to cavity size and thus surface roughness (Webb, 1981; Kandlikar and Spiesman, 1998; Jones, McHale and Garimella, 2009; Kim *et al.*, 2015), and the resin surface has lower surface roughness, the experimental results are explicable and consistent with the present state and limitations of published literature.

Figure 25(b) shows the effect of the additional thermal resistance of the resin microlayer on the copper junction temperature. From an applied perspective, where the objective is to minimise copper temperature, the low thermal conductivity of the resin contributes more significantly to the overall thermal resistance than the reduced boiling heat transfer coefficient.

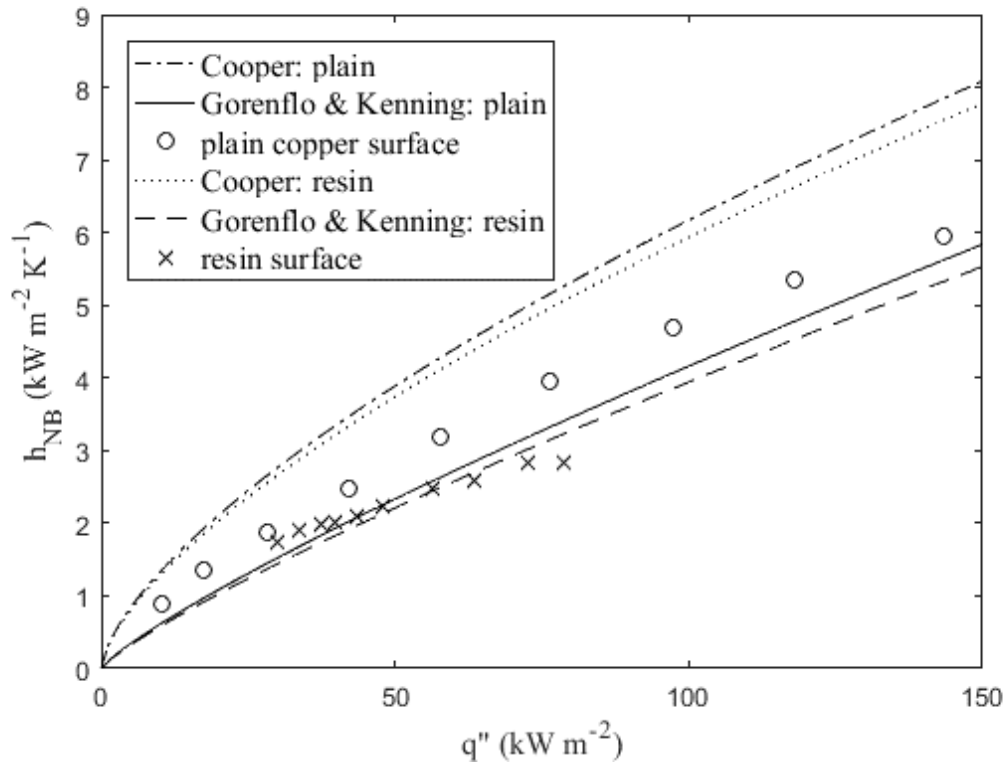


Figure 24: A graph showing the experimental relationship between heat flux and heat transfer coefficient for the plain copper and resin microlayered surfaces, together with the predicted heat transfer coefficients from the Cooper and Gorenflo-Kenning correlations.

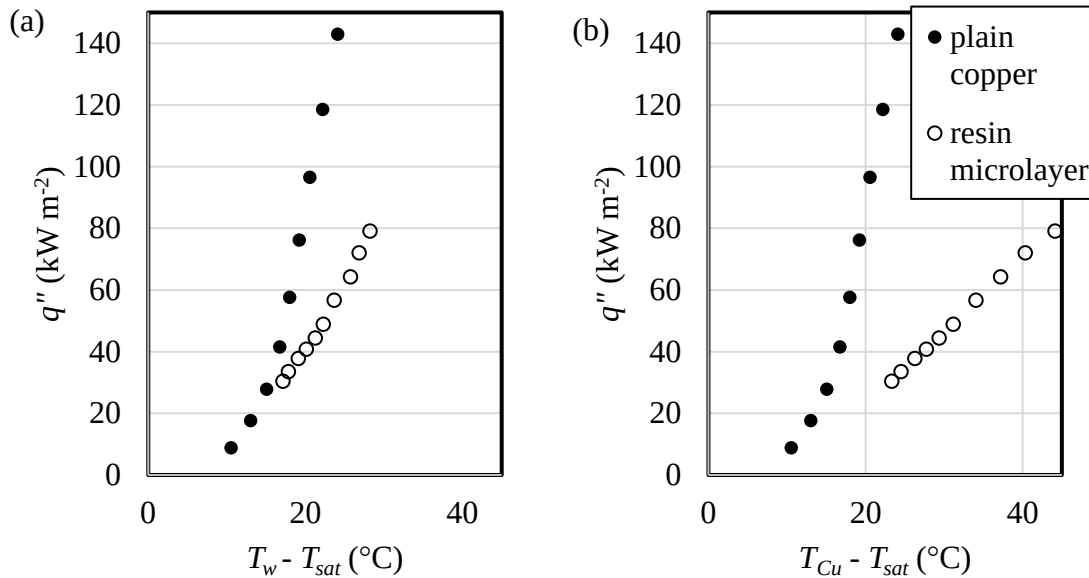


Figure 25: Boiling curves showing the plain copper surface temperatures compared to (a) the temperature on the surface of the resin microlayer, and (b) the temperature at the copper-resin junction of the resin microlayered surface.

#### **4.1.3. Conclusions**

A comparison of pool boiling on plain copper and resin microlayered surfaces was performed. The resin microlayered surface exhibited a lower boiling heat transfer coefficient than the plain copper surface due to surface microgeometry, the effect of which is only partly described in pool boiling correlations that include roughness parameters. Considering the surface as approximating resin-coated windings, the thermal resistance produced by the low thermal conductivity resin is more significant to the temperature increase of the copper than the deteriorated boiling heat transfer coefficient.

## 5. Effect of Spatial Confinement and Liquid Level on Nucleate Boiling

Extended abstract published as: Bartle, R. S., Menon, K. and Walsh, E. J. (2017) ‘Effect of Spatial Confinement and Liquid Level on Nucleate Boiling’, in *Proceedings of the 15th UK Heat Transfer Conference (UKHTC2017)*. Uxbridge, pp. 166–167. The full content, as presented in this Chapter, was given as an oral presentation at the conference.

### Abstract

The effect of spatial confinement ( $C = 0.24$  to  $60$  mm) on nucleate boiling heat transfer from a vertically oriented flat surface is examined. The effect of varying liquid level above an horizontally upwards boiling surface is considered. A modified Bond number  $Bo_v$  is proposed to describe 2D liquid flows bound by a solid and a vapour layer; heat transfer is shown to decrease very significantly when  $Bo_v$  is below unity in the case of decreasing liquid level above an heated surface.

### 5.1. Introduction

The US National Academies of Science, Engineering, and Medicine (2016) reported the historical development of installed electrical generation capacity in aircraft in Figure 26. This suggests that the specific power of electrical machines in an aircraft as of 2016 was around  $2.2 \text{ kW kg}^{-1}$  and expects an increase to around  $9 \text{ kW kg}^{-1}$  by 2036. For an induction machine with 95% efficiency, this translates to a heat dissipation requirement of  $450 \text{ W kg}^{-1}$ . Considering a hypothetical (albeit realistic) machine of  $350$  kg mass with length  $0.5$  m and diameter  $0.7$  m, a mean heat flux of  $34.3 \text{ kW m}^{-2}$  results if the machine is cooled from all surfaces. Minimisation of cooling system mass and volume whilst maintaining reliability is of paramount importance in aeronautical applications. Cooling channels should therefore be as narrow as possible to reduce

volume and minimise coolant inventory without adversely affecting heat transfer coefficient.

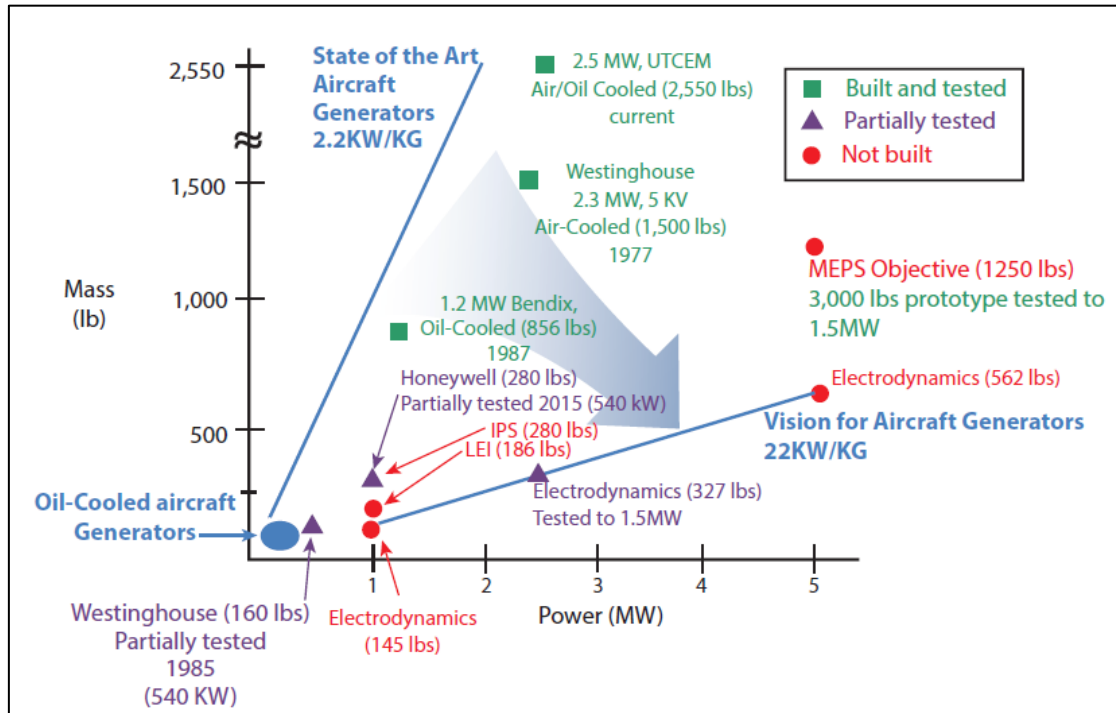


Figure 26: An history of electrical machine development in aircraft (Source: US National Academies of Science, Engineering, and Medicine (2016)).

Prior work has been undertaken on the effects of surface confinement in nucleate boiling, as reviewed in Chapter 2.2.4. Especially pertinent are tests of vertical heated surfaces confined by a parallel unheated surface in passive pool boiling (Ishibashi and Nishikawa, 1969; Fujita *et al.*, 1988; Bonjour and Lallemand, 1998; Geisler and Bar-Cohen, 2009). These show the general trend that moderate confinement augments heat transfer coefficient at low heat flux but decreases it at high heat flux.

In flooded evaporators, system design and application usually determines liquid level. For static applications where space is not at a premium, a cautious high liquid level is often selected (Whitman, Johnson and Tomczyk, 2000) to mitigate against premature dryout on upper tubes. However, excessive fluid inventory increases system mass and, depending on the fluid properties (e.g. flammability, toxicity) can increase the negative

consequence of rupture risk scenarios. Despite this, very little work has been published on the effects of liquid level (or head) in pool boiling. Using liquid potassium, Michiyoshi *et al.* (1958) found the heat transfer coefficient to be adversely affected at liquid levels lower than 50 mm head. Various authors (Moeykens, 1994; Ayub, Chyu and Ayub, 2006; Fernández-Seara and Pardiñas, 2014; Swain and Das, 2014) make passing reference to the effect of liquid head in flooded evaporators in increasing the saturation temperature of the fluid on the tubes underneath by increasing the hydrostatic pressure. The absolute pressure  $p_{abs}$  at a point in a bulk liquid is

$$p_{abs} = p_{atm} + p_H, \text{ where hydrostatic pressure} \quad (5.1)$$

$$p_H = \rho g H, \quad (5.2)$$

and  $H$  is the height of the liquid head above the point in consideration.

## 5.2. Method and Materials

The basic experimental rig and method is as per Chapter 3.3. Modifications were made to enable evaluation of spatial confinement and liquid level effects.

In the spatial confinement experiments, the copper boiling surface was oriented vertically. Parallel to this, an unheated glass surface was introduced. The distance  $C$  between the two surfaces was measured by microscopic camera in all cases except the smallest confinement ( $C = 0.24$  mm), where mechanical spacers were used and situated sufficiently far away from the boiling surface to avoid disrupting fluid flow. This is diagrammed in Figure 27(a). Novec 649 was the coolant used.

The fluid level experiments used the copper boiling surface in horizontal configuration. Firstly, a reference experiment was performed to demonstrate the independence of heat transfer coefficient with time in steady-state conditions. Then, the fluid level  $H$  was gradually decreased by evaporation to an external condensing tank, as

shown in Figure 27(b). A comparison of these two runs is shown in Figure 28. Fluid level measurements were taken by camera. The coolant used in these experiments was Novec 7000.

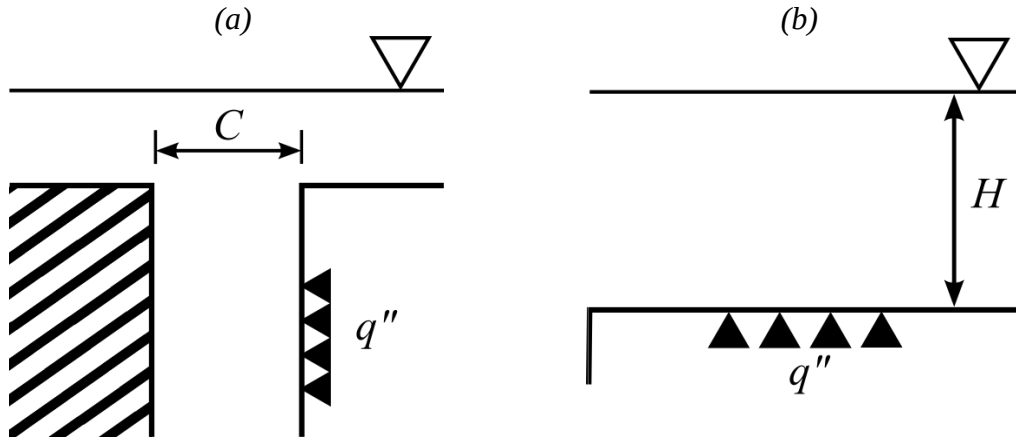


Figure 27: Diagrams to show (a) spatial confinement of a vertical heated surface, and (b) liquid head above an horizontal heated surface.

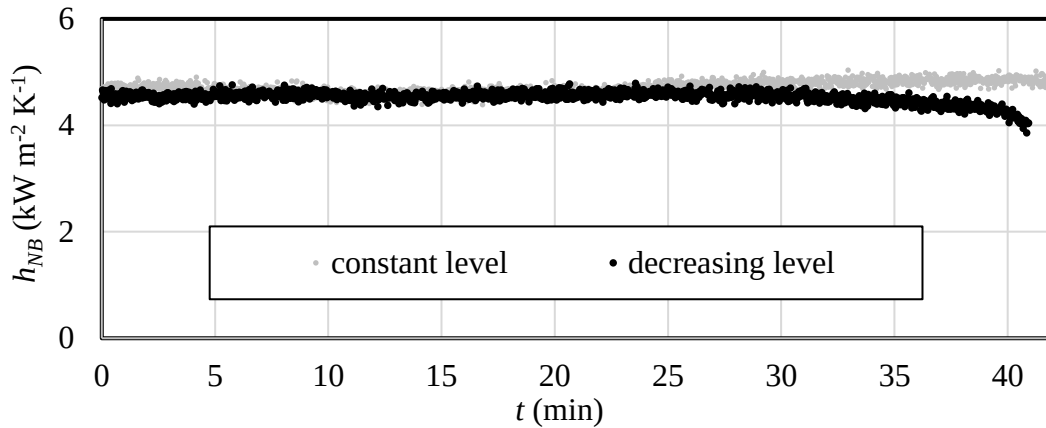


Figure 28: A graph showing the independence of heat transfer coefficient with time in steady-state conditions.

### 5.3. Results and Discussion

The depressing effect of liquid head on heat transfer coefficient is considered by substituting the coolant parameters into eqns. (5.1) and (5.2). The substitution of liquid density  $\rho_l$  in eqn. (5.2) is conservative as, once boiling occurs, the fluid above the point is a mixture of liquid and vapour with a mean density lower than that of pure liquid. For the maximum head tested ( $Bo = 10$ ,  $H = 8.55$  mm) the hydrostatic pressure component is

128 Pa. From the manufacturer's pressure-enthalpy data, this increases the saturation temperature of Novec 7000 by a negligible 0.03 °C, which is less than the temperature measurement uncertainty in this experiment. Nonetheless, for a tall industrial flooded evaporator the hydrostatic pressure acting on the lower tubes would need to be considered.

Figure 29 shows the effect of confinement on boiling heat transfer and excess temperature. Heat transfer coefficient is plotted against excess temperature rather than the more conventional heat flux in order to better emphasise the trends. Spatial confinement had a negligible effect on heat transfer coefficient except in the  $C = 0.5$  mm ( $Bo = 0.58$ ) case where, at high heat fluxes, heat transfer coefficient was depressed below the ordinary as the result of partial surface dryout. The smallest spacing  $C = 0.24$  mm ( $Bo = 0.28$ ) showed a rather different trend, with considerably higher heat transfer coefficient at low heat flux and considerably deteriorated heat transfer coefficient at high heat flux. At low heat fluxes and small spatial confinement, the observation has previously been made of departing vapour filling the centre of the channel whilst liquid adheres to the sides (Fujita *et al.*, 1988; Bonjour and Lallemand, 1998). Kunkelmann (Kunkelmann *et al.*, 2012; Raj *et al.*, 2012) explains that the high heat transfer coefficients result from a buoyancy-driven advancing three-phase contact line consisting of a thin liquid film, with very low thermal resistance, between the solid surface and the vapour bubble i.e. thin film evaporative cooling.

The relationship between liquid head  $H$  and heat transfer coefficient was also explored. To model a boiling fluid bounded vertically by a solid boundary underneath and a vapour boundary above the critical length from the Yao and Chang (1983) Bond number formulation in eqn. (2.37) was modified for liquid head:

$$Bo_H = H / \left( \frac{\sigma}{g(\rho_l - \rho_v)} \right)^{1/2}. \quad (5.3)$$

The effect of liquid head is modelled in terms of  $Bo_H$  in Figure 30. Heat transfer coefficient was found to be adversely affected where  $Bo_H < 5$ , and fell sharply when  $Bo_H < 1$ . Similar trends were observed by Misale et al. (2011) with solid horizontal boundaries of equivalent heights.

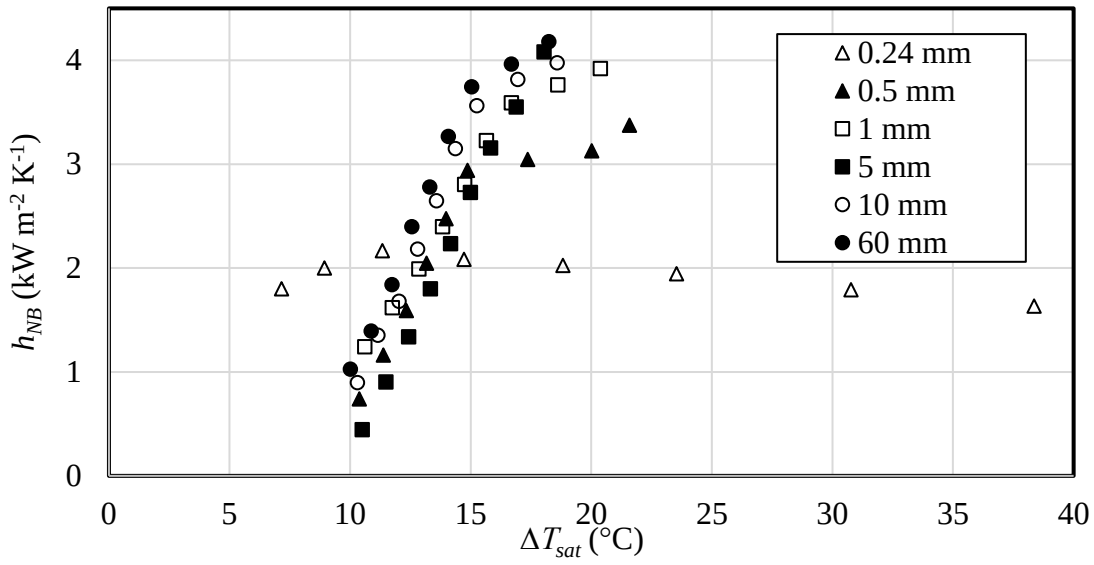


Figure 29: The relationship between temperature difference and heat transfer coefficient at a range of spatial confinements.

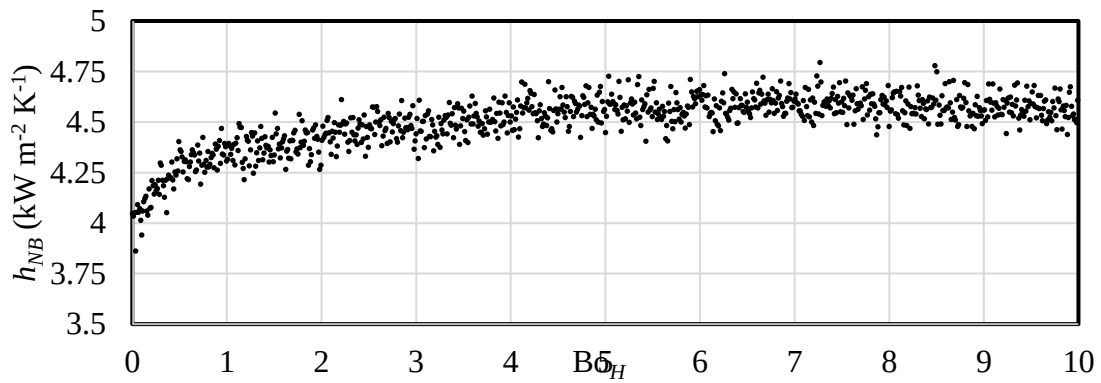


Figure 30: The effect of liquid head, expressed as Bond number, on heat transfer coefficient.

## 5.4. Conclusions

Pool boiling was performed in spatially confined and reducing liquid level configurations, with the following conclusions:

- Highly confined boiling gives higher heat transfer at low heat fluxes but lower heat transfer at high heat fluxes. Regarded in isolation, moderate confinements of planar surfaces negligibly affect heat transfer coefficient.
- The effect of liquid level on heat transfer coefficient in pool boiling is complex. At very low liquid heads ( $Bo_H < 5$ ) heat transfer coefficient is deteriorated, presumably due to the heated surface being starved of liquid replenishment. At very high liquid heads, the hydrostatic pressure acting on the fluid at the boiling surface becomes significant by raising the saturation temperature of the fluid and so decreasing heat transfer coefficient. In most published pool boiling experiments, the effect of fluid head is negligible; for most industrial applications, the fluid level should simply be maintained comfortably above the uppermost heated surface.

## 5.5. Statement of Authorship

### Statement of Authorship for joint/multi-authored papers for PGR thesis

To appear at the end of each thesis chapter submitted as an article/paper

The statement shall describe the candidate’s and co-authors’ independent research contributions in the thesis publications. For each publication there should exist a complete statement that is to be filled out and signed by the candidate and supervisor.

|                    |                                                                                                                                                                                                                                                     |
|--------------------|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| Title of Paper     | Effect of Spatial Confinement and Liquid Level on Nucleate Boiling                                                                                                                                                                                  |
| Publication Status | <input checked="" type="checkbox"/> Published<br><input type="checkbox"/> Accepted for Publication<br><input type="checkbox"/> Submitted for Publication<br><input type="checkbox"/> Unpublished and unsubmitted work written in a manuscript style |

|                     |                                                                                                                                                                                                                                            |
|---------------------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| Publication Details | Bartle, R. S., Menon, K. and Walsh, E. (2017) 'Effect of Spatial Confinement and Liquid Level on Nucleate Boiling', in <i>Proceedings of the 15th UK Heat Transfer Conference (UKHTC2017)</i> . Uxbridge, pp. 166–167. doi: 9781908549327. |
|---------------------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|

#### Student Confirmation

|                                          |                                                                                                                                 |      |            |
|------------------------------------------|---------------------------------------------------------------------------------------------------------------------------------|------|------------|
| Student Name:                            | Roy Sowdagar Bartle                                                                                                             |      |            |
| Contribution to the Paper                | Co-designed experimental rig and method with supervisor;<br>performed test programme;<br>analysed data;<br>prepared manuscript. |      |            |
| Signature [redacted from online version] |                                                                                                                                 | Date | 21/05/2019 |

#### Supervisor Confirmation

By signing the Statement of Authorship, you are certifying that the candidate made a substantial contribution to the publication, and that the description described above is accurate.

|                                               |  |                 |
|-----------------------------------------------|--|-----------------|
| Supervisor name and title: Prof. Edmond Walsh |  |                 |
| Supervisor comments<br>None                   |  |                 |
| Signature [redacted from online version]      |  | Date<br>11/9/19 |

This completed form should be included in the thesis, at the end of the relevant chapter.

## 6. Pool Boiling of Horizontal Mini-Tubes in Unconfined and Confined Columns

Bartle, R. S., and Walsh, E. J. (2019b) 'Pool Boiling of Horizontal Mini-Tubes in Confined and Unconfined Columns', *International Journal of Heat and Mass Transfer*, 145, 118733.

### Abstract

Pool boiling on confined and unconfined tube columns was investigated. A model to predict bulk void fraction in a confined tube column was developed and validated experimentally, returning a mean predictive error of 12.5%. A semi-empirical model was also developed from bubble plume images to evaluate bulk void fraction in an unconfined plume around a tube column. Within an unconfined plume, void fraction increases monotonically with column height, indicating that a column height exists beyond which the mean column heat transfer coefficient is adversely affected by dryness. Pool boiling experiments were undertaken in Novec 649 to investigate the effects of external confinement, tube spacing, and column height on heat transfer. Increasing the column height in a column showed an initial decline in power density to a constant at  $\sim 30 \text{ MW m}^{-3}$ . A tube void fraction threshold was identified as changing the location of critical heat flux from the bottom tube to the top tube in a column, and to cause a deterioration in tube heat transfer coefficient at high heat fluxes.

### 6.1. Introduction

Increasing heat fluxes in electrical machines and power electronics demands more compact, lightweight and efficient cooling solutions (Mudawar, 2001). Compact passive natural convection systems return power densities around  $50 \text{ kW m}^{-3}$  (Meng *et al.*, 2018), and active pumped air systems around  $250 \text{ kW m}^{-3}$ . As the latent heat of vaporisation of

a fluid is typically an order of magnitude larger than the specific heat capacity, phase-changing technologies can similarly improve power density. An industrial sized boiling tube bundle dissipates  $\sim 5 \text{ MW m}^{-3}$  (Gupta, 2005) while a compact microchannel flow boiling heat exchanger typically achieve  $\sim 180 \text{ MW m}^{-3}$  (Kwon *et al.*, 2019).

The minimisation of system mass and volume in safety critical systems is a major challenge to the aeronautical industry. As the installed capacity of electrical systems and power electronics increases in next generation aircraft, so do the heat dissipation requirements. For example, whereas traditional aircraft cooling systems use a chilled bleed air stream, the Boeing 787 uses a liquid system for power electronics cooling (Pacific Design Technologies, 2019). Similar ground-based power electronics cooling applications take advantage of the higher heat transfer coefficients afforded by phase-change heat transfer in a dielectric coolant (Kheirabadi and Groulx, 2016).

The study of tube bundles geometry is relevant not only to shell-and-tube heat exchangers with fluid flow both internal and external to the tubes, but also provides insights into the boiling performance of a horizontally oriented pin finned heat sink, and of a tubular heat sink with heat loads attached at both ends. Historically, research into the pool boiling of tube bundles has been driven by the nuclear industry, where the passive cooling system and high heat transfer coefficient afforded by pool boiling is critical to the operational safety and reliability of reactors (Chung *et al.*, 2015; Liu *et al.*, 2019).

Shell-side flow boiling in a shell-and-tube heat exchanger can demonstrate similar characteristics to pool boiling tube bundles, especially at low mass flow rates. These exist in many applications including flooded refrigerant (Robinson and Thome, 2004), falling film (Fernández-Seara and Pardiñas, 2014), and desalination (Liu and Ishibashi, 2001) evaporators; and kettle reboilers in process distillation systems (McNeil *et al.*, 2010).

Most research undertaken on the boiling of tube bundles has considered flow boiling conditions, with fluid pumped at known flow rates over the tubes, and typically with tube diameters around 19 mm or  $\frac{3}{4}$ ". Various reviews are available (Browne and Bansal, 1999; Ribatski and Thome, 2007; Fernández-Seara and Pardiñas, 2014; Swain and Das, 2014). For the column heights  $N$  tested, within a column at low heat fluxes heat transfer coefficient  $h_{NB}$  increases with tube number (Fujita and Hidaka, 1998), and the addition of tubes to a column increases the overall mean heat transfer coefficient of the column (Kim, Cho and Youn, 2002). Memory *et al.* (1995) defined the former metric as bundle effect ratio  $BE$  and the latter as bundle factor ratio  $BF$ . Adom *et al.* (2010) showed that these trends remain at the 3 mm tube diameter; however the thermocouple supplier OMEGA cautions against the use of unshrouded thermocouples in an electromagnetic field (OMEGA Engineering, 2019).

Limited work has been published on tube bundles in pool boiling conditions without forced convective effects. Bundle effect and bundle factor are confirmed as most pronounced at lower heat fluxes (Memory, Sugiyama and Marto, 1995; Gupta, 2005; Ustinov, Ustinov and Mitrovic, 2011; Kang, 2015). Liu and Qiu (2002, 2006) suggested that, with decreasing tube spacing  $s$ , heat transfer coefficient first increased and then decreased, and that critical heat flux  $CHF$  deteriorated at very small spacings. This was partly corroborated by Gorgy and Eckels (2016) who observed enhanced  $h_{NB}$  for intermediate spacing at medium heat flux. The interaction of plumes from heated parallel surfaces can be investigated by comparing the heat transfer when a single surface is heated to when multiple surfaces are heated. For the effect of unheated spatial confinement on pool boiling heat transfer, extant literature reports predominantly on flat plate geometries and indicates the same trends as for heated tube spacing (Yao and Chang, 1983; Bonjour and Lallemand, 1998; Passos, Da Silva and Possamai, 2005; Bartle,

Menon and Walsh, 2017). Fujita and Hidaka (1998) additionally show that a heated column of tubes surrounded by unheated tubes gives lower  $h_{NB}$  at low heat flux, and higher  $h_{NB}$  at higher heat flux, than the same tube bundle geometry with all tubes heated.

Evaluation of dryness and void fractions in flow boiling conditions has been undertaken experimentally (Aprin, Mercier and Tadrist, 2007; Rooyen *et al.*, 2012), analytically (Robinson and Thome, 2004; Aprin, Mercier and Tadrist, 2011), and computationally (Tian *et al.*, 2016). Additionally, Ahmadpour *et al.* (2018) applied a numerical model to pool boiling over a staggered tube bundle.

This paper investigates nucleate pool boiling from confined and unconfined columns of 2.55 mm diameter mini-tubes in 3M's dielectric, low Global Warming Potential fluid Novec 649. Section 6.2 outlines the experimental methods used in this work. Section 6.3.1 presents an empirical model for unconfined bubble plumes from tube columns. Section 6.3.2 suggests a model for dryness and void fractions in confined plumes, and verifies the model experimentally. Sections 6.3.3 to 6.3.6 present and discuss results for critical heat flux, critical power density, mean column heat transfer coefficient, and column temperature profile.

## **6.2. Method and Materials**

### **6.2.1. Apparatus**

The tube bundle consists of 304 grade stainless steel hollow tubes (2.55 mm outer diameter, 2.15 mm inner diameter, 93 mm heated length) held in position by perforated end walls. Columns of 1 to 10 tubes in height were tested. Tube spacing was varied from 1 to 4 mm, with  $s$  being also the spacing between the bottom tube and the boiling chamber floor. Spatial confinement of the tube column was effected by the 3D printing of spacers that produced an interference fit onto the tube column end walls. These spacers

accommodated a transparent acrylic plate that confined the boiling space whilst retaining visual observation of the tube column. Confinements from 0.25 to 5 mm were tested and compared with the unconfined case. Unconfined and confined tube columns are shown in Figure 31, where the column geometry is diagrammed inset.

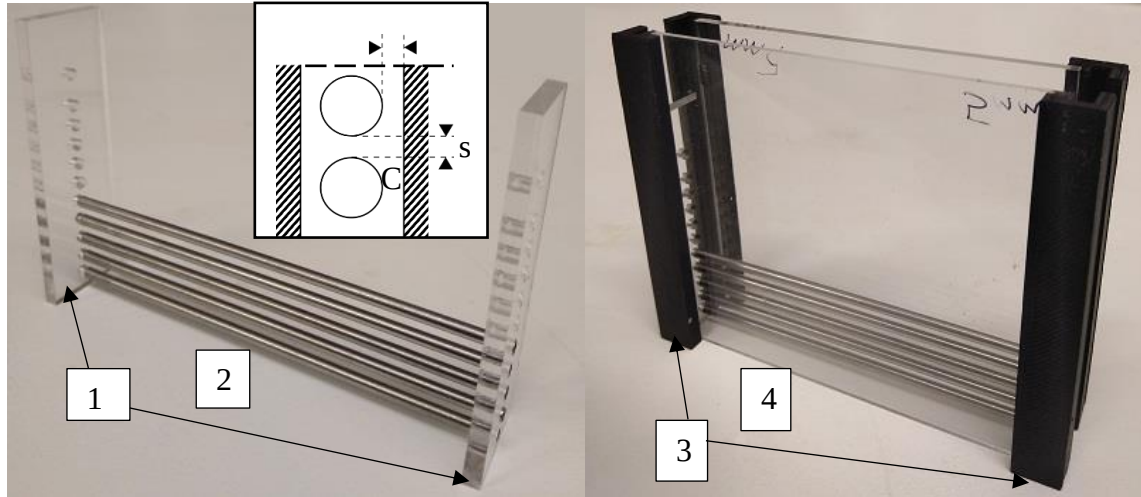


*Figure 31: Images showing (a) unconfined, and (b) confined, 5 tube column. Labels: (1) perforated end walls; (2) tubes; (3) spacing plates; and (4) transparent confinement plates. To aid image clarity, electrical connections and thermistor leads have been removed. Inset: section of a confined tube column showing confinement  $C$  and tube spacing  $s$ .*

Each tube was electrically connected in series, with heat flux controlled by Joule heating. Uniform heating was achieved along the axial length of each tube. A thermistor, calibrated to  $\pm 0.1$  °C against a water bath, was placed within each tube and a Wheatstone bridge configuration used to read the voltage drop. The tubes were sealed so that boiling took place only on the external surfaces.

The tube column was placed within a sealed transparent acrylic chamber. The chamber was filled with coolant to 25 mm above the height of the top tube in the column. A pressure sensor ensured ambient pressure was maintained within the chamber, and  $\pm 0.1$  °C calibrated type K thermocouples monitored pool and vapour temperatures within the chamber and also the room ambient temperature 1 m away from the rig. A relay-

controlled plate heater maintained the liquid pool at saturation temperature. A copper coil with internal water flow condensed the vapour at the top of the chamber. The boiling chamber is diagrammed in Figure 32.

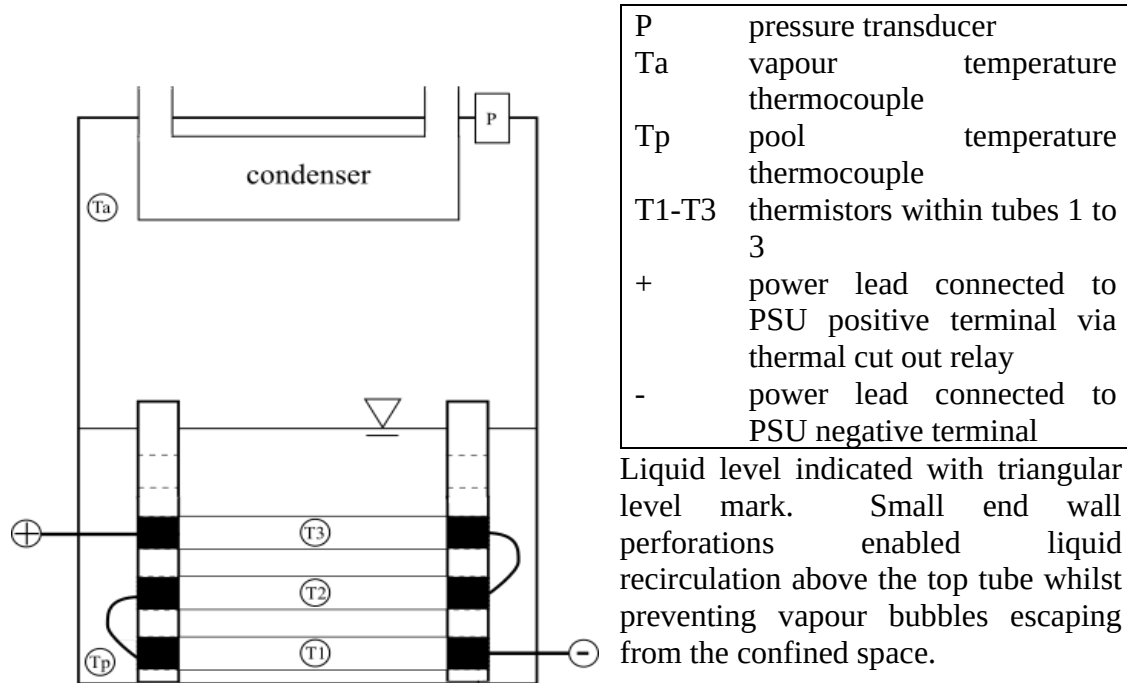


Figure 32: Diagram of an electrically connected 3 tube column with internal thermistors, placed in a liquid pool within the instrumented boiling chamber.

Previous studies have noted that noncondensable gases can have a small effect on the boiling curve of subcooled fluorocarbons (Bar-Cohen, Tong and Simon, 1992; Rainey, You and Lee, 2003). In preparation for this experimental programme, we ran and compared 48 h duration tests of boiling at constant heat flux ( $q'' = 33.8 \text{ kW m}^{-2}$ ) in degassed and aerated fluids at 1 bar and saturation temperature, with venting should an internal overpressure of 0.5 kPa be detected. The degassed fluid was prepared within 15 min sonication followed by 7 cycles of producing a 0.8 kPa vacuum within the boiling and allowing equilibrium pressure to return, so reducing the concentration of noncondensables within the liquid. The aerated fluid was prepared by pumping air from an high pressure line into the liquid for 15 min. Both samples varied by  $\pm 0.35 \text{ }^{\circ}\text{C}$  over

the 48 h duration and there was no statistically significant difference between the pressure or temperature characteristics of the degassed and aerated fluid tests. Ergo when preparing the tube column tests we followed the established practice of using the auxiliary heater in vented boiling for a prolonged period, selected here as 45 min, before commencing the tests.

With smooth boiling surfaces, increasing heat flux from nil can result in an initial thermal hysteresis (El-Genk and Bostanci, 2003). To mitigate against this by activating stable nucleation sites, each test configuration began at a medium power setting ( $q'' = 32.2 \text{ kW m}^{-2}$ ); the power was then decreased to the lowest level tested and then incrementally increased until critical heat flux was obtained.

Heat flux  $q''$  in all the experiments is obtained by Joule heating:

$$q'' = \frac{I^2 R}{A_s}, \quad (6.1)$$

where resistance is measured beforehand, current is adjusted at the power supply, and surface area  $A_s = \pi d_o L$ . The nucleate boiling heat transfer coefficient is

$$h_{NB} = \frac{q''}{\Delta T_{sat}}, \text{ where} \quad (6.2)$$

$$\Delta T_{sat} = T_w - T_{sat}. \quad (6.3)$$

Uncertainty analysis was performed according to the Kline and McClintock (1953) method and returned a constant heat flux uncertainty  $w_{q''}$  of 6.29% (see Appendix C.II for details). The effect of  $\pm 0.5 \text{ kPa}$  pressure deviations from 1 bar on the fluid saturation temperature was noted from fluid PVT data to introduce saturation temperature uncertainty  $\pm 0.25 \text{ }^\circ\text{C}$ . Heat transfer coefficient uncertainty was thus found to range from 6.40% to 6.68% across the experimental matrix.

Surface roughness data obtained for the tubes with a calibrated Alicona InfiniteFocus 3D optical microscope returned isotropic roughnesses  $R_a = 553$  nm (SD = 14.7%) and  $R_q = 675$  nm (SD = 14.0%), which are typical of a smooth surface in a heat exchange application. Images and videos of the boiling were obtained with an high speed camera. For individual non-coalesced spherical bubbles departing from a tube, a mean bubble diameter of 1.57 mm (SD = 12.2%) was measured from images. Significant bubble coalescence was observed, especially for higher tube numbers and at higher heat fluxes, as shown in Figure 33.

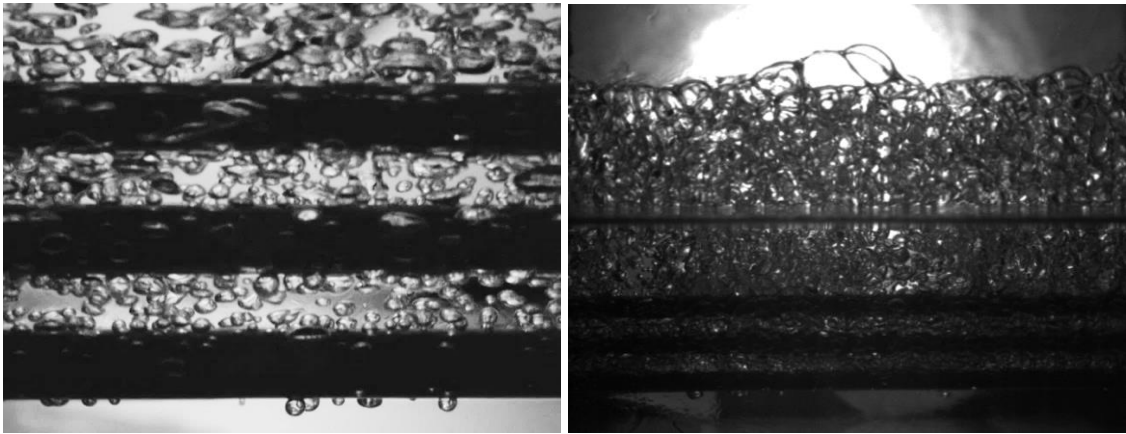


*Figure 33 (left): Image, taken at the onset of nucleate boiling, showing increased bubble coalescence with tube number.*

*Figure 34 (right): Image of vapour head at  $q'' = 32.2 \text{ kW m}^{-2}$  in a confined 3 tube column, taken for void fraction calculations.*

### **6.2.2. Unconfined Plume Measurements**

In Section 6.3.1 the lateral development of an unconfined plume around a tube column, and the associated dryness and void fractions, are investigated with the use of 10 tube columns with  $s = 2$  and 4 mm with transparent end walls (as depicted in Figure 31(a)). For each heat flux, a batch of 60 images at high exposure and 30 FPS were obtained. To evaluate the plume profile, a composite image was produced for each heat flux by using software to average the images within each batch, as shown in Figure 35. The plume profile for each heat flux was then measured from this composite image.

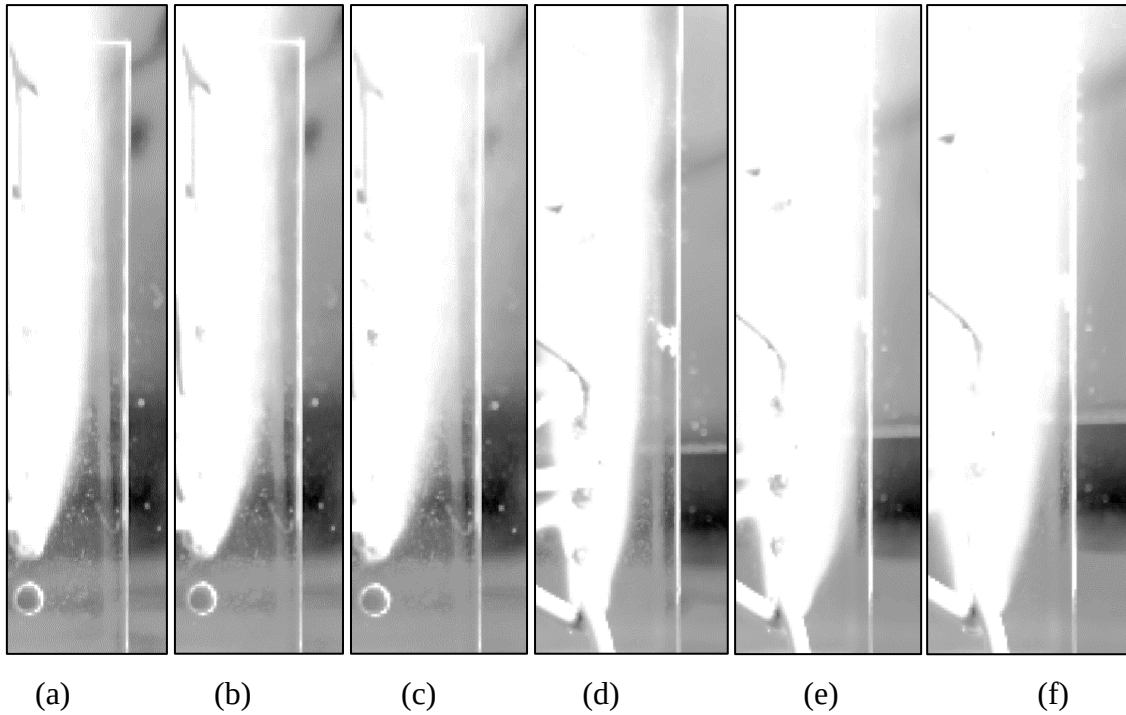


Figure 35: Time-averaged images of unconfined bubble plume profiles from 10 tube columns, for (a)  $q'' = 15.8 \text{ kW m}^{-2}$ ,  $s = 2 \text{ mm}$ ; (b)  $q'' = 32.2 \text{ kW m}^{-2}$ ,  $s = 2 \text{ mm}$ ; (c)  $54.4 \text{ kW m}^{-2}$ ,  $s = 2 \text{ mm}$ ; (d)  $q'' = 15.8 \text{ kW m}^{-2}$ ,  $s = 4 \text{ mm}$ , (e)  $q'' = 32.2 \text{ kW m}^{-2}$ ,  $s = 4 \text{ mm}$ ; and (f)  $q'' = 54.4 \text{ kW m}^{-2}$ ,  $s = 4 \text{ mm}$ .

### 6.2.3. Confined Plume Measurements

To validate the confined plume model presented in Section 6.3.2, a 3 tube column was used. The 2 mm confinement plates were adhered to the tube end walls and to the boiling chamber base to prevent fluid recirculation between the confined column and the liquid outside. Ten images were taken at 7.5 FPS at each heat flux as shown in Figure 34. For each image, five measurements of the distance between the height of the vapour bubble surface and the height of the static liquid pool were taken. The SD for these measurements varied with heat flux level from 5.00% to 10.3%. The loss of liquid head due to evaporation was compensated for in the calculations.

A mean height of vapour head was computed for each heat flux tested to give the total liquid and vapour volumes in the system. The total vapour volume in a column is the sum of a triangular series, therefore the general equation for the total vapour volume in column with  $N$  tubes is

$$V_{v,N} = V_{v,1} \frac{N(N+1)}{2}. \quad (6.4)$$

Rearranging eqn. (6.4) provides the vapour generated by one tube where the total vapour volume is known:

$$V_{v,1} = \frac{2V_{v,N}}{N(N+1)}. \quad (6.5)$$

The vapour volume at tube  $j$  in an  $N$  tube column is thus

$$V_{v,j} = jV_{v,1}. \quad (6.6)$$

The void and dryness fractions at the  $j^{\text{th}}$  tube can then be calculated:

$$\varepsilon_j = \frac{V_{v,j}}{V_{v,j} + V_{l,j}}, \text{ and} \quad (6.7)$$

$$\zeta_j = \frac{V_{v,j}\rho_v}{V_{v,j}\rho_v + V_{l,j}\rho_l}. \quad (6.8)$$

## 6.3. Results and Discussion

### 6.3.1. Unconfined Column Plume Development

The plume profile composite images for 10 tube column at various heat fluxes and two tube spacings are shown in Figure 35. Lateral plume oscillation was observed at high heat fluxes and manifests as a hazy grey area around the plume boundary on the time-averaged images. Videos 1 to 3, embedded in the electronic copy of this paper, show the development of plume oscillation from  $q'' = 15.8$  to  $54.4 \text{ kW m}^{-2}$  for a 5 tube column. The plume nomenclature is shown in Figure 36, and the measurements for the time-averaged plume core are graphed in Figure 37.

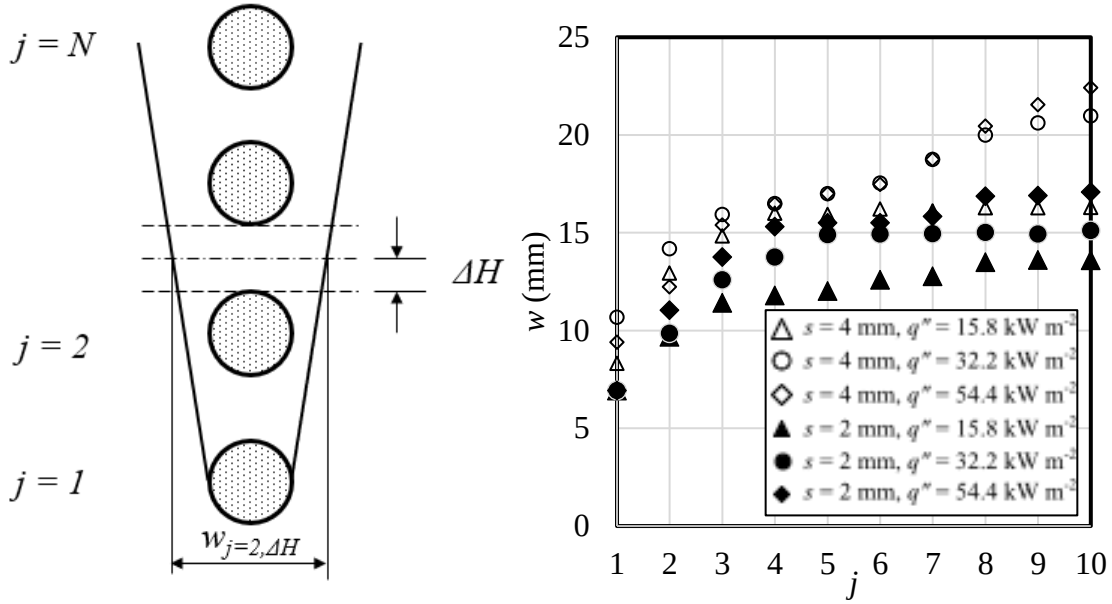


Figure 36 (left): Diagram of an unconfined bubble plume around a tube column.

Figure 37 (right): Plume width at each tube number in a 10 tube column.

In a buoyancy driven system, the vapour mass flow rate at tube  $j$  is

$$G_{v,j} = j \frac{Q}{h_{fg}}. \quad (6.9)$$

where  $Q$  is the heat transferred by a single tube. The liquid mass flow rate at tube  $j$  is

$$G_{l,j} = \rho_l A_j u - G_{v,j}. \quad (6.10)$$

The control area  $A_j$  associated with tube  $j$  in a developing plume is

$$A_j = w_{j,\Delta H} L. \quad (6.11)$$

where  $w_{j,\Delta H}$  is shown in Figure 36 as the plume width at the midpoint between tube  $j$  and tube  $j+1$ .

Bubble velocity was obtained experimentally as a mean  $0.199 \text{ m s}^{-1}$  but, where *a priori* modelling is required, a suitable correlation (e.g. Davies-Taylor (1950), Mendelson (1967), Massarani (1999)) may be employed (1998). Thus, the dryness fraction in the unit volume  $\zeta_j$  is:

$$\zeta_j = \frac{G_{v,j}}{G_{l,j} + G_{v,j}}, \quad (6.12)$$

where  $m_{Tj}$  is the total fluid mass in the unit volume. The void fraction is

$$\varepsilon_j = \frac{\rho_l \zeta_j}{S_j \rho_v (1 - \zeta_j) + \rho_l \zeta_j}. \quad (6.13)$$

Assuming no slip,  $S = 1$ ; else the Chisholm (1983) correlation may be used:

$$S_j = \left( 1 - \zeta_j \left( 1 - \frac{\rho_l}{\rho_v} \right) \right)^{1/2}. \quad (6.14)$$

Figure 38 and Figure 39 graph the dryness and void fractions of an unconfined plume around 10 tube column. With increasing tube number, the incremental width increase is significant at first but tends towards a plateau. The width of the plume does not simply increase by the quantity of vapour added by each succeeding tube; therefore the dryness and void fractions must increase. Figure 38 and Figure 39 show an approximately linear increase in dryness and void fraction with tube number. Heat transfer deterioration can be expected when dryness and voidage exceed a certain threshold. Where this threshold is known, the model provides the tube column height at which heat transfer deterioration would occur, and is therefore of use to engineers to establish the maximum height of a passive boiling heat exchanger column in an unconfined geometry.

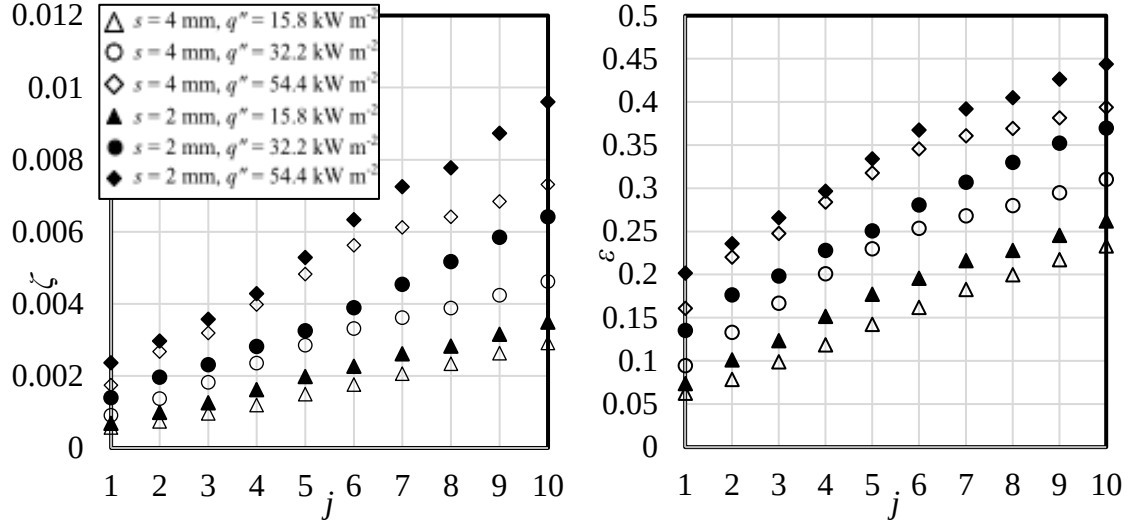


Figure 38 (left): Dryness fraction in unconfined plumes at three representative heat fluxes.

Figure 39 (right): Void fraction in unconfined plumes at three representative heat fluxes.

### 6.3.2. Developed Plume Dryness and Void Fractions in a Confined Tube Column

In a confined tube column, the vapour mass flux generated  $G_{v,j}$  is unchanged from eqn. (6.9). When the plume is developed and spatially constrained, the plume width cannot increase, ergo the dryness and void fractions of the plume must increase by a constant increment. To model this, the control area  $A_l$  in a confined column is

$$A_l = L(2C + d); \quad (6.15)$$

as eqn. (6.11) but with the plume width equal to the channel width  $2C + d$  as shown in Figure 40. The expressions for mass flow rates, dryness and void fractions are unchanged from Section 6.3.1.

Experimental validation of this model was undertaken according to the method outlined in Section 6.2.3 for three heat fluxes, with the results summarised in Table 11. The limitations of the no slip assumption have been noted before (Ribatski and Thome, 2007); compared to the no slip assumption, Chisholm's slip correlation provided a mean

prediction error reduction of 11.1%, with decreased error particularly at elevated heat flux where slip is expected to be most pronounced. The bubble velocity models are semi-empirical and assume single detached bubbles of uniform diameter, neglecting the coalescence around upper tubes at higher heat fluxes. Despite this, a mean predictive error of 12.5% affords confidence in the model.

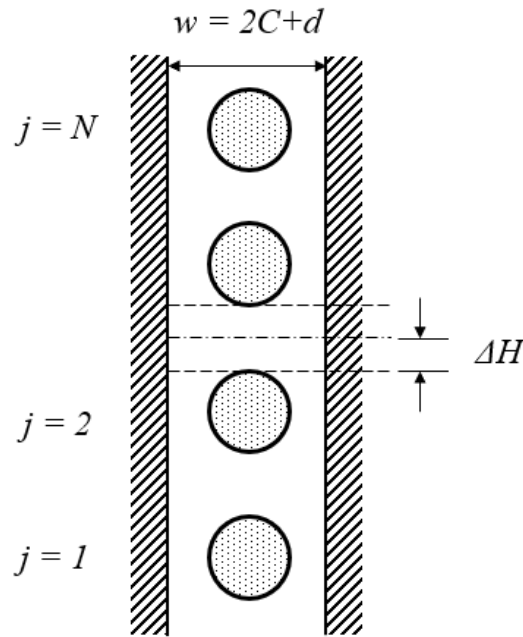


Figure 40: Diagram showing the control volume above a tube in a confined column.

| $q''$ (kW m <sup>-2</sup> )                | 15.8   | 32.2   | 54.4   |
|--------------------------------------------|--------|--------|--------|
| experimental $\varepsilon$                 | 0.154  | 0.338  | 0.414  |
| prediction error with no slip assumption   | +35.1% | +3.55% | +15.2% |
| prediction error using Chisholm slip ratio | +23.3% | -10.4% | -3.86% |

Table 11: Evaluation of vapour fraction theory for a developed plume in a confined  $C = 2$  mm 3 tube column with spacing  $s = 2$  mm.

### 6.3.3. Critical Heat Flux

The relationships between critical heat flux, confinement, and column height are shown in Figure 41(a). The column was said to reach *CHF* when a single tube in the column met the boiling crisis. In Region I, the boiling crisis occurred on the bottom tube. In Region II, the boiling crisis occurred on the top tube (except for the  $C = 0.25$  mm case

in which, at two data points, the 2<sup>nd</sup> tube from top transitioned first). Confinements  $C$  and tube spacings  $s$  may be expressed in terms of Bond number, as formulated by Yao and Chang (1983):

$$Bo = L_c / \left( \frac{\sigma}{g(\rho_l - \rho_v)} \right)^{1/2}, \quad (6.16)$$

where the characteristic length  $L_c$  is substituted by  $C$  and  $s$  to give confinement and tube spacing Bond numbers  $Bo_C$  and  $Bo_s$  respectively.

In Figure 41(a), there is relatively little confinement effect on  $CHF$  at low numbers of tubes  $N$  because the bulk dryness fraction of the flow remains low. As  $N$  increases, greater confinement effect becomes apparent, with decreasing  $C$  resulting in  $CHF$  decrease as bulk dryness becomes important. Of particular interest is the change in dryout mechanism that is observed in Figure 41(b) occurring at  $\varepsilon_{j=N} \sim 0.64$ . Developed plume void fractions were modelled from Section 6.3.2. Where the void fraction of the top tube in the column  $\varepsilon_{j=N} < \sim 0.64$ , the bottom tube in the column reaches CHF first due to the higher surface temperature of the bottom tube causing higher local dryness; this is marked as Region I. However, when  $\varepsilon_{j=N} > \sim 0.64$ , the top tube in the column reached dryout first because of liquid starvation in the bulk fluid; this is marked as Region II.

The  $C = 0.25$  mm ( $Bo_C = 0.29$ ) configuration did not altogether fit the discussion above. At two data points in Region II,  $CHF$  occurred first on the penultimate tube, albeit followed almost immediately by the top tube. Also, the configuration exhibited Region II behavior with dryout first at the top of the column although  $\varepsilon_{j=N}$  was lower than the expected  $\sim 0.64$  threshold. Further, Cornwell and Einarsson (1990) noted that the local void fraction of the fluid around the periphery of an horizontal tube increases from negligible at the bottom to around 0.5 at the sides. Therefore, in this condition a third

*CHF* mechanism is at play due to the development of a vapour clot at the neck between the tube and the column wall. As the top tube is partially replenished with liquid recirculation in the space above, in this case the driest location in the column will be the area underneath  $j=N$  and above  $j=N-1$ . It was observed from images that bubbles were greater than 0.4 mm diameter at the onset of nucleate boiling, and even greater at higher heat fluxes ( $Bo_C = 0.47$ ). A dramatic adverse effect on heat transfer in this  $Bo$  region has been previously observed in confined boiling of pure water on tube bundles (Liu and Qiu, 2006).

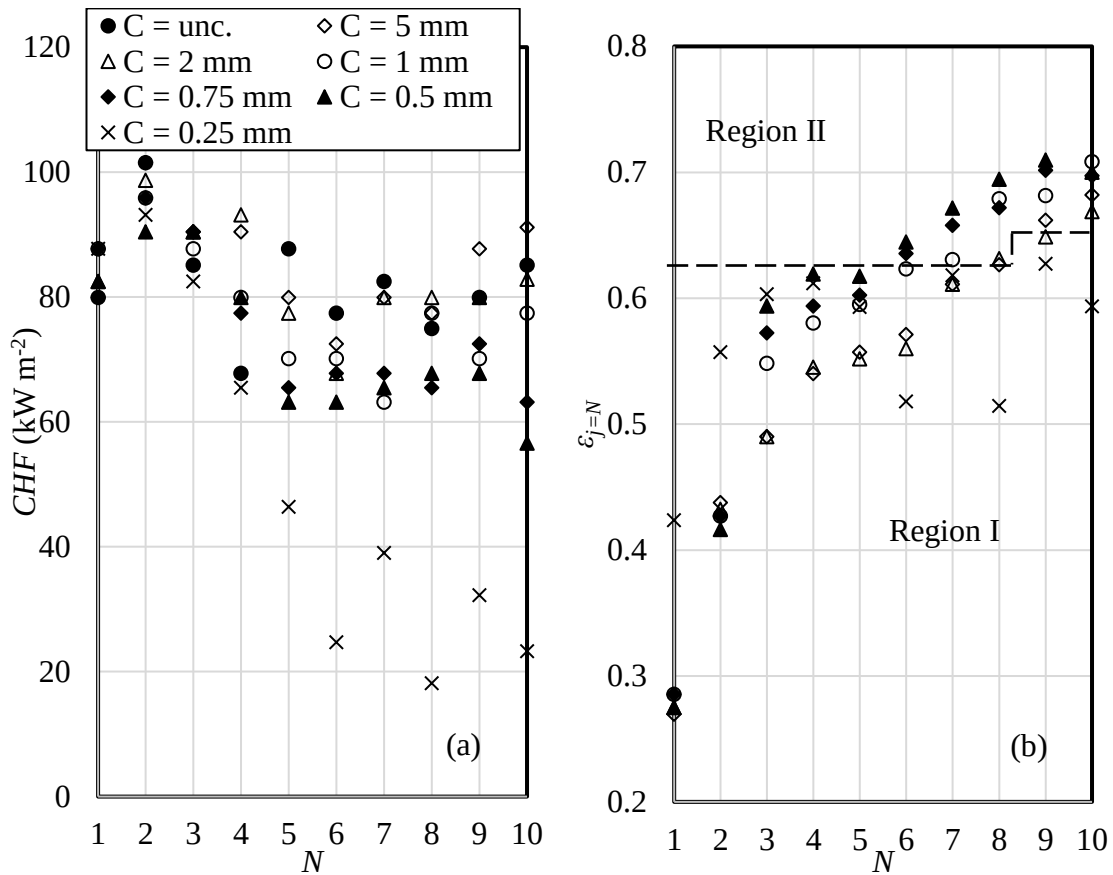


Figure 41: Graphs to show (a) the relationship between critical heat flux, column height, and confinement; and (b) critical location, void fraction at the top tube, and column height; in columns with 2 mm tube spacing.

Figure 42(b) shows the effect of tube spacing on  $CHF$  for unconfined and confined columns. As expected, since a shorter column gives a lower plume dryness fraction a higher  $CHF$  is achieved. At low spatial confinements, the case with the lowest  $s$  and

highest  $N$  ( $s = 1$  mm,  $N = 10$ ) consistently returned low  $CHF$  values; similarly, the case with highest  $s$  and lowest  $N$  ( $s = 4$  mm,  $N = 5$ ) returned the highest  $CHF$  values.

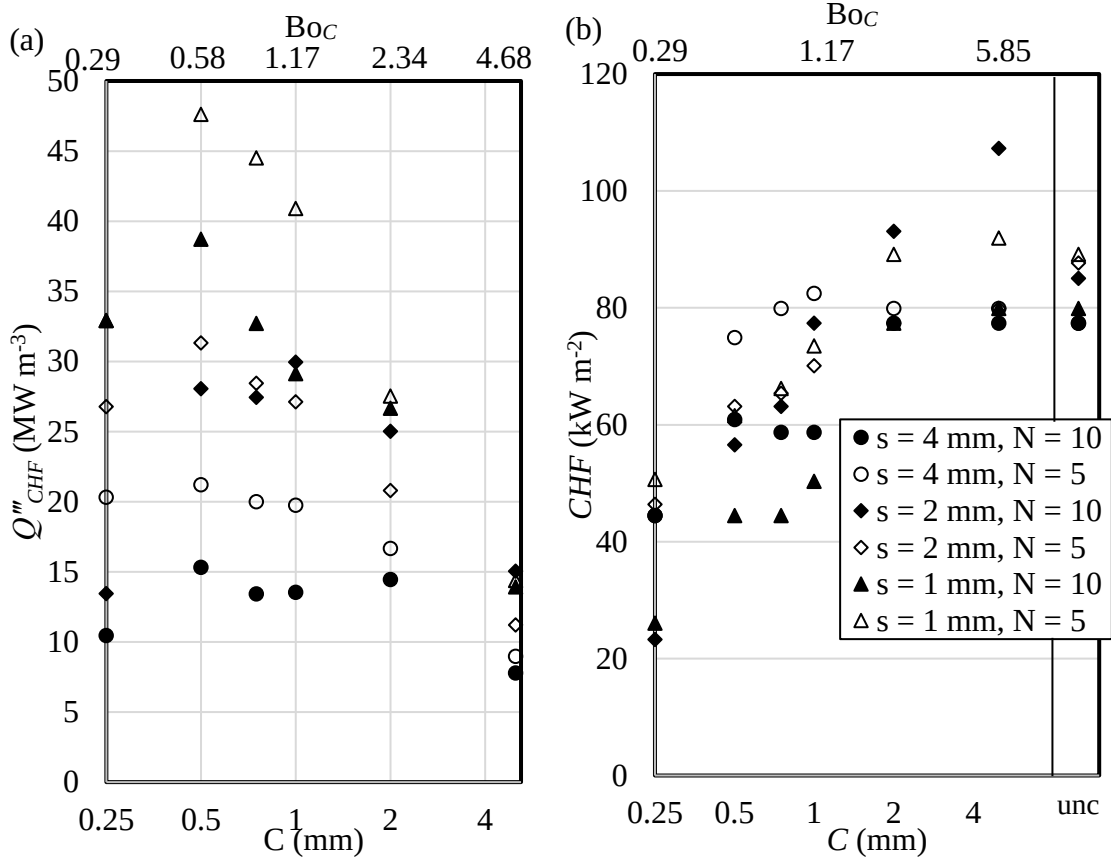


Figure 42: Graphs to show (a) critical power density, and (b) critical heat flux, with respect to confinement, tube spacing, and column height.

#### 6.3.4. Critical Power Density

Critical power density  $Q''_{CHF}$  is defined as the power density of a confined column at critical heat flux, where power density:

$$Q'' = \frac{Q}{V_{col}}, \text{ where internal column volume} \quad (6.17)$$

$$V_{col} = (N(s + d)) \cdot (2C + d) L. \quad (6.18)$$

A comparison of Figure 42 (a) and (b) indicate that, although the absolute value of  $CHF$  generally decreases with decreasing tube spacing and confinement, the critical

power density  $Q'''_{CHF}$  achieved a maximum at  $C = 0.5$  mm ( $Bo_C = 0.58$ ) for all tube spacings  $s$ , but fell away sharply at  $C = 0.25$  mm because of the significantly deteriorated  $CHF$  values discussed in Section 6.3.3. This argument suggests that a cliff edge may similarly be expected to exist for tube spacing  $s$  between  $0.29 < Bo_s < 0.58$ . In the present experiments, the smallest tube spacing tested,  $s = 1$  mm ( $Bo_s = 1.17$ ), returned the highest  $Q'''_{CHF}$ .

The relationships between confinement, column height, and critical power density are shown in Figure 43. Critical power density increases for a multi row column above a column with a single tube but decreases thereafter and levels at  $N > 4$  tubes. This indicates that a maximum power density, which is independent of  $N$ , exists at around  $30 \text{ MW m}^{-3}$  for “tall” columns. For a compact tube bundle under reduced pressure in water, Liu and Liao (Liu and Liao, 2006) showed a similar power density of  $14 \text{ MW m}^{-3}$ . For an industrial sized passive boiling tube bundle, Chung *et al.* (2015) demonstrated a power density of  $12 \text{ kW m}^{-3}$  and Gupta (2005), for a moderate flow boiling tube bundle, returned around  $5 \text{ MW m}^{-3}$ . Geisler and Bar-Cohen (2007) postulate a maximum power density of around  $200 \text{ MW m}^{-3}$  for the pool boiling of dies separated by parallel vertical coolant channels, a geometry that maximises buoyancy effects. Therefore, noting that this is a reliable, passive heat transfer mechanism, favorable comparison is achieved against extant power densities for boiling tube bundles.

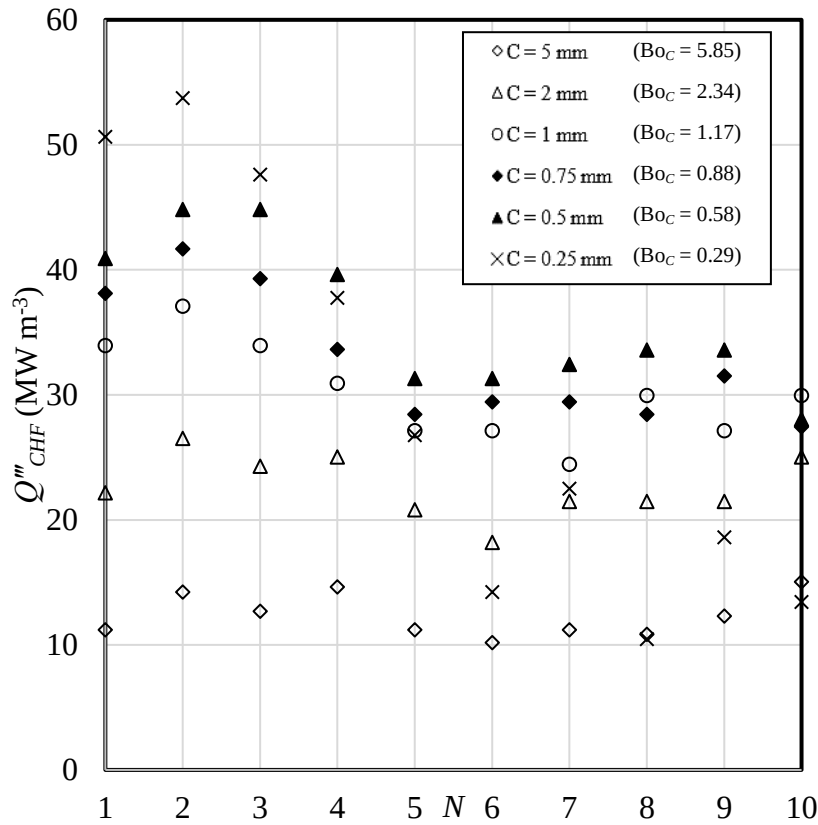


Figure 43: The effect of confinement and column height on critical power density for  $s = 2$  mm.

### 6.3.5. Column Heat Transfer Coefficient

The column heat transfer coefficient was calculated by substituting the mean tube temperature into eqn. (6.2). Results are graphed for representative low and high heat fluxes in Figure 44. To provide a convenient expression for mean heat transfer coefficient, Memory *et al.* (1995) defined bundle factor  $BF$  as the ratio of mean  $h_{NB}$  for the column over  $h_{NB}$  of a single tube. At all heat fluxes, the tube columns returned  $BF$  greater than unity, although the phenomenon was more pronounced at lower heat fluxes as previously reported (Kim, Cho and Youn, 2002). The addition of a tube above a single tube provides the greatest increase in column  $h_{NB}$  and approaches a level as  $N > 3$ . The local heat transfer coefficient and temperature profile within in a column is explored further in Section 6.3.6. At low heat fluxes, more confined columns exhibited greater  $BF$ ; as the bubble plume is spatially confined, the local void fraction within the column

increases and provides the requisite vapour to generate and maintain stable nucleation sites on the tubes above. Confinement also promotes convective heat transfer from increased bubble interaction, although the magnitude of this is not quantified. In the high heat flux case, the mean heat transfer coefficient did not significantly increase beyond the addition of a second tube.

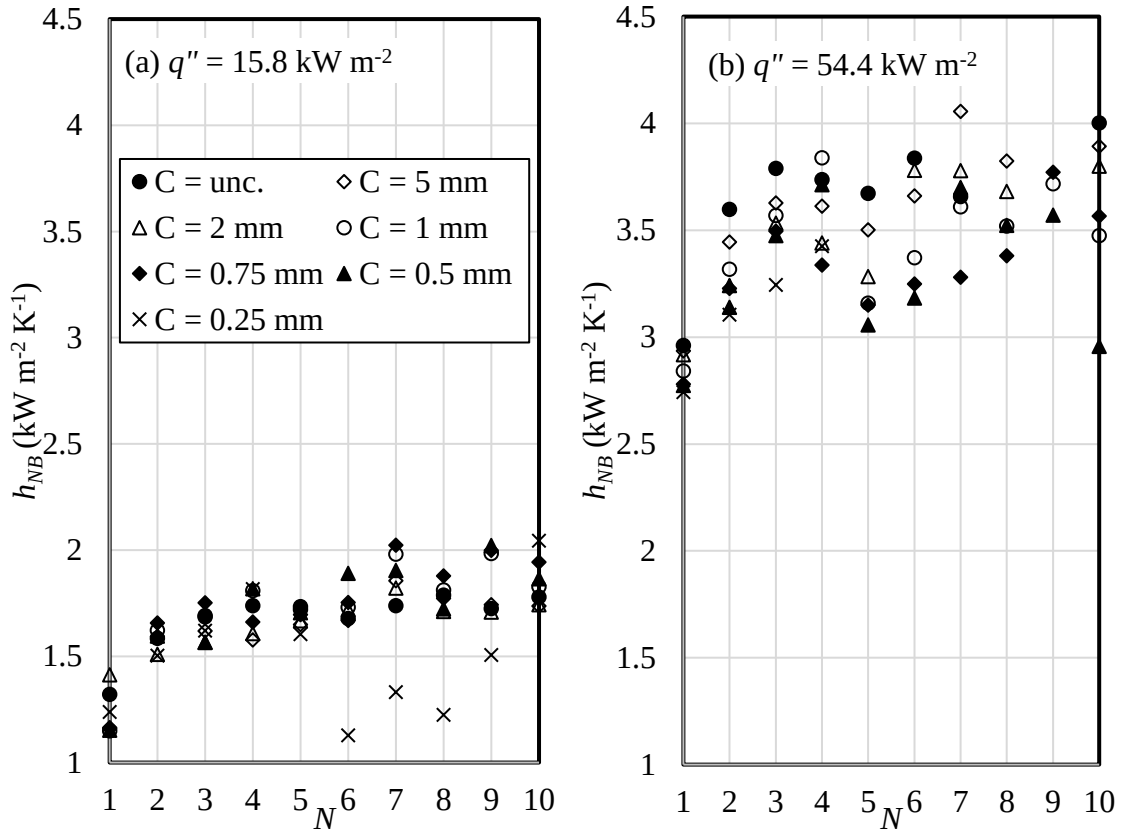


Figure 44: Graphs showing the effect of column height and confinement on mean column heat transfer coefficient for a column with  $s = 2 \text{ mm}$  at (a) low heat flux  $q'' = 15.8 \text{ kW m}^{-2}$ , and (b) high heat flux  $q'' = 54.4 \text{ kW m}^{-2}$ .

The effect of tube spacing on mean  $h_{NB}$  is shown in Figure 45. At low heat fluxes (Figure 45(a)), the intermediate spacing  $s = 2 \text{ mm}$  returned the highest  $h_{NB}$  while  $\text{Bo}_C > 1$ . Below this, increasing spacing  $s$  enhanced heat transfer by decreasing the local void fraction. For the unconfined and  $\text{Bo}_C = 5.85$  confinements,  $s = 2 \text{ mm}$  continued to return the highest  $h_{NB}$ . However, for closer confinements,  $s = 4 \text{ mm}$  was superior. The general trend observed is that moderate spacing provides highest heat transfer for unconfined and

slightly confined columns, whilst large spacing mitigated against local dryout where confinement was significant. The smallest  $s = 1$  mm spacing returned lower heat transfer coefficients at both heat fluxes. Figure 39 shows, from the bulk void fraction increase with decreased tube spacing, that natural convective spreading of a bubbly flow occurs. Also, previous work (Cornwell and Einarsson, 1990; Ahmadpour, Noori Rahim Abadi and Meyer, 2018) has indicated that the void fraction varies significantly in the immediate vicinity of the tube. It is therefore suggested that decreased  $s$  inhibits the natural convective spreading of bubbles and results in a higher void fraction in the bubbly flow impinging on the bottom surface of the tube above.

Figure 45(a) shows that at low heat flux  $h_{NB}$  is augmented by confinement, but with the danger of local choked voidage as in the  $C = 0.25$  mm,  $s = 1$  mm,  $N = 5$  case. At high heat flux in Figure 45(b),  $h_{NB}$  declines where  $C < 1$  mm.

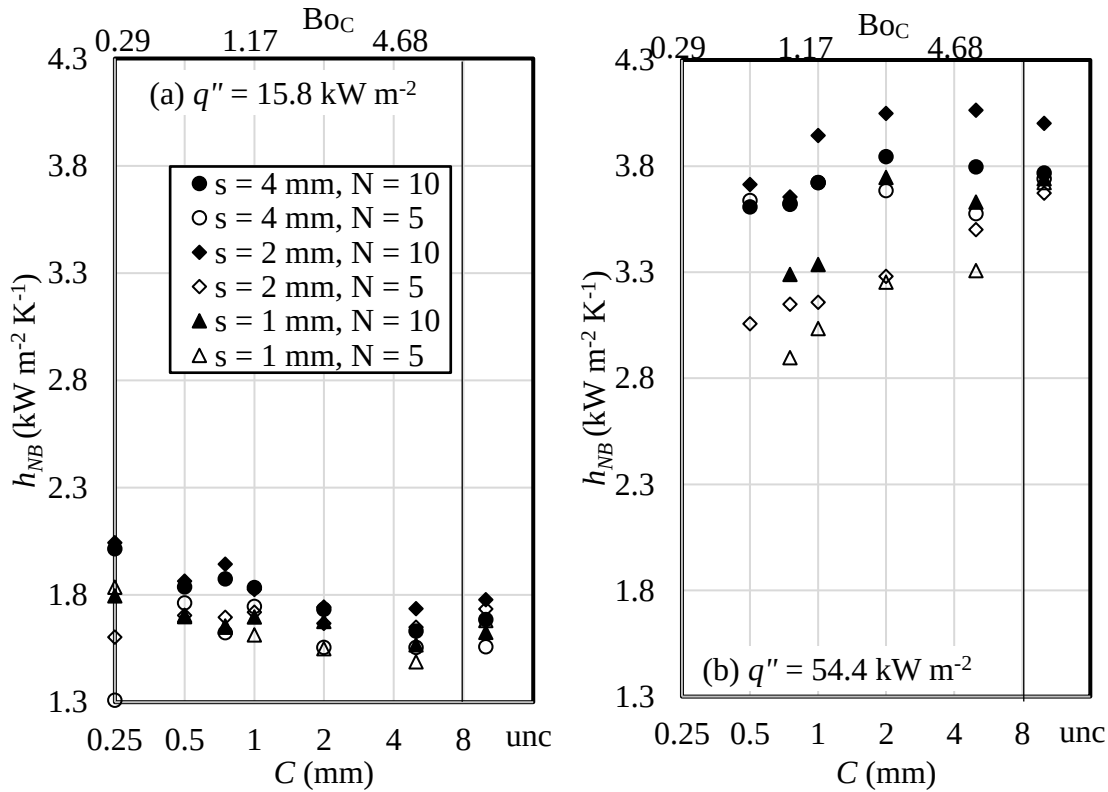


Figure 45: Graphs showing the effect of confinement and tube spacing on heat transfer coefficient at (a)  $q'' = 15.8 \text{ kW m}^{-2}$  and (b)  $q'' = 54.4 \text{ kW m}^{-2}$ .

### 6.3.6. Column Temperature Gradient

In many applications, not only the overall heat transfer coefficient but also the location of hotspots and absolute temperatures within a heat transfer device are important. The fuel rods of boiling water reactors (BWRs) are manufactured with varying reactivity to mitigate against the variation in void fraction with core height (Ikehara *et al.*, 2009). In modelling the thermal failure rate of electronic components, Arrhenius' law indicates an exponential relationship between component temperature and failure rate (Wilcoxon, 2017). Additionally, the electrical properties of materials change at elevated temperatures.

To aid the discussion of the tube temperature variation within the column, void fraction  $\varepsilon$  was graphed in Figure 46 from the model outlined in Sections 6.3.1 and 6.3.2. The temperature variation within 5 and 10 tube confined and unconfined columns at three heat fluxes representing low, medium, and high heat flux conditions are shown in Figure 47. At low  $\varepsilon$ , and low heat flux the pattern for both 5 and 10 tube columns is for tube temperature to decline with increasing tube number  $j$ , whilst at moderate and high heat fluxes tubes above the first row returned a constant temperature independent of  $j$ . Exceptions exist for the  $C = 0.25$  mm cases with the hydrodynamic dryout discussed in Section 6.3.3. With the  $C = 0.25$  mm cases excluded, heat transfer is significantly affected where  $\varepsilon > \sim 0.64$ ; compare  $\varepsilon$  and  $T_w$  for  $C = 2$  mm at high heat flux in Figure 46(c) and Figure 47(c). In flow boiling conditions, previous researchers have reported decreased heat transfer coefficient with increased void fraction in sub CHF conditions (Huo *et al.*, 2004; Copetti *et al.*, 2011). Ergo a column temperature profile that significantly positively correlates with  $j$  predicts that, when CHF occurs, it can be expected at the top of the column.

The increased heat transfer coefficient associated with increasing  $j$  in a column is usually explained by the promotion of convective mixing caused by departing bubbles (Hahne, Chen and Windisch, 1991; Gupta, 2005; Kang, 2015). Additionally Cornwell (1982; 1990; 1990) plausibly suggests liquid microlayer evaporation between the tube surface and the vapour bubbles impinging from below and sliding along the tube surface; this has been widely adopted, alongside convective effects, as contributing to enhanced  $h_{NB}$  (Hsu, Lin and Jensen, 1993; Memory, Sugiyama and Marto, 1995; Aprin, Mercier and Tadrist, 2011).

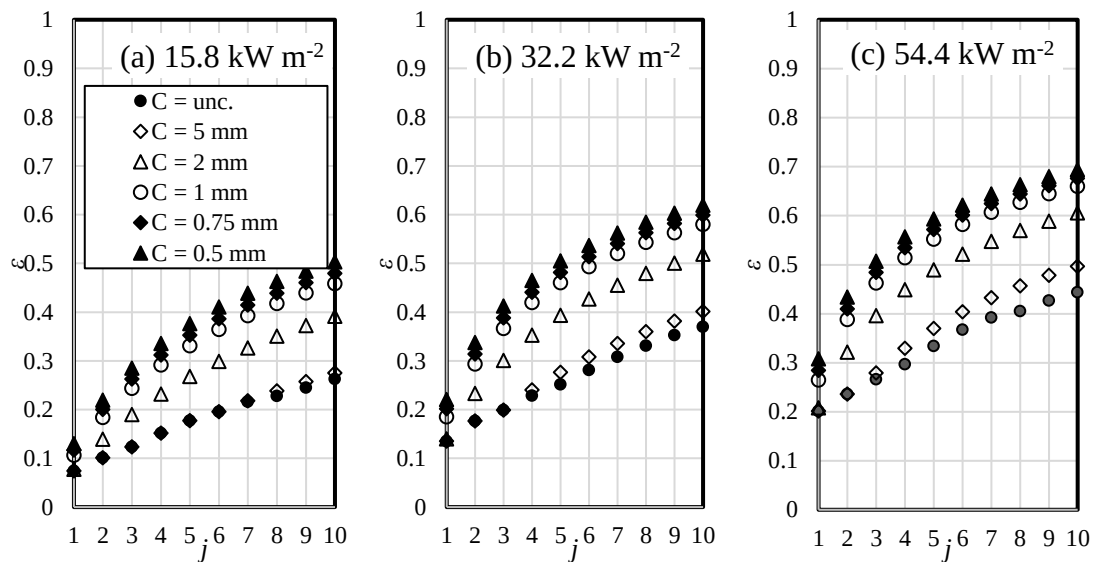


Figure 46: Modelled void fraction in confined and unconfined 10 tube columns at three representative heat fluxes.

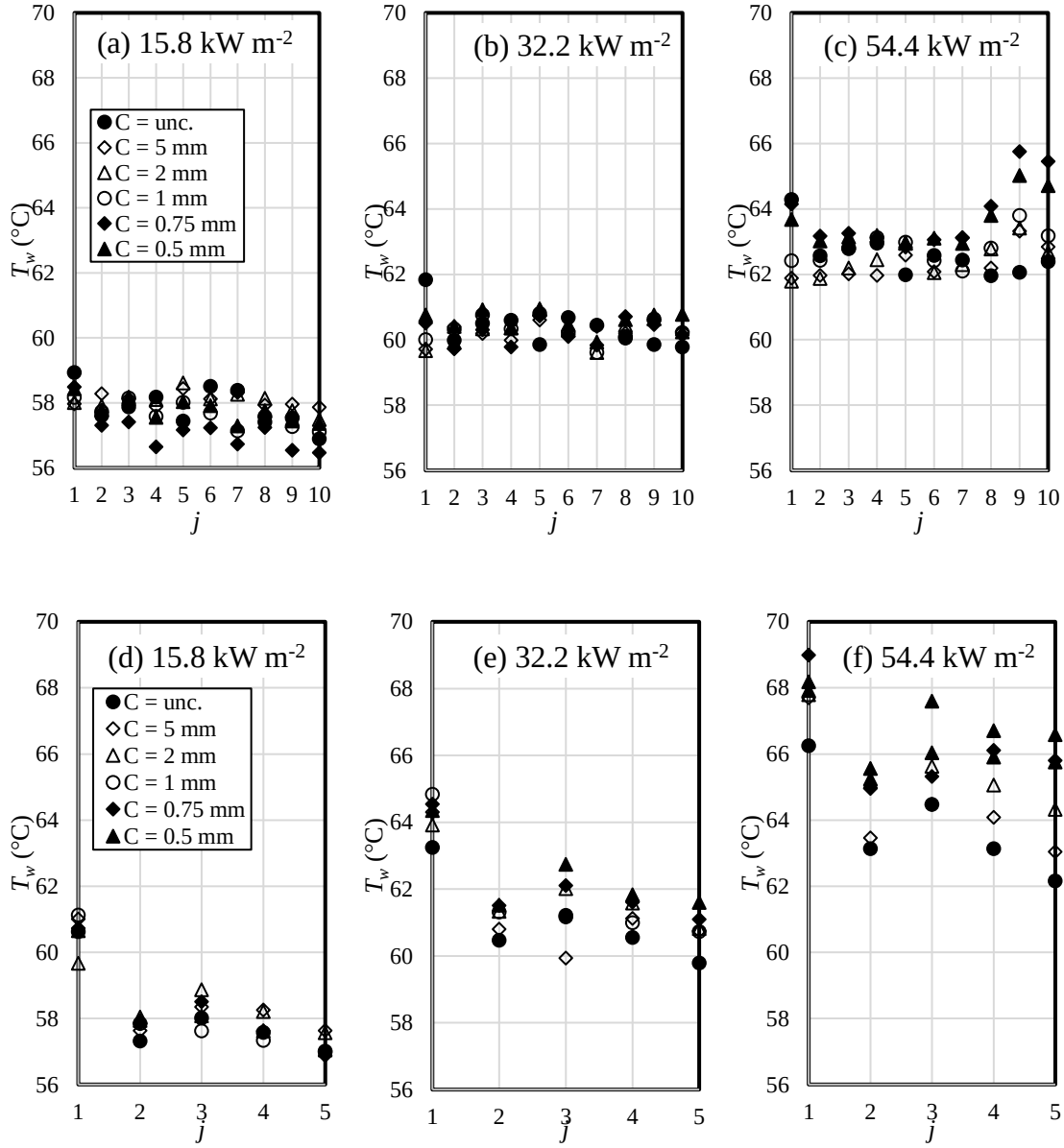


Figure 47: Wall temperature profiles for tube number  $j$  within confined and unconfined columns with (a-c) 10 tubes, and (d-f) 5 tubes, at three representative heat fluxes.

## 6.4. Conclusions

An analytical model to quantify void fraction in confined bubble plumes in pool boiling conditions was presented. Experimental validation showed good agreement with the theory, with mean void fraction predictive error of 12.5%.

Measurements of an unconfined plume in pool boiling were made to produce a semi-empirical model of void fraction in unconfined plumes. Void fraction in an unconfined plume has been showed to increase monotonically with column height.

Pool boiling on confined and unconfined tube columns in Novec 649 was performed with the following key conclusions:

- Three dryout cases were observed. At the very close spatial confinement ( $Bo_C = 0.29$ ), the geometry caused a vapour choke underneath the top tube. Where the void fraction of the top tube was less than  $\varepsilon \sim 0.64$ , critical heat flux occurred on the bottom tube in the column. Where voidage exceeded this threshold, dryout occurred first on top tube, indicating a significant effect of the bulk fluid dryness on *CHF*.
- The same  $\varepsilon \sim 0.64$  dryness threshold also affected heat transfer coefficient in the nucleate boiling regime below *CHF* conditions. Ordinarily in a column the tube temperatures above the second tube are constant with height; however, where  $\varepsilon > \sim 0.64$  the tube temperatures began to increase, with corresponding detriment to heat transfer coefficient.
- Noting that void fraction increases monotonically with column height even in an unconfined plume, and that heat transfer is degraded where  $\varepsilon > \sim 0.64$ , for a given heat flux there exists a maximum unconfined column height beyond which the mean column heat transfer coefficient declines.
- The maximum power density declines from an initial peak to a constant  $Q'' \sim 30 \text{ MW m}^{-2}$  for the tube columns tested.
- The bundle effect augmentation of heat transfer coefficient on higher tubes in a column by bubbles from lower tubes was pronounced at lower heat fluxes and less pronounced at high heat fluxes, which is consistent with extant literature.
- Intermediate tube spacings returned the highest heat transfer coefficients for unconfined and slightly confined columns. Increasing tube spacing mitigated against localised dryout in highly confined columns.

Potential future work includes the application of the confined plume dryness model, and development of the unconfined plume model, against a range of boiling heat transfer fluids. Experimental investigation of slip ratio at for the buoyancy-driven flow in pool boiling conditions would further improve model fidelity.

## 6.5. Statement of Authorship

### Statement of Authorship for joint/multi-authored papers for PGR thesis

To appear at the end of each thesis chapter submitted as an article/paper

The statement shall describe the candidate's and co-authors' independent research contributions in the thesis publications. For each publication there should exist a complete statement that is to be filled out and signed by the candidate and supervisor.

|                     |                                                                                                                                                                                                                                                                                                                             |
|---------------------|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| Title of Paper      | Pool Boiling of Horizontal Mini-Tubes in Unconfined and Confined Columns                                                                                                                                                                                                                                                    |
| Publication Status  | <input checked="" type="checkbox"/> Published<br><input type="checkbox"/> Accepted for Publication<br><input type="checkbox"/> Submitted for Publication<br><input type="checkbox"/> Unpublished and unsubmitted work written in a manuscript style                                                                         |
| Publication Details | Bartle, R. S., and Walsh, E. J. (2019) 'Pool Boiling of Horizontal Mini-Tubes in Unconfined and Confined Columns'. <i>International Journal of Heat and Mass Transfer</i> , 145, 118733.<br><a href="https://doi.org/10.1016/j.ijheatmasstransfer.2019.118733">https://doi.org/10.1016/j.ijheatmasstransfer.2019.118733</a> |

#### Student Confirmation

|                                          |                                                                                                                                 |      |            |
|------------------------------------------|---------------------------------------------------------------------------------------------------------------------------------|------|------------|
| Student Name:                            | Roy Sowdagar Bartle                                                                                                             |      |            |
| Contribution to the Paper                | Co-designed experimental rig and method with supervisor;<br>performed test programme;<br>analysed data;<br>prepared manuscript. |      |            |
| Signature [redacted from online version] |                                                                                                                                 | Date | 11/09/2019 |

#### Supervisor Confirmation

By signing the Statement of Authorship, you are certifying that the candidate made a substantial contribution to the publication, and that the description described above is accurate.

|                                               |  |         |
|-----------------------------------------------|--|---------|
| Supervisor name and title: Prof. Edmond Walsh |  |         |
| Supervisor comments                           |  |         |
| None                                          |  |         |
| Signature [redacted from online version]      |  | Date    |
|                                               |  | 11/9/19 |

This completed form should be included in the thesis, at the end of the relevant chapter.

## 7. Bubble Nucleators to Enhance Pool Boiling in Tube Bundles

Bartle, R. S. and Walsh, E. J. (2019) ‘Bubble Nucleators to Enhance Pool Boiling in Tube Bundles’, in *Proceedings of the 14th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics (HEFAT2019)*. Wicklow, Ireland.

### Abstract

Pool boiling of a column of two horizontal tubes was performed at moderate heat fluxes in the dielectric fluid Novec 649. The effect on heat transfer by the addition of bubble nucleators was examined. Two nucleator devices were compared: a microscale nichrome wire nucleator, and a tubular nucleator of the same dimensions as the column tubes. The effect of nucleator heat flux on column heat transfer was negligible once uniform nucleation was established on the nucleator devices. The heat transfer coefficient of the lower tube in the column was enhanced with the addition of a nucleator and was most pronounced at low heat fluxes. Increased nucleation site density was observed with the use of nucleator devices. Compared to the wire nucleator, the tubular nucleator gave a slightly improved lower tube heat transfer but required an input power 3.4 times higher.

### 7.1. Introduction

The miniaturisation of power electronics places increased demand on engineers to design cooling strategies capable of dissipating high heat fluxes. Coupled with this is the temperature sensitivity of many electronic components, demanding relatively low service temperatures and the minimisation of temperature gradients across the component, in order to increase reliability (Sohel Murshed and Nieto de Castro, 2017).

Nucleate pool boiling offers high heat transfer coefficients and the potential for passive system operation. When a dielectric coolant is used, direct contact can be made

between the hot surface and the fluid, so minimising the thermal path resistance (Bartle, Menon and Walsh, 2018). Pool boiling on tube bundles is also relevant to shell-side heat transfer of shell-and-tube heat exchangers in low and moderate mass flow rate applications including flooded refrigerant (Joo Hong, Seok Wang and Woo Park, 2019), desalination evaporators (Liu and Ishibashi, 2001) and kettle reboilers (McNeil *et al.*, 2010).

In a column or array of horizontal tubes, the upper tubes benefit from increased  $h_{NB}$  compared to the lower tubes (Jensen and Hsu, 1988; Memory, Sugiyama and Marto, 1995; Fujita and Hidaka, 1998; Gupta, 2005; Kang, 2015; Bartle and Walsh, 2019a). Two complementary hypotheses have been advanced to explain this. Minns (1969) suggested that a continuous liquid microlayer is created by a vapour bubble sliding along a surface, which Cornwell (1982; Cornwell and Einarsson, 1990) applied to tube bundles to explain the heat transfer enhancement of upper tubes. Alongside microlayer evaporation, convective enhancement is suggested to include increased local (Cornwell and Einarsson, 1990) and bulk flow velocity (Gupta, 2005; Kang, 2015) and liquid mixing (Kang, 2015). Therefore, in nucleate boiling, the heat transfer coefficient is suggested (Cornwell, 1990; Hahne, Chen and Windisch, 1991; Thome, 2006; Swain and Das, 2014) to be composed of a convective component, and an evaporated bubble component alternating cyclically with heat transfer through a thermal microlayer:

$$h_{NB} = h_c + h_b + h_\delta. \quad (7.1)$$

Upper tube heat transfer enhancement in a column must be due to the enhancement of one or more of these components.

This paper investigates the effect of adding bubble nucleator devices under a tube column. The effects of nucleator type and power dissipation on column heat transfer

coefficient are investigated. High speed imaging provides some further insight into the nucleator heat transfer coefficient enhancement.

## 7.2. Method and Materials

As diagrammed in Figure 48, a column of two 304 grade stainless steel tubes with heated length 93 mm, outer diameter 2.55 mm, and thickness 0.2 mm, were mounted horizontally, electrically connected in series, and uniformly Joule heated. Underneath this were placed two independently adjustable Joule heated bubble nucleators for evaluation, with dimensions shown in the Figure 48 inset. The tubular nucleator was simply another tube of the same specification as the column tubes. The wire nucleator was a 100  $\mu\text{m}$  diameter nichrome wire, stretched taut and soldered at both ends into a brass solder bucket. Spacing plates to hold these components were 3D printed from PTFE. A separate auxiliary heater, used to maintain the fluid at saturation temperature, was situated a distance away from the tube column. The boiling container, into which all these components were placed, was an optically transparent sealed acrylic box that allowed the experiment to be observed by eye and photographed by camera.

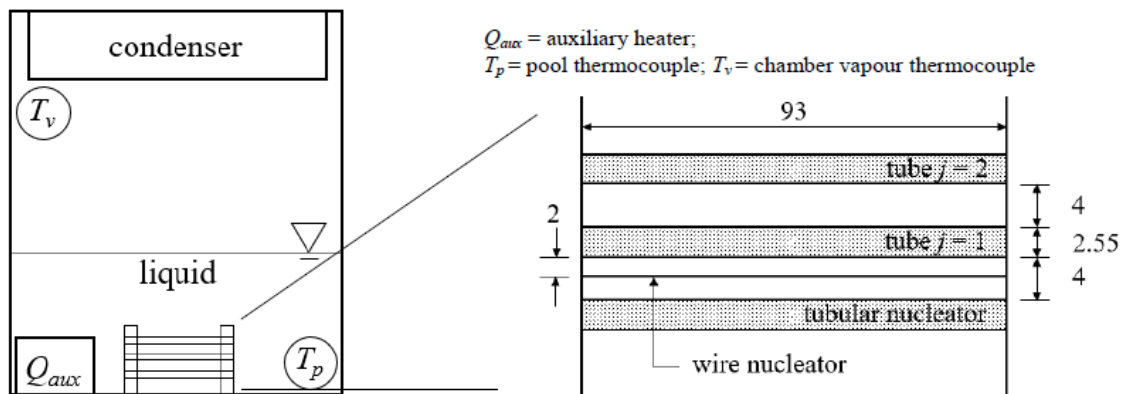


Figure 48 Diagram of experimental apparatus with tube column detail, expanded.

Miniature thermistors, electrically arranged in Wheatstone bridge configuration, were inserted into the tubes to measure the tube surface temperature. Type K

thermocouples read the liquid pool, vapour, and room ambient temperatures. All thermistors and thermocouples were calibrated to  $\pm 0.1$  °C accuracy in a thermal bath. A pressure transducer recorded gauge pressure from a tapping in the boiling container.

The fluid used was Novec 649, a dielectric coolant with acceptable heat transfer and environmental properties. Prior to use, the fluid was degassed with a sonicator for 10 min to encourage cavitation, and then 45 min of vented boiling using the auxiliary heater in the boiling container removed any remaining noncondensibles. Previous studies with low surface tension fluids have noted that a boiling curve obtained by increasing power from nil can show a thermal hysteresis (El-Genk and Bostanci, 2003), where wall temperature increases until stable nucleation sites are generated, and then declines as vaporisation occurs. To mitigate against this, the boiling curves were obtained by beginning at the highest heat flux tested and then incrementally decreasing the power. Three configurations were tested: only the tube column heated, the tube column with the tubular nucleator, and the tube column with the wire nucleator. To evaluate the effect of nucleator power on column heat transfer, an initial experimental programme was run with the wire nucleator and a single tube above.

High speed camera imagery, in conjunction with image analysis software, was used to evaluate bubble dynamics above tube  $j = 1$ . Bubble diameter was measured directly from a still image at each heat test condition, assuming a spherical profile. Bubble velocity was measured by measuring the vertical displacement of a bubble between two images at a fixed frame rate. Ebullition frequency was measured by counting the number of frames required for a nucleating bubble to depart the tube. For each test condition, 10 samples were measured and the mean reported.

Nucleation site density for the lower tube at the lowest heat flux was calculated by simply counting the number of nucleating (attached) bubbles observed on the tube and discounting any bubbles only passing by, as shown in Figure 49. It should be noted that the nucleation site density and ebullition frequency measurements are subject to higher uncertainty than other bubble measurements as a degree of subjectivity exists in judging whether a bubble is attached to the surface or passing over.

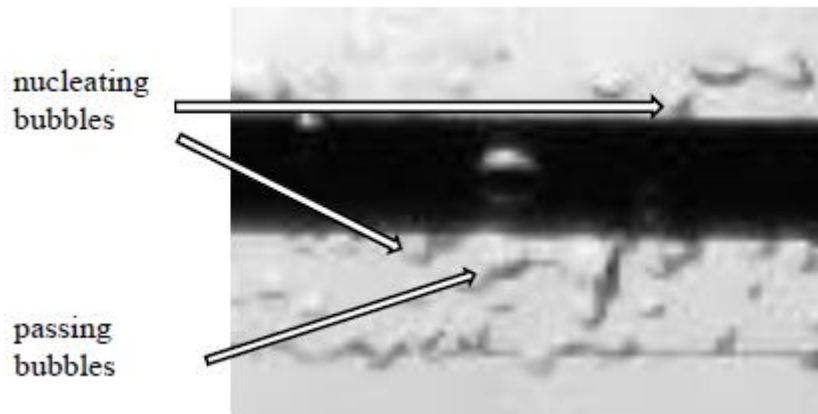


Figure 49: Image to show an attached nucleating bubble versus a passing bubble generated by the wire nucleator below.

Heat flux generated is

$$q'' = \frac{I^2 R}{A_s} \quad (7.2)$$

where  $A_s = \pi dL$ . The nucleate boiling heat transfer coefficient:

$$h_{NB} = \frac{q''}{\Delta T_{sat}}, \quad (7.3)$$

$$\Delta T_{sat} = T_w - T_{sat} \quad (7.4)$$

Kline and McClintock (Kline and McClintock, 1953) uncertainty was computed for the experiment with  $a_I = 0.3\%$  and  $w_L = \pm 0.1$  mm. Tube resistance was measured with  $w_R = 0.0037 \Omega$  and returned heat flux uncertainty of 6.29%; the wire resistance per unit

length is given by the manufacturer as  $137.5 \Omega \text{ m}^{-1}$  and returns a heat flux uncertainty of 0.50%. Heat transfer coefficient uncertainty on the tubes varied from 6.66% to 7.24%.

### 7.3. Results

#### 7.3.1. Bubble Nucleator Heat Flux

An initial test to evaluate the effect of nucleator wire power on the  $h_{NB}$  of a single tube above, with tube heat flux of  $20.6 \text{ kW m}^{-2}$ , was undertaken. Over the nucleator wire heat range tested, corresponding to a nucleator wire heat flux between  $52.6$  and  $130 \text{ kW m}^{-2}$ , the heat transfer coefficient of the tube above varied by less than 1.03%.

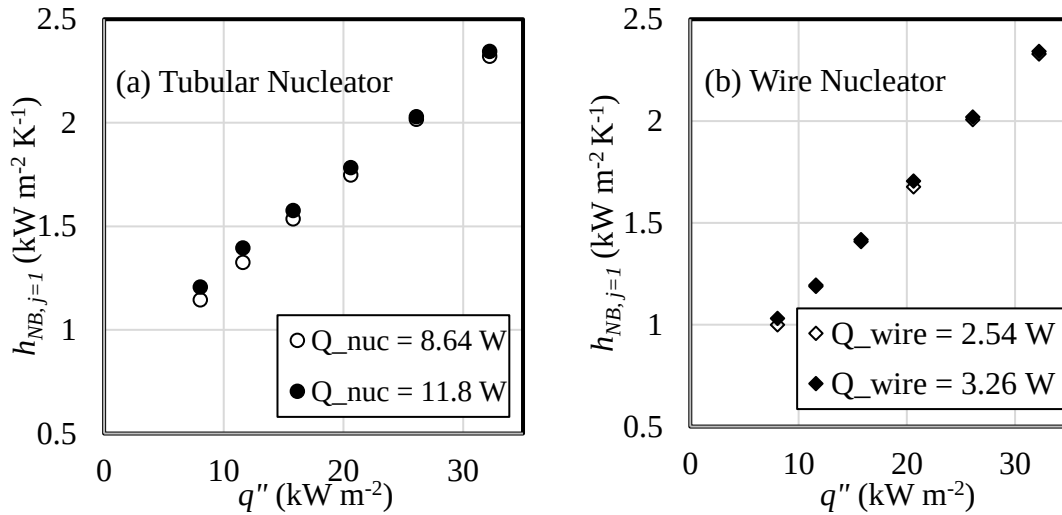


Figure 50: The effect of (a) tubular and (b) wire nucleator heat flux adjustment on the heat transfer coefficient of the lower tube in the column.

This was then evaluated in tube column conditions and with both nucleator device types. No notable heat transfer effect was observed on tube  $j = 2$  for varying heat flux on either the tubular or wire nucleator. Figure 50 shows the effect of varying the tubular and wire nucleator heat fluxes on the heat transfer coefficient of tube  $j = 1$ . Figure 50 (a) and (b) both show the same trend of a slight  $h_{NB}$  increase with increasing heat nucleator heat flux at low column heat fluxes. As more vapour is generated, the probability that a bubble will contact a potential nucleation site on the above tube is increased. In numerical terms, however, only the enhancement caused by the tubular nucleator at the lowest column heat

flux (5.09 %) approaches statistical significance, whilst the heat transfer enhancement caused by increasing nucleator heat flux in the other conditions is lower than the experimental uncertainty.

### 7.3.2. Heat Transfer Coefficient through Column

Having established the negligible effect that varying nucleator heat flux has on column heat transfer, the results presented hereafter are with wire and tubular nucleator  $q''$  set at 87 and 11.6 kW m<sup>-2</sup> respectively.

Figure 51 shows heat transfer coefficient comparisons between the no nucleator, wire nucleator, and tube nucleator cases for the lower and upper tubes in the column. As has been extensively reported, without a nucleator the upper tube returned higher  $h_{NB}$  for all heat fluxes tested, with the difference decreasing with increasing heat flux. At  $q'' = 8.05$  kW m<sup>-2</sup>, the heat transfer coefficient of the lowest tube increased by 204 and 351 W m<sup>-2</sup> K<sup>-1</sup> with the addition of the wire and tubular nucleators respectively, corresponding to tube temperature reductions of 2.07 °C and 3.09 °C.

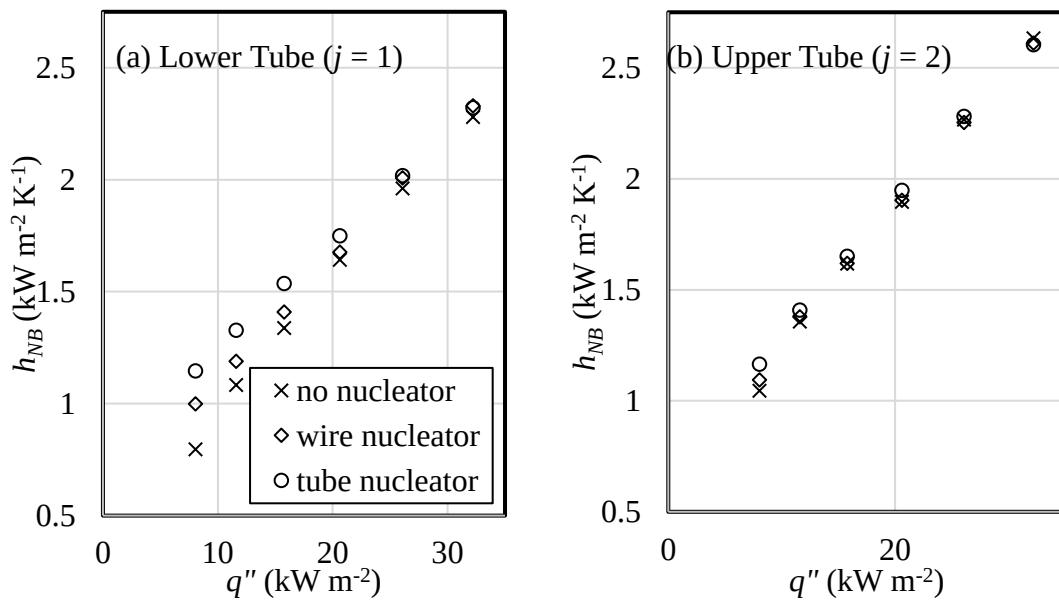


Figure 51: The effects of a bubble nucleator on the heat transfer coefficient of (a) the lower tube and (b) the upper tube in a 2 tube column.

### 7.3.3. Bubble Dynamics

Mean bubble diameter for the experimental conditions tested is shown in column 1 of Table 12. For both heat fluxes, bubble diameter did not change significantly with the use of the wire nucleator but increased moderately with the use of the tubular nucleator. At  $q'' = 8.05 \text{ kW m}^{-2}$  the use of either nucleator increased bubble non-sphericity as shown in Figure 52(a-c). Bubble diameter increased considerably with increased heat flux due partly to coalescence; note in Figure 52(c) the bubbles above the nucleator tube are close to spherical, whereas the bubbles above tube  $j = 1$  have a profile that is more elliptical than spherical.

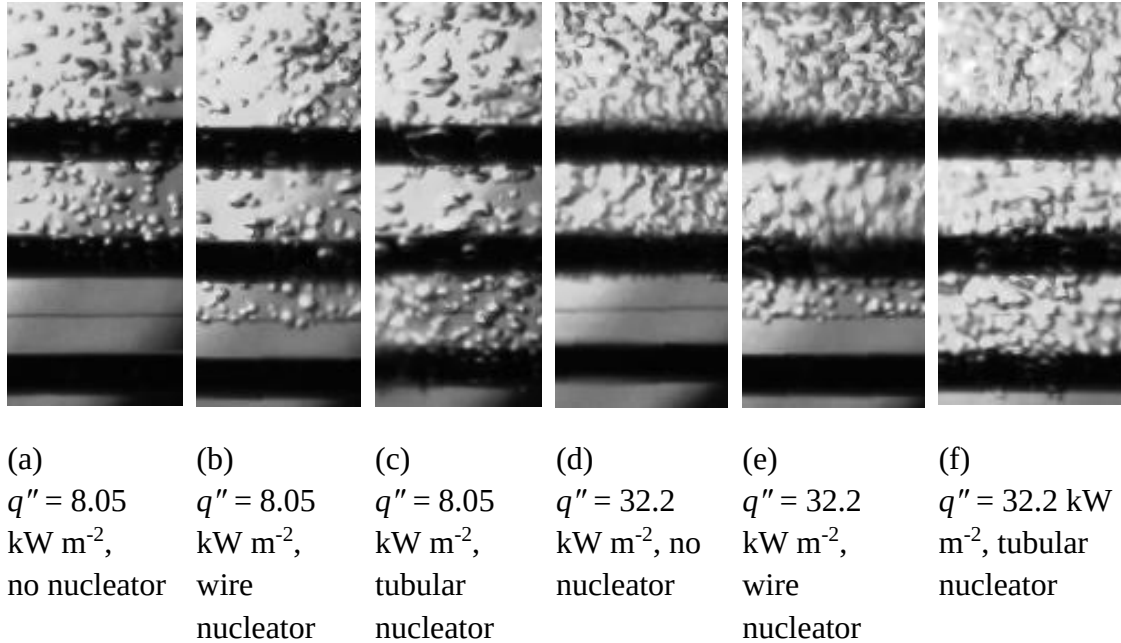


Figure 52: Images to show bubble dynamics for the three nucleator conditions at two representative heat fluxes.

The relationship between bubble diameter and velocity in natural convection has been studied extensively, and most correlations published are empirical. The Massarani correlation, appearing first in English in Scheid *et al.* (1999) predicts that a bubble diameter increase from 1.03 to 1.57 mm increases bubble velocity from  $161 \text{ mm s}^{-1}$  to  $206 \text{ mm s}^{-1}$ , which agrees well with the experimental data above for the no nucleator cases. Ebullition frequency did not vary significantly between test cases.

We note that, although the addition of the tubular nucleator increases mean bubble diameter, there is no corresponding increase of vertical bubble velocity; rather the addition of a nucleator appears to retard the bubble, especially at the lowest heat flux. The horizontal along-tube component of bubble velocity was insignificant in all cases. Ergo the hypothesis of increased convection from higher bubble velocities or increased liquid/vapour mixing contributing to higher  $h_{NB}$  on upper tubes in a column is not substantiated by these tests.

*Table 12: Measured bubble diameter, vertical velocity, and ebullition frequency at different test conditions.*

|                                                        | $d_b$ (mm)        | $u_b$ (mm s <sup>-1</sup> ) | $f_b$ (s <sup>-1</sup> ) | $\Psi$ (k bubbles m <sup>-2</sup> ) |
|--------------------------------------------------------|-------------------|-----------------------------|--------------------------|-------------------------------------|
| $q'' = 8.05$ kW m <sup>-2</sup> ,<br>no nucleator      | 1.03 (SD = 13.3%) | 143 (SD = 7.43%)            | 66.0 (SD = 54.2%)        | 94.3 (SD = 5.11%)                   |
| $q'' = 8.05$ kW m <sup>-2</sup> ,<br>wire nucleator    | 1.01 (SD = 20.3%) | 133 (SD = 9.40%)            | 73.3 (SD = 36.9%)        | 124 (SD = 8.08%)                    |
| $q'' = 8.05$ kW m <sup>-2</sup> ,<br>tubular nucleator | 1.23 (SD = 22.2%) | 124 (SD = 8.30%)            | 73.3 (SD = 36.9%)        | 135 (SD = 5.16%)                    |
| $q'' = 32.2$ kW m <sup>-2</sup> ,<br>no nucleator      | 1.57 (SD = 12.2%) | 199 (SD = 7.23%)            | 65.8 (SD = 44/0%)        | -                                   |
| $q'' = 32.2$ kW m <sup>-2</sup> ,<br>wire nucleator    | 1.59 (SD = 16.6%) | 195 (SD = 12.6%)            | 65.3 (SD = 45.5%)        | -                                   |
| $q'' = 32.2$ kW m <sup>-2</sup> ,<br>tubular nucleator | 1.73 (SD = 11.5%) | 192 (SD = 8.53%)            | 68.3 (SD = 38.5%)        | -                                   |

Nucleation site density is

$$\psi = \frac{N_b}{A_s}. \quad (7.5)$$

At  $q'' = 8.05$  kW m<sup>-2</sup>,  $\psi$  on the lower tube  $j = 1$  increased by 31.5% and 43.2% with the use of the wire and tubular nucleators respectively. The heat removed by bubbles is

$$Q_b = N_b \frac{\pi}{6} d_b^3 f_b \rho_v h_{fg}. \quad (7.6)$$

Hence, where  $d_b$  is produced from a single nucleation site, the evaporated bubble component of eqn. (7.1) is

$$h_b = \frac{Q_b}{A_s \Delta T_{sat}}. \quad (7.7)$$

As the data does not suggest increased convective heat transfer, the increased  $h_{NB}$  from the nucleation devices appears to come from the increased  $h_b$  resulting from greater nucleation site density. The flow of vapour bubbles from below appears to activate and sustain potential nucleation sites. Evaluation of the nucleator effect on the  $h_b$  component can be made by substituting the experimental data into eqn. (7.7) to produce the final column in Table 13. For the  $q'' = 8.05 \text{ kW m}^{-2}$  heat flux,  $h_b$  enhancement around thrice the experimental enhancement shown in Figure 51 (a) is predicted because bubble diameter is cubed, so even a modest change in  $d_b$  corresponds to a large change in  $h_b$ . As this modelled enhancement is not fully realised, some bubble coalescence is suggested to occur. As  $h_b$  and  $h_\delta$  compete for the same heat, and as  $h_c$  does not appear to change significantly, the addition of a nucleator must significantly enhance  $h_b$  at the partial expense of  $h_\delta$ . However, although enhanced nucleation site density is accounted for in the  $h_b$  term, it is the result of the generation of a natural convection bubbly flow by the nucleator device underneath the tube. Hence convective flow effects, albeit not convective heat transfer, can be said to account for the heat transfer enhancement produced by a nucleator device on lower tubes in a column.

*Table 13: Comparison of experimental results and modelled  $h_b$  enhancement at  $q'' = 8.05 \text{ kW m}^{-2}$ .*

| <b>Nucleator type</b> | <b>Experimental <math>h_{NB}</math> augmentation</b> | <b>Nucleation site density <math>N_b</math> change</b> | <b>(Modelled) <math>h_b</math> augmentation</b> |
|-----------------------|------------------------------------------------------|--------------------------------------------------------|-------------------------------------------------|
| wire                  | 25.1%                                                | 31.5%                                                  | 38.6%                                           |
| tubular               | 44.2%                                                | 43.2%                                                  | 125%                                            |

At higher heat fluxes, more homogenous nucleation around the tube can be observed by comparing Figure 52 (a) and (d) as hitherto dormant nucleation sites are activated by the higher local surface temperatures. This has been reported before (Gupta,

2005) and partly explains the reduced effect of nucleator devices on  $h_{NB}$  at elevated heat fluxes.

Producing the column  $h_{NB}$  enhancement at low heat fluxes comes at the cost of a power input into the nucleator devices of 2.54 and 8.64 W for the wire and tubular nucleator respectively. At the lowest heat flux tested, the tubular nucleator increases the column lower tube  $h_{NB}$  by 18.5% above the wire nucleator  $h_{NB}$  enhancement but requires 3.4 times the power. This reinforces that the bubbly flow, rather than the vapour quantity, generated by the nucleator activates potential nucleation sites and dominates the  $h_{NB}$  enhancement resulting from nucleator use.

#### 7.4. Conclusions

Pool boiling of a tube column was performed with bubble nucleators positioned underneath. Within the experimental range, the following was found:

- Once uniform bubble generation was established on the nucleator, the absolute value of heat generated by the nucleator had negligible effect on heat transfer coefficients in the column.
- Increased heat transfer coefficients were observed on the lower tube as the result of nucleator use. The heat transfer enhancement was most pronounced at low column heat fluxes and decreased with increasing heat flux.
- This heat transfer enhancement appears to be due to increased nucleation site activation caused by the convective flow generated by rising bubbles from the nucleator devices.
- The addition of a nucleator device had negligible impact on the upper tube in the column.

## 7.5. Statement of Authorship

### Statement of Authorship for joint/multi-authored papers for PGR thesis

To appear at the end of each thesis chapter submitted as an article/paper

The statement shall describe the candidate's and co-authors' independent research contributions in the thesis publications. For each publication there should exist a complete statement that is to be filled out and signed by the candidate and supervisor.

|                     |                                                                                                                                                                                                                                                                    |
|---------------------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| Title of Paper      | Bubble Nucleators to Enhance Pool Boiling in Tube Bundles                                                                                                                                                                                                          |
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#### Student Confirmation

|                                          |                                                                                                                                                                                    |      |            |
|------------------------------------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|------|------------|
| Student Name:                            | Roy Sowdagar Bartle                                                                                                                                                                |      |            |
| Contribution to the Paper                | Co-designed experimental rig and method with supervisor;<br>performed test programme;<br>analysed data;<br>prepared manuscript;<br>prepared and delivered conference presentation. |      |            |
| Signature [redacted from online version] |                                                                                                                                                                                    | Date | 11/09/2019 |

#### Supervisor Confirmation

By signing the Statement of Authorship, you are certifying that the candidate made a substantial contribution to the publication, and that the description described above is accurate.

|                                               |      |         |
|-----------------------------------------------|------|---------|
| Supervisor name and title: Prof. Edmond Walsh |      |         |
| Supervisor comments<br>None                   |      |         |
| Signature [redacted from online version]      | Date | 11/9/19 |

This completed form should be included in the thesis, at the end of the relevant chapter.

## 8. Discussion

In this Chapter, the pieces of research reported in Chapters 3 to 7 are brought back together. Section 8.1 formally addresses the research questions identified in Section 1.5 in light of the completed research. Section 8.2 discusses these findings with application to the stator windings thermal management problem described in Section 1.4 and then more generally to electrical machines cooling. Section 8.3 considers future research avenues emerging from this work.

### 8.1. Research Questions Review

| No.  | Research Question                                                                                                                                  | Chapter |
|------|----------------------------------------------------------------------------------------------------------------------------------------------------|---------|
| 1(a) | <i>Can the pool boiling performance of a circular-wire motor windings geometry (MWG) surface be acceptably predicted by existing correlations?</i> | 3       |

The Rohsenow (1952) correlation, when used with the curve-fitting parameters derived experimentally, provides a good description of pool boiling heat transfer from an MWG surface. The correlations of Rohsenow (1952), Stephan-Abdelsalam (1980), and Gorenflo-Kenning (2010) provide the most accurate *a priori* prediction of heat transfer coefficient, at 34.9%, 34.4%, and 37.6% mean absolute difference (*MD*). The magnitude of these *MDs* suggests that these correlations are suitable only for initial estimates of heat transfer from an MWG surface, and that experimental testing of a representative surface is required to obtain the curve-fitting parameters for the Rohsenow correlation for more comprehensive heat transfer description.

- 1(b) *How does the pool boiling heat transfer of an MWG surface compare with heat transfer from a flat plate for various surface orientations?* 3

An MWG surface exhibits a higher heat transfer coefficient than a flat surface at all heat fluxes, even when area-compensation is applied. Analysis of photographic evidence shows a longer contact line at the moment prior to bubble departure, requiring a larger force to overcome the surface tension. As the vapour bubble is pinned to the surface for a longer time and over a larger area, it is plausible that any liquid microlayer evaporation contribution to the heat transfer coefficient increases. However, the existence and contribution of a liquid microlayer is itself a matter for further investigation (see Section 8.3).

MWG surfaces agree with the well-reported trends for flat surfaces with respect to surface orientation change. A negligible effect on heat transfer exists by increasing orientation from  $0^\circ$  (horizontal) to  $90^\circ$  (vertical), whilst at  $128^\circ$  heat transfer coefficient was slightly elevated at low heat fluxes and slightly depressed at high heat fluxes.

- 2 *How does removal of the insulative coating on a conductor affect heat transfer in pool boiling conditions?* 4

A bare flat copper surface was tested in pool boiling conditions; this was compared with the same surface covered in a resin coating with ellipsoidal profile and mean thickness of  $60.0\ \mu\text{m}$ , which is representative of typical insulation thicknesses on conductors in electrical machines. Considering first the heat transfer

coefficient between the boiling surface and the fluid, the resin coated surface exhibited lower heat transfer coefficient due partly to lower surface roughness. However, extant correlations evaluating the effect of surface roughness on heat transfer coefficient underpredicted the effect, confirming that single surface roughness parameters inadequately describe the effect of surface microgeometry on heat transfer coefficient. The additional thermal resistance of the insulation also significantly increased the copper junction temperature. Ergo removal of the insulation layer on a conductor improves both the boiling heat transfer coefficient on the heat transfer surface and the conductive thermal path.

- 3     *How do spatial confinement and liquid level affect pool boiling heat transfer from a flat surface?*     5

In Chapter 5, Bond number was formulated as the ratio of characteristic length (confinement, liquid level) to capillary length. For the spatial confinement of a vertical surface, only where  $Bo \leq 0.58$  was a significant relationship between heat flux and heat transfer coefficient observed, with a deteriorated heat transfer coefficient at high heat fluxes. Chapter 6 shows that, where void fraction exceeds a threshold, deteriorated heat transfer occurs, which explains this finding. Chapter 5 showed that, at the confinement  $Bo = 0.28$  ( $C = 0.24$  mm), a marked difference in the boiling curve was observed, with considerably enhanced heat transfer at low heat fluxes and considerably suppressed heat transfer at high heat fluxes.

The effect of liquid head on heat transfer coefficient exhibited similar trends, with a clear decline in heat transfer where  $Bo < \sim 0.5$ . The effect of hydrostatic pressure, due to liquid head, on the saturation temperature of the fluid was negligible for the liquid head range tested ( $0 \leq Bo_H \leq 10$ ).

- 4(a) *How may bulk void fraction be modelled for a bubble plume around a column of conductors in spatially unconfined and confined pool boiling?* 6

For a column of conductors in spatially-confined conditions, modelling of bulk dryness fraction is straightforward once the plume is fully-developed (i.e. the plume fills the channel) by taking the ratio of vapour mass flow rate to total mass flow rate. Where the plume is developing, the plume width is measured to obtain the flow area. To convert dryness to void fraction, either a no-slip assumption is applied, or a slip ratio correlation is used. Slip ratio in buoyancy-driven two-phase flows is eminently worthy of further research.

- 4(b) *How does the bulk void fraction in the vicinity of a single conductor in a column affect its pool boiling heat transfer coefficient and critical heat flux?* 6

Bulk void fraction has an adverse effect on heat transfer coefficient above a specific threshold:  $\varepsilon > \sim 0.64$  for Novec 649. Where void fraction in a column of tubes exceeds this, critical heat flux typically occurs first on the top tube; where void fraction is under this threshold, critical heat flux typically occurs on the bottom tube, where heat transfer coefficient is ordinarily lower.

- 5     *Can the use of a low power nucleator device mitigate “bundle effect”, the reduced heat transfer coefficient of the bottom conductor in a boiling column at low heat flux conditions?*     7

Two nucleator devices were tested: a tube identical to those used in the column, and a 100  $\mu\text{m}$  diameter nichrome wire. These were positioned underneath a two-tube column and Joule heated to generate a buoyancy-driven bubbly flow. With nucleators off, bundle effect was noticeable on the lower tube but decreased with increasing heat flux. Activating either nucleator device enhanced the heat transfer coefficient of the lower conductor by the earlier activation of potential nucleation sites; the magnitude of this enhancement decreased with increasing heat flux. At the lowest heat flux tested,  $q'' = 8.05 \text{ kW m}^{-2}$ , the tubular nucleator increased the column lower tube heat transfer coefficient by 44.2% above the nucleator-off case, corresponding to 18.5% above the wire nucleator enhancement. However, the tubular nucleator required 3.4 times the power to achieve this. It was also observed that, once uniform nucleate boiling was generated on the nucleator device, further increasing power to the device had no impact on the heat transfer coefficient of the tube above.

## 8.2. Pool Boiling in Electrical Machines

The research undertaken in this thesis confirms that the pool boiling of stator end windings is a thermal management solution offering lower maximal windings temperatures and therefore reduced Ohmic losses and improved reliability. The heat transfer coefficient from a circular wire windings geometry surface has been quantified

as superior to a flat surface. However, insulation materials between the coolant and the conductor decrease the boiling heat transfer coefficient, have a high thermal resistance and, where dielectric coolants are used, are unnecessary so long as the conductors are fixed in position and not able to flex to the extent of contacting another conductor. This is difficult to achieve with traditional circular profile windings but plausible with edge wound conductors.

Where the end windings comprises a stack of edge wound conductors, channel spacing surrounding the stack and between each conductor turn should ideally exceed  $Bo \sim 0.5$  to prevent a decline in heat transfer coefficient. Where a channel spacing is less than  $Bo \sim 0.5$ , heat transfer coefficient is enhanced at low heat flux conditions but suppressed at high heat flux conditions; in such a situation critical heat flux, which is catastrophic in the context of stator windings cooling, will occur first in the highly confined channel and therefore dictates the maximum current of the machine. More research is needed into the highly confined pool boiling regime where  $0 < Bo \leq 0.5$  as the channel width created between windings turns by the removal of the insulation layer is typically in this region. Further work is also required on modifying the end windings of edge wound conductors to increase the spacing between each conductor turn.

Additionally, the development of an experimentally-validated model for pool boiling of a tube column, representing a stack of stator conductor turns, shows that heat transfer coefficient deteriorates where the bulk void fraction in the surrounding liquid exceeds a certain void fraction threshold. For Novec 649 this threshold is  $\sim 0.64$ ; it is expected that this threshold will change slightly with the thermofluidic properties of different coolants, but this has yet to be tested. Exceeding this same threshold in the column predicts that critical heat flux will occur on the top tube whilst, underneath the threshold, critical heat flux occurs on the bottom tube. Note that it is possible to exceed

the void fraction threshold, in some cases considerably, before arriving at critical heat flux. Hence the threshold marks the beginning of heat transfer coefficient deterioration and predicts where in the column critical heat flux will occur, but it does not predict that critical heat flux is imminent – critical heat flux can occur at bulk void fractions above or below the threshold. Applying this to the design of stacked end windings in pool boiling, for optimal thermal performance the width of the surrounding coolant channels should be sufficiently large to avoid exceeding the bulk void fraction threshold at any point and, in this situation, the temperature of the bottom turn (the turn closest to the centre of gravity) in the stator should be particularly monitored as critical heat flux will occur there. Where the bulk void fraction threshold is exceeded around the stator stack, critical heat flux will occur in the top turn (furthest from the centre of gravity) so monitoring of the top turn temperature is most needful.

At low to moderate heat fluxes, the bottom turn will exhibit lower heat transfer coefficient than the turns above, because of lower nucleation site density. This can be overcome by the introduction of a nucleator device generating bubbly flow underneath the bottom end windings turn, thus activating dormant nucleation sites. The improved bottom turn heat transfer coefficient will translate in lower windings temperature, meaning reduced Ohmic losses in the turn. However, as the nucleator devices are simply Joule heated conductors themselves, they consume electrical energy. Further work is therefore needed to evaluate whether nucleator device energy consumption is less than the reduction in Ohmic losses.

Before passing from this discussion, the limitations of end windings pool boiling must also be noted. Where end windings pool boiling is employed an inverse logarithmic relationship, as shown previously in Figure 8, exists between machine length and maximum current. As a considerable axial temperature gradient exists, end windings

pool boiling is not ideally suited to long machines. However, for many designs of high power density machines, it offers high heat transfer coefficients coupled with system simplicity and the potential for passive operation. Indeed, coupled with a well-designed system involving vapour condensing on the outer surfaces of an aircraft, it is possible to conceive of a fully passive system with no moving parts required in routine operation. This intrinsic reliability is especially attractive in aeronautical applications, for which safety and reliability are paramount.

### **8.3. Future Work Directions**

Several avenues of future work emerge from this thesis.

#### **8.3.1. Pool Boiling Fundamentals**

- a. Development of a model to predict slip ratio in a buoyancy-driven bubble plume, for confined and unconfined conditions, would increase the accuracy of void fraction modelling and be useful to the design of end windings for pool boiling, and various heat exchangers and boiling water reactors.
- b. Exploration of the relationship between confinement and heat transfer coefficient in highly confined vertical and horizontal spaces with  $Bo < 0.25$ . This has been explored to a limited extent in flow boiling but not in pool boiling. Additionally, more spacing steps in the domain  $0.25 < Bo < 1$  will help more accurately identify the spacing at which heat transfer coefficient begins to deteriorate; from this, the maximum tube column power density of  $30 \text{ MW m}^{-3}$  identified in Section 6.3.4 may be improved upon.
- c. Exploration of the reality and heat transfer contribution of a liquid microlayer under a nucleating bubble. Some authors (e.g. Yao and Chang, 1983; Gupta, 2005; Kang, 2015; Kim *et al.*, 2016) present pool boiling heat transfer as

composed only of convective and evaporative heat transfer contributions, whilst others also include a liquid microlayer component (e.g. Cornwell and Schuller, 1982; Nishikawa *et al.*, 1984; Thome, 2006; Swain and Das, 2014). In pool boiling, except at the lowest heat fluxes the evaporative component dominates the heat transfer coefficient, so this question remained outside the scope of this thesis. More concentrated investigation of this question may indicate to what extent the present definitions of a liquid microlayer are interchangeable and, indeed, the reality, characteristics, and heat transfer contribution of a liquid microlayer. Analytical modelling, coupled with image analysis of a range of nucleating bubbles in low and high surface tension fluids, may be a useful approach.

### **8.3.2. Electrical Machines**

- a. Combined thermal and electromagnetic investigation of the effect of end windings heat transfer surface augmentation techniques. Can the spacing between turns in edge-wound end windings be increased for optimal pool boiling performance without adverse electromagnetic effects?
- b. It has been demonstrated that, at low heat fluxes, nucleator devices positioned under the bottom conductor in a stack of end winding turns increases the heat transfer coefficient of the bottom turn. Iterative numerical integration or finite element modelling of the stator windings fed by the extant experimental data obtained will show whether, at low power conditions, the power loss due to Ohmic heating can be reduced by a greater amount than the additional power required to activate the nucleators. This work is presently in progress by the author for future publication.
- c. The use of end windings pool boiling as a thermal management strategy is limited by the machine length. Convective cooling by flowing a fluid through channels

made in the stator windings or around the stator slot edges are presently being investigated (Popescu and Cavagnino, 2015; Schiefer and Doppelbauer, 2016; Wohlers *et al.*, 2018). Flow boiling offers superior heat transfer coefficients and, so long as certain parameters are met, produces a negligible thermal gradient along the length of the machine. The author has given some attention to this during his DPhil studies but, as it was felt to be outside the scope of the thesis, the research is not included here. Further work is required to generalise the findings and then undertake experimental validation.

### **8.3.3. Miscellaneous**

- a. Progress in aircraft electrification involves ancillary systems as well as electrical motors. Some of these devices use high heat flux control electronics. Finned heat sinks with a high conductivity thermal path to the heat source can be used to allow power electronics modules to be sited in restricted spaces. Knowledge acquired in this thesis on spacing effects can be applied to produce an algorithm for the design and optimisation of finned heat sinks, which can then be verified experimentally.
- b. More detailed research into the wider system will aid application of both pool and flow boiling in aircraft. Identification and configuration of condensation surfaces and modelling of thermodynamic profiles through a flight cycle is necessary. Perhaps the greatest distinctive of pool boiling is that fully passive operation is possible, which requires the design of surface geometries and traps to maintain flow circulation. Hence, attention is needed to aspects of system design and integration for phase-changing thermal solutions in aircraft. Some initial work in this area was done by the author during industrial secondment.

## 9. Conclusions

The research conducted in this thesis has filled several gaps in the literature. Pool boiling heat transfer characterisation of wire-wound motor windings geometry (MWG) surfaces have been characterised, with MWG surfaces returning higher heat transfer even with area-compensation. Extant correlations were evaluated against both surfaces; the Rohsenow correlation with curve-fitting parameters was the best descriptor for both surfaces, whilst the Gorenflo-Kenning and unfitted Rohsenow provided good *a priori* predictive accuracy. Pool boiling of a heated vertical surface parallel to an unheated surface was shown to considerably enhance heat transfer coefficient at low heat flux, and considerably degrade heat transfer coefficient at high heat flux, when  $Bo < 0.58$ . Reducing liquid level below a similar level produced a marked deterioration in heat transfer from a horizontal surface. A semi-empirical model was developed to evaluate bulk void fraction for confined and unconfined mini-tube bundles. Heat transfer coefficient was shown to decline where void fraction exceeded 0.64. Exceeding the same bulk void fraction threshold anywhere in the column was shown to predict that dryout would occur first at the top tube in the column rather than at the bottom tube. The maximum power density of a mini-tube bundle was obtained around  $30 \text{ MW m}^{-3}$ . Bubble nucleator devices were shown to be effective at mitigating against bundle effect by activating dormant nucleation sites.

These novel discoveries enable stator end windings geometry and packaging to be optimised to maximise pool boiling heat transfer, in turn allowing the design of machines with higher power density and efficiency and cooled by a system that has the potential for entirely passive operation.

As the scope and time available on any project is limited, there remain several avenues of research to explore. In pool boiling fundamentals, evaluation of slip ratio in buoyancy-driven bubble plumes will improve the void fraction model accuracy. Investigating vertical and horizontal confinements in pool boiling where  $0 < Bo < 1$  will help identify the mechanisms by which heat transfer deterioration occurs towards the lower part of this confinement envelope. In electrical machines, consideration of flow boiling around or within the stator windings reduces the thermal path and allows an essentially flat axial temperature gradient, potentially affording significant efficiency and power density gains. Finally, further systems integration work is required to consider also the condensation and recirculation aspects of the system, including evaluation of whether an entirely passive cooling system is feasible.

In summary, this research has contributed novel insights into pool boiling fundamentals, applied these to the pool boiling of electrical systems, and identified possible future research directions.

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## Appendices

### A. Thermal Analysis of Radial Conduction Path

This Appendix shows the full radial conduction path working. The radial thermal path through the stator slot and casing is shown in Figure 4. The heat equation with internal heat generation for the  $x$  axis only is

$$\frac{\partial^2 T}{\partial x^2} + \frac{\dot{q}}{k} = 0. \quad (1.7)$$

Integrating once:

$$\frac{\partial T}{\partial x} = -\frac{\dot{q}}{k}x + C_1. \quad (A.1)$$

Integrating again for the general solution:

$$T(x) = -\frac{\dot{q}}{2k}x^2 + C_1x + C_2. \quad (A.2)$$

To obtain the particular solution for turn  $i = 1$ , boundary conditions are applied. The first boundary condition is that the maximum temperature of the stator windings occurs on the bottom surface of the bottom winding:

$$T(x=0) = T_{\max}. \quad (A.3)$$

Substituting this into (A.2) gives

$$C_2 = T_{\max}. \quad (A.4)$$

The second boundary condition is that the boundary at  $x = 0$  is isothermal because heat transfer is entirely in the positive  $x$  direction:

$$\left. \frac{\partial T}{\partial x} \right|_{x=0} = 0. \quad (A.5)$$

Substituting this into (A.1) gives

$$C_1 = 0. \quad (\text{A.6})$$

Substituting the coefficients  $C_1$  and  $C_2$  from (A.6) and (A.4) into (A.2) provides the particular solution:

$$T(x) = -\frac{\dot{q}}{2k} x^2 + T_{\max}. \quad (\text{A.7})$$

If we are interested simply in the temperature at  $\Delta x$ , which is the temperature at the top face of the conductor  $T_{1,con}$ , we substitute this into the particular solution:

$$T_{1,con} = -\frac{\dot{q}}{2k_{con}} \Delta x_{con}^2 + T_{\max}. \quad (\text{A.8})$$

Substituting this into the expression for internal heat generation from (1.5), and rearranging to give temperature difference:

$$\Delta T_{1,con} = \frac{Q}{2k_{con}A} \Delta x_{con}. \quad (\text{A.9})$$

The temperature difference through the insulation layer is simply

$$\Delta T_{1,ins} = \frac{Q}{k_{ins}A} \Delta x_{ins}. \quad (\text{A.10})$$

Therefore the temperature difference across turn  $i = 1$ , comprising the conductor and the insulation, is

$$\Delta T_1 = \Delta T_{1,con} + \Delta T_{1,ins}; \quad (\text{A.11})$$

$$\Delta T_1 = \frac{Q}{A} \left( \frac{\Delta x_{Cu}}{2k_{Cu}} + \frac{\Delta x_{ins}}{k_{ins}} \right). \quad (\text{A.12})$$

Returning to the general solution, eqn. (A.2). For windings layer 2, the first boundary condition is that the temperature at the bottom of the layer ( $x = 0$ ) is  $T_1$ ; returning

coefficient  $C_2 = T_1$ . The second boundary condition is that the heat flux at  $x = 0$  is the same as the heat flux through the insulation layer:

$$-k \frac{\partial T}{\partial x} \Big|_{x=0} = \frac{Q}{A}. \quad (\text{A.13})$$

Substitution into (A.1) returns

$$C_1 = -\frac{Q}{kA}. \quad (\text{A.14})$$

Hence the particular solution for the temperature difference across conductor  $i = 2$  is

$$\Delta T_{2,con} = \frac{\dot{q}}{2k_{con}} \Delta x_{con}^2 \frac{Q}{kA} \Delta x_{con}. \quad (\text{A.15})$$

Including the  $i = 2$  insulation layer:

$$\Delta T_2 = \Delta T_{2,con} + \Delta T_{2,ins}; \quad (\text{A.16})$$

$$\Delta T_2 = \frac{Q}{A} \left( 3 \frac{\Delta x_{con}}{2k_{con}} + 2 \frac{\Delta x_{ins}}{k_{ins}} \right). \quad (\text{A.17})$$

The temperature difference from the bottom of turn  $i = 1$  to the top of turn  $i = 2$  is thus

$$\Delta T_{1 \rightarrow 2} = \Delta T_1 + \Delta T_2; \quad (\text{A.18})$$

$$\Delta T_{1 \rightarrow 2} = \frac{Q}{A} \left( 4 \frac{\Delta x_{con}}{2k_{con}} + 3 \frac{\Delta x_{ins}}{k_{ins}} \right). \quad (\text{A.19})$$

By continuing this evaluation for more turns, it can be shown that the temperature profile through a stator slot is

$$\Delta T_{1 \rightarrow i} = \frac{Q}{A} \left( i^2 \frac{\Delta x_{con}}{2k_{con}} + i \left( \frac{i+1}{2} \right) \frac{\Delta x_{ins}}{k_{ins}} \right). \quad (1.8)$$

Depending on the design of the stator windings, the top turn can comprise a half thickness of insulation rather than the full thickness between the other turns. Then, the temperature difference across the windings to the top turn  $N$  is

$$\Delta T_{1 \rightarrow N} = \frac{N^2}{2} \cdot \frac{Q}{A} \left[ \frac{\Delta x_{con}}{k_{con}} + \frac{\Delta x_{ins}}{k_{ins}} \right]. \quad (1.9)$$

## B. Thermal Analysis of Axial Conduction Path

This Appendix shows the full axial conduction path working. Assuming heat transfer only in the machine axial axis  $z$ , the heat equation is identical in form to eqn. (1.7):

$$\frac{\partial^2 T}{\partial z^2} + \frac{\dot{q}}{k} = 0. \quad (1.13)$$

Integrating once:

$$\frac{\partial T}{\partial z} = -\frac{\dot{q}}{k} z + C_1. \quad (B.1)$$

Integrating again for the general solution:

$$T(z) = -\frac{\dot{q}}{2k} z^2 + C_1 z + C_2. \quad (B.2)$$

If  $L$  is here defined as the half-length of the stator, as shown in Figure 6, the boundary conditions are:

$$T(-L) = T_{w,1}, \text{ and} \quad (B.3)$$

$$T(L) = T_{w,2}. \quad (B.4)$$

where  $T_{w,1}$  and  $T_{w,2}$  are temperatures, at either end of the stator, at the point at which the windings enter the slot. Substituting the boundary conditions into eqn. (B.2), and solving, returns the coefficients

$$C_1 = \frac{T_{w,2} - T_{w,1}}{2L}, \text{ and} \quad (\text{B.5})$$

$$C_2 = \frac{\dot{q}}{k} L^2 + \frac{T_{w,1} + T_{w,2}}{2}. \quad (\text{B.6})$$

The particular solution describing the temperature variation along the axial length of the stator is therefore

$$T(z) = -\frac{\dot{q}}{2k} z^2 + \frac{T_{w,2} - T_{w,1}}{2L} z + \frac{\dot{q}}{k} L^2 + \frac{T_{w,1} + T_{w,2}}{2}. \quad (\text{B.7})$$

The maximum temperature exists where

$$\frac{\partial T}{\partial z} = 0. \quad (\text{B.8})$$

By substituting eqns. (B.5) and (B.8) into (B.7), the location of the maximum temperature is obtained as

$$z_{\max} = \frac{2\dot{q}L}{k(T_{w,2} - T_{w,1})}. \quad (\text{B.9})$$

Substitution of this location into the particular solution (B.7), and algebraically simplifying, provides the maximum temperature in a stator:

$$T_{\max} = \frac{\dot{q}}{k} + \frac{L^2 \dot{q}}{k} - \frac{2L^2 \dot{q}^3}{k^3 (T_{w,1} - T_{w,2})^2} + \frac{T_{w,1} + T_{w,2}}{2}. \quad (\text{B.10})$$

If identical cooling strategies are employed on both end windings, the temperature variation will be symmetrical about the half-length of the axis, and the expressions are simpler. The particular solution can be simplified by invoking the boundary condition

$$\left. \frac{\partial T}{\partial z} \right|_{z=0} = 0, \quad (\text{B.11})$$

which specifies that no thermal gradient exists at the axial midpoint of the stator. The boundary condition described in (B.4) remains. Substituting into (B.1) and then into (B.2) gives

$$C_1 = 0, \text{ and} \quad (\text{B.12})$$

$$C_2 = \frac{\dot{q}}{k} L^2 - T_{w,2}, \quad (\text{B.13})$$

with which the particular solution for the temperature variation along a symmetrically cooled stator winding may be assembled:

$$T(z) = \frac{\dot{q}}{k} (L^2 - z^2) + T_{w,2}. \quad (\text{B.14})$$

By setting  $z = 0$ , the expression for the maximum temperature is obtained:

$$T_{\max} = \frac{\dot{q}}{k} L^2 + T_{w,2}. \quad (\text{B.15})$$

The effect of the heat transfer coefficient associated with the end windings cooling method can be incorporated into the analysis. Convective heat flux is

$$q'' = h(T_w - T_\infty), \quad (\text{B.16})$$

where  $T_\infty$  is the bulk temperature of the cooling medium. However, the conduction and convection areas are not identical because end windings connect from one slot to the next, and so the additional heat transfer area of the end windings is significantly larger than the conduction area. Ergo the conductor cross sectional area  $A$  and the end windings heat transfer area  $A_{\text{end}}$  must be included into the equations. Thus, at  $z = L$ , the heat due to conduction through the solid equals the convective heat from the solid to the cooling medium:

$$-kA \frac{\partial T}{\partial z} \bigg|_{z=L} = hA_{\text{end}} (T_w - T_\infty). \quad (\text{B.17})$$

Combining (B.16) and (B.12):

$$-k \frac{\partial T}{\partial z} \bigg|_{z=L} = -k \frac{\dot{q}}{k} L, \text{ which simplifies to}$$

$$-k \left. \frac{\partial T}{\partial z} \right|_{z=L} = \dot{q}L. \quad (\text{B.18})$$

Invoking (B.17) and rearranging gives

$$T_w = T_\infty + \frac{\dot{q}LA}{hA_{\text{end}}}. \quad (\text{B.19})$$

Substitution of this into (B.14) gives the expression for temperature variation along a symmetrically cooled stator turn with end windings heat transfer coefficient  $h$ :

$$T(z) = \frac{\dot{q}}{k} \left( L^2 - z^2 \right) + T_\infty + \frac{\dot{q}LA}{hA_{\text{end}}}, \quad (1.16)$$

and the maximum temperature expression from (B.15) similarly becomes

$$T_{\text{max}} = \dot{q}L \left( \frac{L}{k} + \frac{A}{hA_{\text{end}}} \right) + T_\infty. \quad (1.17)$$

## C. Uncertainty Analysis

### I. Planar Surfaces

For Chapters 3 to 5, uncertainty analysis was performed according to the Kline and McClintock (1953) method. The model suggests that a measured uncertainty has an associated probability (e.g. the probability that a pressure transducer measures within 1 kPa is 95%), which is deduced from experimental observation. This method uses partial derivatives to give the total uncertainty of a calculated value when the inputs to the calculation, and their uncertainties, are known. If the calculated value  $R$  is a function of independent variables  $x_1, x_2 \dots x_n$ , and  $w_R$  is the total uncertainty where  $w_1, w_2 \dots w_n$  are individual uncertainties, then,

$$w_R = \left( \sum_{i=1}^n \left( \frac{\partial R}{\partial x_i} w_i \right)^2 \right)^{1/2} \quad (\text{C.1})$$

which holds while: (1) the measurements are made independently, (2) repeated observations of the measurements would display Gaussian distribution, and (3) the uncertainties are expressed with the same probability (Moffat, 1988). Evidently, this root-sum-square method means that the total uncertainty can be dominated by one or a few large individual uncertainties.

Heat flux  $q''$  in this experiment was obtained from Fourier's law

$$q'' = k \nabla T \quad (C.2)$$

With reference to Figure 10, eqn. (C.2) becomes

$$q'' = -k \left( \frac{3(z_2 T_2 + z_3 T_3 + z_4 T_4) - (z_2 + z_3 + z_4)(T_2 + T_3 + T_4)}{3(z_2^2 + z_3^2 + z_4^2) - (z_2 + z_3 + z_4)^2} \right) \quad (C.3)$$

where  $T_i$  is the thermocouple temperature of the  $i$ -th thermocouple row from top, and  $z_i$  is the distance of row  $i$  in the  $z$ -axis from the top of the copper block. Partial differentiation with respect to temperature and dimension provides the uncertainty components of these parameters:

$$\frac{\partial q''}{\partial T_2} = \frac{k(z_3 + z_4 - 2z_2)}{3z_2^2 + 3z_3^2 + 3z_4^2 - (z_2 + z_3 + z_4)^2} \quad (C.4)$$

$$\frac{\partial q''}{\partial T_3} = \frac{k(z_2 + z_4 - 2z_3)}{3z_2^2 + 3z_3^2 + 3z_4^2 - (z_2 + z_3 + z_4)^2} \quad (C.5)$$

$$\frac{\partial q''}{\partial T_4} = \frac{k(z_2 + z_3 - 2z_4)}{3z_2^2 + 3z_3^2 + 3z_4^2 - (z_2 + z_3 + z_4)^2} \quad (C.6)$$

$$\begin{aligned} \frac{\partial q''}{\partial z_2} = & \frac{k(2z_3 + 2z_4 - 4z_2)(3T_2 z_2 + 3T_3 z_3 + 3T_4 z_4 - (T_2 + T_3 + T_4)(z_2 + z_3 + z_4))}{2} \\ & - \frac{k(T_3 + T_4 - 2T_2)}{3z_2^2 + 3z_3^2 + 3z_4^2 - (z_2 + z_3 + z_4)^2} \end{aligned} \quad (C.7)$$

$$\frac{\partial q''}{\partial z_3} = \frac{k(2z_2 + 2z_4 - 4z_3)(3T_2z_2 + 3T_3z_3 + 3T_4z_4 - (T_2 + T_3 + T_4)(z_2 + z_3 + z_4))}{2} - \frac{k(T_2 + T_4 - 2T_3)}{3z_2^2 + 3z_3^2 + 3z_4^2 - (z_2 + z_3 + z_4)^2} \quad (C.8)$$

$$\frac{\partial q''}{\partial z_4} = \frac{k(2z_2 + 2z_3 - 4z_4)(3T_2z_2 + 3T_3z_3 + 3T_4z_4 - (T_2 + T_3 + T_4)(z_2 + z_3 + z_4))}{2} - \frac{k(T_2 + T_3 - 2T_4)}{3z_2^2 + 3z_3^2 + 3z_4^2 - (z_2 + z_3 + z_4)^2} \quad (C.9)$$

These area in turn substituted into Eqn. (C.1) to give

$$w_{q''} = \left( \left( \frac{\partial q''}{\partial T_2} \right)^2 + \left( \frac{\partial q''}{\partial T_3} \right)^2 + \left( \frac{\partial q''}{\partial T_4} \right)^2 + \left( \frac{\partial q''}{\partial z_2} \right)^2 + \left( \frac{\partial q''}{\partial z_3} \right)^2 + \left( \frac{\partial q''}{\partial z_4} \right)^2 \right)^{1/2} \quad (C.10)$$

A numerical solution to each partial differential in eqns. (C.4) to (C.9) is calculated with substitution of the experimental data. These solutions are then substituted into Eqn. (C.10) to return the uncertainties reported in Table 6. The uncertainty of fluid properties was neglected for the experiments in Chapters 3 to 5.

## II. Tube Columns

Kline and McClintock uncertainty analysis was again performed for the experimental setup presented in Chapters 6 and 7. In this experiment, the heat transfer coefficient is calculated as

$$h_{NB} = \frac{I^2 R}{\pi d_o L (T_w - T_{sat})} \quad (C.11)$$

Current measurement accuracy  $a_I = 0.3\%$  and resistance uncertainty  $w_R = \pm 0.0037 \, \Omega$ . The thermistors are calibrated to  $w_{T_w} = \pm 0.1 \, ^\circ\text{C}$ ; uncertainty of the tube diameter was negligible and tube length uncertainty  $w_L = \pm 0.1 \, \text{mm}$ . The effect of  $\pm 0.5 \, \text{kPa}$  pressure

deviations from 1 bar on the fluid saturation temperature was noted from Novec 649 PVT data to introduce saturation temperature uncertainty  $w_{T_{sat}} = \pm 0.25$  °C. The partial derivatives of eqn. (C.11) are

$$\frac{\partial h_{NB}}{\partial I} = \frac{2IR}{\pi d_o L (T_w - T_{sat})}, \quad (C.12)$$

$$\frac{\partial h_{NB}}{\partial R} = \frac{I^2}{\pi d_o L (T_w - T_{sat})}, \quad (C.13)$$

$$\frac{\partial h_{NB}}{\partial L} = -\frac{I^2 R}{\pi d_o L^2 (T_w - T_{sat})}, \quad (C.14)$$

$$\frac{\partial h_{NB}}{\partial T_w} = \frac{I^2 R}{\pi d_o L (T_w - T_{sat})^2}, \text{ and} \quad (C.15)$$

$$\frac{\partial h_{NB}}{\partial T_{sat}} = -\frac{I^2 R}{\pi d_o L (T_w - T_{sat})^2}. \quad (C.16)$$

These are substituted into eqn. (C.1) to give the heat transfer coefficient uncertainties  $w_{h_{NB}}$  reported in the papers in Chapters 6 and 7.

## D. Tube Heating Uniformity

Prior to designing the experimental tube setup used in this Chapter, preliminary analysis of axial temperature uniformity was performed. The temperature profile along the axial dimension  $x$  of the tube is described by

$$\frac{\partial^2 T}{\partial x^2} - \frac{h_{NB} P}{k A_{CS}} (T_w - T_{sat}) = -\frac{\dot{q}}{k} \quad (D.1)$$

where rate of internal heat generation  $\dot{q} = Q/V$ . Defining

$$m = \sqrt{\frac{hP}{kA_{CS}}} , \quad (D.2)$$

$$\phi = \frac{\dot{q}}{k}, \text{ and} \quad (D.3)$$

$$T_e = T_w - T_{sat} . \quad (D.4)$$

Substituting into (D.1):

$$\frac{\partial^2 T_e}{\partial x^2} - m^2 T_e = -\phi \quad (D.5)$$

From the characteristic equation

$$r^2 - m^2 = 0, \text{ thus} \quad (D.6)$$

$$r = \pm m . \quad (D.7)$$

The general solution to the corresponding homogenous equation is

$$y(x) = c_1 e^{mx} + c_2 e^{-mx} \quad (D.8)$$

The particular solution  $y_p$  is of the form:

$$y_p = c_3 \text{ i.e. a constant.} \quad (D.9)$$

$$\frac{\partial y_p}{\partial x} = 0 \text{ and } \frac{\partial^2 y_p}{\partial x^2} = 0 \quad (\text{D.10})$$

Multiplying (D.5) by the guessed particular solution derivatives gives the particular solution:

$$0 - m^2 c_3 = -\phi . \text{ Rearranging gives} \quad (\text{D.11})$$

$$c_3 = \frac{\phi}{m^2} \quad (\text{D.12})$$

The general solution of the non-homogenous equation is the general solution of the homogenous equation (D.8) plus the particular solution of the non-homogenous equation (D.12):

$$T_e(x) = c_1 e^{mx} + c_2 e^{-mx} + \frac{\phi}{m^2} . \quad (\text{D.13})$$

The boundary conditions are that the perforated plates holding the tubes in position at  $x = 0$  are at saturation temperature, and that the temperature gradient at the midpoint  $L_{1/2}$  of the tube is 0; thus:

$$T_e(x=0) = 0 \text{ and} \quad (\text{D.14})$$

$$\left. \frac{\partial T_e}{\partial x} \right|_{x=L_{1/2}} = 0 . \quad (\text{D.15})$$

These are used to solve for constants  $c_1$  and  $c_2$ . Differentiating eqn. (D.13) gives

$$\frac{\partial T_e}{\partial x} = mc_1 e^{mx} - mc_2 e^{-mx} . \quad (\text{D.16})$$

Applying the boundary condition (D.15) into (D.16):

$$\left. \frac{\partial T_e}{\partial x} \right|_{x=L_{1/2}} = mc_1 e^{mL_{1/2}} - mc_2 e^{-mL_{1/2}} = 0 ;$$

$$m(c_1 e^{mL_{1/2}} - c_2 e^{-mL_{1/2}}) = 0.$$

From (D.2),  $m \neq 0$ , thus

$$c_1 e^{mL_{1/2}} - c_2 e^{-mL_{1/2}} = 0. \text{ Rearranging gives}$$

$$c_1 = c_2 e^{-2mL_{1/2}}, \text{ or} \quad (D.17)$$

$$c_1 = c_2 (\cosh 2mL_{1/2} - \sinh 2mL_{1/2}) \quad (D.18)$$

Next, applying the boundary condition (D.14) into eqn. (D.13):

$$T_e(x=0) = c_1 e^{m(0)} + c_2 e^{-m(0)} + \frac{\phi}{m^2} = 0. \quad (D.19)$$

$$e^0 = 1, \text{ therefore:}$$

$$c_1 + c_2 + \frac{\phi}{m^2} = 0. \quad (D.20)$$

Substituting  $c_1$  from eqn. (D.18) into eqn. (D.20), rearranging, and simplifying gives:

$$c_2 = \frac{-\phi}{m^2 (1 + e^{-2mL_{1/2}})}, \text{ or} \quad (D.21)$$

$$c_2 = \frac{-\phi}{m^2 (1 + \cosh 2mL_{1/2} - \sinh 2mL_{1/2})}. \quad (D.22)$$

Substitution of  $c_2$  from eqn. (D.21) into eqn. (D.20) and simplifying gives

$$c_1 = \frac{-\phi}{m^2} e^{-2mL_{1/2}}, \text{ or} \quad (D.23)$$

$$c_1 = \frac{-\phi}{m^2} (\cosh 2mL_{1/2} - \sinh 2mL_{1/2}). \quad (D.24)$$

The final expression for the variation in tube excess temperature with dimension  $x$  can now be obtained. The coefficients  $c_1$  and  $c_2$  from eqns. (D.23) and (D.21) respectively are substituted into (D.13) thus:

$$T_e(x) = -\frac{\phi}{m^2} e^{-2mL_{1/2}} e^{mx} - \frac{\phi}{m^2} (1 + e^{2mL_{1/2}}) e^{-mx} + \frac{\phi}{m^2}. \quad (\text{D.25})$$

Simplifying and applying trigonometric identities reduces this to the final expression,

$$T_e(x) = \frac{\phi}{m^2} (\cosh mx - \sinh mx - 1). \quad (\text{D.26})$$

To evaluate whether end effects would be significant in the tube bundle experimental setup, this expression was applied to three representative cases corresponding to low ( $q'' = 20 \text{ kW m}^{-2}$ ,  $h_{NB} = 1.5 \text{ kW m}^{-2} \text{ K}^{-1}$ ), medium ( $q'' = 40 \text{ kW m}^{-2}$ ,  $h_{NB} = 2.85 \text{ kW m}^{-2} \text{ K}^{-1}$ ), and high ( $q'' = 80 \text{ kW m}^{-2}$ ,  $h_{NB} = 5 \text{ kW m}^{-2} \text{ K}^{-1}$ ) heat flux conditions; and tube parameters  $k = 15 \text{ W m}^{-1} \text{ K}^{-1}$ ,  $d_o = 2.55 \text{ mm}$ ,  $d_i = 2.15 \text{ mm}$ ,  $L = 100 \text{ mm}$ . The resultant temperature profiles are graphed in Figure 53.

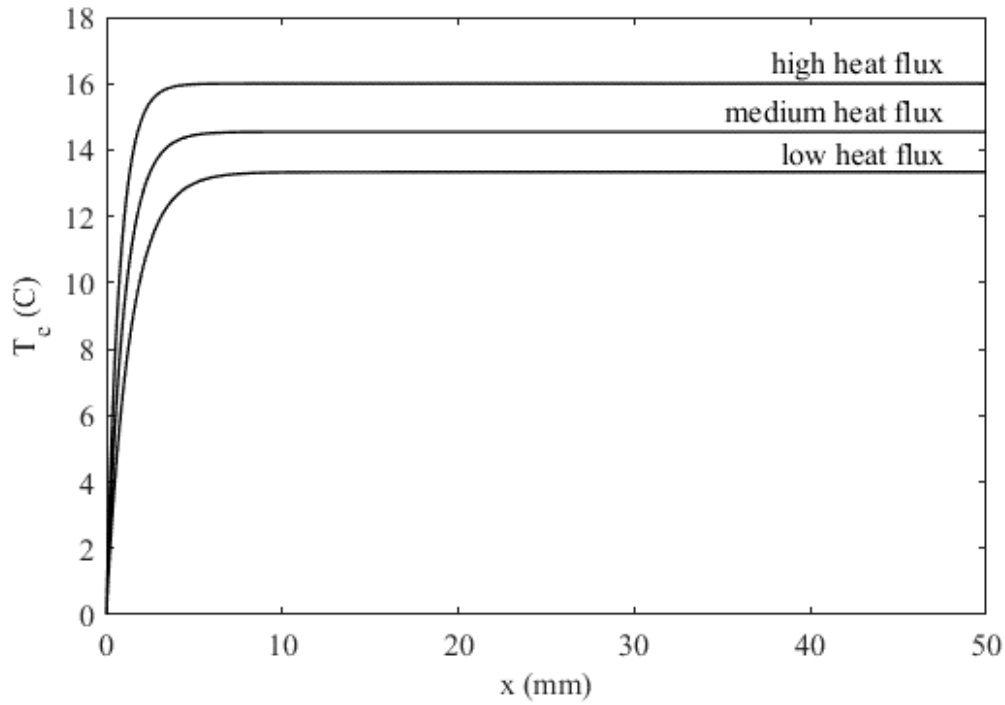


Figure 53: A graph to show the axial excess temperature profile of a Joule heated 304 SS tube with parameters representative of experimental runs at three heat flux levels.

End effects are shown to be less pronounced with increasing heat flux as more heat is produced to counteract the effect of the unheated tube end at  $x = 0$ . However, when applied to a real-world configuration shortcomings of the analytical model must be

acknowledged. In particular, the assumes a uniform heat transfer coefficient along the tube length; in reality, the local boiling heat transfer coefficient is dependent on the local surface temperature. Additionally, the experimental setup used in Chapters 6 and 7 involve a short 3 mm section of tube inserted into an acrylic spacing plate, and banana plugs of 4 mm length inserted into the tube, at either end. This introduces axial nonuniformities of heat transfer coefficient, thermal conductivity, and electrical resistivity. The analytical derivation to include these nonuniformities with respect to  $x$  is very complex and the development of a numerical model to capture their effects would be simpler. Nonetheless, the analytical solution presented shows that end effects are small and the temperature along most of the tube is constant for the representative heat fluxes used.