ON THE THERMAL AND FLUIDIC CHARACTERISTICS
OF STEAM CONDENSATION IN AN AIR-COOLED
CONDENSER

Ph.D THESIS

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DECLARATION

The substance of this thesis is the original work of the author, and due reference and acknowledgement has been made, where necessary, to the work of others. No part of this thesis has already been accepted for any degree, and it is not being currently submitted in candidature for any other degree.

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ABSTRACT

With anthropogenic climate change being one of the dominant socio-economic topics of the last decade, renewable energy has benefited from policies and legislation limiting carbon emissions. The technologies to exploit renewable energy sources have experienced significant growth and development as a result. One of the foremost technologies is Concentrated Solar Power (CSP), which has the potential to provide 25% of the world’s electricity by 2050. Successful, wide-scale deployment of CSP, however, will be dependent on the development of enhanced air-cooling strategies for use in the plant’s Rankine cycle.

This thesis is focused on the condensate-side performance of a modular air-cooled condenser for use in CSP plants. The overarching objective is to enhance understanding of condensing flows of steam in air-cooled condensers (ACCs) at typical Rankine cycle operating conditions. This was principally achieved through an experimental programme, consisting of thermal and hydrodynamic measurements, which was predominantly carried-out on circular tube condensers. The experimental programme was divided into two main streams - experimentation relating to full-scale multi-row condensers, and experimentation relating to a reduced-scale equivalent condenser. In both cases, measurements were generated by investigating parameters related to ACCs, such as steam/condensate mass flow rate, air mass flow rate, and condenser inclination angle.

Throughout an experimental programme that evolved in response to various condensate-side phenomena encountered, a test facility capable of investigating the condensate-side characteristics of a full-scale condenser at realistic Rankine-cycle conditions was developed. The hydrodynamic characteristics were quantified by the condensing pressure loss which, for the experimental conditions examined in this study, was found to be quite small, in the range of 120 Pa - 280 Pa over a vapour Reynolds number range of 1890 - 5150. It was shown that such relatively small magnitudes were due to momentum recovery offsetting the frictional losses in the flow - a phenomenon which appears unique to condensing flows. However, a parametric investigation concluded that this will not always be the case, and that a threshold point around \( D = 0.02 \) m and \( L_t = 4 \) m exists, after which the frictional losses tend to exceed the momentum recovery. Thermal characteristics were expressed by the condensate-side thermal resistance, which was shown to vary in its contribution to the overall thermal resistance with vapour Reynolds number. The average contribution was quantified as 26% at \( Re_v = 2280 \) to 13% at \( Re_v = 4420 \).

The reduced-scale condenser allowed for a more robust investigation into the thermal and fluidic mechanisms, which were not possible on the full-scale. Through a novel, non-invasive measurement technique, the predominant two-phase flow regimes were inferred. In general, it was found that annular flow exists nearest the tube inlet, with the flow deviating to stratified-wavy as the flow progresses through the tube. Local heat transfer measurements were related to this flow topology, in that large condensing Nusselt numbers were measured at the tube inlet, progressively decreasing towards the outlet, and ultimately tending to converge as the tube exit was approached. A multi-dimensional element to the heat transfer was observed as the Nusselt number was measured around the inner tube circumference. It was seen that the Nusselt number decreased from the top to the bottom of the tube - suggesting a deterrent to heat transfer, in the form of a condensate pool, resides towards the bottom.

Providing context to the overall investigation in this thesis is a thermodynamic model, which incorporates measured results to highlight limitations in current modelling approaches in the literature. It was seen that neglecting to account for the condensate-side thermal resistance in modelling approaches can lead to very different results, in terms of net plant output.
PUBLICATIONS

Some of the findings reported in this thesis have been published in the following articles.

PEER-REVIEWED JOURNAL ARTICLES


CONFERENCE PROCEEDINGS

To my parents, James and Geraldine, for getting me to where I am today.
The easiest part of this thesis is acknowledging those who contributed to it.

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# CONTENTS

1 INTRODUCTION ............................. 1
  1.1 Motivation for research ................. 2
  1.2 Application of research ................ 4
    1.2.1 Modular Air-Cooled Condenser ........ 6
  1.3 Research objectives .................... 9
  1.4 Thesis structure ........................ 10

2 CONDENSATION LITERATURE REVIEW AND THEORY 12
  2.1 External Film Condensation ............... 12
  2.2 Internal Convective Condensation ......... 15
    2.2.1 Two-phase flow overview ............... 16
    2.2.2 Two-phase flow regimes ................ 17
    2.2.3 Gravity-driven condensation .......... 21
    2.2.4 Shear-driven condensation ............. 25
    2.2.5 Summary ................................ 28
  2.3 Two-Phase Pressure Drop ................ 29
    2.3.1 General two-phase pressure drop model 29
    2.3.2 Two-phase frictional pressure drop empirical methods . 32
    2.3.3 Calculation methodology ............... 35
    2.3.4 Summary ................................ 36
  2.4 Summary .................................. 37

3 HEAT EXCHANGER THEORY .................... 39
  3.1 Aerodynamic and thermal heat exchanger theory 39
    3.1.1 Axial fan theory ...................... 39
    3.1.2 Heat exchanger aerodynamic theory .... 41
    3.1.3 Heat exchanger thermal theory ......... 44
  3.2 Heat exchanger modelling ................ 47
  3.3 Thermodynamic modelling ................ 49
    3.3.1 Modelling methodology ............... 51
    3.3.2 Semi-empirical modelling methodology 52
  3.4 Summary .................................. 53

4 EXPERIMENTATION - FULL-SCALE MACC ...... 54
  4.1 Overview of pre-existing test facility .. 54
  4.2 Experimental set-up for condensate characterisation 58
    4.2.1 Issues: steam-side phenomena ........ 58
    4.2.2 Modifications: implementation of dephlegmator 60
    4.2.3 Mass flow rate measurement .......... 64
  4.3 Parametric analysis ..................... 66
  4.4 Experimental arrangement ............... 68
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Figure 1.1</td>
<td>Modes of condensation on a horizontal flat plate</td>
<td>2</td>
</tr>
<tr>
<td>Figure 1.2</td>
<td>Example of a typical conventional A-frame ACC cell. Source of image is [1]</td>
<td>5</td>
</tr>
<tr>
<td>Figure 1.3</td>
<td>MACC concept in a range of configurations</td>
<td>7</td>
</tr>
<tr>
<td>Figure 1.4</td>
<td>Illustration of the variation in gross power, fan power consumption, and net power with fan rotational speed. Note the existence of an optimum operating point</td>
<td>8</td>
</tr>
<tr>
<td>Figure 2.1</td>
<td>Nusselt condensation on vertical flat plate</td>
<td>13</td>
</tr>
<tr>
<td>Figure 2.2</td>
<td>Idealised liquid-vapour two-phase flow in a circular tube</td>
<td>15</td>
</tr>
<tr>
<td>Figure 2.3</td>
<td>Idealised model of two-phase liquid-vapour flow in an inclined tube</td>
<td>16</td>
</tr>
<tr>
<td>Figure 2.4</td>
<td>Taitel and Dukler flow regime map for horizontal gas-liquid flow. Map is re-created from [2]</td>
<td>18</td>
</tr>
<tr>
<td>Figure 2.5</td>
<td>El Hajal et al. flow regime map for horizontal condensing liquid-vapour flow. Map is re-created from [3]</td>
<td>20</td>
</tr>
<tr>
<td>Figure 2.6</td>
<td>Illustration of gravity-driven stratified condensation from tube end-view</td>
<td>21</td>
</tr>
<tr>
<td>Figure 2.7</td>
<td>Illustration of shear-driven annular condensation from tube end-view</td>
<td>26</td>
</tr>
<tr>
<td>Figure 2.8</td>
<td>Idealised model of momentum transport in liquid-vapour two-phase flow</td>
<td>30</td>
</tr>
<tr>
<td>Figure 3.1</td>
<td>An example of an axial fan characteristic curve</td>
<td>40</td>
</tr>
<tr>
<td>Figure 3.2</td>
<td>An example of an axial fan characteristic curve at different fan rotational speeds, where $\omega_1 &gt; \omega_2$</td>
<td>40</td>
</tr>
<tr>
<td>Figure 3.3</td>
<td>Intersection of fan curves with plate-finned heat exchanger resistance curve</td>
<td>45</td>
</tr>
<tr>
<td>Figure 3.4</td>
<td>Nusselt number solution as given by the addition of asymptotes method</td>
<td>46</td>
</tr>
<tr>
<td>Figure 3.5</td>
<td>CSP plant steam turbine performance characteristics</td>
<td>50</td>
</tr>
<tr>
<td>Figure 4.1</td>
<td>Prototype MACC module in experimental test configuration</td>
<td>55</td>
</tr>
<tr>
<td>Figure 4.2</td>
<td><em>ebm-papst HyBlade</em> axial fan details</td>
<td>55</td>
</tr>
<tr>
<td>Figure 4.3</td>
<td>MACC module with heat exchanger geometries considered in this study. Pertinent geometrical parameters are highlighted</td>
<td>57</td>
</tr>
<tr>
<td>Figure 4.4</td>
<td>Example of vacuum decay observed in MACC prior to modifications</td>
<td>59</td>
</tr>
<tr>
<td>Figure 4.5</td>
<td>IR images showing backflow and non-condensable zones in leading tube row</td>
<td>60</td>
</tr>
</tbody>
</table>
Figure 4.6  
*Busch R5 RA 0100 F* vacuum pump performance curve reproduced from manufacturer’s data sheet  
61

Figure 4.7  
Plot of steady-state pressure which was established through the installation of a dephlegmator-vacuum pump system. Y-axis is scaled in insert.  
62

Figure 4.8  
Dephlegmator-vacuum pump arrangement on MACC module  
63

Figure 4.9  
Schematic of test facility for steam-side characterisation; 1-steam boiler; 2-steam separator; 3-steam trap; 4-pressure-reducing valve; 5-inlet valve; 6-MACC; 7-dephlegmator; 8-vacuum pump; 9-condensate tank; 10-float level switches; 11-condensate pump; 12-non-return valve  
64

Figure 4.10  
Data from calibration of mass in condensate tank  
65

Figure 4.11  
Condensate tank showing the transient nature in the measured pressure data that resulted from the condensate pump on/off cycle  
66

Figure 4.12  
MACC module with location of pressure transducers highlighted  
69

Figure 4.13  
Thermistor array at MACC air inlet plane during an induced draft air flow  
70

Figure 4.14  
Thermal resistance network for single annular-finned, circular tube  
76

Figure 4.15  
Thermal resistance network for single plate-finned, rectangular tube  
77

Figure 5.1  
Final form of reduced-scale annular-finned circular tube  
82

Figure 5.2  
Overview of reduced-scale ACC final design  
83

Figure 5.3  
Schematic of reduced-scale ACC with 8412 NG axial fan highlighted  
85

Figure 5.4  
Pressure-flow characteristics for single 8412 NG fan operating at 3100 rpm  
85

Figure 5.5  
Supply voltage - fan speed calibration results  
86

Figure 5.6  
Modified fan performance test facility to measure system resistance  
88

Figure 5.7  
Measured system resistance curve for annular-finned tube  
89

Figure 5.8  
Fan-heat exchanger operating points from intersection of measured pressure drop and measured system resistance curve  
90

Figure 5.9  
Variation of air flow through the reduced-scale ACC for range of fan speeds investigated  
90

Figure 5.10  
Schematic of test facility for air-side thermal characterisation  
92

Figure 5.11  
Variation in overall air-side heat transfer coefficient with air mass flow rate  
94

Figure 5.12  
Variation in air-side heat transfer coefficient with air mass flow rate  
95
Figure 5.13 Graphical summary of the mean air-side thermal characteristics ........................................... 96
Figure 5.14 Overview of steam and condensate loop with reduced-scale ACC installed ................................. 98
Figure 5.15 Plot of vacuum decay measured with initial industrial fittings .................................................. 100
Figure 5.16 Plot of steady-state vacuum achieved with vacuum-grade fittings ............................................. 101
Figure 5.17 Experimental arrangement for single-phase vapour pressure drop testing .................................. 104
Figure 5.18 Experimental arrangement for single-phase liquid pressure drop testing .................................... 105
Figure 5.19 Location and arrangement of thermistors in instrumented tube, with angular tube rotation to alter measurement platform highlighted .............................................................. 107
Figure 5.20 Reduced-scale ACC inclination angle ....................................................................................... 108
Figure 5.21 Resistance-temperature response characteristics of thermistors ............................................... 109
Figure 5.22 Plot of thermistor temperature as a function of power dissipated ................................................ 110
Figure 5.23 Wheatstone bridge measurement circuit with thermistor $R_x$ forming one arm .......................... 111
Figure 5.24 Elemental control volume, for tube cross-section, with thermal resistance network for 1-D radial heat flow ............................................................................................................. 114
Figure 5.25 Thermal resistance network and heat flow analysis when tube is rotated .................................. 116
Figure 5.26 Local measurement grid ........................................................................................................ 117
Figure 5.27 Global analysis of reduced-scale ACC ....................................................................................... 118
Figure 5.28 Heat-energy balance from condensation measurements for range of steam/condensate flow rates investigated ........................................................................................................... 119
Figure 6.1 Variation in measured pressure as a function of fan speed for a range of condensate flow rates for the six row circular tube condenser ................................................................. 123
Figure 6.2 Variation in partial pressure of air in the four row circular tube MACC ........................................ 124
Figure 6.3 Comparison between the variation in pressure contributions with fan speed for Pump A (a) and Pump B (b) - (d), for the four row circular tube condenser ......................................................... 126
Figure 6.4 Variation in measured and predicted pressure with air mass flow rate for a range of condensate flow rates for the four row circular tube condenser ...................................................... 127
Figure 6.5 Variation in measured and predicted temperature with air mass flow rate for a range of condensate flow rates for the four row circular tube condenser ...................................................... 127
Figure 6.6 Variation in measured and predicted pressure as a function of air mass flow rate for a range of condensate flow rates for the single row rectangular tube condenser .......................... 129
Figure 6.7 Variation in measured and predicted temperature as a function of air mass flow rate for a range of condensate flow rates for the single row rectangular tube condenser .......................... 129
Figure 6.8 Measured pressure drop as a function of air mass flow rate for a range of condensate flow rates for the six row circular tube condenser .................................................. 132
Figure 6.9 Inferred inlet vapour velocity as a function of air mass flow rate for a range of condensate flow rates for the six row circular tube condenser .................................................. 132
Figure 6.10 Measured pressure drop as a function of vapour Reynolds number for the circular tube condensers at the nominal air flow rate 134
Figure 6.11 Momentum recovery, inferred from measurements, as a function of vapour Reynolds number for the circular tube condensers .................................................. 135
Figure 6.12 Frictional pressure loss, inferred from measurements, as a function of vapour Reynolds number for the circular tube condensers .................................................. 135
Figure 6.13 Constituent components of condensing two-phase pressure drop as a function of vapour Reynolds number for the circular tube condensers. The data marker style follows those presented previously; black - 6 row, grey - 4 row. ............................................. 136
Figure 6.14 Frictional pressure drop, inferred from measurements, as a function of vapour Reynolds number plotted with the pressure drop predicted from Lockhart and Martinelli [4], Friedel [5], Grönnnerud [6], and Müller-Steinhagen and Heck [7] ............................................. 137
Figure 6.15 Comparison between experimentally-derived frictional pressure drop data and predicted data from Lockhart and Martinelli [4] and Müller-Steinhagen and Heck [7] ............................................. 138
Figure 6.16 Response surface illustrating the variation in the ratio of frictional pressure drop to momentum pressure recovery as a function of tube diameter and length ............................................. 140
Figure 6.17 Response surfaces illustrating the variation in the ratio of frictional pressure drop to momentum pressure recovery as a function of steam temperature ............................................. 141
Figure 6.18 Measured pressure drop (a) and pressure ratio (b) as a function of vapour Reynolds number for the rectangular tube condenser .......................... 143
Figure 6.19 Variation in air-side thermal resistance with air mass flow rate for the four row circular tube (CT) and the single row rectangular tube (RT) condenser. Also plotted is the predicted variation from the ε-NTU model given in Chapter 3.

Figure 6.20 Variation in thermal resistances with air mass flow rate for a range of vapour Reynolds numbers for the four row circular tube condenser.

Figure 6.21 Variation in condensate-side thermal resistance with air mass flow rate for a range of vapour Reynolds numbers for the four row circular tube condenser.

Figure 6.22 Comparison between the condensate-side thermal resistance, as a function of air mass flow rate and vapour Reynolds number, at two different inclination angles on the four row condenser.

Figure 6.23 Variation in thermal resistances with air mass flow rate and vapour Reynolds number for the single row rectangular tube condenser.

Figure 6.24 Variation in condensate-side thermal resistance (a), and thermal resistance ratio (b), with air flow rate for a range of vapour Reynolds numbers for the single row rectangular tube condenser.

Figure 6.25 Condensate-side thermal resistance as a function of vapour Reynolds number, combined from the four row circular tube condenser and single row rectangular tube condenser.

Figure 7.1 Variation in steam pressure with air mass flow rate for a range of condensate mass flow rates.

Figure 7.2 Variation in steam temperature with air mass flow rate for a range of condensate mass flow rates.

Figure 7.3 Comparison between the variation in temperature as a function of air-side heat transfer coefficient achieved on the lab-scale ACC and full-scale MACC.

Figure 7.4 Measured steam temperature as a function of condensate mass flow rate for the range of air mass flow rates investigated.

Figure 7.5 Variation in isothermal heat rejection with condensate mass flow rate.

Figure 7.6 Variation in steam temperature, and inlet vapour velocity, as a function of air flow rate at a condensate flow rate of 0.65 g/s.

Figure 7.7 Flow regime map of Taitel and Dukler [2] for horizontal co-current gas-liquid flow, with representative sample of experimental data range superimposed.

Figure 7.8 Flow regime map of El Hajal et al. [3], evaluated for steam at $T_s = 70$ °C and $G = 1.9 \text{ kg/sm}^2$, in a horizontal tube of $D = 25.3$ mm.
Figure 7.9 Variation in measured temperature difference from steam core (≈ constant) to tube wall around the tube circumference at an axial plane $x/L = 0.5$, with $Re_v = 2690$. Graphic illustrates direction of tube rotation .......................... 167

Figure 7.10 Temperature difference from steam core to tube wall around the tube circumference at the axial planes investigated for $Re_v = 3025$ ................................. 169

Figure 7.11 Variation in measured temperature difference from steam core to tube wall around the tube circumference for a range of vapour Reynolds numbers ............................... 170

Figure 7.12 Measured temperature difference history over a 30 second data acquisition period for the lowest ($Re_v = 2030$) and highest ($Re_v = 4185$) flow rates investigated (a) - (e). Variation in standard deviation from the mean temperature difference as a function of rotation angle (f). All measurements were acquired at $x/L = 0.667$ .................................................. 173

Figure 7.13 Single-phase pressure drop measurements, presented in terms of friction factor, as a function of Reynolds number .......................... 176

Figure 7.14 Measured two-phase condensing pressure drop (a), and pressure gradient (b), as a function of vapour Reynolds number .......................... 177

Figure 7.15 Momentum recovery in the condensing flow (a), offsetting frictional losses (b), as a function of vapour Reynolds number .......................... 178

Figure 7.16 Inferred frictional pressure drop as a function of vapour Reynolds number compared with the prediction of Lockhart and Martinelli [8] ............................. 179

Figure 7.17 Measured pressure drop data presented in terms of Martinelli parameter ($X$) and two-phase multiplier ($\phi$), compared with the solution of Lockhart and Martinelli [8] .............................. 180

Figure 7.18 Comparison between experimentally-deduced frictional pressure drop and predicted frictional pressure drop of Lockhart and Martinelli [8] .............................. 181

Figure 7.19 Local circumferential condensing Nusselt number as a function of non-dimensional tube axial length, for a range of vapour Reynolds numbers (a) - (e). Circumferentially-averaged Nusselt number as a function of non-dimensional tube axial length (f) ............................. 183

Figure 7.20 Local circumferential Nusselt number for a range of positions near the bottom of the tube as a function of non-dimensional tube axial length, coupled with the single-phase liquid Nusselt number solution .............................. 185
List of Figures xvii

Figure 7.21 Variation in local circumferential Nusselt number with non-dimensional angular position inside the tube for a range of vapour Reynolds numbers at axial locations $x/L = 0.167$ and $x/L = 0.334$ ........................................ 187

Figure 7.22 Variation in local circumferential Nusselt number with non-dimensional angular position inside the tube for a range of vapour Reynolds numbers at axial locations $x/L = 0.5$, $x/L = 0.667$, and $x/L = 0.834$ ........................................ 188

Figure 7.23 Series of contour plots illustrating bi-directional variation of condensing Nusselt number for range of vapour Reynolds numbers. All contour plots are given with an absolute scale and original contour level ........................................ 191

Figure 7.24 Contour plots re-created from figure 7.23 with common Nusselt number scale and increased contour level ........................................ 192

Figure 7.25 Variation in global condensing Nusselt number with vapour Reynolds number compared with correlations from Shah [9], Ananiev et al. [10], Cavallini and Zecchin [11], and Dobson [12] 194

Figure 7.26 Variation in global condensing Nusselt number with vapour Reynolds number for a range of condenser inclination angles . 197

Figure 7.27 Variation in global condensing Nusselt number with vapour Reynolds number for a range of airflow Reynolds numbers . . 199

Figure 7.28 Variation in global condensing Nusselt number with airflow Reynolds number for a range of vapour Reynolds numbers . . 200

Figure 7.29 Response surfaces illustrating the variation in condensing Nusselt number for a range of vapour Reynolds numbers, airflow Reynolds numbers, and tube inclinations ............... 201

Figure 7.30 Variation in thermal resistances with airflow Reynolds number and vapour Reynolds number ........................................ 202

Figure 7.31 Variation in thermal resistance ratio as a function of vapour Reynolds number ........................................ 203

Figure 7.32 Variation in condensate-side thermal resistance with both airflow Reynolds number and tube inclination for a range of vapour Reynolds numbers ........................................ 204

Figure 7.33 Variation in thermal resistance ratio with both airflow Reynolds number and tube inclination for a range of vapour Reynolds numbers ........................................ 205

Figure 8.1 Variation in condenser performance and related thermodynamic quantities, as function of fan speed, for two condenser sizes and a range of representative ambient temperatures. Note all calculations were based on a 2 m x 2 m four-row circular tube MACC with four fans ........................................ 209
Figure 8.2 Variation in measured and predicted condenser performance, and related thermodynamic quantities, as function of fan speed over a range condenser sizes corresponding to the experimental steam flow rates investigated .......................... 212
Figure 8.3 Disparity between the net plant output, as given by the air-side approach and the experimental approach .......................... 213
Figure 8.4 Variation in condenser temperature as a function of fan speed for 200 and 300 modules, across a range of ambient temperatures .......................... 215
Figure 8.5 Variation in net plant output as a function of fan speed for 200 and 300 modules, across a range of ambient temperatures .......................... 216
Figure 8.6 Comparison between the optimum net plant output and optimum fan speed for the standard, unconstrained case and the constrained condition .......................... 217
Figure A.1 Overview of MACC steam supply line and ancillary components .......................... 241
Figure B.1 Sample thermocouple calibration curve .......................... 242

LIST OF TABLES

Table 2.1 Experimental data range for internal convective condensation in horizontal and slightly inclined tubes published to date, compared with author’s data .......................... 29
Table 2.2 Separated flow regimes constants for use in Lockhart and Martinelli model .......................... 32
Table 2.3 Comparison of thermophysical properties of various working fluids in condensation studies, where all properties are taken from Rogers and Mayhew [13] at $T_s = 40 \degree C$ and $G = 5 \text{ kg}/\text{sm}^2$ .......................... 37
Table 2.4 Dimensionless parameters utilised throughout this study .......................... 38
Table 3.1 Dimensionless pressure drop empirical correlations and analytical model .......................... 43
Table 3.2 Dimensionless heat transfer empirical correlations and analytical model .......................... 45
Table 3.3 Nominal exit conditions of ~ 20 MW and ~ 50 MW steam turbines .......................... 51
Table 4.1 Geometrical features of heat exchanger geometries examined herein .......................... 56
Table 4.2 Vacuum pump specifications .......................... 62
Table 4.3 Range of flow conditions investigated in this study .......................... 68
Table 4.4  Uncertainties associated with primary, measured variables  . . . 79
Table 4.5  Uncertainties associated with derived, calculated parameters  . 79
Table 5.1  Comparison of reduced-scale and full-scale annular-finned cir-
cular tube geometrical parameters  . . . . . . . . . . . . . . . . . . . . . 82
Table 5.2  *ebm-papst 8412 NG* fan specifications  . . . . . . . . . . . . . 84
Table 5.3  Summary of aerodynamic and thermal characteristics on the
air-side of the reduced-scale ACC  . . . . . . . . . . . . . . . . . . . . . . . 96
Table 5.4  Effect of low pressure on steam/water thermophysical proper-
ties for a mass flux of $G = 5 \text{ kg/s} m^2$  . . . . . . . . . . . . . . 102
Table 5.5  Range of flow conditions investigated in reduced-scale ACC  . 103
Table 5.6  Complete experimental test matrix for condensation heat trans-
fer measurements  . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . 113
Table 5.7  Uncertainties associated with the reduced-scale ACC primary
variables  . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . 120
Table 5.8  Uncertainties associated with the reduced-scale ACC calcu-
lated parameters  . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . 120
Table 7.1  Two-phase flow regimes identified  . . . . . . . . . . . . . . . . . . . . . 175

**NOMENCLATURE**

- $A$: area (m$^2$)
- $A_o$: free flow area (m$^2$)
- $b$: fin spacing (m)
- $C$: Chisholm constant (-)
- $c_p$: specific heat capacity (Jkg$^{-1}$K$^{-1}$)
- $D$: internal tube diameter (m)
- $d_e$: external tube diameter (m)
- $d_f$: fin diameter (m)
- $dP/dz$: pressure gradient (Pam$^{-1}$)
- $f$: friction factor (-)
- $F_i$: friction force (N)
- $Fr$: Froude number (-)
- $G$: mass flux (kgs$^{-1}$m$^{-2}$)
- $g$: gravitational acceleration (ms$^{-2}$)
- $H$: height (m)
- $h$: heat transfer coefficient (Wm$^{-2}$K$^{-1}$)
- $h_{fg}$: enthalpy of vaporization (Jkg$^{-1}$)
- $hd$: head height (m)
- $Ja$: Jakob number (-)
\( k \) thermal conductivity \((\text{Wm}^{-1}\text{K}^{-1})\)  
\( K_c, K_e \) entrance and loss co-efficients (-)  
\( L \) length (m)  
\( L_a^* \) dimensionless aerodynamic entrance length (-)  
\( L_{th}^* \) dimensionless thermal entrance length (-)  
\( \dot{m} \) mass flow rate \((\text{kgs}^{-1})\)  
\( m \) mass (g)  
\( N \) number of fans, fins, modules, tubes, etc. (-)  
\( NTU \) number of transfer units (-)  
\( Nu \) Nusselt number (-)  
\( P \) power (W)  
\( P \) pressure (Pa)  
\( P_c \) critical pressure (Pa)  
\( P_r \) reduced pressure (-)  
\( Pe \) Péclet number (-)  
\( Pr \) Prandtl number (-)  
\( \dot{Q} \) heat transfer rate (W)  
\( R \) individual gas constant \((\text{JK}^{-1}\text{mol}^{-1})\)  
\( r \) radius (m)  
\( r_h \) hydraulic radius (m)  
\( R_x \) resistance (Ω)  
\( R_{th} \) thermal resistance \((\text{KW}^{-1})\)  
\( Re \) Reynolds number (-)  
\( S_f \) fin spacing (m)  
\( St \) Stanton number (-)  
\( T \) temperature (K)  
\( t \) time (s)  
\( U \) overall heat transfer co-efficient \((\text{Wm}^{-2}\text{K}^{-1})\)  
\( u \) velocity \((\text{ms}^{-1})\)  
\( \dot{V} \) volumetric flow rate \((\text{m}^3\text{s}^{-1})\)  
\( V \) volume \((\text{m}^3)\)  
\( V_x \) voltage (volts)  
\( W \) width (m)  
\( We \) Weber number (-)  
\( X \) Martinelli parameter (-)  
\( x \) steam quality (-)  
\( X_{tu} \) Lockhart-Martinelli parameter (-)  

**ABBREVIATIONS**  
ACC Air-Cooled Condenser  
CSP Concentrated Solar Power  
DAQ Data Acquisition  
DNI Direct Normal Irradiance
HTF       Heat Transfer Fluid
IR        Infrared
MACC      Modular Air-Cooled Condenser
NCG       Non-Condensable Gas
WCC       Water-Cooled-Condenser

GREEK SYMBOLS
α         void fraction
δ         thickness (m)
ΔP        pressure difference (Pa)
ΔT        temperature difference (K)
ε         effectiveness
η         efficiency
θ         inclination (°)
μ         dynamic viscosity (kgm$^{-1}$s$^{-1}$)
ρ         density (kgm$^{-3}$)
σ         contraction ratio
τ         shear stress (Nm$^{-2}$)
ϕ         two-phase multiplier (-)
φ         rotation angle (°)
ω         fan rotational speed (rpm)

SUBSCRIPTS
∞         ambient
a         air
avg       average
c         condensate
$D_h$     hydraulic diameter
dev       developing
e         exit
f         fan
f         fin
fd        fully-developed
fr         friction
ft         fan total
gr         gross output
h         homogeneous
i         inlet
l         liquid-phase
ll        laminar-laminar
lo        liquid-only
lt        laminar-turbulent
m         measured
\begin{tabular}{ll}
\textit{mom} & momentum \\
\textit{net} & net output \\
\textit{o} & outlet \\
\textit{out} & heat exchanger air outlet \\
\textit{p} & predicted \\
\textit{rec} & recieved \\
\textit{rej} & rejection \\
\textit{s} & saturated steam \\
\textit{ST} & steam turbine \\
\textit{st} & static \\
\textit{tot} & total \\
\textit{tt} & turbulent-turbulent \\
\textit{v} & vapour-phase \\
\textit{vo} & vapour-only \\
\textit{w} & wall \\
\end{tabular}
INTRODUCTION

Liquid-vapour phase change processes, including evaporation, boiling, and condensation, are amongst the most important transport mechanisms encountered in engineering. Owing to the practically isothermal heat transfer process, condensation pervades many ubiquitous technologies, being particularly attractive in power and refrigeration cycles - from a thermodynamic efficiency perspective. Coupled to this are the high heat transfer coefficients associated with condensation, which have rendered it an increasingly attractive mechanism for the thermal control of compact devices with high heat dissipation rates. Such widespread application of condensation has resulted in the liquid-vapour phase change process assuming different forms in different configurations, and it is generally classified accordingly.

Depending on the application, the working fluid being condensed can vary from some class of refrigerant in vapour-compression cycles, to organic fluids in organic Rankine cycles, and water in traditional Rankine cycles. However, as shown in figure 1.1, irrespective of the working fluid there are two very distinct modes in which the liquid phase forms, with each one being so distinctive that it completely characterises that particular condensation process. Dropwise condensation, which can be seen in figure 1.1a, occurs when a vapour condenses to form individual liquid droplets on a surface. In such a scenario, the surface is said to be poorly-wetted, as a result of the liquid’s lack of affinity, and hydrophobicity is observed. Dropwise condensation is typically characterised by very high heat transfer coefficients, which are often one or two orders of magnitude greater than those achieved by the alternative mechanism - filmwise condensation [14, 15]. However, continuous dropwise condensation rarely occurs naturally in process equipment as the individual droplets tend to grow and coalesce with adjacent droplets to, ultimately, form a film of liquid. Continuous dropwise condensation, therefore, needs to be promoted and sustained using specialist techniques. This is typically achieved through the use of hydrophobic coatings applied to the surface or, alternatively, specialist surfaces fabricated with micro-scale structures [16, 17]. The main issue with these solutions is that they are difficult to incorporate on a large scale, or maintain in typical condensation environments. As a result, filmwise condensation, although not preferential, is by far the more commonly encountered mode of condensation in typical engineering applications.

Filmwise condensation is classified as the process whereby the conversion of vapour to liquid assumes the form of a liquid film which fully wets the surface in contact with the vapour. As seen in figure 1.1b, the surface is blanketed by the liquid film and, as the name implies, the condensation heat transfer occurs at the liquid-vapour interface of the liquid film covering this surface. As more mass is transferred from
the condensing vapour flowing parallel to the surface, the film grows in thickness in the downstream direction. This leads to the development of a temperature gradient across the film, where the film represents a thermal resistance to the heat transfer process. The rate of condensation, therefore, depends on the amount of condensate accumulated on the surface. This contrasts with dropwise condensation where a significant portion of the surface is free of liquid and is directly exposed to vapour, where there is no impedance to heat flow.

Dropwise, or filmwise, condensation generally occurs in one of two scenarios - externally or internally. External condensation refers to condensation in the absence of any well-defined geometrical boundaries. Examples include condensation on vertical plates, horizontal tubes, tube bundles, and enhanced surfaces. Shell-and-tube surface condensers are a prominent application for external condensation. Conversely, internal condensation refers to condensation inside a well-defined passage. Even though the fundamental heat transfer mechanism is almost identical to external condensation, the geometrical restrictions imposed upon internal condensation render it considerably more complex. These complexities mainly arise from the relatively strong interactions between the two phases which, usually, do not exist when a vapour condenses in an unconfined space. Confined spaces in which internal condensation usually occurs are tubes and ducts of circular, rectangular, and elliptical cross-sections. One of the main applications for internal condensation in such geometries are air-cooled condensers (ACCs), which are an intrinsic component in a wide variety of power and refrigeration cycles. This thesis is focused on the internal, filmwise condensation of steam in circular and rectangular tubes in ACCs.

1.1 motivation for research

The installation of ACCs has increased exponentially in the last 15 years [18] - mainly in fossil-fuel thermoelectric power plants. This has occurred despite a lack of informa-
motivation for research

1.1 motivation for research

The majority of research to date has tended to focus on the air-side performance, with little attention given to the consequences inside the tubes. This is understandable as, in general, the air-side resistance is the dominant, controlling thermal resistance in ACCs, with it representing more than 80% of the total resistance according to Shah and Sekulić [19]. Shah and Sekulić attribute the remaining 20% of the total resistance to the condensate-side thermal resistance, which is comprised of the condensate film and fouling film. However, this 20% figure is an approximation, which needs to be explicitly verified and established in order to satisfy the assumption that the air-side is largely dominant. There is little information in the literature regarding this. Hence, this thesis seeks to supplement current literature by experimentally quantifying the condensate-side thermal resistance over a range of parameters related to operational ACCs.

The lack of condensate-side literature extends to ACC condensation pressure losses. Studies addressing the pressure drop of two-phase, condensing flows of steam through air-cooled tube bundles are not well-documented in the literature. The potential for research has been explicitly acknowledged by Palen et al. [20], who recognised the lack of hydrodynamic data at sub-atmospheric (vacuum) conditions - at which full-scale condensers in application nominally operate at. It is known that pressure losses on the condensing-side have the potential to inhibit condenser performance and can, ultimately, curtail plant output by increasing the steam turbine backpressure. Hence, in the design and testing of condensers, excessive condensing-side pressure losses should be avoided. This thesis presents measurements which quantify the losses expected at typical ACC operational conditions, and presents a general methodology for determining such losses.

Although, traditionally, there has been extensive research carried out on condensation transport phenomena, almost none of the work reported in the literature pertains to conditions encountered in Rankine cycle-based ACCs. As will be shown in later chapters, examining condensing flows at typical ACC operating conditions leads to some interesting effects, none of which have been disclosed to date in the literature. Indeed, the author could only find two studies which examined steam condensation at somewhat similar conditions - that of Caruso et al. [21] who employed air-cooling as a means of investigating steam condensation in the presence of NCGs, and that of Berrichon et al. [22] who identified the heat transfer benefits when condensing steam at low pressures, albeit for a vertical tube configuration. In keeping with the majority of the literature, however, both of these studies quantified their measurements on a global-scale, offering little insight into the underlying mechanisms.

A novel feature of the work presented in this thesis was the characterisation of condensing flows locally, along the length, and around the circumference, of an air-
cooled circular tube. Irrespective of specific operating conditions relevant to ACCs, there are only two studies in the whole of the condensation literature, of which the author is aware, which present local convective condensation measurements. Namely, these are the studies of Rosson and Meyers [23], who examined the circumferential distribution of heat transfer in one axial plane, and Goodykoontz and Dorsch [24], who presented the axial variation of heat transfer for a fixed circumferential plane. However, neither of these studies can individually boast the measurement resolution of this thesis, which presents the simultaneous distribution of heat transfer around the circumference along the axial direction. Ultimately, this offers a unique insight into condensing flows, particularly at conditions applicable to operational ACCs. Further context on the measurements presented in this thesis is given in Chapter 2, which contains a comprehensive review of the relevant literature to-date.

1.2 APPLICATION OF RESEARCH

The Intergovernmental Panel on Climate Change (IPCC) stated in 2013 that “it is extremely likely (95–100% probability) that human influence has been the dominant cause of the observed warming since the mid-20th century”. Chief culprit amongst these anthropogenic activities is fossil-fuel combustion for power generation, which accounted for 42% of total global CO$_2$ emissions in 2014 [25]. Therefore, it is the high-level priority of many environmental lobbyists, climatologists, and policy-makers that current CO$_2$ emissions being released from fossil-fuel based thermoelectric plants are rapidly curtailed. This has provoked a renewed emphasis on alternative energy solutions, predicated on the fact that renewable energy could provide over half the world’s energy requirements by 2050 [26]. One of the technologies with the capability to exploit this potential is Concentrated Solar Power (CSP).

Similar to the overwhelming majority of electricity generating plants, CSP is based on the Rankine cycle - with the heat source being solar radiation. As the primary energy source is the direct normal irradiance (DNI), the ideal site for a CSP plant is in the “sun belt” of the Earth. The sun belt is characterised by largely arid, desert regions, exposed to high concentrations of DNI. Consequently, there is a distinct lack of natural water supplies available in such regions for cooling in the condenser. This issue is exacerbated by the extortionate cost of transporting and supplying water to a site, to the point where using water as a cooling medium is prohibitive. This has instigated the transition from wet-cooling systems to dry-cooling methodologies, which typically reduce the water requirement of a plant by approximately 99% over that required by a once-through wet-cooling system [27]. At the forefront of current dry-cooling strategies are direct ACCs, which have been proposed instead of shell-and-tube water-cooled surface condensers for use in CSP plants. The current industry standard is the A-frame ACC, an example of which is given in figure 1.2.

An ACC, such as that shown in figure 1.2, is generally comprised of a steam duct, a bank of finned tubes, an array of axial fans, and a condensate return line. In addition,
there are a number of ancillary components which are installed in conjunction with the ACC, such as a condensate pump and vacuum pump (evacuation unit). The steam duct transports the steam from the exit of the turbine to the inlet of the condenser, and forms the apex of the A-frame structure. From here the steam is distributed through the bank of finned tubes, which form the sloping sides of the structure. Air, from a number of large axial fans installed at the base of the unit, is blown through the tube bank where the steam is, consequently, condensed. The resulting condensate is collected in the condensate return line at the base of the bank of tubes. Also forming part of the overall ACC cell is a reflux (counterflow) condenser, sometimes referred to as a dephlegmator, which is responsible for displacing non-condensable gasses (NCGs) from the unit. A typical ACC cell has a footprint of 12 m x 12 m, with finned tubes 9 - 12 m in length, and an apex angle of approximately 60° [28].

The ACC represents a critical component of the thermodynamic cycle, as it operates in conjunction with the upstream steam turbine to govern the power output of a given plant. Gross power output is a function of the enthalpy drop (steam expansion) through the turbine where, in a Rankine cycle, the outlet of said turbine is generally coupled to the inlet of a condenser. Hence, it follows that the enthalpy at the turbine outlet is regulated by the enthalpy at the inlet of the condenser. However, the conditions in the condenser are dictated by the cooling medium, where there are inevitable drawbacks associated with ACCs. These drawbacks emanate from the unfavourable thermal transport properties of air, relative to water. To compensate, ACCs incorporate much larger heat exchangers than water-cooled condensers (WCCs) and, thus, require substantially higher capital investment. Current estimates for the total installed cost of an ACC are 3 - 5 times larger than those for a WCC [27, 29]. Furthermore, typical levelized power production costs for a plant with an ACC are approximately 15% higher than for plants utilising water-cooling.
However, Maulbetsch and Barker [30] and Zhai and Rubin [31] found that increasing water costs could eliminate this gap - a scenario which could very well be realised due to water shortages, environmental legislation, etc.

The increased popularity of A-frame ACCs has occurred despite an array of empirical evidence to suggest that such designs suffer from inherent design flaws. These design flaws are manifested by non-uniform air flows [32, 33, 34], steam-flow maldistribution [35], susceptibility to inclement weather conditions - particularly wind [36, 37, 38, 39, 40], and vapour backflow [41, 42, 43, 44]. The net result of these deficiencies is a reduction in condenser effectiveness and performance, ultimately, propagating through the thermodynamic cycle and culminating in plant losses. Such losses can threaten the viability of CSP plants as a competitor to fossil-fuel power generation. Such issues, however, pale in significance to the underlying issue of air as a cooling medium in hot, arid locations suitable for CSP sites. As the air dry bulb temperature is the thermodynamic limit of the heat sink, an increase in ambient temperature increases the temperature, and pressure, in the ACC, thereby reducing the gross output of the plant. In fact, the backpressure in an ACC can be approximately 2.1 times higher than that with a wet-cooling solution according to Tawney et al. [27]. The net effect of this on a CSP plant is a reduction in electrical power output of about 4 - 7% [27, 45], and an increase in cost of electricity by approximately 8 - 10% [45, 46], relative to wet-cooling.

More worryingly still is the limited ability of current ACC designs to respond to any variations in the ambient temperature. Studies by Chuang and Sue [47] and Pieve and Salvadori [48] both present the explicit degradation in ACC performance as a result of increasing ambient temperature. According to the European Union’s Strategic Energy Technology (SET) plan [49], a 25% loss in power output can occur in CSP plants on warm days due to condenser deficiencies. Such losses are mainly attributed to the large axial fans utilized in current ACCs. These fans have a very limited range of rotational speeds, normally just a low speed setting and a high speed setting. There is little in the way of sensor feedback and control to alter the fan rotational speed in response to the ambient temperature fluctuations. This lack of control can have a detrimental effect on steam turbine output, where an increase in condenser temperature, as a consequence of increased ambient temperature, cannot be mitigated against by increasing fan speed. It is imperative that any losses in CSP plant performance due to inadequate condenser designs be minimised to ensure CSP is cost-competitive with fossil-fuel power. In light of this, it is clear that there is scope and motivation for improvement in current ACC designs, if they are to become the pre-eminent cooling strategy for CSP power generation.

1.2.1 Modular Air-Cooled Condenser

In an attempt to alleviate current issues and, hence, minimise losses associated with conventional ACCs, a novel Modular Air-Cooled Condenser (MACC), is introduced
as part of the work in this thesis. As the name suggests, the MACC concept illustrated in figure 1.3 adopts a modular approach to the traditional large, fixed A-frame ACC. Not only is this modular nature advantageous from a practical perspective, as it allows for easier installation, but it is also more flexible, as the number of MACC modules can be tailored and customized to suit the specific requirements of a given plant.

![Exploded view of single MACC 8 m x 2 module](image)

(a) Exploded view of single MACC 8 m x 2 module

![Eleven MACC modules arranged to form a “street”](image)

(b) Eleven MACC modules arranged to form a “street”

![MACC modules arranged in A-frame configuration](image)

(c) MACC modules arranged in A-frame configuration

![MACC A-frame arrangement surrounded by wind wall](image)

(d) MACC A-frame arrangement surrounded by wind wall

Figure 1.3: MACC concept in a range of configurations

As shown by figure 1.3a an individual MACC module is a cross flow heat exchanger - consisting of a compact heat sink (tube bundle) coupled with an axial fan array. The close proximity of the axial fans to, and their distribution across, the tube bundle shows the MACC as an integrated solution of heat exchanger and fans, rather than separate, individual components - as is the case in A-frame ACCs. The tube bundle design can be specified from the outset to suit the desired application, whereby the geometry can range from a variety of multi-row circular tube designs, or a single-row rectangular tube design - which appears to be the current state-of-the-art. One of the key features of the MACC concept are the axial fans, which deliver the air flow. In comparison to the large (ø = 9 - 12 m), mechanically-driven, axial fans in conventional A-frame ACCs, the fans fitted on a MACC module are relatively small, with a diameter of approximately 1 m. Furthermore, these fans are electrically-
1.2 APPLICATION OF RESEARCH

Commutated. The benefits of small, compact, lightweight fans, in conjunction with electrical excitation, are manifested in the ability to vary the rotational speed of the fans instantaneously, with little lag time. This ability to continually vary fan speed is a critical feature, as varying the fan speed of an ACC leads to a variation in the air-side thermal resistance which, ultimately, results in a change in condenser temperature, and pressure. Increasing fan speed reduces condenser temperature and pressure, and vice-versa. Hence, the rotational speed of the MACC fans is continuously variable in an effort to maintain optimum condenser conditions, irrespective of ambient temperature, wind effects, steam flow rate, etc. This ability to control and adjust condenser temperature and pressure ensures maximum steam turbine output which, in turn, maintains optimum plant output. It is proposed that this defining characteristic of the MACC can minimise the parasitic losses associated with current air-cooled power plants. This characteristic is graphically summarised in figure 1.4.

![Figure 1.4: Illustration of the variation in gross power, fan power consumption, and net power with fan rotational speed. Note the existence of an optimum operating point](image-url)

Figure 1.4 illustrates that, through manipulation of MACC fan speed, the gross output and net output can, to an extent, be controlled. As an increase in fan speed is accompanied by a reduction in condenser temperature and pressure, the enthalpy drop through the upstream turbine will inevitably be larger. Hence, gross output is increased. In effect, increasing fan speed lowers the heat sink working temperature. However, the objective is not to simply infinitely increase fan speed in an effort to reduce condenser temperature and pressure because, as shown in figure 1.4, there are penalties in the form of high fan power consumption rates when operating at high rotational speeds. In fact, at excessively high fan speeds, the increase in power consumed by the fans is actually greater than, and offsets, the corresponding increase in gross power. The net effect of this is a reduction in net plant output. Hence, there exists an optimum operating point for the plant, at which the net plant output is maximised. The optimum fan speed shown in the example in figure 1.4, however,
varies with ambient air temperature and steam mass flow rate. In general, ACCs are always subjected to fluctuations in ambient temperature and steam flow rate and, hence, to continually maintain the optimum operating point, fan speed must be variable. This emphasises the importance of variable speed fans in an ACC and, whilst constantly operating at the optimum point will require precise fan speed control, it certainly offers an advantage over current state-of-the-art ACCs.

This thesis adopts the MACC to frame the work undertaken, which primarily seeks to improve understanding of condensing flows of steam, and condensate-side behaviour, for a range of operating conditions typical of those which the MACC will be expected to experience in application. This is principally achieved through thermal and hydrodynamic characterisation of a full-scale prototype MACC, and a reduced-scale equivalent. Not only does establishing the condensate-side characteristics of the MACC supplement current literature, as outlined in Section 1.1, but the characteristics are doubly important to ensure the main advantages of the MACC concept are realised in application. To achieve the optimum operating point in a potential power plant, the complete thermal performance must be established. This consists of the air-side performance, quantified in a separate study by Moore [50], and the condensate-side performance, which this thesis seeks to quantify. Ultimately, empirical relationships from these studies will be incorporated into fan speed control algorithms responsible for maintain the optimum operating point.

Underpinning the MACC concept and, thus, the work presented in this thesis are the thermodynamics of the Rankine cycle. For a potential end-user, plant output and plant efficiency are far more telling and desirable metrics than any thermal or hydrodynamic quantity. Therefore, a thermodynamic model, into which condensate-side thermal measurements can be integrated, was developed to explicitly quantify the thermodynamic performance of the MACC for a range of conditions. Several similar approaches are available in the literature for standard ACCs [51, 48, 46, 29], however all of these models neglect to account for, or dismiss, the condensate-side thermal resistance. Instead these models rely upon the assumption that the air-side thermal resistance is the only concern in ACC modelling and, hence, simply pursue an air-side only approach. It is speculated that disregarding the condensate-side resistance could lead to erroneous and misleading thermodynamic results. Ultimately, this will result in non-optimal plant operation and, therefore, inaccurate prediction of annual electricity production. This thesis investigates this hypothesis by comparing the thermodynamic plant performance based on condensate-side measurements, to that given solely by adopting an air-side only approach.

1.3 RESEARCH OBJECTIVES

The primary goal of this thesis was to enhance understanding of condensing flows of steam in ACCs in general and, in particular, at typical Rankine-cycle operating conditions. This primary objective was accomplished through the completion of a
The number of constituent objectives, each of which were based on the need to address specific shortcomings in the literature. These objectives are listed as follows:

1. Design and develop an experimental platform which can replicate typical Rankine cycle-based ACC operating conditions to examine condensing flows. This includes generation of guidelines to mitigate against air ingress, and to eliminate backflow and trapping of NCGs in tubes.

2. Examine the effects of system-level variables, such as size of vacuum pump, condenser inclination, and fan orientation, on condenser performance.

3. Establish quantitative and qualitative relationships between air-side heat transfer coefficient and absolute condenser temperature and pressure.

4. Determine the hydrodynamic losses associated with steam condensation in air-cooled tubes. Quantify the specific contributions to the losses, investigate the validity and accuracy of existing published correlations, and present a general methodology for predicting the relative magnitudes of condensation pressure drop and absolute condenser pressure.

5. Quantify the contribution of condensate-side thermal resistance to the total thermal resistance for a range of parameters, such as steam/condensate flow rate and air flow rate, related to full-scale ACCs.

6. Identify the prevailing two-phase flow regimes encountered in circular ACC tubes through inference of local temperature distribution measurements. Compare these findings with current state-of-the-art flow regime maps.

7. Enhance understanding of convective condensation through local condensation heat transfer measurements along the axial length and circumference of a circular ACC tube. Investigate such local distributions for a range of steam/condensate flow rates, air flow rates, and condenser inclination angles.

8. Develop a thermodynamic model to quantify condenser performance in terms of plant performance. Investigate the practical implications of condensate-side thermal resistance by integrating the measurements into the thermodynamic model, and comparing with theoretical approaches.

1.4 THESIS STRUCTURE

The remainder of this thesis is comprised of seven main chapters. Following this chapter is the literature review, where a comprehensive overview of existing literature relating to condensation heat transfer and pressure drop is presented. Chapter 3 proceeds the literature review, and presents aerodynamic and thermal theory relating to the air-side of ACCs. A thermodynamic model for predicting plant performance is subsequently, presented, into which the air-side theory can be integrated. The next
four chapters can be sub-divided into two main streams - those based on full-scale MACC research, and those based on the reduced-scale equivalent. In this regard, Chapters 4 and 6 present the experimental facilities and methodologies, and results, respectively, for the full-scale MACC. Likewise, Chapters 5 and 7 present the experimentation and results, respectively, for the reduced-scale ACC. The penultimate chapter (8), presents the outcomes from thermodynamic modelling efforts, and offers an element of context to the overall work presented in this thesis. Finally, the thesis is concluded in Chapter 9, accompanied by recommendations for future work. References and the appendices bookend the thesis.
CONDENSATION LITERATURE REVIEW AND THEORY

The heat transfer and fluid flow mechanisms associated with condensation are typically among the more complex transport circumstances encountered in engineering applications. In addition to such single-phase complexities as non-linearity, transition to turbulence, three-dimensional behaviour, and time-variance, additional effects arising from the presence of a second phase must be considered. These include interfacial effects, wetting characteristics of the liquid, and momentum exchange between the liquid and vapour phases. As a result of the complex nature of condensing flows, development of predictive tools has proven to be a formidable task. Generalised analytical treatment of condensing flows is extremely difficult and simplified models are often used in lieu. Numerous experimental studies have also been carried-out in an attempt to characterise condensing flows. Therefore, there exists a wide range of experimental data and associated correlations to predict the heat transfer and pressure drop during condensation.

The purpose of this chapter is to present, describe, and review studies undertaken in the field of condensation to date. In the interest of chapter cohesiveness and conciseness, and due to the sheer volume of studies available on condensation, this chapter is separated into a series of sub-sections with each one focusing on a different aspect of condensation. The first section presents an overview of external (vapour-space) condensation. A number of important approximate analytical solutions have been developed for external condensation, which are implicit in, and form the basis for, studies on internal (tube-side) condensation. The second section in this chapter addresses those investigations into internal condensation. As internal condensation is the focus of much of the work presented in this thesis, it is explored and analysed in considerable depth. Concluding this chapter is an overview of two-phase pressure drop literature, with a particular emphasis on approaches to quantifying the losses in condensing flows. To close each individual section, a summary of current knowledge and work published to date in that specific field is presented. This identifies aspects of condensation that requires further investigation, hence, providing scope for the research outlined in this thesis.

2.1 EXTERNAL FILM CONDENSATION

External (or vapour-space) film condensation usually occurs when a stagnant vapour comes into contact with some solid surface which is at a lower temperature than the vapour saturation temperature. In 1916, Nusselt [52] published an analytical solution for external film condensation of a pure, quiescent vapour on a vertical flat plate. His
pioneering work laid the foundations for further studies on condensation, and is still considered the quintessential analysis in the field. This is emphasised by the fact that Nusselt’s analysis is still the basis on which those working in the field of condensation build their research. Nusselt’s solution, however, is a highly simplified approach to a complex process, with a number of assumptions imposed to allow the physical behaviour to be modelled. In order to solve the continuity, momentum and energy equations, Nusselt assumed the following:

- Steady-state laminar flow of the liquid phase, with constant properties.
- Subcooling of the liquid phase is negligible in the energy balance.
- Inertial forces and associated effects are negligible in the momentum balance.
- The vapour phase is stationary - imposes negligible drag on the liquid phase.
- The liquid-vapour interface is smooth with no interfacial waves.
- Heat transfer across the liquid is by conduction only (convection is neglected).

Based on these assumptions, Nusselt developed an analytical model for external film condensation, the system for which is shown in figure 2.1.

![Figure 2.1: Nusselt condensation on vertical flat plate](image)

The surface in figure 2.1, exposed to a motionless ambient of saturated vapour, is taken to be isothermal with a temperature below that of the vapour saturation
temperature. Consequently, a film of liquid condensate forms on the surface. Due to the neglect of inertial forces, the liquid film is drained solely by gravitational forces. This is referred to as falling-film condensation. The thickness of the condensate film increases along the length of the plate due to mass transfer at the liquid-vapour interface. By applying force and energy balances, Nusselt was able to formulate an expression for the film thickness, $\delta$, given in equation 2.1.

$$\delta = \left[ \frac{4k_l \mu_l (T_s - T_w) z}{\rho_l (\rho_l - \rho_v) gh_f g} \right]^{1/4}$$

(2.1)

where $z$ is the local length scale, $T_s$ is the vapour saturation temperature, and $T_w$ is the wall temperature. All other terms in equation 2.1 are given by their usual definitions in the nomenclature.

Nusselt assumed in his analysis that the convection of heat along the length of the plate, in the longitudinal direction, is negligible compared to conduction across the liquid film. He also assumed a linear temperature gradient across the film. Based on these assumptions, the local Nusselt number, $Nu_z$, at any point along the length of the plate, $z$, and the average Nusselt number, $Nu_L$, over the full length of the plate, $L$, were defined as in equation 2.2 and 2.3, respectively;

$$Nu_z = \frac{h_l z}{k_l} = \left[ \frac{\rho_l (\rho_l - \rho_v) gh_f g z^3}{4k_l \mu_l (T_s - T_w)} \right]^{1/4}$$

(2.2)

$$Nu_L = \frac{h_l L}{k_l} = 0.943 \left[ \frac{\rho_l (\rho_l - \rho_v) gh_f g L^3}{k_l \mu_l (T_s - T_w)} \right]^{1/4}$$

(2.3)

Whilst simple to use and incorporate into models, the Nusselt solution outlined above is, nevertheless, an idealised approach which fails to account for some commonly encountered transport phenomena. To overcome these limitations, a number of investigators have attempted to refine Nusselt’s work and relax many of his assumptions in a systematic manner. One of the first to do so was Bromley [53] and, subsequently, Rohsenow [54] who investigated the effect of subcooling and heat convection in the energy equation. Their integral analyses relaxed Nusselt’s assumption of a linear temperature profile by incorporating correction factors in the form of modified latent heat terms. However, under normal operating conditions, the effects that Bromley, and Rohsenow, examined had a negligible impact for pure saturated vapours. They concluded that the assumption of a linear temperature profile is quite acceptable for Jakob numbers less than unity ($Ja < 1$) - a condition most condensing fluids adhere to, with the Jakob number being defined as the ratio of sensible heat transfer to latent heat transfer.

In their 1959 publication, Sparrow and Gregg [55] presented a boundary-layer analysis of film condensation on a flat plate and formulated an expression practically identical to Nusselt’s. This is despite the fact that their analysis included the inertial terms in the momentum equation and the convective terms in the energy...
equation. Their solution showed that the inertial terms are negligible as Prandtl number approaches infinity, and that convective heat transfer is negligible as Jakob number approaches zero. This confirmed the validity of the basic Nusselt analysis for Prandtl numbers greater than unity and Jakob numbers less than unity. In a study which was an extension of Sparrow and Gregg’s, Koh et al. [56] analytically investigated the influence of waves, caused by vapour drag, at the liquid-vapour interface. Their results again showed that for common condensing fluids, with relatively high Prandtl numbers, there is no significant deviation from the Nusselt solution.

From the aforementioned studies, it is clear that despite the alterations and refinements that have been made to Nusselt’s analysis in the ensuing years, it is still expected to give reasonably accurate results for the case of external film condensation. However, the caveat to this is that the applicability of external condensation analyses to internal condensation is questionable due to the differences in the fluid dynamics [57]. It is for this reason that the review of literature relating to external condensation has not been exhaustive. Internal condensation is, however, reviewed in detail in the following section.

2.2 INTERNAL CONVECTIVE CONDENSATION

Internal (or tube-side) convective condensation is most commonly encountered when a flowing vapour comes into contact with a circular tubular surface which is at a lower temperature than the vapour saturation temperature. As a flowing vapour condenses in a circular tube, liquid is invariably formed. The liquid and vapour phases in simultaneous motion inside the tube constitutes a two-phase flow, a generic example of which is illustrated in figure 2.2.

![Idealised liquid-vapour two-phase flow in a circular tube](image)

Figure 2.2: Idealised liquid-vapour two-phase flow in a circular tube

Due to the relatively straightforward geometry and widespread application, studies of condensation in circular tubes are numerous in the technical literature, with vertical tubes being the most popular configuration. Studies comprise of analytical efforts to model the underlying physics, experimental investigations to measure and quantify the heat transfer characteristics, and various combinations of the two. While analytical models treat the physics of the flow, they tend to be mathematically complex. As a result, somewhat simpler empirical relations that correlate condensation heat transfer data are often preferred. The issue with empirical relations is that most
investigators collect data over a limited range - chosen to suit their application(s) of interest. Thus, empirical results tend to be specific to the test circumstances under which they were developed. Efforts to match empirical data with existing correlations, or to develop correlations to fit empirical data have been met with varying degrees of success. Exacerbating the issue is that many correlations come with no explicit range of parameters over which they can be expected to give accurate results. These issues are addressed in this section.

2.2.1 Two-phase flow overview

A range of different liquid-vapour flow combinations, known as flow regimes, can exist during convective condensation in a passage. There is general agreement in the literature that the condensation mechanism is intimately linked with the prevailing two-phase flow regime. The flow dynamics must then be considered when discussing convective condensation. However, before proceeding to a detailed discussion of the morphology of two-phase flow, it is necessary to establish a framework for describing two-phase flow using an idealised model. This is shown graphically in figure 2.3.

Figure 2.3: Idealised model of two-phase liquid-vapour flow in an inclined tube

From figure 2.3, the total mass flow rate through the tube is given by $\dot{m}$. This is equivalent to the sum of the mass flow rates of the individual phases;

$$\dot{m} = \dot{m}_v + \dot{m}_l \tag{2.4}$$

where $\dot{m}_v$ and $\dot{m}_l$ are the vapour and liquid flow rates, respectively. The ratio of vapour flow to total mass flow is given as;

$$x = \frac{\dot{m}_v}{\dot{m}} \tag{2.5}$$

where $x$ is known as the dryness fraction, or quality of the condensing flow. For a channel with a cross-sectional area, $A$, the mass flux, $G$, is defined as;
\[ G = \frac{\dot{m}}{A} \quad (2.6) \]

Another commonly encountered quantity in two-phase flow analyses is the void fraction, \( \alpha \), which is defined as the ratio of the vapour flow cross-sectional area, \( A_v \), to the total cross-sectional area, \( A \);

\[ \alpha = \frac{A_v}{A} \quad (2.7) \]

where \( A \) must equal the sum of the cross-sectional areas occupied by each individual phase, i.e.:

\[ A = A_v + A_l \quad (2.8) \]

The Reynolds number is a dimensionless quantity representing the ratio of the inertial to viscous forces. In two-phase flow analyses, this is frequently defined by the superficial Reynolds number, which assumes that the phase of interest occupies the entire tube alone - at the specified mass flow rate for that particular phase. This is given by;

\[ Re_l = \frac{G(1-x)D}{\mu_l} \quad \text{(liquid)} \quad Re_v = \frac{GxD}{\mu_v} \quad \text{(vapour)} \quad (2.9) \]

2.2.2 Two-phase flow regimes

The arbitrary two-phase flow regime shown in figure 2.3 is a relatively simple, idealised fluidic arrangement. In general, the morphology of the flow tends to be quite complex, where the arrangement of the phases is highly dependent on the fluid pair and their respective properties, flow conditions, and channel geometry. The two-phase flow regime in a circular tube will generally depend on the mean velocity of each phase and their respective properties. During convective condensation, the variation in properties over the flow length is quite small due to isothermal conditions. However, due to the phase change, a significant variation in the relative flow velocities tends to occur, and the flow regime can change dramatically over the length of the tube as a result. In addition, the flow regimes will vary according to the tube orientation. In the literature, tube orientation is normally classified as either vertical or horizontal. As this study primarily focuses on the latter, horizontal tubes forms the basis for the flow regime discussion.

For any two-phase flow pattern, the local heat transfer is intimately linked to that particular flow arrangement. Thus, the sequence of flow regimes that occur along a tube has a very strong impact on the heat transfer characteristics associated with the condensation process. Consequently, significant research has been conducted on the ability to predict a given flow regime under a prescribed set of conditions. This has resulted in the development of predictive techniques known as flow maps. These
maps graphically describe the sequence of flow regimes by separating the map into a series of distinct regions, where each region corresponds to a particular flow regime. Each region is separated from another by a boundary line, which is defined by a set of equations which may be empirically-based or analytically-based - depending on the choice of flow pattern map.

One of the first flow maps for horizontal tubes was that of Baker [58]. Baker’s map was based on observations of adiabatic gas-liquid flows in circular tubes, where the fluid combinations included air-water and oil-water. These combinations provided a relatively wide range of fluid properties to examine. Despite this, Baker’s map is rarely used as a predictive tool. However, it retains historical significance as the first widely recognised flow regime map.

Mandhane et al. [59] developed a flow map similar to Baker’s but used a much larger database of 5935 air-water observations, upon which to base their map. The Mandhane map correctly predicted the flow regime for 68% of the observations in their database, as opposed to 42% for the Baker map, perhaps a testament to the empirical nature of flow regime mapping. Nevertheless, numerous modifications to the Mandhane map have been necessary to ensure its applicability to conditions other than those for which it was developed.

Taitel and Dukler [2] proposed a map which was strongly based on theoretical arguments. They suggested that each flow regime transition was based on a different set of competing forces and that no single parameter should be expected to predict all flow regime transitions. The Taitel and Dukler map includes five flow regimes and was developed for adiabatic flows. An example of the Taitel and Dukler map is shown in figure 2.4.

![Taitel and Dukler flow regime map for horizontal gas-liquid flow. Map is re-created from [2]](image)

In the map of Taitel and Dukler shown in figure 2.4, the abscissa, \( X \), is the Martinelli parameter. This represents the ratio between the theoretical frictional pressure gradients defined in equation 2.10.
where \((dP/dz)_l\) and \((dP/dz)_v\) are the frictional pressure gradients for the liquid and vapour phases flowing alone in the pipe, respectively, defined in Section 2.3. The ordinates on the Taitel and Dukler map, \(F_{TD}\), \(T_{TD}\), and \(K_{TD}\), are the transition parameters - based on semi-theoretical derivations for different flow pattern transitions. These parameters are defined as follows:

\[
F_{TD} = \frac{G}{\rho_v (\rho_l - \rho_v) D g \cos \theta}^{1/2}
\]

\[
T_{TD} = \left[ \frac{(dP/dz)_l}{(\rho_l - \rho_v) g \cos \theta} \right]^{1/2}
\]

\[
K_{TD} = \frac{G}{\nu_l \rho_v (\rho_l - \rho_v) g \cos \theta}^{1/2}
\]

The choice of co-ordinate system in the map of Taitel and Dukler depends on the flow regime transition being considered. For example, Taitel and Dukler suggested \(K_{TD}\) vs. \(X\) for the transition from stratified-smooth to stratified-wavy flow. The \(F_{TD}\) vs. \(X\) relationship was proposed for the transitions between stratified-wavy, annular-dispersed, dispersed bubble, and intermittent flows. Finally, \(T_{TD}\) vs. \(X\) was recommended for defining the transition between dispersed bubble and intermittent flow regimes.

One of the main issues with the maps outlined thus far is that they were developed for adiabatic two-phase flow conditions. For condensing flows, factors such as mass transfer - which increases the liquid film thickness, and deceleration of the flow - due to phase change, have a considerable effect on transition between regimes. In response to this, authors such as Breber et al. [60] and Tandon et al. [61] developed maps specifically for condensation, based on experimental data from a range of refrigerants. These maps account for the influence of heat flux on the flow pattern transitions. A more recent effort by Kattan et al. [62] is based on over 1000 local evaporating flow pattern observations for seven different refrigerants. This is, essentially, a modification of the Taitel and Dukler map. The map of Kattan et al. is presented with the vapour quality, \(x\), as abscissa and mass velocity, \(G\), as ordinate. These co-ordinates are convenient as they facilitate prediction of the evolution of the flow pattern, at fixed mass velocities, with increasing/decreasing vapour quality along the length of a tube.

Most recently, El Hajal et al. [3] presented a new version of the Kattan et al. map specifically for condensing flows. This map was developed in accordance with the experimental database of Cavallini et al. [63], which includes 425 data points from six different refrigerants. The map of El Hajal et al. uses the same co-ordinates as the Kattan et al. map and, presently, appears to be the current standard for condensing flows.
flow maps. Current research is on-going into evaluating its applicability across a wider range of operating conditions. Figure 2.5 illustrates the El Hajal et al. map.

Figure 2.5: El Hajal et al. flow regime map for horizontal condensing liquid-vapour flow. Map is re-created from [3]

The map of El Hajal et al. classifies the two-phase flow patterns into five regimes; mist, annular, intermittent, stratified-wavy, and stratified flow, where intermittent flow commonly refers to some combination of plug flow and slug flow. The equations which define the transitions between these regimes are listed as follows. In the interest of conciseness, the reader is referred to the Nomenclature for the definition of specific terms in the equations.

\[
G_{\text{wavy}} = \left\{ \frac{16A_{s_d}^3 \rho_l \rho_v}{x^2 \pi^2 \left[ 1 - \left( \frac{2h_{ld}}{h_{ld} - 1} \right)^2 \right]^{0.5} x} \left[ \frac{\pi^2}{25h_{ld}^2} \left( \frac{We}{Fr} \right)_{l}^{-1.023} + 1 \right] \right\}^{2.14}
\]

\[
G_{\text{strat}} = \left\{ \frac{226.3^3 A_{ld} A_{sd}^2 \rho_l \rho_v (\rho_l - \rho_v) \mu_l g}{x^2 (1 - x) \pi^3} \right\}^{1/3} + 20x
\]

\[
x_{IA} = \left\{ 0.2914 \left( \frac{\rho_v}{\rho_l} \right)^{-1/1.75} \left( \frac{\mu_l}{\mu_v} \right)^{-1/7} + 1 \right\}^{-1}
\]

\[
G_{\text{mist}} = \left\{ \frac{7680 A_{sd}^2 \rho_l \rho_v}{x^2 \pi^2 \zeta} \left( \frac{Fr}{We} \right)_{l}^{0.5} \right\}
\]

Comparing the map of El Hajal et al. with that of Taitel and Dukler, it can be seen that flow regimes can be defined differently, depending on the author. Thus, the subjective nature of classifying flow pattern observations, and even the difference in opinion on the actual flow pattern definitions, ensures that a quantitative comparison between different maps is not always realistic. There remains a degree of ambiguity surrounding two-phase flow mapping and debate continues as to which flow pattern map should be used for a prescribed set of circumstances. However,
one factor that is generally acknowledged in the literature is that a given flow map should provide a good indication as to the predominant condensation mechanism, i.e. gravity-dominated or vapour-shear dominated. For the purpose of analysing condensing heat transfer behaviour, the various flow regimes discussed hitherto are divided into these two broad categories of gravity-dominated and shear-dominated.

2.2.3 Gravity-driven condensation

In line with the description of two-phase flows, the review of the literature relating to gravity-dominated condensation is limited to horizontal, or slightly inclined, tubes. As the flow regimes associated with vertical tubes are markedly different from those in horizontal tubes, the heat transfer characteristics cannot be expected to be similar. Therefore, theory relating to horizontal and slightly inclined tubes only is presented in this section.

Gravity-driven condensation, as the name suggests, is characterised by low vapour velocities and gravitational forces which are much stronger than vapour shear forces. Consequently, in the absence of appreciable shear, vapour that condenses on the top of the tube wall tends to flow downwards, around the circumference and, ultimately, accumulates in the bottom of the tube as illustrated in figure 2.6.

![Figure 2.6: Illustration of gravity-driven stratified condensation from tube end-view](image)

In gravity-driven condensation, the liquid pool residing in the bottom of the tube transports the condensed liquid through the tube, in the direction of the mean flow. This is manifested as one of three flow regimes discussed in the previous section; stratified, stratified-wavy, or wavy flow. These regimes are lumped together not just because of their visual similarities and analogous flow dynamics, but also because the prevailing heat transfer mechanism is conduction across the thin film at the top of the tube. This is very similar to the external falling-film condensation discussed in Section 2.1. For this reason, heat transfer analyses for gravity-driven internal condensation rely heavily on the theory developed for external condensation. Some
One of the first experimental investigations into condensation in horizontal and inclined tubes were carried-out by Jakob [64] and his colleagues. They found good agreement between data from their steam condensation experiments and the Nusselt theory, despite ignoring the axially-flowing condensate in the bottom of the tube. Tepe and Mueller [65] examined condensation of benzene and methanol inside inclined tubes but evaluated heat transfer coefficients that did not compare well with those predicted using Nusselt’s analysis. Akers et al. [66], Akers and Rosson [67], and Rosson and Myers [68] correlated a large number of heat transfer data as a function of a density-corrected Reynolds number. In general, these correlations over-predicted the corresponding Nusselt values, particularly at high flow rates. In one of the first analytical studies on condensation in horizontal tubes, Chaddock [69] combined the Nusselt solution with an empirical relation for the depth of the liquid condensate to evaluate the heat transfer characteristics. None of these aforementioned studies, however, examined the scenario where the condensate layer in the bottom of the tube is sufficiently appreciable, such that the heat transfer is affected. This was rectified by Chato [57, 70].

Chato developed a detailed analytical model for internal condensation in horizontal and inclined tubes, concentrating in particular on stratified-annular flows at low vapour velocities and $Re_v < 35,000$. He developed a similarity solution for the condensate film, which was based on Chen’s [71] analysis of falling-film condensation on the exterior of a horizontal cylinder. The approach of Chato was to apply the similarity solution to the upper portion of the tube, where falling-film condensation prevailed. The remainder of the tube circumference was designated to the condensate pool, the depth of which was predicted using a separate model based on open channel hydraulics. Chato found his experimental data from the refrigerant R-113 to agree quite well with his analytical model, expressed in the form of average Nusselt number:

$$\bar{Nu} = \frac{\tau D}{k_t} = 0.555 \left[ \frac{\rho_l (\rho_l - \rho_v) gh_f g D^3}{k_t \mu_l (T_s - T_w)} \right]^{1/4}$$

(2.18)

Apart from the multiplying prefactor of 0.555, equation 2.18 is very similar to the relation obtained from the classic Nusselt analysis given previously by equation 2.3 in Section 2.1. Chato incorporated the prefactor to account for the reduction in heat transfer due to the liquid condensate pool in the bottom of the tube. He also argued that, based solely on conduction, the heat transfer through the condensate pool was negligible compared to the transport across the thin film on the remainder of the tube perimeter. Nevertheless, the similarity of equation 2.18 and equation 2.3 is consistent with the interpretation that the condensation process over the top portion of the inside tube wall is very similar to Nusselt’s falling-film condensation over a flat plate. Chato’s model was compared with 210 experimental data points.
for refrigerants that satisfied the criterion of $Re_v < 35,000$, and was found to have a mean deviation of about 13%.

In the ensuing years since Chato’s analysis, numerous investigators have attempted to refine and build upon his work. In their 1966 study, Rufer and Kezios [72] presented a model for stratified flow under an imposed pressure gradient. This was in response to Chato’s liquid flow model, which assumed the flow was driven by a hydrostatic gradient. Rufer and Kezios disagreed with this assumption as they believed it was not representative of behaviour in an actual condenser. Their model contradicted Chato’s insofar that it predicted the depth of the condensate pool to increase along the length of tube. The model of Rufer and Kezios also predicted that, as the quality of the flow approached zero, the liquid level approaches the height of the channel. Inherently, this approach appears to model the actual flow circumstances in a more representative fashion. However, it is rarely used as numerous authors, such as Jaster and Kosky [73], have found it computationally-intense and overly cumbersome for practical use.

Jaster and Kosky proposed a correlation similar to Chato’s for stratified flow condensation. However, following the findings of Rufer and Kezios, they accounted for the variation in the condensate pool depth in a manner consistent with pressure-driven flow. To achieve this, Jaster and Kosky replaced the prefactor in Chato’s equation with a void fraction function. Jaster and Kosky’s correlation had a mean deviation of about 37% with the authors data from their steam condensation experiments - carried out over a mass flux range of 12 - 145 kg/sm$^2$. However, of greater concern is the fact that both correlations neglect the heat transfer that occurs in the condensate pool at the bottom of the tube. Chato proposed that, by considering conduction only, this heat transfer is negligible in comparison to that through the upper portion of the tube where the liquid film is much thinner. The assumption of conduction-only is appropriate for the low-speed stratified flows which Chato focussed on. However, for situations where the vapour velocity is appreciable, even in stratified flows, substantial convective heat transfer can occur through the condensate pool, according to Dobson and Chato [74].

In what appears to be one of the only studies to-date on quantifying the heat transfer through the condensate film, Rosson and Myers [23] collected local measurements in the intermittent flow regime, which they defined as encompassing stratified flow, wavy flow, and various combinations of the two. These conditions were examined over a vapour Reynolds number range of 2000 - 40,000 and liquid Reynolds number range of 60 - 1500. Rosson and Myers measured the variation in the condensing heat transfer coefficient for methanol and acetone around the inside perimeter of a horizontal tube. They found, as expected, that the heat transfer coefficient decreased continuously from the top to the bottom of the tube. They proposed to predict the condensing coefficient at the top of the tube by incorporating the effect of vapour shear in Nusselt’s solution. This was achieved through the use of an empirically determined function of the vapour Reynolds number as defined in equation 2.19.
2.2 Internal convective condensation

\[ Nu_{\text{top}} = \frac{h_{\text{top}} D}{k_l} = 0.31 R e_v^{0.12} \left[ \frac{\rho_l (\rho_l - \rho_v) g h f_g D^3}{k_l \mu_l (T_s - T_w)} \right]^{1/4} \]  \hfill (2.19)

For the heat transfer in the bottom of the tube, Rosson and Myers suggested that forced-convection heat transfer was occurring as a result of the motion of the condensate pool. By means of a heat and momentum transfer analogy, they proposed the following correlation:

\[ Nu_{\text{bot}} = \frac{h_{\text{bot}} D}{k_l} = \frac{\phi_{l,ht} \sqrt{8 R e_l}}{5 \left[ 1 + \ln \left( 1 + 5 P r_l \right) / P r_l \right]} \]  \hfill (2.20)

where \( \phi_{l,ht} \) is the two-phase multiplier for laminar liquid flow and turbulent vapour flow as given in Lockhart and Martinelli’s [4] pioneering paper on two-phase pressure drop. It is defined in Section 2.3, which proceeds this. As Rosson and Myers defined two distinct and separate heat transfer mechanisms, a parameter was necessary to distinguish between the regions which are governed by falling-film, or forced-convective, condensation. They defined a parameter, \( \beta \), which represents the fraction of the tube perimeter over which falling-film condensation occurs. Inherently, the remainder must, therefore, be dominated by forced-convection. By adopting this approach, Rosson and Myers defined a circumferentially-averaged, local streamwise Nusselt number as:

\[ Nu = \beta \cdot Nu_{\text{top}} + (1 - \beta) Nu_{\text{bot}} \]  \hfill (2.21)

When compared with their own experimental data, Rosson and Myers’ correlation showed reasonable agreement. The condensation heat transfer coefficient in the upper region of the pipe, bottom region of the pipe, and streamwise local was within \( \pm 27\% \), \( \pm 40\% \) and \( \pm 30\% \) of the predicted values, respectively. It is not known if these discrepancies are a result of inaccuracies in their experimental methods or, simply, theoretical deficiencies.

Dobson [12, 74] examined the condensation of various refrigerants in the wavy flow regime at mass fluxes ranging from 25 - 800 kg/sm\(^2\). Dobson found that as the vapour velocities increased from very low values, the vapour shear leads to an increase in the convective heat transfer in the liquid pool residing in the bottom of the tube. He also postulated, similarly to Rosson and Myers, that increased vapour velocities generate axial shear components which tend to augment heat transfer all around the inside tube perimeter. From the experimental data, Dobson proposed a correlation which separates the heat transfer by film condensation in the upper part of the tube from forced convection in the bottom pool. This correlation is recommended when \( G < 500 \) kg/sm\(^2\).

More recently, Lips and Meyer have overseen a substantial body of experimental and theoretical research into gravity-driven convective condensation. Initially, they identified the lack of, and need for more, data relating to condensation in inclined tubes [75]. They noted that Chato [57] was the first to observe the effect of tube
inclusion, but since then experimental studies were very limited in their range of test conditions. In response, Lips and Meyer experimentally investigated the flow pattern and heat transfer characteristics of the refrigerant R134a across the full range of inclination angles (from vertical downwards to vertical upwards) [76, 77, 78]. They found that the inclination angle had a significant effect on the liquid distribution in the tube and found an optimal angle of -15° for downward flow, which increased the heat transfer by approximately 20%. In the same study, the authors also observed that, at high mass fluxes, shear forces are dominant which render the heat transfer coefficient somewhat independent of inclination angle. Following this, Lips and Meyer [79] presented a model for stratified flow convective condensation in inclined tubes, which was in good agreement with their previously published data.

In a very recent study, Shah [80], similar to Lips and Meyer, recognised that comparatively few studies have been carried out on inclined tube condensers, and that there is no well-verified method for the prediction of heat transfer at such conditions. Shah presented an empirical approach, which had a mean absolute deviation of 15.7% with a database consisting of eleven individual experimental studies containing 669 data points in all. In his concluding remarks, Shah stressed the need for further research into condensation heat transfer in inclined tubes. Despite the fact that the equations presented in this section are not directly referenced, or compared with the experimental data, in Chapters 6 and 7, the accompanying discussions provide context to the results presented in those chapters.

2.2.4 **Shear-driven condensation**

At sufficiently high vapour velocities, interfacial shear stresses dominate the flow. These stresses lead to a two-phase flow topology characterised by an almost axisymmetric thin film of liquid around the inside tube circumference, within which flows a high-speed vapour core. As illustrated in Section 2.2.2, this is commonly referred to as annular flow, with a cross-sectional view of this fluidic arrangement illustrated in figure 2.7. The annular flow regime tends to prevail for the majority of the condensing length and, as such, a range of experimental and theoretical studies have been carried out to investigate and establish the heat transfer characteristics of such flows. This section discusses some of the most prominent of these studies.

One of the earliest experimental studies can be traced back to Ananiev et al. [10], who examined steam condensation in a horizontal tube. Operating conditions during the experiments of Ananiev et al. included a pressure range of 12 - 99 bar, a temperature range of 189 - 310 °C, and steam velocities from 5 - 38 m/s. Ananiev et al. correlated their heat transfer coefficients with the following relation:

\[ h = h_1 \sqrt{\frac{\rho_l}{\rho_m}} \]  \hspace{1cm} (2.22)
2.2 INTERNAL CONVECTIVE CONDENSATION

Figure 2.7: Illustration of shear-driven annular condensation from tube end-view

where \( h_l \) is the single-phase heat transfer coefficient for the entire flow as liquid, as given by Dittus-Boelter [81] in equation 2.23:

\[
h_l = 0.023 \text{Re}_l^{0.8} \text{Pr}_l^{0.4} \left( \frac{k_l}{D} \right)
\]  

(2.23)

Ananiev et al. noted that the large steam velocities in their experiments tended to “blow-off” the condensate which formed on the tube wall and a comparatively thin film remained behind. This study was followed by that of Boyko and Kruzhilin [82] who investigated steam condensation in both a horizontal tube and in a bundle of horizontal tubes. Boyko and Kruzhilin’s experiments were carried-out at similar conditions to those of Ananiev et al., with a pressure range of approximately 12 - 88 bar and a large vapour Reynolds number range of 10,000 - 100,000.

In high velocity flows, it is generally recognised that the shear at the liquid-vapour interface, and at the tube wall, is directly linked to the transport of heat across the liquid film. Shear-based correlations were first proposed by Carpenter and Colburn [83], who postulated that the resistance to heat transfer in turbulent liquid flow was entirely inside the laminar sub-layer. This work was extended upon by Soliman et al. [84], who developed a semi-empirical correlation which explicitly acknowledged the importance of shear effects in high velocity flows. This correlation is given by:

\[
\frac{h_l \mu}{k_l \rho_l^{1/2}} = 0.036 \text{Pr}_l^{0.65} \tau_w^{1/2}
\]

(2.24)

where \( \tau_w \) is the shear stress at the wall, the calculation of which is quite complicated through semi-empirical means. Soliman et al. compared the predictions from their correlation with experimental data from various refrigerants and observed reasonable agreement. However, due to the complex nature of the correlation - particularly the shear stress, it is rarely found in the literature.

Cavallini and Zecchin [11] used the results of a theoretical analysis of annular flows to deduce the variables that should be present in an annular flow correlation.
The authors subsequently employed regression analysis to justify neglecting variables which did not form part of their correlation. Their final correlation is expressed by equation 2.25:

\[
Nu = 0.023 Re_l^{0.8} Pr_l^{0.33} \left\{ 2.64 \left[ 1 + \left( \frac{\rho_l}{\rho_v} \right)^{0.5} \frac{x}{1-x} \right] \right\}^{0.8}
\]  

(2.25)

Equation 2.25 represents a two-phase multiplier approach to predicting the heat transfer, where the bracketed term is the multiplier acting on the single-phase liquid heat transfer coefficient - given previously by equation 2.23. Cavallini and Zecchin compared their correlation with experimental data from six different studies employing refrigerants including R-113, R-12, and R-22 as the working fluid. The standard deviation between their correlation and the experimental data sets varied from 8% to 47%.

Based purely on an empirical approach, Shah [9] proposed a correlation as a best-fit to a wide range of convective condensation heat transfer data for round tubes. This correlation was developed from Shah’s original correlation for saturated boiling heat transfer [85], by observing the similarity between the mechanisms of heat transfer during film condensation and boiling without bubble nucleation (evaporation). This similarity is valid provided the entire pipe surface remains wetted by the liquid during condensation. The correlation of Shah is expressed as:

\[
Nu = 0.023 Re_l^{0.8} Pr_l^{0.4} \left[ 1 + 3.8 \left( \frac{x}{1-x} \right)^{0.76} \right]^{0.8}
\]  

(2.26)

where \( P_r \) is the reduced pressure - that is the ratio of actual, operating pressure to the critical pressure of the fluid of interest, defined in equation 2.27. As the reduced pressure decreases, the properties of the liquid and vapour phases become more dissimilar, and Shah’s two-phase multiplier expression increases.

\[
P_r = \frac{P}{P_c}
\]  

(2.27)

Shah found good agreement between his correlation and film condensation data for a wide variety of fluids including water, various refrigerants, and organic fluids condensing in horizontal, vertical, and inclined tubes. The data examined spanned saturation temperatures from 21 - 310 °C, vapour velocities from 3 - 300 m/s, and mass fluxes from 11 - 211 kg/sm². In all, Shah examined 473 data points from 21 independent experimental studies and correlated these with a mean deviation of approximately 15%.

Recently, Shah [86] extended his original correlation [9] to encompass a wider range of parameters including mass fluxes from 4 - 820 kg/sm². Nevertheless, the extended correlation retained its accuracy, as demonstrated by a mean deviation with the expanded database of approximately 14.5%. Successive studies by Shah [87, 88] have focussed on refining the correlation through flow mapping.
One of the most recent developments in annular-driven condensation was the correlation proposed by Dobson [12]. Similar to Cavallini and Zecchin, and Shah, this correlation was a two-phase multiplier approach expressed as:

\[
Nu = 0.023 Re^{0.8} Pr^{0.4} \left[ 1 + \frac{2.22}{X_{tt}^{1.889}} \right] \tag{2.28}
\]

where \( X_{tt} \) is the Lockhart-Martinelli parameter defined in equation 2.29. At a quality of zero, this parameter approaches infinity and, thus, equation 2.28 becomes the single-phase liquid Nusselt number.

\[
X_{tt} = \left( \frac{\rho_v}{\rho_l} \right)^{0.5} \left( \frac{\mu_l}{\mu_v} \right)^{0.1} \left( \frac{1 - x}{x} \right)^{0.9} \tag{2.29}
\]

Dobson compared his correlation to annular flow data from four refrigerants - R-12, R-22, R-134a, and and azeotropic blends of R-32/R-125. In general, he observed excellent overall agreement with the data, emphasised by a mean deviation of \( \pm 4.5\% \). Furthermore, over 67\% of the data points were predicted to within \( \pm 5\% \), and 96\% of the points were predicted to within \( \pm 15\% \).

2.2.5 Summary

The aforementioned literature comprise the bulk of heat transfer studies undertaken on single-component, internal condensation in horizontal, and slightly inclined, round tubes to-date. As described, the majority of these studies are either analytically-based, or deeply-rooted in theory. The most widely-cited experimental investigations on condensing flows such as Carpenter [89], Goodykoontz and Dorsch [24, 90], Blangetti and Schlunder [91], and Kuhn et al. [92] are all for vertical tube configurations. The flow characteristics associated with a vertical tube are significantly different from those in a horizontal tube and, hence, the heat transfer characteristics are not expected to be similar.

Experimental investigations on condensing flows in horizontal tubes have been carried-out, for example, by Altman et al. [93], Bae et al. [94], Azer et al. [95], and Cavallini et al. [63], but all these studies, and more recent efforts [96, 97, 98, 99, 100, 101, 102], are based on refrigerants at mass fluxes and Reynolds numbers much larger than those examined in this thesis. In fact, there appears to be a significant scarcity of experimental data for condensing flows of steam, particularly at typical condenser operating conditions. This was surprising, given the widespread application of steam condensation in thermoelectric power plants. Motivation for the research presented in this thesis was partly based on this lack of information. The deficiencies in the literature are summarised in table 2.1, which presents a comparison between internal convective condensation data published to date and the author’s own data.

From table 2.1, it is clear that steam condensation in horizontal and inclined tubes has, hitherto, been examined over a very limited range of parameters. Indeed, the
Table 2.1: Experimental data range for internal convective condensation in horizontal and slightly inclined tubes published to date, compared with author’s data

<table>
<thead>
<tr>
<th>Parameter</th>
<th>All fluids</th>
<th>Water only</th>
<th>Author’s data</th>
</tr>
</thead>
<tbody>
<tr>
<td>$G$ (kg/sm$^2$)</td>
<td>13 - 3368</td>
<td>13</td>
<td>0.6 - 2.5</td>
</tr>
<tr>
<td>$Re_l$</td>
<td>670 - 115362</td>
<td>1808</td>
<td>22 - 120</td>
</tr>
<tr>
<td>$Re_v$</td>
<td>37258 - 599510</td>
<td>54415</td>
<td>1480 - 5145</td>
</tr>
<tr>
<td>$u_v$ (m/s)</td>
<td>3 - 40</td>
<td>5 - 38</td>
<td>8 - 42</td>
</tr>
<tr>
<td>$P_r$</td>
<td>0.0023 - 0.907</td>
<td>0.0023 - 0.44</td>
<td>0.00035 - 0.002</td>
</tr>
</tbody>
</table>

author could only find two experimental studies based on condensing flows of steam in horizontal and slightly inclined tubes - that of Ananiev et al. [10] and Varma [103]. This echoed the findings of Shah when he was gathering data for development of his correlation [9, 86]. Shah noted a distinct lack of experimental investigations for steam-water flows in horizontal tubes and, consequently, stressed the need for more data. Taking this into consideration, combined with the preceding review of the contemporary relevant literature, there is still scope for further research in the field of internal convective condensation. This is particularly true for steam condensation at conditions typical of those in Rankine cycle condensers.

2.3 TWO-PHASE PRESSURE DROP

It is generally recognised in the literature that the main issue associated with the topic of two-phase pressure drop is that no universal method is available that will reliably predict the fluidic losses. This issue persists despite numerous theoretical treatments and experimental investigations into adiabatic, boiling, evaporating, and condensing flows of various fluid combinations. Theoretical approaches are only feasible through narrowing the range of applicability and, in some cases, still incorporate some form of empiricism to close the solution. Experimental investigations give rise to empirical correlations to predict the pressure drop. However, the applicability of any given empirical correlation is restricted by the very nature of its empiricism - which is the range of its underlying database. Accordingly, it follows that no single correlation is currently available to provide reasonable accuracy for general use. This section presents and reviews the approaches most commonly found in literature.

2.3.1 General two-phase pressure drop model

The simplest model of two-phase transport treats the flow as steady and one-dimensional, in an approach commonly referred to as the separated flow model. This model quantifies the contribution of each phase separately by considering the specific thermophysical properties and velocities associated with each phase. The resultant pressure
2.3 Two-Phase Pressure Drop

Drop is, therefore, the sum of the losses associated with each phase. The separated flow model is derived here with reference to figure 2.8.

\[
\dot{m} = \dot{m}_v + \dot{m}_l \quad (2.30)
\]

By differentiation of equation 2.30, the following expression is determined:

\[
d\dot{m}_v = -d\dot{m}_l \quad (2.31)
\]

The second governing equation for the flow illustrated in figure 2.8 is derived from a force-momentum balance on the differential element. The balance on the vapour phase is expressed as:

\[
PA_v - \left[ (P + dP) (A_v + dA_v) \right] - dF_v - dF_{i,v} - A_v \rho_v g \sin \theta = \left[ (\dot{m}_v + d\dot{m}_v) (u_v + du_v) \right] - \dot{m}_v u_v - d\dot{m}_v u_l \quad (2.32)
\]

where the first two terms on the left-hand side of equation 2.32 represent the pressure forces on the element dz. \(dF_v\) and \(dF_{i,v}\) represent the frictional effect of the vapour on the channel wall and the interfacial shear force, respectively. The first two terms on the right-hand side account for the momentum change due to acceleration/deceleration of the fluid continuum. Finally, \(d\dot{m}_v u_l\) represents the momentum exchange between the liquid and the vapour, due to the phase change occurring at the interface. In a completely analogous way, a force-momentum balance on the liquid phase is expressed as follows:

\[
PA_l - \left[ (P + dP) (A_l + dA_l) \right] - dF_l - dF_{i,l} - A_l \rho_l g \sin \theta = \left[ (\dot{m}_l + d\dot{m}_l) (u_l + du_l) \right] - \dot{m}_l u_l - d\dot{m}_l u_l \quad (2.33)
\]
For steady flow, the interfacial shear forces must balance, i.e.;

$$dF_{i,v} = -dF_{i,l}$$  \hspace{1cm} (2.34)

From Section 2.2.1, it follows that the differential area equates to the sum of the individual differential areas occupied by each phase. This is expressed as;

$$dA = dA_v + dA_l$$  \hspace{1cm} (2.35)

Summing equations 2.32 and 2.33, and incorporating the arguments laid forth in equations 2.34 and 2.35, the following relation for the overall two-phase force-momentum balance is derived;

$$-AdP - dF_l - dF_v - (A_l \rho_l + A_v \rho_v) gdz \sin \theta = d(\dot{m}_v u_v + \dot{m}_l u_l)$$  \hspace{1cm} (2.36)

A fictitious pressure gradient, \( (dP/dz)_{fr} \), is introduced to account for the combined frictional effect of the liquid and vapour phase on the wall;

$$dF_l + dF_v = -(dP/dz)_{fr} Adz$$  \hspace{1cm} (2.37)

Incorporating equation 2.37, and equation 2.7 from Section 2.2.1, the momentum balance relation can be expressed as;

$$-dP/dz = -(dP/dz)_{fr} + \frac{d}{dz} \left[ \frac{G^2 x^2}{\rho_v \alpha} + \frac{G^2 (1-x)^2}{\rho_l (1-\alpha)} \right] + [(1-\alpha) \rho_l + \alpha \rho_v] g \sin \theta$$  \hspace{1cm} (2.38)

In equation 2.38, it can be seen that there are three contributing components which constitute the overall pressure gradient. Namely, these terms represent the frictional, momentum, and gravitational (static) effects experienced by the two-phase condensing flow. The frictional effects arise from the shearing of each phase on the channel wall. Momentum effects and gravitational effects reflect the change in kinetic energy and potential energy, respectively, of the fluid continuum. Through application of the chain rule on the momentum term, equation 2.38 can be expanded to assume the following form;

$$-\frac{dP}{dz} = -\left( \frac{dP}{dz} \right)_{fr} + G^2 \left\{ \left[ \frac{(1-x)^2}{\rho_l (1-\alpha)} + \frac{x^2}{\rho_v \alpha} \right]_e - \left[ \frac{(1-x)^2}{\rho_l (1-\alpha)} + \frac{x^2}{\rho_v \alpha} \right]_i \right\}$$

$$+ [(1-\alpha) \rho_l + \alpha \rho_v] g \sin \theta$$  \hspace{1cm} (2.39)

The most contentious aspect of equation 2.39 is in the calculation of the frictional pressure gradient. The standard approach is to incorporate a two-phase multiplier
with a well-established single-phase pressure drop expression. It is at this point, however, that empiricism is introduced to the model. Empirical correlations, or relations derived from simplified theory, are used to determine the two-phase multipliers. The empirical methods are outlined in the ensuing section. Ultimately, equation 2.39 can be expressed in shorthand as follows;

$$\Delta P = \Delta P_{fr} + \Delta P_{mom} + \Delta P_{st}$$ (2.40)

2.3.2  Two-phase frictional pressure drop empirical methods

This section presents a range of two-phase frictional pressure drop correlations which can be integrated into equation 2.39 to predict the frictional pressure drop component. The correlations chosen are reflective of their popularity in literature and are amongst the most widely-cited for frictional pressure losses in circular tubes.

2.3.2.1  Lockhart and Martinelli [4]

Lockhart and Martinelli [4] proposed a generalised correlation method for determining the two-phase multiplier, $\phi_l$ or $\phi_v$, from which the frictional pressure drop can be predicted. They found their correlation to be in good agreement with experimental data for adiabatic and isothermal two-phase flows of various air and liquid combinations. Lockhart and Martinelli correlated this data by identifying four types of specific flow, dependent on whether each phase was laminar or turbulent. The two-phase multipliers proposed by Lockhart and Martinelli were expressed as a function of the Martinelli parameter, $X$ - previously given by equation 2.10 in Section 2.2, and are related as shown by the following relationships;

$$\phi_l = \left(1 + \frac{C}{X} + \frac{1}{X^2}\right)^{1/2} \quad \phi_v = \left(1 + CX + X^2\right)^{1/2}$$ (2.41)

The value of $C$ in equation 2.41 depends on the particular flow regime associated with the liquid and vapour phases. The constants recommended by Chisholm and Laird [104] for each of the four possible combinations are given in table 2.2.

<table>
<thead>
<tr>
<th>Liquid</th>
<th>Gas</th>
<th>Subscript</th>
<th>$C$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent</td>
<td>Turbulent</td>
<td>tt</td>
<td>20</td>
</tr>
<tr>
<td>Viscous</td>
<td>Turbulent</td>
<td>vt</td>
<td>12</td>
</tr>
<tr>
<td>Turbulent</td>
<td>Viscous</td>
<td>tv</td>
<td>10</td>
</tr>
<tr>
<td>Viscous</td>
<td>Viscous</td>
<td>vv</td>
<td>5</td>
</tr>
</tbody>
</table>

The two-phase multipliers, given in equation 2.41, are applied to the single-phase pressure drop term, for either phase, to predict the frictional pressure drop as follows;
\[ \Delta P_{fr} = \phi_l^2 \Delta P_l \quad \Delta P_{fr} = \phi_v^2 \Delta P_v \] (2.42)

where \( \Delta P_l \) and \( \Delta P_v \) is the pressure drop that would exist if the the liquid or vapour phase, respectively, was assumed to flow alone in the entire cross-section of the channel. These terms are defined as;

\[ \Delta P_l = \frac{2 f_l G^2 (1 - x)^2 L}{\rho_l D} \quad \Delta P_v = \frac{2 f_v G^2 x^2 L}{\rho_v D} \] (2.43)

where the single-phase Fanning friction factors, \( f_l \) and \( f_v \), for round tubes are given by the classical definitions;

\[ f_l = B Re_l^{-n} \quad f_v = B Re_v^{-n} \] (2.44)

The constants \( B \) and \( n \) in equation 2.44 depend on the flow conditions. If the flow is laminar, \( B = 16 \) and \( n = 1 \), whereas for turbulent conditions, \( B = 0.079 \) and \( n = 0.25 \).

Lockhart and Martinelli’s method is applicable to the full range of vapour qualities, i.e. \( 0 \leq x \leq 1 \). According to Quiben [105], as the correlation was developed for horizontal two-phase flow of two-component systems at low pressures, its application outside this range is not recommended. Whalley [106] recommends use of the Lockhart and Martinelli method at mass velocities less than 100 kg/sm^2.

2.3.2.2 Friedel [5]

Friedel [5] used a database of approximately 25,000 points to develop a highly-empirical correlation for a two-phase multiplier. The multiplier is utilised as follows to predict the frictional pressure drop;

\[ \Delta P_{fr} = \phi_{lo}^2 \Delta P_{lo} \] (2.45)

where \( \Delta P_{lo} \) is the pressure drop that would result if liquid only flowed through the tube at the total mass flow rate, defined as follows;

\[ \Delta P_{lo} = \frac{2 f_l G^2 L}{\rho_l D} \] (2.46)

\( \phi_{lo} \) in equation 2.45 is the Friedel two-phase multiplier, correlated as;

\[ \phi_{lo}^2 = C_{F1} + \frac{3.42 C_{F2}}{Fr^{0.045} We^{0.035}} \] (2.47)

\( C_{F1} \) and \( C_{F2} \) in equation 2.47 are correlating parameters used by Friedel. \( Fr \) and \( We \) are the Froude number and Weber number, respectively. These terms are defined as;

\[ C_{F1} = (1 - x)^2 + x^2 \left( \frac{\rho_l}{\rho_v} \right) \left( \frac{f_{vo}}{f_{lo}} \right) \] (2.48)
\[ C_{F2} = x^{0.78} (1 - x)^{0.24} \left( \frac{\mu_l}{\mu_v} \right)^{0.91} \left( \frac{x}{\rho_v} \right) \left( \frac{1 - \mu_v}{\mu_l} \right)^{0.7} \] (2.49)

\[ Fr = \frac{G^2}{gD\rho_h} \quad We = \frac{G^2D}{\rho_h\sigma_{st}} \] (2.50)

In both the Froude number and Weber number, the homogeneous density, \( \rho_h \), is introduced. This is sometimes called the mean density as it represents the average density of both phases, combined, and is given as;

\[ \rho_h = \left( \frac{x}{\rho_v} + \frac{1-x}{\rho_l} \right)^{-1} \] (2.51)

Whalley [106] recommended the use of Friedel’s correlation when \( \mu_l/\mu_v < 1000 \) and \( G < 2000 \text{ kg/sm}^2 \). For most condensing fluid combinations \( \mu_l/\mu_v < 1000 \).

2.3.2.3 Grönnerud [6]

The method of Grönnerud [6] was developed specifically for refrigerants. Grönnerud, similarly to previous methodologies, applied a two-phase multiplier to the single-phase pressure drop term to account for increased losses. This expression is given in equation 2.52;

\[ \Delta P_{Fr} = \phi_{gd} \Delta P_l \] (2.52)

The two-phase multiplier proposed by Grönnerud, \( \phi_{gd} \), is defined in equation 2.53;

\[ \phi_{gd} = 1 + \left( \frac{dP}{dz} \right)_{Fr} \left[ \left( \frac{\rho_l}{\rho_v} \right)^{0.25} - 1 \right] \] (2.53)

where the pressure gradient used in Grönnerud’s multiplier is dependent on the Froude number as shown;

\[ \left( \frac{dP}{dz} \right)_{Fr} = f_{Fr} \left[ x + 4 \left( x^{1.8} - x^{10} f_{Fr}^{0.5} \right) \right] \] (2.54)

as \( f_{Fr} \) is the friction factor based on the Froude number, given by;

\[ f_{Fr} = Fr_l^{0.3} + 0.0055 \left( \frac{1}{Fr} \right) \] (2.55)

The correlation of Grönnerud is applicable over the range of \( 0 \leq x \leq 1 \) and is particularly suited to predicting the frictional pressure drop of refrigerants.

2.3.2.4 Müller-Steinhagen and Heck [7]

Müller-Steinhagen and Heck [7] developed their correlation from a data bank of 9000 pressure drop measurements of various fluid pairs, including steam-water and the refrigerant R12. Mass velocities associated with the data varied from 50 - 2490
The authors proposed that the two-phase frictional pressure gradient is an empirical interpolation between liquid only flow and vapour only flow, and can be calculated as follows:

\[
\left(\frac{dP}{dz}\right)_{fr} = \left\{ \left(\frac{dP}{dz}\right)_{lo} + 2 \left[ \left(\frac{dP}{dz}\right)_{vo} - \left(\frac{dP}{dz}\right)_{lo} \right] x \right\} (1 - x)^{0.33} + \left(\frac{dP}{dz}\right)_{vo} x^3 \tag{2.56}
\]

The method of Müller-Steinhagen and Heck is applicable over the full range of vapour qualities. In a relatively recent study comparing several different methods for predicting the two-phase frictional pressure drop of liquid-gas flows, including air-water, steam-water, and air-oil, Tribbe and Müller-Steinhagen [107] found the method of Müller-Steinhagen and Heck to give the best results.

2.3.3 Calculation methodology

The empirical correlations presented hitherto are, largely, applicable to adiabatic flows. As a result, in most cases for which the correlations were developed, the steam quality was constant from inlet to outlet without varying in the manner seen in a typical diabatic condensing flow - such as that studied in this thesis. Therefore, in order to compare the correlations with experimentally-derived frictional data, the following calculation methodology was employed:

- The frictional pressure drop was calculated on a per-tube basis, i.e. for a single, individual tube
  - For the single tube, reduced-scale ACC, this methodology was inherent.
  - For the multi-row, full-scale MACC, this methodology was applied to reduce the problem to the case of a single tube in the MACC tube bundle.
    * To satisfy this, it was assumed that the total mass flow rate through the MACC module was distributed evenly among the tubes in the bundle.

- With an individual mass flow rate and, hence, Reynolds number defined, an individual tube was discretised by dividing it into 100 individual segments to calculate the local steam qualities using an iterative procedure.
  - 100 elements were found to be sufficient as the calculation converged at this discrete resolution.

- Using this method, the vapour quality leaving one element was equal to the quality at the inlet to the following element.

- The frictional pressure drop was calculated for each individual incremental length, for which the quality was constant, using equation 2.42, 2.45, 2.52, or 2.56, depending on choice of correlation.
The total frictional pressure drop was the summation of each individual pressure drop, as given by equation 2.57:

\[ \Delta P_{fr, to} = \sum_{i=1}^{n} (\Delta P_{fr})_i \]  

(2.57)

2.3.4 Summary

Due to the sheer number of studies on frictional pressure drop in two-phase flows, not all methods and/or correlations could be included in this thesis. Those that were chosen, however, are amongst the most widely-cited in the literature. Despite this, it is difficult to ascertain the applicability of any one correlation for predicting the frictional pressure drop in a condensing flow. This is primarily due to a lack of condensing steam-water experimental data. A limited number of studies [108, 109] present data on steam condensation in plate heat exchangers, but such geometries are markedly different to the ones examined in this thesis. It appears that the majority of the data available in the literature for round tubes is either for adiabatic flows or for evaporating flows. Furthermore, refrigerants seem to be the working fluids most commonly employed in two-phase pressure drop studies.

A limited number of comparative studies have been carried-out to deduce the most suitable correlation for a particular set of conditions. Idsinga et al. [110] assessed the applicability of eighteen correlations for adiabatic steam-water flows at mass fluxes in the range of 270 - 4340 kg/sm². Leung et al. [111] presented measurements of steam-water flows in heated tubes spanning mass fluxes of 1000 - 10,000 kg/sm² and also compared their data, in terms of the two-phase multiplier. Tribbe and Müller-Steinhagen [107] compared some of the leading correlations to a large database of experimental data containing such fluid combinations as air-oil, air-water, water-steam, and several refrigerants. They found that, statistically, the method of Müller-Steinhagen and Heck gave the most accurate and reliable results. In another relatively recent comparative study, Ould Didi et al. [112] examined pressure drop data for evaporation of refrigerants over mass fluxes of 100 - 500 kg/sm² and found that the method of Müller-Steinhagen and Heck [7] and Grönnerud [6] compared best. The same authors also noted that disparity in predictions from various leading methods can be as large as 100%, thereby highlighting that no single correlation provides acceptable accuracy for general use. However, the caveat with these studies is the fact that the mass fluxes examined were two - three orders of magnitude greater than those examined in this thesis - which are representative of typical condensing flows in Rankine cycle ACCs. Therefore, it is difficult to apply any conclusions from those studies to the pressure drop measurements presented in this thesis. Hence, it was one of the aims of the work presented in this thesis to determine the most applicable correlation for predicting the frictional pressure drop of condensing flows of steam. The ability to predict the frictional component of a two-phase pressure drop will, ultimately, facilitate prediction of the overall pressure drop.
To close the review of literature relating to condensation heat transfer and pressure drop, table 2.3 is presented to provide an element of context to the work presented in this thesis. This table presents a comparison between the thermophysical properties of a sample range of popular refrigerants and the water working fluid examined in this thesis. As stated in Sections 2.2 and 2.3, the predominant working fluid in condensation studies in the literature is some variant of refrigerant, with little research undertaken on steam-water flows - particularly in horizontal tubes.

Table 2.3: Comparison of thermophysical properties of various working fluids in condensation studies, where all properties are taken from Rogers and Mayhew [13] at $T_s = 40 \, ^\circ C$ and $G = 5 \, kg/sm^2$

<table>
<thead>
<tr>
<th>Parameter</th>
<th>R-22</th>
<th>R-134a</th>
<th>R-410A</th>
<th>Ammonia</th>
<th>Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho_v$ (kg/m$^3$)</td>
<td>66.19</td>
<td>50.09</td>
<td>102.6</td>
<td>12.03</td>
<td>0.051</td>
</tr>
<tr>
<td>$\rho_l$ (kg/m$^3$)</td>
<td>1129</td>
<td>1147</td>
<td>979</td>
<td>579</td>
<td>992</td>
</tr>
<tr>
<td>$\mu_v$ (kg/s.m)</td>
<td>$13.52 \times 10^{-6}$</td>
<td>$12.55 \times 10^{-6}$</td>
<td>$15.21 \times 10^{-6}$</td>
<td>$10.33 \times 10^{-6}$</td>
<td>$10.31 \times 10^{-6}$</td>
</tr>
<tr>
<td>$\mu_l$ (kg/s.m)</td>
<td>$139 \times 10^{-6}$</td>
<td>$163 \times 10^{-6}$</td>
<td>$98 \times 10^{-6}$</td>
<td>$114 \times 10^{-6}$</td>
<td>$653 \times 10^{-6}$</td>
</tr>
<tr>
<td>$u_v$ (m/s)</td>
<td>0.07</td>
<td>0.11</td>
<td>0.05</td>
<td>0.42</td>
<td>98</td>
</tr>
</tbody>
</table>

From table 2.3, it can be seen that the thermophysical properties for the four different refrigerants are, by and large, quite similar. In all cases, there is at least an order of magnitude similarity. This also appears to be the case when comparing the thermophysical properties of the refrigerants with those of water. However, there is one noticeable discrepancy - the vapour density, $\rho_v$, which is 2-3 orders of magnitude smaller than the vapour densities of the refrigerants. This is important, as the relatively low value of vapour density results in a significant vapour velocity of $\approx 98 \, m/s$, in this instance. This value is much larger than the comparable refrigerant velocities, which are 2-3 orders of magnitude smaller. The implications of this on the literature reviewed in this section is discussed as follows.

The effect of vapour velocity has been well established in literature relating to two-phase flows. Throughout this chapter, these effects have been highlighted and described. In short, large vapour velocities lead to shear forces which tend to alter the morphology of the flow by reducing the tendency for stratification through distributing the liquid film around the inside tube perimeter. This action leads to a thinner film of liquid. Increased shear also forces the liquid phase towards the tube exit - again reducing the film thickness. In terms of heat transfer, increased vapour velocities promote the advection component of convection, as the bulk motion of the fluid continuum is heightened. As the vapour velocities associated with condensing flows of steam-water are much larger than those for refrigerants, at similar operating conditions, it is apparent that more studies on such flows are needed to augment current literature. Vapour velocity is a critical parameter which influences flow to-
condensing flows of steam-water need to be quantified in more detail. This is one of the primary objectives of this thesis.

Throughout the remainder of this thesis, a number of dimensionless parameters introduced in this chapter will be referred to. These parameters have been defined hitherto, but are listed here in table 2.4 to highlight the parameters which are of importance to this study.

Table 2.4: Dimensionless parameters utilised throughout this study

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Interpretation</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reynolds number, $Re$</td>
<td>Ratio of inertial to viscous forces</td>
<td>$\frac{GD}{\mu}$</td>
</tr>
<tr>
<td>Nusselt number, $Nu$</td>
<td>Ratio of convective to conductive heat transfer</td>
<td>$\frac{hD}{k}$</td>
</tr>
<tr>
<td>Prandtl number, $Pr$</td>
<td>Ratio of momentum diffusivity to thermal diffusivity</td>
<td>$\frac{\mu C_p}{k}$</td>
</tr>
<tr>
<td>Jakob number, $Ja$</td>
<td>Ratio of sensible to latent heat transfer</td>
<td>$\frac{C_p(T_s-T_w)}{h_fg}$</td>
</tr>
<tr>
<td>Froude number, $Fr$</td>
<td>Ratio of inertial to gravitational forces</td>
<td>$\frac{(G/\rho_l)^2}{gD}$</td>
</tr>
<tr>
<td>Weber number, $We$</td>
<td>Ratio of inertial to surface tension forces</td>
<td>$\frac{G^2D}{\rho_e\sigma}$</td>
</tr>
</tbody>
</table>
This chapter serves two main purposes - to present theory and relationships associated with the performance of heat exchangers, and to present a thermodynamic model to predict the performance of a given Rankine cycle thermoelectric power plant. The heat exchanger theory presented in this chapter is restricted to that relating to the air-side only, separate from the condensate-side theory which was presented in Chapter 2. This theory forms part of the thermodynamic model, in that it is incorporated to predict the performance of the condenser. Using a combination of fan-flow theory and empirical correlations/analytical solutions for forced convection, the aerodynamic and thermal characteristics on the air-side of various heat exchanger designs can be determined. This chapter addresses this theory and, subsequently, presents the thermodynamic model into which it is incorporated, to predict Rankine cycle performance.

3.1 Aerodynamic and thermal heat exchanger theory

Within this section, equations which express the performance of a single axial fan, and axial fans in parallel, are presented. Thereafter, empirical correlations to predict the pressure drop and heat transfer characteristics of annular-finned tube bundles are reproduced from the literature. Lastly, analytical models for predicting the pressure drop and heat transfer characteristics of plate-finned heat sinks are presented.

3.1.1 Axial fan theory

Generally, the aerodynamic performance of an axial fan is expressed in terms of volumetric flow rate, $\dot{V}$, and static pressure rise across the fan, $\Delta P_f$. These terms are usually given as ordered pairs which define the fan characteristic curve - a generic example of which is shown in figure 3.1. A characteristic curve expresses the pressure-flow performance of a given fan, at a given rotational speed, and is usually generated by the fan manufacturer in accordance with a standardized testing method [113, 114].

In the case of a variable speed fan, the characteristic curve at the nominal fan speed can be extrapolated to express the pressure-flow performance for any given rotational speed. Application of the fan laws, equations 3.1 & 3.2 from Bleier [115], for dimensionally similar fans allow the nominal fan characteristic curve to be generated for a range of fan speeds. An example illustrating this is given in figure 3.2.

$$\dot{V} \propto \omega D^3$$
3.1 Aerodynamic and Thermal Heat Exchanger Theory

Figure 3.1: An example of an axial fan characteristic curve

\[ \dot{V}_2 = \left( \frac{\omega_2}{\omega_1} \right) \dot{V}_1 \]  
\[ \Delta P_f \propto \rho \omega^2 D^2 \]  
\[ \Delta P_{f2} = \left( \frac{\omega_2}{\omega_1} \right)^2 \Delta P_{f1} \]

Figure 3.2: An example of an axial fan characteristic curve at different fan rotational speeds, where \( \omega_1 > \omega_2 \)

In addition, the power consumed by a fan is usually specified by the fan manufacturer, based on the nominal fan speed. Inherently, this power varies with fan speed, according to the following relationship from Bleier [115]:

\[ P_f \propto \rho \omega^3 D^5 \]
In equations 3.1 - 3.3, the air density, $\rho$, and fan diameter, $D$, terms cancel as they are constant for a given fan. Therefore the relationships defined in equations 3.1 - 3.3 are, for a given fan, solely a function of fan rotational speed.

For fans in parallel, i.e. fans arranged side-by-side in a common axial direction, each fan experiences the same magnitude of fluidic resistance and, thus, the static pressure rise for the system of fans remains the same as that for one individual fan. However, the total flow rate is equal to the sum of each individual fan flow rates. In this regard, the overall system characteristic curve retains the y-axis magnitude of a single axial fan but the x-axis is scaled through multiplication by the number of fans in the system.

For dimensionally dissimilar fans, the aerodynamic performance can be compared by scaling the fan flow rate and static pressure with fan diameter and rotation speed. Doing so yields the non-dimensional flow and pressure coefficients, as defined in the following equations taken from Bleier [115];

$$\phi = \frac{\dot{V}}{\omega D^3} \quad \psi = \frac{\Delta P_f}{\rho \omega^2 D^2}$$

(3.4)

When fans are integrated in a heat exchanger to form a forced-convection cooling solution, the flow of air experiences fluidic losses, due to friction, as the air flows through the heat exchanger geometry. This loss of energy is manifested as a pressure loss, and is compensated for by an increase in static pressure across the fan - thus satisfying conservation of energy. The following section is primarily concerned with quantifying the aerodynamic losses inherent in the heat exchanger designs studied in this thesis, and presents the material which is used to predict and calculate these losses.

### 3.1.2 Heat exchanger aerodynamic theory

The total pressure drop incurred by a flow of air through a given heat exchanger design is represented by equation 3.5 - from Shah and Sekulić [19].

$$\Delta P = \frac{\rho_i \bar{u}_i^2}{2g_c} \left[ \frac{1 - \sigma^2 + K_e}{1 - \sigma^2 + K_e} + \frac{2}{\rho_i - 1} \left( \frac{\rho_i}{\rho_o} \right) \right] + \frac{fL}{\tau_h \rho_m} \left( \frac{\rho_i}{\rho_o} \right) \left( 1 - \sigma^2 + K_e \right)$$

(3.5)

As can be seen, equation 3.5 is comprised of four constituent components, each one accounting for a different effect experienced by the flow through the heat exchanger. Namely, these effects are the entrance losses, momentum effects, core friction, and exit effects. The entrance losses arise from flow convergence as it enters the reduced
free-flow area of the heat exchanger. Flow separation, followed by irreversible free expansion, are manifested as fluidic losses at the entrance. The momentum term accounts for the decrease in density, and the consequent increase in mean velocity of the flow, due to heating. Pressure losses are incurred as a result of the flow acceleration. Core friction results from the flow experiencing skin friction and form drag, at the leading and trailing edges of individual fins, in the heat exchanger core. Both of these effects, inherently, lead to pressure losses. Finally, the exit effects represent the flow separation and divergence, again due to the change in free-flow area as the flow leaves the heat exchanger core. Exit effects result in a pressure rise, or loss, the advent of which generally depends on the flow conditions and heat exchanger geometry.

In equation 3.5, the momentum term is easily calculable through knowing the inlet and outlet air densities. The entrance losses and exit effects, however, are dependent on the contraction ratio, $\sigma$, Reynolds number, and flow cross-sectional geometry. The loss coefficients, $K_c$ and $K_e$, are graphically presented in Kays and London [116] for multiple heat exchanger geometry types. However, not all geometries are represented in the charts of Kays and London. Thus, Culham and Muzychka [117] presented an alternative approach for determining the loss coefficients, which was based on a solution originally proposed by White [118] for flow in pipes. These terms are given as follows:

\[
K_c = 0.42 \left(1 - \sigma^2\right) \quad (3.6)
\]

\[
K_e = \left(1 - \sigma^2\right)^2 \quad (3.7)
\]

For the specific case of heat exchanger cores comprised of staggered, multi-row, banks of tubes (such as the annular-finned circular tube MACC designs), it is recognised that additional flow losses occur inside the core due to the flow periodically accelerating and decelerating through the tube bundle as the free flow area changes. This phenomenon is similar in principle to the bulk flow convergence and divergence on entering and exiting the core, respectively. Due to this similarity, all fluidic effects induced by flow area discontinuities are often lumped together into the friction factor, $f$. In doing so, the friction factor accounts for all losses other than momentum effects. This approach simplifies equation 3.5 to the following form:

\[
\Delta P = \frac{\rho_i \bar{u}_i^2}{2 g_c} \left[\frac{\text{momentum effect}}{2 \left(\frac{\rho_i}{\rho_o} - 1\right)} + \frac{f L}{r_h \rho_m} \right] \quad (3.8)
\]

Equation 3.8 is specifically employed for calculating the pressure drop associated with multi-row heat exchangers, whereas equation 3.5 is the more general solution, applicable to single row heat exchangers such as the plate-finned rectangular tube MACC design. Ultimately, the respective equation is incorporated into the thermo-
dynamic model in Section 3.3, depending on the heat exchanger geometry specified. The dominant term in equations 3.5 and 3.8 and, hence, the main contributor to pressure loss is the core friction. Generally, this accounts for more than 90% of the total pressure drop [19]. The friction factor in the core friction term is usually derived experimentally and, hence, a range of empirical correlations are available to predict this - depending on the application of interest. Alternatively, analytical models exist, for somewhat simpler heat exchanger geometries, which allow the friction factor to be calculated. Correlations and analytical solutions relevant to the geometries examined in this thesis are given in table 3.1, as follows.

Table 3.1: Dimensionless pressure drop empirical correlations and analytical model

<table>
<thead>
<tr>
<th>Heat Exchanger</th>
<th>Correlation/Equation</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>6 row - annular</td>
<td>[ f = 9.465Re^{-0.316} \left( \frac{X_t}{d_e} \right)^{-0.927} \left( \frac{X_t}{X_d} \right)^{0.515} \left( \frac{n_r D_L}{L} \right) ]</td>
<td>[119]</td>
</tr>
<tr>
<td>4 row - annular</td>
<td>[ f = 2.12Re^{-0.25} \left( \frac{A_o}{A_t} \right)^{-0.55} \left( \frac{d_f}{d_e} \right)^{-0.25} K_p ]</td>
<td>[120]</td>
</tr>
<tr>
<td>1 row - plate</td>
<td>[ f \cdot Re_{Dh} = \left[ \left( f \cdot Re_{Dh(dev)} \right)^2 + \left( f \cdot Re_{Dh(fd)} \right) \right]^{1/2} ]</td>
<td>[121]</td>
</tr>
</tbody>
</table>

For the six and four row circular tube heat exchanger designs, empirically-based correlations from Robinson and Briggs [119] and Nir [120], respectively, are recommended in the literature for calculating the friction factor. The terms in each correlation listed in table 3.1 are given in the nomenclature. However, an additional noteworthy term in the correlation of Nir is the correction factor - \( K_p \). As Nir’s correlation is for the row-averaged friction factor, row effects are not explicitly accounted for. \( K_p \) is, hence, included to account for the effects of different numbers of tube rows and is defined as:

\[ K_p = 2.08 - 0.83 \left( \frac{A_o - d}{A_o - t} \right) \]  

(3.9)

As opposed to multi-row heat exchanger designs, the flow through the single row plate-finned design is considerably more straightforward and, therefore, can be modelled. Essentially, the overall, bulk flow consists of a series of local, sub-flows through adjacent rectangular channels - formed by the extended fin surface. As such, the flow is modelled as laminar channel flow, which is developing in the entrance region and is fully-developed at some point downstream. Muzychka and Yovanovich [121] developed a general model, which employs the Churchill and Usagi [122] addition of asymptotes method to combine the aerodynamically developing and fully-developed equations. This composite solution is given in terms of the friction factor-Reynolds number product shown in table 3.1. The developing term is independent of the duct shape and is expressed as follows;
3.1 AERODYNAMIC AND THERMAL HEAT EXCHANGER THEORY

\[
f \cdot Re_{Dh(dev)} = \frac{3.44}{\sqrt{L^*_a}}
\]  

(3.10)

where \( L^*_a \) is the dimensionless aerodynamic axial entrance length for developing flow, and is defined as:

\[
L^*_a = \frac{L}{D_h Re_{Dh}}
\]  

(3.11)

The fully-developed term is dependent on the duct geometry - in this case a function of the aspect ratio of the rectangular channel as shown in equation 3.12;

\[
f \cdot Re_{Dh(fd)} = 24 - 32.527 \left( \frac{b}{H} \right) + 46.721 \left( \frac{b}{H} \right)^2 - 40.829 \left( \frac{b}{H} \right)^3 \\
+ 22.954 \left( \frac{b}{H} \right)^4 - 6.089 \left( \frac{b}{H} \right)^5
\]  

(3.12)

The accuracy of the correlations, and model, outlined thus far, and their applicability to the relevant heat exchanger, has been examined in two separate, but complimentary, studies of which the author was a part [123, 124]. In each case, the use of the correlation/model was validated through comparison with experimental measurements. Therefore, the friction factor for each heat exchanger design can be calculated with confidence in its accuracy. Subsequently, the relevant friction factor can be incorporated into equation 3.5 and 3.8 to calculate the pressure drop for the single row rectangular tube MACC and the multi-row circular tube MACC, respectively.

The pressure drop associated with a particular heat exchanger design equates to the resistance to flow through that given system, at a certain flow rate. This system resistance can be calculated for a range of arbitrary flow rates - giving the system resistance curve. When this is superimposed on the fan characteristic plot, the intersection between the two curves defines the operating point of the system - that is the pressure drop and flow rate at a particular fan speed. As outlined in Section 3.1.1, for the case of a variable speed fan, a range of characteristic curves can be plotted. Therefore, as can be seen in figure 3.3, a range of heat exchanger operating points exist from the intersection between the series of fan curves and the single system resistance curve.

The abscissa of each operating point establishes the flow rate of air, provided by the fan(s), through the heat exchanger. Once established, this allows the thermal characteristics, on the air-side, to be determined as outlined in the following section.

3.1.3 Heat exchanger thermal theory

Similar to the aerodynamic characteristics, the air-side heat transfer characteristics differ greatly depending on the heat exchanger design. The flow through the six row and four row annular-finned heat exchangers is chaotic and characterised by
separation, recirculation, and reattachment. It is very difficult to accurately model such flows and, as a result, experimentally-based correlations are relied upon to predict the heat transfer characteristics. The most robust and relevant correlations for the multi-row, annular-finned heat exchanger geometries examined in this thesis are presented in terms of the dimensionless Nusselt number and Stanton-Prandtl product, given in table 3.2.

In contrast to the multi-row tube bundles, the flow through the individual rectangular channels in the plate-finned heat exchanger is much more structured and ordered. As a consequence, the heat transfer, similar to the aerodynamics, can be modelled analytically. As the combined entry problem (simultaneously developing flows) is encountered, the model takes the same form as that for the aerodynamic model, i.e. asymptotic blending of the developing and fully-developed equations. This model is given in terms of the Nusselt number in table 3.2.

Table 3.2: Dimensionless heat transfer empirical correlations and analytical model

<table>
<thead>
<tr>
<th>Heat Exchanger</th>
<th>Correlation/Equation</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>6 row - annular</td>
<td>$Nu = 0.134Re^{0.681}Pr^{0.33} \left[ \frac{2(S_f-\delta_f)}{d_f-d_e} \right]^{0.2} \left( \frac{S_f-\delta_f}{\delta_f} \right)^{0.1134}$</td>
<td>[125]</td>
</tr>
<tr>
<td>4 row - annular</td>
<td>$StPr^{2/3} = Re^{-0.4} \left( \frac{d_f}{d_e} \right)^{-0.266} R_b^{-0.4} \left( \frac{d_f}{d_e} \right)^{-0.4} K_h$</td>
<td>[120]</td>
</tr>
<tr>
<td>1 row - plate</td>
<td>$Nu = \left[ (Nu_{dev})^3 + (Nu_{fd})^3 \right]^{1/3}$</td>
<td>[126]</td>
</tr>
</tbody>
</table>

The correlation of Briggs and Young [125] is one of the most widely-cited heat transfer correlations for annular-finned tube bundles, and is recommended for geometries consisting of four tube rows or more. The correlation of Nir [120] is expressed
in terms of the Stanton-Prandtl product (the Colburn factor, $j$), which is analogous to the Nusselt number. Nir correlated experimental data with a dimensionless characteristic length of the heat transfer surface - $R_b$. This is a ratio of the total free-flow area to the free-flow area inside a fin tip, and is defined in equation 3.13. Nir also introduced a correction factor, $K_h$, to compensate for the reduction in heat transfer associated with fewer tube rows. For a four row heat exchanger, $K_h = 0.97$.

$$R_b = \frac{X_t - d_f + [(d_f - d_e) (1 - \delta_f N_f)]}{[(d_f - d_e) (1 - \delta_f N_f)]}$$

(3.13)

The form of the analytical solution for modelling the heat transfer in the plate-fin design was proposed by Teertstra et al. [126]. They adopted the same approach as Muzychka and Yovanovich [121] in employing Churchill and Usagi’s [122] addition of asymptotes technique to form the composite solution given in table 3.2. A graphical example illustrating this solution is given in figure 3.4.

![Nusselt number solution as given by the addition of asymptotes method](image)

For the developing flow region, Teertstra et al. used the analytical expression originally formulated by Sparrow [127] using the Karman-Pohlhausen integral method for laminar forced convection in flat, rectangular ducts. This is given as;

$$Nu_{dev} = \frac{0.664}{\sqrt{L_{th} Pr^{1/6}}} \left( 1 + 7.3 \sqrt{Pr L_{th}} \right)^{1/2}$$

(3.14)

where $L_{th}^*$ is the dimensionless thermal axial entrance length for developing flow, and is equivalent to the inverse of the Graetz number. Equation 3.15 defines this as follows;

$$L_{th}^* = G z^{-1} = \frac{L}{D_h Pr Re_{Dh}}$$

(3.15)

For the fully-developed flow region, the Nusselt number is constant for a given duct but is, however, dependent on the particular duct geometry. Shah and Sekilić
[19] proposed an analytical expression for rectangular passages, based on the aspect ratio. This is given as;

\[
Nu_{fd} = 7.541 \left[ 1 - 2.61 \left( \frac{b}{H} \right) + 4.97 \left( \frac{b}{H} \right)^2 - 5.119 \left( \frac{b}{H} \right)^3 \\
+ 2.702 \left( \frac{b}{H} \right)^4 - 0.548 \left( \frac{b}{H} \right)^5 \right]
\] (3.16)

From table 3.2, the Nusselt number and/or Stanton number can be calculated for any of the heat exchanger designs investigated in this study. Consequently, the air-side heat transfer coefficient for any of the designs can be determined. The Nusselt number, and Stanton number, is related to the heat transfer coefficient through equation 3.17, and 3.18, respectively.

\[
Nu = \frac{hL}{k}
\] (3.17)

\[
St = \frac{Nu}{RePr} = \frac{h}{\rho \bar{u} C_p}
\] (3.18)

It is important to note that the validity and applicability of the heat transfer correlations, and equation, given in table 3.2 were examined in separate studies, contributed to by the author, in Moore et al. [123, 124]. Good agreement was generally observed between the experimental measurements and the relevant correlation/equation, ensuring confidence in their use in any thermodynamic modelling. Depending on the heat exchanger geometry set in Section 3.3, either equation 3.17 or equation 3.18 is incorporated into the thermodynamic model to evaluate the air-side heat transfer coefficient and, ultimately, condenser temperature.

### 3.2 HEAT EXCHANGER MODELLING

This section presents a novel effectiveness-NTU (\(\varepsilon\)-NTU) method for predicting the performance of a variable-speed ACC, in terms of terminal temperature and pressure. This model makes use of the heat exchanger theory presented thus far, with the main parameter incorporated being the air-side heat transfer coefficient. The \(\varepsilon\)-NTU model is based solely on air-side theory, with the analysis purposely and consciously neglecting to account for condensate-side phenomena such as the condensate film, tube fouling, air leakage, etc. In doing so, the condensate-side thermal resistance is disregarded, thereby providing an “idealised” response for comparison with experimental measurements from the MACC. It should be noted that a number of studies, related to modelling and predicting ACC performance, in the literature seem to be based around this approach of air-side theory only [51, 48, 46]. Through comparison with experimental measurements presented in Chapter 6, Section 6.1, this thesis
seeks to highlight the inherent flaws in those studies, that arise from neglecting to account for condensate-side thermal resistance in ACC modelling.

To satisfy the energy balance, this analysis is based on the assumption that only isothermal heat rejection is occurring during condensation - that is sensible heat rejection (de-superheating/subcooling) is neglected. Therefore, assuming steady-state, application of the 1st law of thermodynamics on the condensing-side yields the following enthalpy rate equation;

\[ \dot{Q}_{\text{rej}} = \dot{m}_c \Delta h = \dot{m}_c h_{fg} \]  

(3.19)

where the change in enthalpy, \( \Delta h \), for an isothermal, two-phase condensing flow is quantified by the loss in enthalpy of vaporization - \( h_{fg} \). In a similar fashion, applying the 1st law on the air-side yields;

\[ Q_{\text{rec}} = \dot{m}_a \Delta h = \dot{m}_a C_p (T_{\text{out}} - T_{\infty}) \]  

(3.20)

where, for a single-phase flow, a temperature change from inlet to outlet occurs as a result of sensible heating. An energy balance on the ACC imposes that the energy rejected during condensation must be absorbed by the air flow. Equating the loss of enthalpy from the steam to the gain in enthalpy in the air;

\[ Q = \dot{m}_c h_{fg} = \dot{m}_a C_p (T_{\text{out}} - T_{\infty}) \]  

(3.21)

For a cross-flow air-cooled heat exchanger, the effectiveness is defined as;

\[ \varepsilon = \frac{Q}{Q_{\text{max}}} = \frac{\dot{m}_c h_{fg}}{\dot{m}_a C_p (T_s - T_{\infty})} = \frac{\dot{m}_a C_p (T_{\text{out}} - T_{\infty})}{\dot{m}_a C_p (T_s - T_{\infty})} \]  

(3.22)

where the effectiveness of a condenser can also be expressed in terms of the Number of Transfer Units (NTU) as follows;

\[ \varepsilon = 1 - \exp^{-NTU} = 1 - \exp^{-\frac{\eta_o a h_a A_a}{\dot{m}_a C_p}} \]  

(3.23)

From equation 3.23, it can be seen that only terms relating to the air-side constitute the NTU. The air-side film coefficient, \( h_a \), is calculated using the previously presented equations 3.17 or 3.18, depending on the type of heat exchanger being modelled. The mass flow rate of air, \( \dot{m}_a \), is evaluated from the appropriate pressure drop equation (equation 3.5 or 3.8) combined with the fan characteristics, as shown heretofore in figure 3.3. The air-side effectiveness, \( \eta_{o,a} \), is a weighted average of the effectiveness of the prime surface, and less than 100% of the fin surface, as shown in equation 3.24;

\[ \eta_{o,a} = 1 - \frac{A_f}{A} (1 - \eta_f) \]  

(3.24)

where the fin effectiveness, \( \eta_f \), accounts for temperature gradients across the fin. Gardner [128] provides a simplified method for evaluating the fin efficiency, based on the fin geometry and air-side heat transfer coefficient;
\[ \eta_f = \frac{\tanh (mL)}{mL}; \quad \text{where} \quad m = \sqrt{\frac{2h_a}{k_f}} \quad (3.25) \]

Equating the effectiveness terms in equations 3.22 and 3.23, and re-arranging to derive an expression for \( T_s \), the condenser temperature:

\[ T_s = T_{\text{HEX}} = \frac{\dot{m}_c h_f}{1 - \exp^{-NTU}} + T_\infty \quad (3.26) \]

where \( T_s \) is actually the temperature at the boundary of the air film (condenser wall temperature), as this analysis is based upon air-side aerodynamic and thermal theory only. Hence, it is designated \( T_{\text{HEX}} \). However, as condensate-side phenomena, and their associated thermal resistances, inside the tubes are purposely neglected in this analysis, the condenser wall temperature and steam temperature are assumed equal. Equation 3.26, therefore, permits the theoretical condenser temperature to be calculated for a given condenser size and ambient temperature. The variation in temperature with fan speed is accounted for through the air-side conditions (mass flow rate and heat transfer coefficient), which are implicit in the model as they make up the \( NTU \) term in equation 3.26. Ultimately, the condenser pressure can easily be determined from the calculated temperature using equation 3.27, which assumes saturation conditions, and is derived from a curve-fit to data taken from the steam tables \[13\].

\[ P_s = 0.001095T_s^4 - 0.0595T_s^3 + 5.291T_s^2 - 21.686T_s + 881.17 \quad (3.27) \]

Finally, the air-side thermal resistance can be predicted from the calculated temperature, and the heat rejection rate and ambient temperature upon which it is based, through equation 3.33 as follows:

\[ (R_{th})_a = \frac{\Delta T}{Q} = \frac{T_{\text{HEX}} - T_\infty}{\dot{m}_c h_f} \quad (3.28) \]

### 3.3 Thermodynamic Modelling

The core aim of the thermodynamic modelling was to investigate the effects of various condenser parameters on plant performance. Thus, the variables included in, and varied systematically during, the analysis were limited to those solely relating to the condenser. In this regard, only the relationship between the condenser and the steam turbine was considered. Steam turbine inlet conditions, such as steam temperature and flow rate, were assumed constant at values representative of those according to the given steam turbine application.

To ensure a realistic analysis, the performance characteristics from two operational steam turbines can be incorporated into the model. Both turbines are typical of those found in small-medium CSP plants - the target application for the MACC. Indeed,
the smaller of these turbines (~ 20 MW) is currently installed in a central tower type CSP plant - “Gemasolar” - in southern Spain. The performance data for this turbine were provided by a MACCSol consortium partner, and are presented in figure 3.5a, where the turbine gross output and condenser heat rejection are expressed as a function of condenser temperature. The characteristics of the larger turbine (~ 50 MW), which were previously presented in [129, 130], were generated from a parabolic trough CSP plant, and are re-produced here in figure 3.5b. These characteristics were also provided by a MACCSol project partner. All thermodynamic calculations presented in this thesis are based on the ~ 50 MW turbine, unless otherwise stated.

The nominal operating exit conditions of the turbines are summarised in table 3.3.
Table 3.3: Nominal exit conditions of ~ 20 MW and ~ 50 MW steam turbines

<table>
<thead>
<tr>
<th>Parameter</th>
<th>~ 20 MW turbine</th>
<th>~ 50 MW turbine</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{m}_s$ (kg/s)</td>
<td>9.98</td>
<td>39.1</td>
</tr>
<tr>
<td>$T_s$ (K)</td>
<td>313.5</td>
<td>338.3</td>
</tr>
<tr>
<td>$P_s$ (Pa)</td>
<td>7500</td>
<td>25,200</td>
</tr>
</tbody>
</table>

3.3.1 Modelling methodology

The following series of steps is an overview of the modelling procedure to generate thermodynamic data for a Rankine cycle thermoelectric power plant with a MACC system installed. This sequence of steps was part of an algorithm, which was run in a user-defined programme in MATLAB, the m-file script for which is available in Appendix C.

1. The plant size is set, i.e. the steam turbine characteristics from either figure 3.5a or 3.5b are integrated into the model.

2. The heat exchanger geometry (MACC design) is chosen.

3. Ambient temperature and condenser size, i.e. number of MACC modules, are set.

4. The fan rotational speed is set, and the corresponding fan characteristic curve is generated from the fan laws (equations 3.1 & 3.2) given in Section 3.1.

5. The system resistance curve for the chosen heat exchanger geometry is calculated for a range of arbitrary flow rates using equation 3.5 or 3.8, from Section 3.1.

6. The fan characteristic curve and the system resistance curve are plotted, with the point of intersection outputting the actual flow rate through the heat exchanger at that given fan speed, set in Step 4. This step was previously shown in graphical form in figure 3.3.

7. The relevant dimensionless heat transfer characteristic for the chosen heat exchanger is calculated from equation 3.17 or 3.18 in Section 3.1, for the flow rate established in Step 6. From this, the air-side heat transfer coefficient is, subsequently, calculated.

8. The condenser temperature is determined using equation 3.29, which governs the relationship between condenser temperature, condenser heat rejection and air-side heat transfer coefficient. $\dot{Q}_{rej}$ in equation 3.29 is a function of $T_n$, being defined from a polynomial fit to the heat rejection data given in figure 3.5a or 3.5b, depending on Step 1.
\[ \frac{Q_{rej}}{\dot{m}_a C_p (T_s - T_\infty)} = 1 - \exp \left( -\frac{\dot{m}_a h_a \Delta u}{\dot{m}_a C_p} \right) \] (3.29)

9. The steam turbine outlet temperature is taken to be equal to the condenser temperature and, hence, the gross output, \( P_{gr} \), is determined from a polynomial fit to the gross power data for the respective steam turbine, given in figure 3.5a or 3.5b.

10. Fan power consumption, \( P_f \), is calculated for the set fan speed using equation 3.3, in conjunction with the fan manufacturer’s data. Subsequently, the total fan power, \( P_{ft} \), is calculated using equation 3.30 as follows - where the number of fans, \( N_f \), is dictated by the condenser size (number of modules);

\[ P_{ft} = P_f \times N_f \] (3.30)

11. Net plant output is calculated by subtracting the total fan power consumption from the gross power output, i.e;

\[ P_{net} = P_{gr} - P_{ft} \] (3.31)

12. A plot of net plant output as a function of fan speed is generated by repeating Steps 4-11.

13. The plot of net plant output versus fan speed, outputted in Step 12, is generated for a range of ambient temperatures and/or condenser sizes by changing Step 3, and repeating Step 12.

14. A different steam turbine (plant size) and/or heat exchanger geometry can be modelled by editing Step 1 and/or Step 2, respectively, and repeating Step 13.

3.3.2 Semi-empirical modelling methodology

The following procedure is very similar to the one just outlined in the previous section, albeit with one main difference - experimental data is incorporated. Experimental measurements taken from full-scale MACC condensers, which are presented in Chapter 6, are integrated into the model to establish the condenser temperature. This methodology is described as follows;

- Steps 1 - 3 outlined in Section 3.3.1 are repeated.
- Steps 4 - 8 are replaced by importing the measured variation in temperature, as a function of fan speed, data for the chosen condenser design.
  - Measured data is modified for the thermodynamic analysis. The variation in temperature measurements were obtained from a single MACC module, where the steam at the inlet was deliberately maintained in a superheated
state. In application, however, the steam will not be completely dry at the condenser inlet, and this must be accounted for to ensure measurements are applicable to a plant analysis.

– Accounting for the actual variation in condenser heat rejection as condenser temperature varies is achieved using equation 3.32:

\[ T_s^* = (Q_{\text{rej}} \times R_{th}) + T_\infty \] (3.32)

– where \( T_s^* \) is the modified condenser temperature. \( Q_{\text{rej}} \) is the heat rejection which, as in Step 8 in the previous modelling procedure, is defined from a polynomial curve-fit to the heat rejection data in figure 3.5a or figure 3.5b - depending on choice of turbine. \( R_{th} \) is the absolute thermal resistance, established from the measurements and is expressed as;

\[ R_{th} = \frac{\Delta T_m}{Q_m} = \frac{T_s - T_\infty}{\dot{m}_c \times h_{fg}} \] (3.33)

- Step 9 in the previous methodology is modified insofar that the gross output is now determined from the temperatures given by equation 3.32.

- Steps 10 - 14 remain unchanged.

3.4 summary

This chapter serves to compliment the condensate-side theory, presented in Chapter 2, by presenting theory related to the air-side of heat exchangers only. This theory was identified through a comprehensive review of the literature, with its applicability and validity established through experimentation in several separate studies of which the author was a part. A novel \( e \)-NTU model was developed, into which the air-side theory can be integrated to purposely predict the performance of ACCs solely based on an air-side analysis. The results from this model are compared with experimental measurements of condenser performance in Chapter 6 - to highlight the limitations of current air-side only modelling approaches. To close the chapter, a thermodynamic model was presented. Again, air-side theory can be integrated into this, in conjunction with experimentally-derived measurements to determine the implications on plant performance. The outcomes from this investigation are given in Chapter 8.
This chapter describes the experimental facilities used, and methodologies employed, to investigate the condensate-side characteristics and performance of various prototype MACC modules. To open the chapter, the MACC prototype design is introduced, accompanied by an overview of the original, pre-existing test facility. A number of unforeseen fluidic and thermal phenomena were observed during the preliminary stages of testing, which had an adverse effect on condenser performance. As such, significant alterations and modifications to the pre-existing design were carried-out to facilitate the condensate-side characterisation measurements. These issues and subsequent resolutions are outlined herein. Following this, the specific experimental arrangement and procedure for condensate-side pressure drop and heat transfer measurements is described, accompanied by the data analysis. Closing this chapter is the experimental uncertainty.

4.1 Overview of Pre-Existing Test Facility

Measurements were carried-out on a full-scale prototype MACC module to investigate the condensing characteristics at a system level. All measurements were acquired under representative Rankine cycle conditions and, thus, the measurements provided valuable insight into the MACC performance expected in operation. The MACC prototype module, on which the measurements were taken, was a 2 m x 2 m cross-flow ACC, essentially consisting of a compact heat exchanger coupled to a bank of variable speed axial fans. Steam flowed through the inside of the tubes, in the heat exchanger core, whilst air flowed in a direction transverse to this, through the fin channels on the exterior of the tubes. Steam entered the tubes via an inlet manifold and exited as condensate through the condensate manifold, where it was subsequently collected in a condensate tank. The module was integrated into a steel support structure which provided the capability to vary the condenser angle of inclination. An overview of the MACC module is shown in figure 4.1.

A bank of four *ebm-papst HyBlade* axial fans provided the air flow. The fans were nominally fixed in an induced-draft (pulling air over the tubes) configuration - as indicated in figure 4.1a. The main advantage of the fans is their small size - relative to conventional ACC fans. With a diameter of 0.91 m, the fans are considerably smaller than the traditional 9 - 12 m fans found in typical A-frame ACCs. The *HyBlade* fan is a proprietary hybrid design where an aluminium support structure is mated with fibre-glass reinforced plastic. This lightweight, compact design, coupled with a high efficiency EC (electronically commutated) motor, allows for the rotational...
speed of the fans to be varied instantaneously, with little lag. Therefore, a range of fan speeds are possible, the effect of which on the condensate-side characteristics is not well documented in the literature. Hence, fan speed was one of the main parameters investigated in this study. Figure 4.2 is an image of one such fan installed on the MACC, accompanied by the pressure-flow characteristics - provided by the fan manufacturer for a nominal rotational speed of 1000 rpm.

In this study, the condensate characteristics of a series of heat exchanger geometries were investigated, with each geometry corresponding to a specific MACC design. In its initial form, the MACC heat sink consisted of 183 annular-finned circular tubes. These tubes were arranged in a staggered 6 row configuration. A second iteration was manufactured by removing two of these tube rows, thus reducing the MACC to a 4 row heat exchanger consisting of 122 annular-finned circular tubes. These tubes retained the staggered configuration of the 6 row design. The final heat
exchanger geometry investigated was a plate-finned, rectangular tube design, which was markedly different from the previous geometries and is more akin to the current industry standard in conventional A-frame ACCs. This design was arranged in a standard 1 row configuration. Figure 4.3 illustrates the two different heat exchanger geometries, with the main geometrical features of each particular geometry highlighted. Table 4.1 summarises the pertinent geometrical parameters and their dimensions, with material properties taken from Kays and London [116].

Table 4.1: Geometrical features of heat exchanger geometries examined herein

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A ) (m(^2))</td>
<td>Tube cross-sectional area</td>
<td>5.94×10(^{-4})</td>
</tr>
<tr>
<td>( D ) (m)</td>
<td>Inside tube diameter</td>
<td>0.0275</td>
</tr>
<tr>
<td>( d_e ) (m)</td>
<td>External tube diameter</td>
<td>0.03175</td>
</tr>
<tr>
<td>( d_f ) (m)</td>
<td>Fin diameter</td>
<td>0.0635</td>
</tr>
<tr>
<td>( \delta_f ) (m)</td>
<td>Fin thickness</td>
<td>0.00041</td>
</tr>
<tr>
<td>( H_f ) (m)</td>
<td>Fin height</td>
<td>0.01588</td>
</tr>
<tr>
<td>( k_{\text{fin}} ) (W/mK) (al)</td>
<td>Thermal conductivity of fin</td>
<td>200</td>
</tr>
<tr>
<td>( k_{\text{tube}} ) (W/mK) (st)</td>
<td>Thermal conductivity of tube</td>
<td>50</td>
</tr>
<tr>
<td>( L_t ) (m)</td>
<td>Tube length</td>
<td>2.15</td>
</tr>
<tr>
<td>( N_r ) (-)</td>
<td>Number of tube rows</td>
<td>6/4</td>
</tr>
<tr>
<td>( N_t ) (-)</td>
<td>Number of tubes</td>
<td>183/122</td>
</tr>
<tr>
<td>( S_f ) (m)</td>
<td>Fin spacing</td>
<td>0.00254</td>
</tr>
<tr>
<td>( X_d ) (m)</td>
<td>Diagonal tube spacing</td>
<td>0.0645</td>
</tr>
<tr>
<td>( X_l ) (m)</td>
<td>Longitudinal tube spacing</td>
<td>0.0559</td>
</tr>
<tr>
<td>( X_t ) (m)</td>
<td>Transverse tube spacing</td>
<td>0.0645</td>
</tr>
</tbody>
</table>

(b) Plate-finned, rectangular tube geometry

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A ) (m(^2))</td>
<td>Tube cross-sectional area</td>
<td>0.0044</td>
</tr>
<tr>
<td>( b ) (m)</td>
<td>Fin spacing</td>
<td>0.00235</td>
</tr>
<tr>
<td>( \delta_f ) (m)</td>
<td>Fin thickness</td>
<td>0.00025</td>
</tr>
<tr>
<td>( H_f ) (m)</td>
<td>Fin height</td>
<td>0.0186</td>
</tr>
<tr>
<td>( H_t ) (m)</td>
<td>Tube height</td>
<td>0.22</td>
</tr>
<tr>
<td>( k_{\text{fin}} ) (W/mK) (al)</td>
<td>Thermal conductivity of fin</td>
<td>200</td>
</tr>
<tr>
<td>( k_{\text{tube}} ) (W/mK) (st)</td>
<td>Thermal conductivity of tube</td>
<td>50</td>
</tr>
<tr>
<td>( L_f ) (m)</td>
<td>Fin length</td>
<td>0.2</td>
</tr>
<tr>
<td>( L_t ) (m)</td>
<td>Tube length</td>
<td>2.15</td>
</tr>
<tr>
<td>( N_r ) (-)</td>
<td>Number of tube rows</td>
<td>1</td>
</tr>
<tr>
<td>( W ) (m)</td>
<td>Tube width</td>
<td>0.02</td>
</tr>
</tbody>
</table>
Figure 4.3: MACC module with heat exchanger geometries considered in this study. Pertinent geometrical parameters are highlighted.
The MACC module was integrated into a steam and condensate loop which provided steam, and facilitated the removal of condensate. Slightly superheated steam (4 - 5 K > $T_s$) was supplied to the MACC from a commercial steam boiler, via a series of valves and ancillary steam components - to ensure satisfactory steam conditions. The degree of superheat was controlled using a water injector system, located upstream of the MACC inlet manifold. The condensate (exit) manifold of the MACC was connected to a condensate tank, into which flowed subcooled condensate (7 - 14 K < $T_s$). Directly downstream of the tank was a condensate pump which pumped the condensate into the boiler feedwater tank, thereby creating a closed loop. A non-return valve located downstream of the condensate pump, in the return line, prevented any backflow from the higher pressure reservoir into the condensate tank or MACC module. Specific details on the steam and condensate loop are provided in Appendix A.

4.2 EXPERIMENTAL SET-UP FOR CONDENSATE CHARACTERISATION

The test facility described thus far largely pre-existed the author’s involvement, and was originally deemed to be satisfactory for quantifying the condensate-side characteristics. However, during the course of preliminary testing carried-out by the author, a number of issues arose which needed to be resolved through certain modifications to the test configuration. These issues, and the resulting modifications and resolutions, are described in this section.

4.2.1 Issues: steam-side phenomena

One of the biggest issues encountered during initial experiments was air ingress. This is an issue that appears to be inherent in all operational condensers, whether they be ACCs or shell-and-tube, and has been highlighted in ACC design guidelines as a problem which must be resolved for efficient ACC operation [41]. Generally, all condensers operate at sub-atmospheric pressure (vacuum conditions) to maximise thermodynamic efficiency. However, as the pressure inside the condenser is much lower than the surrounding air at atmospheric pressure, a negative pressure differential exists which promotes air leakage into the condenser. This issue was prevalent in the MACC and, despite best efforts to seal various leakage sites, air ingress persisted. The main disadvantage of air leaking into the system was an inability to maintain the desired vacuum. The vacuum in the MACC would simply decay over time as air constantly leaked into the system, with the system eventually reaching atmospheric pressure. This vacuum decay issue is depicted in the measurements in figure 4.4.

The measurements presented in figure 4.4 were acquired by condensing a fixed quantity of steam in the MACC, and observing the variation in pressure and temperature. This was done by isolating the MACC from the steam flow through closing
4.2 Experimental Set-Up for Condensate Characterisation

the main inlet and exit valves. As a result, a fixed mass of steam remained in the MACC, which was condensed by the cooling air flow from the fans. As can be seen in figure 4.4, after a brief period of pressure reduction, the measured pressure gradually increases due to the accumulation of incoming air. Ideally, it would follow the trend of the saturation pressure - which is calculated from the measured steam temperature. The difference between the saturation pressure and measured pressure must, therefore, be due to the contribution of the partial pressure of air.

The issue of air ingress was compounded by the presence of a phenomenon known as backflow, which predominantly occurred in the multi-row, circular tube MACC prototypes. This phenomenon was not just specific to the MACC however, as all single-pass, multi-row, cross-flow ACCs suffer from a design flaw known as the “row effect” which, ultimately, is responsible for backflow. The row effect is a condition whereby the ambient air gets warmer as it progresses downstream, through successive tube rows. In a single-pass, multi-row, cross-flow ACC, the first row in the air flow direction will be exposed to the lowest air temperature and, hence, the largest temperature difference will be present in this row. Subsequent rows will be subjected to elevated air temperatures, with the temperature increasing from row-to-row as heat is added from the condensing steam. Since the quantity of steam condensed in any given row is proportional to the temperature difference between the saturated steam and the ambient air, it follows that the rows exposed to lower air temperatures (i.e. the leading rows) will condense more steam than the downstream rows. The differing condensation rates lead to non-uniform pressure gradients in the tube rows, with the leading rows having a larger pressure drop than the downstream rows. The non-uniform pressure gradients cause non-condensed steam, from downstream rows which have a smaller pressure drop, to flow into the common condensate manifold and “back-up” the leading tube rows which have a larger pressure drop. Thus, a situation arises where steam enters tubes in the leading rows from both ends. This undesirable phenomenon is known as backflow and is illustrated in figure 4.5.
Backflow, combined with air ingress leads to the formation of “air pockets” which are non-condensable zones within the condenser tubes. These regions develop as air becomes trapped between the condensing flows of steam entering from both ends of a given tube. Very little heat transfer takes place in these non-condensable zones and, accordingly, the effective heat transfer area of the condenser is reduced. In this respect, backflow has an adverse effect on condenser performance, with numerous researchers identifying this \([41, 42, 131, 43, 44]\). Infra-red (IR) images taken of the MACC leading tube row during initial tests confirmed this was an issue, as seen in figure 4.5.

![Figure 4.5: IR images showing backflow and non-condensable zones in leading tube row](image)

**4.2.2 Modifications: implementation of dephlegmator**

The most widely employed strategy to prevent backflow and, hence, the formation of non-condensable zones is the use of a dephlegmator. A dephlegmator generally refers to a secondary heat exchanger which is installed in series with the primary condenser. This is accompanied by a vacuum pump located at some point downstream. The dephlegmator-vacuum pump arrangement is, essentially, a mechanism for separating the air from the steam and, subsequently, displacing the air from the system. As a consequence of this, it also facilitates condensation of the uncondensed (excess) steam, which would otherwise flow back into the leading tube rows. The net effect of this is that vacuum decay, backflow, and the resulting formation of non-condensable zones are all eliminated.

The first step to installing the dephlegmator-vacuum pump system on the MACC was to establish the rate of air leakage - in order to determine an appropriately-sized
vacuum pump. The flow rate of air leaking into the MACC was approximated using the ideal gas law, given in equation 4.1.

\[ PV = mRT \]  

(4.1)

where \( P \), \( V \), \( m \), \( R \), and \( T \) is the pressure, volume, mass, individual gas constant, and temperature, respectively.

Equation 4.1 can be manipulated to equate the mass and, when expressed in terms of time derivatives, assumes the following form;

\[ \frac{dm}{dt} = \dot{m} = \frac{dP}{dt} \frac{V}{RT} \]  

(4.2)

where \( \dot{m} \) is the mass flow of air leaking into the MACC. The slope of the measured pressure line given in figure 4.4 is equal to the rate of pressure rise, \( dP/dt \), and was evaluated as approximately 13 Pa/s for the period of \( 200 \text{ s} \leq t \leq 800 \text{ s} \). The volume of the MACC under vacuum was calculated to be approximately 0.8 m\(^3\), which included the volume of the inlet and outlet headers, the heat exchanger tubes and condensate tank. Substituting these, along with the individual gas constant for air, into equation 4.2 gives a mass flow rate of 0.122 g/s which equates to a volumetric flow rate of 0.365 m\(^3\)/hr. This figure was used to source and install a suitably-sized vacuum pump to continually displace the air from the MACC, namely \textit{Busch R5 RA 0100 F}. The performance of this pump is expressed by its characteristic curve - given in figure 4.6, with the main characteristics summarised in table 4.2.

![Busch R5 RA 0100 F vacuum pump performance curve re-produced from manufacturer’s data sheet](image)

Figure 4.6: \textit{Busch R5 RA 0100 F} vacuum pump performance curve re-produced from manufacturer’s data sheet

The vacuum pump performance depicted in figure 4.6 is given in terms of inlet pressure at the pump (x-axis) and flow rate displaced from the system to which the pump is connected (y-axis). As the outlet-side of the pump is open to atmosphere, a smaller pressure at the inlet-side means there is a greater pressure rise across the pump. It can be clearly seen that as the inlet pressure reduces (greater pressure
Table 4.2: Vacuum pump specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Magnitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{V}_f$ (m$^3$/hr)</td>
<td>Free air displacement</td>
<td>100</td>
</tr>
<tr>
<td>$P_u$ (Pa)</td>
<td>Ultimate pressure</td>
<td>10</td>
</tr>
<tr>
<td>$\dot{V}_{100}$ (m$^3$/hr)</td>
<td>Displacement at 100 Pa</td>
<td>60</td>
</tr>
</tbody>
</table>

difference), the flow rate which the pump can evacuate from the system reduces. This is a fundamental characteristic of any rotary vane vacuum pump. The lowest possible pressure at the inlet to the pump is termed the ultimate pressure, at which point only very low flow rates are possible. Conversely, as the inlet pressure approaches atmospheric pressure (10$^5$ Pa), the pressure rise across the pump reduces, thereby increasing the pumping capacity of the pump. Under such conditions, maximum flow rate of the pump is achieved, otherwise known as the free air displacement of the pump.

With the dephlegmator-vacuum pump system installed, a vacuum decay test, similar to that carried-out to obtain the measurements in figure 4.4, was performed. The results from this are given in figure 4.7, which shows that, instead of the pressure rise (vacuum decay) that was initially observed, a steady-state pressure was achieved.

![Figure 4.7: Plot of steady-state pressure which was established through the installation of a dephlegmator-vacuum pump system. Y-axis is scaled in insert.](image)

As can be seen from the plot in figure 4.7, a certain period of time elapses before a steady-state vacuum is reached. The pressure is still reducing until approximately 15 minutes (900 seconds) have elapsed, after which time a constant pressure exists. Generally, a period of approximately 15 minutes was required during a typical vacuum test for the vacuum pump to reach a state of equilibrium, whereby the air leaking into the system was continuously exhausted to atmosphere.
The vacuum pump was installed downstream of the dephlegmator - which itself was mounted downstream of the MACC condensate manifold, as can be seen in figure 4.8, particularly figure 4.8b. Very little information is available in open literature regarding the size of the dephlegmator heat transfer area, relative to the primary condenser heat transfer area. Data procured from an operational A-frame ACC containing a dephlegmator, suggested a ratio of ACC area to dephlegmator area of approximately 7:1. Owing to a lack of any viable alternative design guidelines, a dephlegmator was sized, loosely based on this criterion. The dephlegmator heat transfer area was approximately 38 m$^2$ which, although not strictly adhering to the 7:1 ratio, was deemed to be satisfactory based on the fact that the ratio was not a proven relationship and more of a guideline. Furthermore, the primary objective was to quantify the condensate-side characteristics of the MACC, with the dephlegmator simply seen as a means to facilitate this.

![Dephlegmator-vacuum pump arrangement on MACC module](image)

Figure 4.8: Dephlegmator-vacuum pump arrangement on MACC module

The dephlegmator installed was a secondary air-cooled condenser, consisting of a compact heat exchanger and an axial fan. The inlet to the dephlegmator was mounted to the condensate manifold of the MACC, with the dephlegmator exit connected to the inlet of the vacuum pump. The MACC, dephlegmator, and vacuum pump were connected in series. This configuration ensured that the non-condensed steam and air present in the MACC tended to flow towards the lowest pressure point, i.e., the vacuum chamber in the pump. The mixture first flows through the dephlegmator, where the excess steam flows vertically upwards and is condensed by the cooling air, with the condensate returning to the MACC condensate manifold under the influence of gravity. This operating principle is illustrated in figure 4.8a. It was important to install the dephlegmator so as to ensure condensate flow would not be inhibited and could return to the MACC condensate manifold to be measured as part of the
overall mass flow. The air in the system passes through the dephlegmator, where it is displaced to the atmosphere via the vacuum pump.

It is important to note that the implementation of the dephlegmator-vacuum pump system was not just a feature specific to the MACC but is, in fact, ubiquitous in all commercially operating ACCs. With the dephlegmator and vacuum pump added to the MACC, the desired conditions for steam-side characterisation were achieved. These conditions were realistic insofar that they were representative of typical ACC conditions in a thermoelectric power plant. The final layout during experimentation is shown in the schematic in figure 4.9.

4.2.3 Mass flow rate measurement

The measured mass flow rate was the condensate mass flow rate, which is equal to the vapour mass flow rate - provided all steam is condensed. During any given set of measurements, the condensate which exited the MACC tubes accumulated in the common condensate manifold, where it was joined by the relatively smaller quantity of condensate from the dephlegmator. Ultimately, the condensate flowed downstream, via gravity drainage, from the condensate manifold into the condensate tank. Two float level switches were installed in the tank to regulate the condensate pump on/off cycle. The level between these switches was set, thus ensuring that the mass of condensate that could accumulate, between the switches, was fixed. By establishing this mass, the condensate mass flow rate could be determined by simply dividing the known mass by the time taken for it to accumulate during a given test. The mass between the two switches was established by repeatedly filling the tank,
between the switches, with water. On each occasion, the pump commenced once the upper level was reached and emptied the tank to the lower level, at which point the pump disengaged. The mass of water emptied from the tank was collected and weighed. This procedure was repeated six times to ensure consistency of the mass pumped in each cycle. The results of these six tests are given in figure 4.10, where variations in the mass of water collected can be seen to be no more than 0.01%, demonstrating the precision in the measurement.

![Figure 4.10: Data from calibration of mass in condensate tank](image)

The mass flow rate of condensate could, thus, be inferred during measurements by monitoring the time taken to fill between the two level switches. This time frame began when the pump ceased (at the lower level) until the point at which the pump commenced again (the higher level). Using the mean mass of condensate from figure 4.10, equation 4.3 expresses the mass flow rate as:

\[
m_c = \frac{\bar{m}_c}{t}
\]  

(4.3)

An important characteristic of the system, that manifested itself during testing, was the transient response that occurred during the pump on/off cycle. An example of transience in the measured pressure is given in the graph in figure 4.11. As can be seen, there are five distinct cycles shown in the data, each with a relatively flat region of pressure data - the measurement cycle, followed by a juncture of dramatic reduction and subsequent increase in pressure - the pumping cycle. During a typical set of pressure drop, or heat transfer measurements, the liquid level in the condensate tank raises until the upper float switch closes. At this point, the pump commences and empties the tank which results in the sharp reduction in MACC pressure seen in figure 4.11. Once the liquid level reaches the lower float switch, the pump ceases and the pressure recovers again.

For the case presented in figure 4.11, the average pressure of the five cycles was evaluated as approximately 5443 Pa, with a local maximum and minimum of about
Figure 4.11: Condensate tank showing the transient nature in the measured pressure data that resulted from the condensate pump on/off cycle.

5632 Pa and 5058 Pa, respectively. This results in a variation of + 3.5% & - 7.1% about the mean. For all reported full-scale measurements in this thesis, the transient response necessitated that, at any given fan speed, five pump on/off cycles be recorded. This permitted accurate measurement of the mass flow rate and also ensured confidence in the measured pressure and temperature by compensating for any fluctuations. Furthermore, the average/mean of those cycles was used, along with the standard deviation for the same range of cycles, to analyse the measured data. Accordingly, equation 4.3 assumes the following form;

\[
\dot{m}_c = \frac{1}{n} \sum_{i=1}^{n} \left( \frac{\bar{m}_{c,i}}{t_i} \right)
\]  

(4.4)

where \( \bar{m}_c \) is the average mass of condensate, which was established from the condensate tank calibration and is constant for each cycle. \( t \) is the time period for each measurement cycle and is the variable in equation 4.4. It was recorded using a stopwatch, from the point at which the condensate pump ceases to the point at which it commences again.

4.3 Parametric Analysis

It was envisaged that the MACC concept would, ultimately, be commercialised, leading to its installation in thermoelectric power plants. It was necessary, therefore, that the MACC characterisation experiments be carried-out at conditions similar to those expected in operation. By doing so, established performance characteristics,
such as the variation of pressure with fan speed, would provide a realistic indication
of the ability of the MACC to control and manipulate steam turbine back-pressure
in application. Furthermore, the performance characteristics could also be used with
confidence in thermodynamic analyses and plant performance predictive tools.

Testing at representative conditions meant that the ability to establish and main-
tain sub-atmospheric (vacuum) conditions was crucial. Hence, the importance of
installing the dephlegmator-vacuum pump arrangement, outlined in Section 4.2.2.
Based on data procured from an operational A-frame ACC, a design point of 0.068
bar (6800 Pa) and 38 °C (311 K) was used as a reference for steam conditions dur-
ing testing. Clearly, this was subject to variation during testing due to changes in
ambient temperature and mass flow rate, etc. but, nevertheless, was useful to refer
to when initially establishing vacuum conditions.

One of the most crucial test variables was the mass flow rate of steam through
the MACC. For realistic conditions, it was imperative that a range of representa-
tive flow rates were employed. In order to ensure this, a 50 MW steam turbine was
used as the benchmark plant size. The characteristics of this turbine are given in the
thermodynamic modelling section in Chapter 3. The nominal mass flow rate exiting
this turbine, \( \dot{m}_{ST} \), and, consequently, entering a downstream condenser is approx-
imately 39 kg/s. Based on a techno-economic analysis carried out by Poullikkas et
al. [129, 130], an optimum number of MACC modules, \( N_{MACCs} \), for this plant size
was calculated to be approximately 650. This number varies slightly depending on
the module type of interest. Nevertheless, this figure allowed for an approximate
representative range of steam flow rates to be investigated in the various MACC
modules.

The nominal flow rate through a single MACC module, such as the prototype, was
calculated to be about 0.06 kg/s/module, using equation 4.5.

\[
\dot{m}_{MACC} = \frac{\dot{m}_{ST}}{N_{MACCs}} \tag{4.5}
\]

A range of flow rates based around this nominal flow rate were chosen for investig-
ating the condensate-side characteristics of the various MACC prototypes. Assuming
a uniform flow rate through each tube in the module, the mass flow rate per tube,
\( \dot{m}_t \), can be determined from the total module mass flow rate as follows;

\[
\dot{m}_t = \frac{\dot{m}_{MACC}}{N_t} = \frac{\dot{m}_c}{N_t} \tag{4.6}
\]

where \( N_t \) is the total number of tubes in the given MACC module of interest.
Ultimately, the flow rate per tube can be expressed non-dimensionally in terms of
the Reynolds number for either phase, as previously shown in Chapter 2, Section 2.2.
In this case, however, the mass flux term in the Reynolds number is defined for a
single tube as shown in equation 4.7;

\[
G = \frac{\dot{m}_t}{A} \tag{4.7}
\]
Determining the flow rate per tube is predicated on the assumption that the total module mass flow rate is distributed evenly amongst the tubes in the module. It was difficult to know with any great degree of certainty if this was, in fact, the case. However, it was a necessary assumption to permit calculation of the mass flux and Reynolds numbers, as these terms are dependent upon a characteristic length scale - in this case the tube diameter. Defining a mass flux or Reynolds number for the entire MACC module would not be possible due to the lack of an obvious and well-defined length scale. Ultimately, the range of flow rates and corresponding Reynolds numbers examined in this study are summarised in table 4.3.

Table 4.3: Range of flow conditions investigated in this study

<table>
<thead>
<tr>
<th>Module</th>
<th>$\dot{m}_{\text{MACC}}$ (kg/s)</th>
<th>$G$ (kg/s.m$^2$)</th>
<th>$\text{Re}_v$</th>
<th>$\text{Re}_l$</th>
</tr>
</thead>
<tbody>
<tr>
<td>6 row, circular tube</td>
<td>0.10 - 0.22</td>
<td>0.951 - 1.963</td>
<td>2543 - 5145</td>
<td>40 - 93</td>
</tr>
<tr>
<td>4 row, circular tube</td>
<td>0.05 - 0.10</td>
<td>0.690 - 1.384</td>
<td>1893 - 3706</td>
<td>24 - 57</td>
</tr>
<tr>
<td>1 row, rectangular tube</td>
<td>0.06 - 0.10</td>
<td>0.983 - 1.641</td>
<td>2650 - 4420</td>
<td>46 - 79</td>
</tr>
</tbody>
</table>

The vapour Reynolds number range of 1893 - 5145 investigated herein, is noticeably smaller than those in current literature, which can often be one to two orders of magnitude greater. The main reason for this is attributed to the greater number of tubes needed in an ACC - to compensate for air as a cooling medium - in comparison to water-cooled shell-and-tube condensers. Generally, there are more tubes in an ACC and, thus, the flow rate per tube in an ACC will be lower than the flow rate per tube in a comparable shell-and-tube condenser coupled to the same steam turbine. Clearly, if the flow rate per tube is less, the Reynolds number will be smaller. In any case, the fact that the range of Reynolds numbers outlined in table 4.3 have not been studied in any great detail to-date provides a motivating factor for the research to be carried-out.

4.4 EXPERIMENTAL ARRANGEMENT

The general test facility described thus far, in Sections 4.1 and 4.2, formed the basis of the measurement platform for pressure drop and heat transfer measurements. As both sets of measurements were acquired simultaneously, the experimental set-up was common to both. However, to supplement the details provided heretofore, additional information on the specific experimental arrangement for the pressure drop and heat transfer measurements is presented in this section.

4.4.1 Pressure drop

Pressure drop data was acquired using two different sets of pressure transducers, which were installed in the MACC as shown in figure 4.12. A set of 0 - 4 bar absolute
(0 - 4×10⁵ Pa) pressure transducers, with an accuracy of 0.25% of full-scale deflection (FSD), were used to measure atmospheric pressure and above. A separate set of 0 - 0.35 bar absolute (0 - 3.5×10⁴ Pa) transducers were used to measure sub-atmospheric pressure, as these transducers had a greater accuracy over this range of ±0.08% of FSD. However, the drawback of such accuracy was the limited range and for occasions when high pressure steam was flowing through the MACC, the 0 - 4 bar absolute pressure transducers were employed. They were also used to indicate the onset of vacuum conditions, at which point the high accuracy transducers were engaged.

As can be seen in figure 4.12, the pressure transducers were arranged in pairs, with each pair consisting of a single 0 - 4 bar absolute and 0 - 0.35 bar absolute pressure transducer. Three pairs each were installed in the inlet and condensate manifolds - constituting twelve pressure transducers in total. Utilising this paired arrangement, the 0 - 4 bar absolute pressure transducers were constantly exposed to the fluidic pressure but the high accuracy transducers were protected and isolated by the closed valve when the pressure exceeded their 0.35 bar limit.

![Figure 4.12: MACC module with location of pressure transducers highlighted](image)

Pressure transducers located in the inlet manifold and condensate manifold measured the inlet and exit pressure, respectively. Ultimately, this allowed the difference in pressure through the condenser tubes to be determined. Installing an array of transducer pairs across the manifolds served two primary functions. The first was to increase the measurement resolution. The second was to compensate for any potential fluidic effects across the manifolds. Indeed, as reported in their theses, losses in the inlet manifold have been observed by Honing [132] and Owen [133]. However, measurements confirmed this was not an issue in the MACC as pressure variation across the inlet manifold generally remained within ±2% of the mean. At all times, the pressure difference remained below that measured through the tubes.

All pressure transducers were wired, in a differential manner, to a National Instruments (NI) Data Acquisition (DAQ) system containing a 24-bit, 10 channel card,
namely NI-9207. The 0 - 5 V output signal from the pressure transducers was scanned, amplified, conditioned, and sampled by a single analog-to-digital converter (ADC) in the card. The digital signal was, ultimately, recorded by a custom LabVIEW program.

4.4.2 *Heat transfer*

The main additional features for the heat transfer measurements comprised of K-type thermocouples installed in the inlet and condensate manifolds to measure the steam temperature and condensate temperature, respectively. An array of thermistors located at the heat exchanger air inlet - to measure the ambient air temperature, and heat exchanger air outlet - to measure the exit air temperature, were also mounted on the MACC. The thermocouples were installed in close proximity to the pressure transducers, shown in figure 4.12, so that saturation conditions, or otherwise, could be monitored. Therefore, for each pressure transducer pair installed in the MACC, a corresponding thermocouple was installed. Calibration details relating to these thermocouples are provided in Appendix B. All thermocouples were connected to a 24-bit, 6 channel, NI-9211 DAQ card, where signal conditioning occurred. The resultant digital signal was, subsequently, recorded by LabVIEW.

Thermistors were mounted on two rails, with one rail each located at the air inlet and air outlet plane of the heat exchanger tube bank. The air inlet plane during an induced draft scenario is shown in figure 4.13. As can be seen, the instrumented rail of thermistors could be positioned along the length of the MACC, via a traversing rail. This arrangement enabled measurement of the air temperature at various positions along the plane. As measurements were usually carried-out under calm conditions, wind effects were not an issue, nor was hot air re-circulation. Therefore, the spatial variation in air inlet temperature was found to be less than 2 °C and the rail of thermistors usually occupied a constant position as a result.

![Figure 4.13: Thermistor array at MACC air inlet plane during an induced draft air flow](image)

Thermistors are thermally sensitive resistors, in that they are electrical resistors whose resistance varies with temperature. For NTC thermistors, such as those moun-
ted on the MACC, the resistance decreases with increasing temperature. The chosen NTC thermistors had a nominal resistance of 2252 $\Omega$ at 25 °C and an accuracy of ±0.2 °C. These thermistors were integrated into one arm in a Wheatstone bridge, with the resulting voltage signal acquired by a 16-bit, 32 channel NI 9205 DAQ card. Further details of this measurement system and calibration are available in [34].

4.5 EXPERIMENTAL PROCEDURE

As the pressure drop and heat transfer measurements were acquired simultaneously, a single experimental procedure was common to both. A typical set of measurements began by establishing a steady-state vacuum, similar to that presented in figure 4.7. This involved “purging” the condenser of any non-condensables at start-up and, subsequently, creating a vacuum in the system. Thereafter, the pressure and temperature data was recorded. The following series of steps outline the procedure to, firstly, establish adequate vacuum conditions and, secondly, acquire the desired data.

4.5.1 System preparation

1. The experimental facility was arranged as described thus far and as shown in figure 4.9. Before any test, all equipment was checked to ensure correct operation and all valves were set to the desired positions.

2. The steam boiler was switched on and, once ready, the boiler valve was fully opened to release steam from the boiler chamber.

3. Steam at a pressure and temperature of approximately 1.5 bar and 115 °C, respectively, was allowed into the MACC by opening the main inlet valve. This steam flowed through the condenser tubes, into the condensate tank and, eventually, back into the boiler feedwater tank. This forced any residual air pockets from the system.

4. After a period of 5 - 10 minutes of steam flow it was assumed all non-condensables were removed, at which point an isothermal heat exchanger surface should exist. This was confirmed using an IR camera.

5. Once isothermal conditions were achieved, the MACC main inlet valve was closed. This created a closed system between the inlet valve and non-return valve, located downstream of the condensate pump.

6. The MACC fans were set to maximum speed of $\approx$ 1000 rpm and the reduction in volume as the steam collapses to form condensate creates the initial vacuum.

7. The vacuum pump was started and the dephlegmator fan was activated.
8. The valve connecting the condensate manifold to the dephlegmator was opened. This allows the vacuum pump access to the MACC and permits air removal.

9. A period of about 15 - 20 minutes was allowed for a steady-state vacuum to be attained, as shown in figure 4.7.

10. Steps 1 - 9 were repeated at the start of each measurement test.

4.5.2 Data acquisition

1. Once steady-state vacuum was achieved, steam flow into the MACC was resumed by opening the main inlet valve. This was set to a specific position, thus fixing the mass flow into the MACC.

2. Mass flow rate was recorded by monitoring the condensate pump on/off cycle as described in Section 4.2.3. The time period for the known mass of condensate to accumulate between the level switches was recorded using a stopwatch.

3. The pressure and temperature at the MACC inlet and condensate manifold were recorded using LabVIEW. The ambient temperature was also recorded.

4. Fan speed was reduced incrementally from $\approx 1000$ rpm, in steps of $\approx 100$ rpm, and step 3 was repeated.

5. Step 4 was repeated for the desired range of fan speeds.

6. Steps 1 - 5 were repeated for the predetermined range of mass flow rates.

7. Step 6 was repeated for a range of condenser inclinations.\footnote{Step 7 was only applicable to heat transfer measurements}

The main addition to the procedure for heat transfer measurements was the inclusion of Step 7, which was the investigation of the condenser inclination. This was examined to determine if there was any appreciable effect of varying inclination angle on the condensate characteristics and, if so, was there an optimal angle to promote heat transfer.

4.6 Data reduction

This section describes the steps taken to extract and analyse useful data from the raw measurements obtained using the procedure outlined in Section 4.5. Firstly, the pressure drop data reduction is presented. This includes the method for processing the measured data, and the data reduction technique to evaluate the various components which constitute the overall two-phase pressure drop. Following this, the data reduction for the heat transfer measurements is presented.
4.6.1 Pressure drop

4.6.1.1 Processing measured data

The two-phase condensing pressure drop through the tube bundle is equivalent to the pressure difference between the inlet manifold and condensate manifold. This differential pressure was calculated by subtracting the mean of the absolute pressure at the inlet manifold from the mean of the absolute pressure at the condensate manifold. This is expressed in equation 4.8:

$$\Delta P_m = \bar{P}_i - \bar{P}_e$$  \hspace{1cm} (4.8)

The mean of the absolute pressure at either manifold was the result of five pump on/off cycles, which were necessary to compensate for the transient issue discussed in Section 4.2. Therefore, for a given fan speed, five pumping cycles were allowed which meant the inlet pressure and exit pressure given in equation 4.8 were evaluated from the raw data as follows:

$$\bar{P}_i = \frac{1}{n} \sum_{i=1}^{n} P_{i,i} \hspace{1cm} \bar{P}_e = \frac{1}{n} \sum_{i=1}^{n} P_{e,i}$$  \hspace{1cm} (4.9)

For all pressure drop calculations, complete condensation occurred in the MACC. This was ensured by monitoring the steam conditions in the inlet and condensate manifolds during testing. Slightly superheated steam ($4 - 5 \text{ K} > T_s$) was maintained in the inlet manifold through the steam separator, and isentropic expansion and throttling across the main inlet valve, in conjunction with a water injection system to prevent excess superheat. Subcooled condensate ($7 - 14 \text{ K} < T_s$) was confirmed by temperature measurements in the condensate manifold. Hence, the quality could be assumed to vary from unity at the inlet to zero at the exit.

4.6.1.2 Two-phase pressure drop analysis

For a two-phase flow, such as condensing steam, the measured pressure drop, $\Delta P_m$, is the sum of three contributions: the frictional pressure drop, $\Delta P_{fr}$, the momentum pressure drop/recovery, $\Delta P_{mom}$, and the static (gravitational) pressure drop, $\Delta P_{st}$. This is expressed as:

$$\Delta P_m = \Delta P_{fr} + \Delta P_{mom} + \Delta P_{st}$$  \hspace{1cm} (4.10)

Equation 4.10 can be re-arranged to determine the frictional component:

$$\Delta P_{fr} = \Delta P_m - \Delta P_{mom} - \Delta P_{st}$$  \hspace{1cm} (4.11)

where the momentum pressure loss/recovery and static pressure loss need to be evaluated in order to calculate frictional losses. The momentum pressure loss/recovery reflects the change in kinetic energy of the fluid continuum. For the usual case...
of an evaporating flow (as with refrigerants), this is referred to as the acceleration pressure drop due to the losses associated with the conversion of pressure energy to kinetic energy as the evaporating fluid accelerates through the tubes. However, for the case of a condensing flow, the fluid continuum decelerates from a buoyant vapour at inlet to a dense liquid at exit. Thus, acceleration losses are not incurred and, instead, a pressure recovery occurs. Momentum recovery for a two-phase flow is represented by equation 4.12;

\[
\Delta P_{\text{mom}} = G^2 \left\{ \left[ \frac{1 - x^2}{\rho_l (1 - \alpha)} + \frac{x^2}{\rho_v \alpha} \right]_e - \left[ \frac{1 - x^2}{\rho_l (1 - \alpha)} + \frac{x^2}{\rho_v \alpha} \right]_i \right\} \quad (4.12)
\]

As slightly superheated steam was entering the tubes and slightly subcooled condensate was leaving the tubes, equation 4.12 can be expressed in the form of a single-phase flow. Slightly superheated steam is a single-phase gas and, due to the absence of any liquid, inherently has a steam quality of unity \((x = 1)\). Also, as a single-phase gas, it occupies the entire tube cross section, resulting in a void fraction equal to one \((\alpha = 1)\). Conversely, subcooled condensate is a single-phase liquid with steam quality and void fraction both equal to zero \((x = 0, \alpha = 0)\). These conditions reduce equation 4.12 to the following single-phase expression;

\[
\Delta P_{\text{mom}} = G^2 \left\{ \left[ \frac{1}{\rho_l} \right]_e - \left[ \frac{1}{\rho_v \alpha} \right]_i \right\} \quad (4.13)
\]

The static pressure drop expresses the variation in potential energy of the fluid continuum and, for a two-phase flow, is given as;

\[
\Delta P_{\text{st}} = [(1 - \alpha) \rho_l + \alpha \rho_v] g \sin \theta \quad (4.14)
\]

In a similar fashion to the momentum recovery term, the static pressure loss term reduces to the following as a result of single-phase conditions at the inlet and condensate manifolds;

\[
\Delta P_{\text{st}} = \rho_h g \Delta z \sin \theta \quad (4.15)
\]

where \(\rho_h\) is the homogeneous (average) density of both phases combined, defined previously in equation 2.51 in Chapter 2. Here, it was determined for the average steam quality from inlet to outlet of \(x = 0.5\). \(\Delta z\) is the distance between the inlet and exit pressure transducers. \(\theta\) is the inclination angle, relative to the horizontal. For horizontal tubes there is no change in static pressure head and, generally, for inclined tubes the losses will be small - relative to the frictional or momentum terms. This is the case unless a large vertical head height, \(hd\), exists for the liquid level in a given tube. However, only estimates of the gravitational pressure drop could be calculated as it was practically impossible to know the head height of the liquid in any given tube at any given time. Therefore, using equation 4.15, the gravitational pressure loss for the largest possible head height of \(hd = 1.38\) m was estimated to be
approximately 2 Pa. As this is the worst-case scenario, the gravitational losses can be assumed negligible and, accordingly, the contribution of the static pressure drop to the overall pressure drop can be ignored. Thus, the final expression for determining the frictional pressure drop for comparison with predictive correlations is given by equation 4.16.

\[
\Delta P_{fr} = \Delta P_m - \left( G^2 \left\{ \frac{1}{\rho l} c - \frac{1}{\rho v} i \right\} \right)
\]  

(4.16)

4.6.2 Heat transfer

4.6.2.1 Processing measured data

In a similar manner to the absolute pressure measurements, the temperature in the inlet manifold and condensate manifold was recorded over five pumping cycles, at each fan speed setting. Therefore, the inlet temperature and condensate temperature referred to throughout is, in fact, the average of each respective temperature across five cycles. This is expressed in equation 4.17:

\[
T_s = \frac{1}{n} \sum_{i=1}^{n} T_{i,s}, \quad T_c = \frac{1}{n} \sum_{i=1}^{n} T_{i,c}
\]

(4.17)

where \(T_s\) and \(T_c\) are the steam and condensate temperature, respectively.

The air inlet and outlet temperature was measured using thermistor arrays, with each rail consisting of thirty-two thermistors. Each individual thermistor measures a local air temperature. Thus, the bulk air inlet temperature, \(T_\infty\), and bulk air outlet temperature, \(T_{out}\), was determined as shown in equation 4.18.

\[
T_\infty = \frac{1}{n} \sum_{i=1}^{n} T_{\infty,i}, \quad T_{out} = \frac{1}{n} \sum_{i=1}^{n} T_{out,i}
\]

(4.18)

As data was recorded over a prolonged testing period, the ambient temperature was not constant for each data set due to the fact that it intrinsically varied from day-to-day. Therefore, the measured data was normalised to ensure consistency and to allow measurements be compared using a common ambient temperature. The data normalisation was carried-out by evaluating the temperature from a set, predetermined and common ambient value as shown in equation 4.19:

\[
T^*_s = \Delta T + T^*_\infty
\]

(4.19)

where \(T^*_s\) is the normalised temperature based on a prescribed ambient value of \(T^*_\infty\), which is set and is common to all measured data. \(\Delta T\) is the measured temperature difference between the steam and bulk ambient air given as:

\[
\Delta T = T_s - T_\infty
\]

(4.20)
4.6.2.2 Thermal resistance analysis

The thermal characteristics of an ACC are frequently quantified in terms of thermal resistance. Thermal resistance is analogous to electrical resistance in that it is an impedance to flow. Generally, in a condenser, there are a number of thermal resistances in series through which the heat must flow as energy is rejected from the vapour to the cooling medium. Figures 4.14 and 4.15 illustrate the media, which constitute the flow path for heat transfer, and the accompanying thermal resistance network for the circular tube and rectangular tube designs, respectively.

![Thermal Resistance Network](image)

Figure 4.14: Thermal resistance network for single annular-finned, circular tube

As seen in figures 4.14 and 4.15, there are a number of sequential thermal resistances, through which the heat must flow. The overall resistive path can be separated into two distinct, separate resistances - that inside the tube (the steam-side thermal resistance, or condensate-side thermal resistance) and that on the tube exterior (the air-side thermal resistance). On the steam-side, the air present in the system, due to the leakages under vacuum, and the condensate film are the primary barriers to heat transfer. The tube wall also acts as an impediment but, due to the high thermal conductivity of aluminium ($k \approx 200$ W/m.K), it offers significantly less resistance. These three resistances can be grouped together in series, the result being the steam-side thermal resistance. At this point, it should be noted that tube-side fouling was not considered in this study. As the condenser tubes were new and were not used outside of the characterisation tests, fouling was not expected to be a significant factor. Thus, all steam-side thermal resistance values reported in this thesis are assumed to be solely due to fluidic effects. Outside the tube, the air-side thermal resistance is that resulting from the film of air (the boundary layer) blanketing the tube wall and fin channels, combined with the conductive resistance through the fins. The air-side is considered to be the dominant thermal resistance in ACCs.
The experimental testing permitted measurement of the global steam and condensate temperature but, due to the large scale of the MACC facility and difficulty in mounting instrumentation in appropriate sites, local measurements were not possible. Local temperatures such as the wall temperature, $T_{w,i}$, shown in figures 4.14 and 4.15, could not be acquired as a result. However, using the following data reduction technique, the global steam-side thermal resistance was inferred. The total resistance to heat flow, from the steam core to the bulk ambient is given by equation 4.21:

$$\frac{(R_{th})_{tot}}{Q_m} = \frac{\Delta T_m}{\dot{m}_c \times h_{fg}}$$

(4.21)

$\Delta T$ in equation 4.21 is the measured temperature difference between the steam core and ambient air. $Q$ is the measured energy transfer rate, determined from the isothermal heat rejection, $\dot{Q}_{rej}$, with any sensible heat rejection ignored. Sensible heat rejection occurred due to de-superheating and subcooling. However both these terms are less than 5% of the latent heat rejection and, thus, could be considered negligible without introducing any error in the data reduction.

There are two approaches to determining the air-side thermal resistance. Firstly, through calculation of the heat exchanger temperature, $T_{HEX}$, using the effectiveness-NTU model described in Chapter 3, Section 3.2, the air-side resistance can be determined as shown in equation 4.22. The effectiveness-NTU model incorporates well-established correlations from the literature, for the relevant heat exchanger design, to calculate the air-side heat transfer coefficient.

$$\frac{(R_{th})_a}{Q} = \frac{T_{HEX} - T_{\infty}}{\dot{m}_a C_p (T_{out} - T_{\infty})}$$

(4.22)
where the temperature difference in equation 4.22 is that outside the tubes, i.e. from the heat exchanger surface to the bulk ambient. The accuracy of the air-side resistance determined in this manner has been validated by measurements taken on the MACC in a separate set of experiments, as reported by the authors in [124, 50]. The second approach to establishing the air-side thermal resistance is to simply use the measured values from [124, 50]. In either case, the global steam-side resistance can be inferred from equation 4.23 as follows:

\[(R_{th})_s = (R_{th})_{tot} - (R_{th})_a\]  

(4.23)

4.7 uncertainty analysis

To ensure a satisfactory level of confidence in the experimental results presented in this thesis, an uncertainty analysis was applied to all reported measurements. The uncertainty in the experimental results was calculated using the method described by Holman in [134], which is, in fact, based on the popular method first presented by Kline and McClintock [135]. This method is based on specifying the uncertainties in the various primary experimental measurements. These uncertainties propagate through the calculations with a combination of the different measurement variables - a methodology summarised as follows.

The result, \(R\), of an experiment can be a calculated parameter which is a function of a number of independent, measured variables \(x_1, x_2, \ldots, x_n\). This is expressed as,

\[R = f(x_1, x_2, \ldots, x_n)\]  

(4.24)

Uncertainty in the result, \(\omega_R\), is dependent on the uncertainty of the individual variables \(-\omega_1, \omega_2, \ldots, \omega_n\). If the uncertainties in the independent variables are all given with the same odds, then the uncertainty in the result having these odds is given as:

\[\omega_R = \left[ \left( \frac{\partial R}{\partial x_1} \omega_1 \right)^2 + \left( \frac{\partial R}{\partial x_2} \omega_2 \right)^2 + \ldots + \left( \frac{\partial R}{\partial x_n} \omega_n \right)^2 \right]^{1/2}\]  

(4.25)

Table 4.4 lists the measured experimental variables, and the error associated with them. By applying the analysis just outlined, and equation 4.25 in particular, experimental uncertainty associated with the calculated parameters was determined. The analysis was assessed over the range of experimental variables investigated, with the results presented in table 4.5.

4.8 summary

The full-scale prototype MACC module, and the accompanying experimental test facility, was introduced in this chapter. It was shown that, in its initial form, the
facility was not capable of providing, or maintaining, adequate experimental test conditions. A number of adverse condensate-side phenomena were observed during preliminary tests, which were alleviated through significant alterations, as described in this chapter. Through a number of design modifications, a facility capable of characterising the condensate-side performance of full-scale MACC modules was established. The methodology for acquiring hydrodynamic and thermal measurements from this facility was described in detail, accompanied by the appropriate data reduction techniques. The results corresponding to this experimental chapter are presented in Chapter 6.

Table 4.4: Uncertainties associated with primary, measured variables

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Units</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{m}_c$</td>
<td>Condensate flow rate</td>
<td>kg/s</td>
<td>2.1%</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure</td>
<td>Pa</td>
<td>0.8%</td>
</tr>
<tr>
<td>$T_s$</td>
<td>Steam temperature</td>
<td>K</td>
<td>0.5%</td>
</tr>
<tr>
<td>$T_\infty$</td>
<td>Ambient air temperature</td>
<td>K</td>
<td>0.5%</td>
</tr>
<tr>
<td>$\Delta P$</td>
<td>Pressure drop</td>
<td>Pa</td>
<td>1.8%</td>
</tr>
</tbody>
</table>

Table 4.5: Uncertainties associated with derived, calculated parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Units</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$G$</td>
<td>Mass flux (velocity)</td>
<td>kg/sm$^2$</td>
<td>5.3%</td>
</tr>
<tr>
<td>$Q$</td>
<td>Heat transfer rate</td>
<td>W</td>
<td>6.9%</td>
</tr>
<tr>
<td>$R_{th}$</td>
<td>Thermal resistance</td>
<td>K/W</td>
<td>5.8%</td>
</tr>
<tr>
<td>$Re_v$</td>
<td>Vapour Reynolds no.</td>
<td>-</td>
<td>4.5%</td>
</tr>
<tr>
<td>$\Delta P_f$</td>
<td>Frictional pressure drop</td>
<td>Pa</td>
<td>6.7%</td>
</tr>
<tr>
<td>$\Delta P_{mom}$</td>
<td>Momentum pressure recovery</td>
<td>Pa</td>
<td>8.3%</td>
</tr>
</tbody>
</table>
Full-scale experimentation on the MACC provided the opportunity to investigate and establish the condensate-side characteristics on a global scale. However, the disadvantage of such a large-scale facility was that it proved difficult to maintain a controlled environment, where certain variables could be manipulated in a predetermined manner. Furthermore, measurement of properties at the local-scale was not possible. Local measurements are, generally, more desirable as they provide a more detailed insight into the underlying physics and phenomena which, ultimately, improves understanding. As a consequence of these inhibitions at the full-scale, a reduced-scale ACC was designed and fabricated. This facility allowed for a more detailed investigation into both the global, and local, condensing characteristics to be undertaken in a systematic manner.

Complete fluidic and thermal characterisation of an ACC requires that both the air-side and condensate-side characteristics be established. This chapter is divided into those two main streams, where the experimental test facilities, methodologies, and data analyses are described in detail for each characterisation case. Prior to this, however, a general overview of the heat exchanger is presented. Experimental uncertainty closes the chapter.

5.1 Heat Exchanger Overview

The top level objective when designing the reduced-scale ACC was to ensure test conditions similar to those on the full-scale MACC prototypes could be achieved. The first step to replicating the test conditions was to ensure that the heat exchanger geometry be almost identical. As two MACC modules, and a number of additional potential designs, were comprised of annular-finned circular tubes, this geometry was chosen as the experimental candidate. For a reduced-scale experimental programme, a single tube was deemed to be sufficient for investigating the condensing characteristics. The pertinent geometrical dimensions of the candidate tube corresponded almost exactly with the full-scale tube dimensions, but for the fact that the reduced-scale tube was, accordingly, shorter in length. Furthermore, the tube was made from aluminium, as opposed to the steel full-scale tubes. This was to ensure that the reduced-scale tube wall thermal resistance was minimised.

The total length of the tube was determined by the finned length - that is the length of tube over which the fins were applied to give the area necessary for heat transfer. An elementary heat transfer analysis, outlined as follows, permitted approximation of the finned length, based on the thermal loading and nominal operating conditions.
The length of finned tube, $L_{fd}$, is given by the product of the number of fins, $N_f$, and the fin spacing, $S_f$, as shown in equation 5.1;

$$L_{fd} = N_f \cdot S_f$$ (5.1)

where the fin spacing was constrained to match the spacing on the full-scale tubes. The number of fins in equation 5.1 is equivalent to the total heat transfer area, $A_T$, divided by the area of one fin, $A_f$:

$$N_f = \frac{A_T}{A_f}$$ (5.2)

where the total heat transfer area is the area required to achieve a given heat rejection, for a specific heat transfer coefficient and temperature difference, and is expressed as:

$$A_T = \frac{Q}{h_a (T_s - T_\infty)}$$ (5.3)

In equation 5.3, the heat rejection term was the maximum heat transfer rate to be employed during experimentation ($Q \approx 2250 \text{ W}$). The heat transfer coefficient was chosen from the lower end of the range measured on the MACC, available in Moore et al. [124, 123]. As will be discussed in later sections, this was mainly due to a lack of suitable correlations or analytical solutions, from which the heat transfer coefficient could be predicted. The temperature difference was based on the expected nominal operating conditions. As in Chapter 4, data procured from an operational ACC was used as a reference point. This data specified a nominal operating temperature of 38 °C (311 K). Accordingly, a temperature difference of 20 K was based off this as it was anticipated that temperature differences any lower would not be encountered during experimentation. Choosing minimal values of $h_a$ and $\Delta T$ satisfied the required area for heat transfer for all heat transfer coefficients greater than the lowest value chosen and inlet temperature differences greater than 20 K.

The area of one fin was calculated using equation 5.4, in accordance with the geometrical dimensions listed in table 5.1. Again, the pertinent dimensions were constrained to reflect those on the full-scale tubes.

$$A_f = \left[ \frac{\pi (d_f^2 - d_e^2)}{4} + \pi d_e (S_f - \delta_f) \right]$$ (5.4)

where $d_f$ and $d_e$ are the fin diameter and tube external diameter, respectively. $\delta_f$ is the fin thickness. Incorporating equations 5.3 and 5.4 into equation 5.2 permits the number of fins to be calculated as follows:

$$N_f = \frac{Q}{[h_a (T_s - T_\infty)] \left[ \frac{\pi (d_f^2 - d_e^2)}{4} + \pi d_e (S_f - \delta_f) \right]}$$ (5.5)
Ultimately, the number of fins calculated allowed equation 5.1 to be solved, thus giving a finned length of approximately 0.9 m. In order to accommodate a cooling solution and, ultimately, incorporate the tube into a steam and condensate loop, the overall tube length was kept longer than the finned length. The final dimensions of the reduced-scale tube are compared with those from which it was based (the full scale annular-finned circular MACC tube) in table 5.1.

Table 5.1: Comparison of reduced-scale and full-scale annular-finned circular tube geometrical parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Full-scale</th>
<th>Reduced-scale</th>
</tr>
</thead>
<tbody>
<tr>
<td>D (m)</td>
<td>Inside tube diameter</td>
<td>0.0275</td>
<td>0.0253</td>
</tr>
<tr>
<td>d_e (m)</td>
<td>External tube diameter</td>
<td>0.03175</td>
<td>0.03175</td>
</tr>
<tr>
<td>d_f (m)</td>
<td>Fin diameter</td>
<td>0.0635</td>
<td>0.0635</td>
</tr>
<tr>
<td>( \delta_f ) (m)</td>
<td>Fin thickness</td>
<td>0.00041</td>
<td>0.00041</td>
</tr>
<tr>
<td>L_{fd} (m)</td>
<td>Finned length</td>
<td>2</td>
<td>0.9</td>
</tr>
<tr>
<td>L_t (m)</td>
<td>Tube length</td>
<td>2.15</td>
<td>1.15</td>
</tr>
<tr>
<td>N_f (m^{-1})</td>
<td>Number of fins per m</td>
<td>394</td>
<td>394</td>
</tr>
<tr>
<td>S_f (m)</td>
<td>Fin spacing</td>
<td>0.0025</td>
<td>0.0025</td>
</tr>
</tbody>
</table>

Figure 5.1 illustrates the final form of the annular-finned circular tube, with some of the pertinent geometrical parameters highlighted.
In order to ensure similarity with the full-scale MACC test conditions, air was employed as a cooling medium. The tube presented in figure 5.1 was, therefore, integrated into a housing with a bank of axial fans - which provided a cross-flow of cooling air. This facility constituted the heat exchanger, an overview of which is shown in figure 5.2. As can be seen, the configuration of the fans installed in the housing ensured an induced-draft air flow, whereby air was “pulled” over the finned tube. This induced-draft condition was primarily employed to simulate the induced-draft air flow conditions from the MACC. Achieving conditions identical to a single tube from the MACC tube bundle would have required a similar tube bundle to be fabricated, whilst also using the same fan and fan-heat exchanger spacing specifications from the MACC. Forgoing fabricating such a facility, the simpler arrangement illustrated in figure 5.2 was deemed to be the best solution. In particular, it provided a reasonable representation of a single tube in the leading tube row, subjected to an induced-draft air flow, in the MACC.

Figure 5.2: Overview of reduced-scale ACC final design

Air-cooling also avoided the reduced condensation length which would have resulted, for the given heat load, if water-cooling was employed. Such a condition would have made the condensation experiments much more difficult. In one of the only other studies employing air as a cooling medium for condensation experiments, Caruso et al. [21] favoured air-cooling due to the resultant smooth variation of condensation parameters, such as vapour and liquid Reynolds numbers, along the tube. This maintains uncertainties to a minimum for local heat transfer evaluation.

It is recognised, however, that air-cooling may introduce considerably large experimental errors as a result of the relatively low heat transfer coefficients on the air-side. Indeed, the air-side heat transfer coefficient can often be two orders of magnitude smaller. Therefore, in an effort to reduce experimental errors, an accurate experimental characterisation analysis of the air-side heat transfer coefficient was undertaken. This program is described in detail in the following section.
This section describes the methodologies for establishing the air-side properties of the reduced-scale ACC. The air-side characterisation encompasses quantifying both the aerodynamic and thermal boundary conditions on the ACC exterior. As the air-side is generally thought of as the dominant, or controlling, thermal resistance in ACCs, it is paramount that the air-side characteristics be established with a high degree of certainty, accuracy, and precision. As any inaccuracies in the air-side characteristics could be manifested, and exacerbated, as errors in the condensing characteristics, it was important that these inaccuracies be minimised.

An experimental approach was favoured for evaluating the air-side properties. This was mainly due to a lack of reliable theory with which to predict the properties. The chaotic nature of a fan-induced air flow across a finned tube, coupled with the unique duct geometry formed by the annular fins, meant that general analytical solutions for flow in channels, such as [136, 121, 137, 138], could not be trusted. Empirical correlations were not feasible either due to their prohibitive underlying database. Some of the most widely-cited correlations, given in [125, 139, 120], are only applicable to tube banks and are not relevant for a single, isolated finned tube. Therefore, an experimental program was devised to, firstly, determine the aerodynamic characteristics and, subsequently, the thermal characteristics on the air-side of the condenser.

### 5.2.1 Axial fan performance

The reduced-scale ACC cooling solution was comprised of a bank of eleven *ebm-papst* DC axial fans, namely the 8412 NG model. The fans were installed in the housing as shown in figure 5.3, with a schematic of a standalone fan highlighted. Specific details on these fans are summarised in table 5.2.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\omega$ (rpm)</td>
<td>3100</td>
</tr>
<tr>
<td>D (m)</td>
<td>0.076</td>
</tr>
<tr>
<td>H (m)</td>
<td>0.08</td>
</tr>
<tr>
<td>L (m)</td>
<td>0.08</td>
</tr>
<tr>
<td>W (m)</td>
<td>0.025</td>
</tr>
</tbody>
</table>

The fans were chosen to replicate the air flow conditions on the full-scale MACC. The air flow conditions provided by any given fan are quantified by the fan characteristic curve, which expresses the pressure-flow aerodynamics. Two different pressure-flow data sets for the 8412 NG fan were obtained from the fan manufacturer. Each data set was developed in accordance with ISO 5801 [114], which specifies free-inlet,
Figure 5.3: Schematic of reduced-scale ACC with 8412 NG axial fan highlighted

free-outlet test conditions. Figure 5.4 presents the characteristic curves plotted from these data sets, both of which are for the nominal fan speed of 3100 rpm. Also included in figure 5.4 is measured data from an isolated test fan at 3100 rpm. This measured data adhered to BS 848 [113] - again under free-inlet, free-outlet conditions. Details on the facility to acquire the fan data are given in Grimes et al. [140, 141].

Figure 5.4: Pressure-flow characteristics for single 8412 NG fan operating at 3100 rpm

Figure 5.4 indicates that the measured pressure-flow performance of the fan was very similar to that provided by the fan manufacturer’s data sets. There are some slight differences, most likely attributable to experimental error but, in general, the measured data agrees quite well with the data supplied by the manufacturer.
At all times, the fans were fixed in an induced-draft configuration, as the resulting flow field has been shown by Moore et al. [142] to be much more uniform than that from a forced draft air flow. The fans were located approximately 0.08 m downstream of the trailing edge of the tube fins. The fan-heat exchanger spacing was also shown by Moore et al. [142] to have a largely negligible effect on the aerodynamic characteristics in a fan-induced air flow. Hence, the 0.08 m distance was chosen somewhat arbitrarily, whilst nevertheless ensuring that the flow was unobstructed. A nominal operating fan speed of 3100 rpm was recommended by the manufacturer. However, to achieve variable air-side boundary conditions similar to the MACC, a range of fan speeds above and below this nominal value were used throughout.

The variation in fan speed was controlled by the fan power source. The bank of eleven axial fans, installed in the heat exchanger housing, were wired in parallel, from a common power source. Hence, the supply voltage and supply current to each individual fan was expected to be uniform across the fan bank. Provided each fan is subjected to the same supply voltage and current, the fan speed across the fan bank should, therefore, be uniform. One of the first tests was to confirm that this was the case. Through use of a laser tachometer, the rotational speed of each fan in situ was measured for a range of supply voltages - which were measured using a voltmeter. The voltage range examined and employed during testing was 8 - 15 V, which did not exceed the range recommended by the fan manufacturer in order to ensure longevity in the fans. The results from this voltage-fan speed calibration are given in figure 5.5.

![Figure 5.5: Supply voltage - fan speed calibration results](image)

The results given in figure 5.5 demonstrate that the rotational speed of each individual fan in the fan bank is quite uniform at any given supply voltage. The standard deviation varies from as low as 23 rpm to a maximum of 41 rpm, which emphasises the consistency in fan speed. This calibration was checked intermittently, throughout the duration of the testing programme, to ensure that degradation in fan performance
was not an issue. The calibration remained essentially identical to that presented in figure 5.5 throughout, hence ensuring uniformity of the air velocity along the length of the tube.

5.2.2 Aerodynamic characterisation

The aerodynamic characteristics of a heat exchanger are usually expressed in terms of the pressure drop and air flow rate. For relatively straightforward heat exchanger geometries such as circular, rectangular, and elliptical ducts, analytical solutions exist which permit calculation of the pressure drop based on some friction factor-Reynolds number relationship [143, 144, 145, 121, 138]. Additionally, there are a number of friction factor empirical correlations available in the literature for heat exchangers consisting of tube bundles [125, 119, 146, 139, 120]. Under ideal circumstances, either an analytical solution or experimental correlation could be used to predict the pressure drop for the reduced-scale ACC. In such a case, the flow rate would subsequently be determined from the operating point - which is the point of intersection between the fan characteristic curve and the predicted pressure drop. However, the following reasons negated the use of this approach.

Firstly, as stated in Section 5.2.1, the fan characteristic curve given in figure 5.4 was developed under specific free-inlet, free-outlet conditions. Therefore, the use of the fan characteristic curve in any analysis assumes that the fan performance during operation will be consistent with the pressure-flow characteristics expressed by that curve. However, as can be seen in figure 5.3, the axial fans in the heat exchanger were configured in a ducted-inlet, free-outlet scenario. Furthermore, the multiple fans are arranged in parallel. Therefore, it is highly likely that the actual fan performance will deviate from that given by the characteristic curve and, as such, the fan curves could not be used with any degree of confidence in evaluating the aerodynamic characteristics. Compounding this issue was the fact that no analytical solution was applicable to the unique duct geometry formed by the fin channels, and correlations for a single, isolated tube are non-existent. Hence, an alternative approach was used where, firstly, the system resistance curve was established, followed by pressure drop measurements from the fan generated air flow. Ultimately, the flow rates through the heat exchanger were established from the intersection between the system resistance curve and pressure drop measurements.

5.2.2.1 Experimental details

The system resistance curve simply expresses the pressure drop characteristics of a given heat exchanger geometry - in this case the fluidic losses incurred as air flows through the fin channels. This was established by measuring the pressure drop across a section of the tube for a range of arbitrary air flow rates. A fan performance test facility, previously described in detail by Grimes et al. [140, 141] and built in accordance with BS 848 [113], was modified and mounted to the inlet-side of the heat
exchanger, as shown in figure 5.6. This facility provided reliable and standardized air flow conditions.

![Diagram of test facility to measure system resistance](image)

**Figure 5.6: Modified fan performance test facility to measure system resistance**

The auxiliary fan in this facility provided the air flow which progressed through a series of straightening vanes and settling screens. This configuration straightened the flow by removing the swirl component emanating from the auxiliary fan. Therefore, upon entering the heat exchanger, the air flow field was representative of a more uniform induced-draft flow. The air flow rate was varied through adjusting the auxiliary fan speed and, at any given fan speed, the flow rate was measured using the inlet orifice plate. Corner tappings, located downstream of the orifice plate, and arranged around the perimeter of the airway, were connected to a Furness Controls FCO510 micro-manometer to measure the average differential pressure between the four pressure tappings and the ambient. The mass flow rate through the orifice plate was, subsequently, calculated from equation 5.6 taken from BS 848 [113];
\[ \dot{m}_a = \tau \varepsilon \frac{\pi d^2}{4} \sqrt{2\rho_a \Delta P} \]  

(5.6)

where \( \tau \) is the flow coefficient, and \( \varepsilon \) is the expansibility factor, both of which are obtained from data charts in [113]. \( d \) is the orifice plate throat diameter, \( \rho_a \) is the atmospheric air density, and \( \Delta P \) is the measured pressure drop across the orifice plate.

In conjunction with the orifice plate pressure drop measurements, another identical Furness Controls FCO510 micro-manometer was used to measure the pressure drop across the finned tube section as shown in figure 5.6b. Tappings were constructed, conforming to BS 848, and were fitted upstream and downstream of the finned tube in the heat exchanger housing - to permit differential pressure drop measurement. The tappings were installed flush with the inside wall - to ensure non-invasive measurement. Consequently, parallel streamlines were maintained throughout. For the range of flow rates generated by the auxiliary fan, the pressure drop measured across the finned tube section is presented in figure 5.7.

![Figure 5.7: Measured system resistance curve for annular-finned tube](image)

The actual flow rates through the heat exchanger, as generated by the bank of axial fans, were determined through further pressure drop measurements across the finned tube. In this instance, the air flow was generated by the bank of axial fans in-situ, and the fan performance test facility was removed. Eight different fan speeds, calibrated from the eight supply voltages in Section 5.2.1, were employed to generate the air flow. This resulted in eight distinct flow rates through the heat exchanger and, accordingly, eight pressure drop measurements were obtained. The intersection between these pressure drop measurements and the system resistance curve shown in figure 5.7 established the heat exchanger operating points, from which the flow rates generated by the fan bank were determined. These measurements are presented in figure 5.8, in the following results section.
5.2.2.2 Results

Figure 5.8: Fan-heat exchanger operating points from intersection of measured pressure drop and measured system resistance curve

For the specific range of fan speeds examined in this thesis, the operating points illustrated in figure 5.8 quantify the aerodynamic characteristics of the reduced-scale ACC. The abscissa and ordinate of each operating point define the pressure drop and flow rate through the heat exchanger, respectively. The mass flow rates determined from the operating points in figure 5.8 were scaled to account for the entire heat exchanger, as only an incremental length was examined in the experimentation described thus far. Therefore, the mass flow rate for the entire bank of eleven axial fans is presented, as a function of mean fan bank speed, in figure 5.9.

Figure 5.9: Variation of air flow through the reduced-scale ACC for range of fan speeds investigated
The mass flow of air, $\dot{m}_a$, can be expressed non-dimensionally, in terms of the air flow Reynolds number, $Re_a$, using equation 5.7:

$$Re_a = \frac{d_e \dot{m}_a}{\mu A_o}$$

(5.7)

where $A_o$ is the free flow area of the heat exchanger. The aerodynamic flow characteristics for the reduced-scale ACC are summarised in table 5.3.

### 5.2.3 Thermal characterisation

The thermal characteristics on the air-side of an ACC are, generally, quantified in terms of the air-side heat transfer coefficient, or air-side thermal resistance. Similar to the aerodynamic characterisation described in Section 5.2.2, the thermal characteristics could not be calculated from simplified theory, analytical solutions, or empirical correlations. This was mainly due to the nature of the test-facility, where the fan-induced air flow through the atypical fin channels restricted the use of theoretical models or analytical solutions such as [147, 127, 145, 136, 126, 137]. Furthermore, empirical correlations, such as [125, 119, 139, 120], are generally limited to finned tube banks consisting of multiple rows. Accordingly, no correlation exists for a single finned tube and, hence, the thermal characteristics were determined experimentally.

The experimental methodology employed was based on the technique pioneered by Kays and London [148, 149, 116], which has since become a standardized means of characterising the air-side thermal performance of compact heat exchangers. The methodology of Kays and London is based on a steady-state test facility, consisting of a heat exchanger test core within which steam is condensed, and a cooling solution - usually a wind tunnel. The main feature of the test methodology is that an excess quantity of slightly superheated steam is passed through the test core in a concerted effort to minimise the steam-side thermal resistance - arising from the formation of condensate in the tubes. Thermal resistance is further minimised by inclining the heat exchanger, which promotes condensate drainage via gravity. The excess steam prerequisite ensures that an isothermal wall condition is always maintained throughout testing. Combined, the negligible steam-side thermal resistance and isothermal conditions, allow for the steam core temperature to be assumed equal to the tube wall temperature, thus facilitating calculation of the wall to bulk-air heat transfer coefficient. The main drawback of this technique is that some steam is always present at the heat exchanger exit, which must be separated from the condensate and vented to atmosphere. This method has been successfully employed by the author in studies by Moore et al. [123, 124], thus reinforcing confidence in its use.

#### 5.2.3.1 Experimental details

Figure 5.10 is an overview of the experimental arrangement during a typical test. The reduced-scale ACC described thus far constituted the main component of the
test facility. Condensing steam was the working fluid inside the tube and the fan-induced, forced convective air flow was the cooling solution. Steam was supplied to the condenser by a boiler. Before entering the condenser, the steam passed through a baffle-plate steam separator and a series of ball-valves. The separator removed any liquid entrained in the steam flow and improved the quality to guarantee slightly superheated steam (3 - 5 K > $T_s$) at the condenser inlet. Providing steam in a superheated state not only ensures isothermal conditions, but also permits confident calculation of the enthalpy at the inlet. The magnitude of superheat was regulated throughout and was limited to approximately 5 K, to ensure that the resultant sensible cooling was negligible ($\leq 1\%$) - relative to the latent cooling. A series of valves were used to control the flow rate, and hence pressure of the steam, at the condenser inlet. The pressure was kept slightly elevated above atmospheric pressure (1.1 - 1.5 bar absolute) to prevent air from entering the condenser, which would otherwise lead to temperature non-uniformities.

The same range of fan speeds established in Section 5.2.1 were employed throughout the testing. As fan speed was increased from the lowest applicable speed, the flow rate of steam was incrementally increased, all the while ensuring slightly superheated and slightly pressurised conditions at the inlet to the condenser. Steam temperature and pressure were recorded at the condenser inlet and exit using K-type thermocouples and 0 - 4 bar absolute pressure transducers, respectively. The instrumentation was interfaced with LabVIEW 2011 via NI DAQ equipment, where signal conditioning occurred. At the condenser exit, the excess steam and condensate entered a separator. Due to density difference, the steam was vented to atmosphere and the condensate was collected, via gravity drainage, in a tank. The condensate was weighed and the time taken to fill the tank was recorded to infer the mass

Figure 5.10: Schematic of test facility for air-side thermal characterisation
flow rate. The acquired data was, ultimately, used to determine the air-side thermal properties using the data analysis outlined in the proceeding section.

5.2.3.2 Data reduction

The energy rejected by the condenser can be expressed by the product of the mass flow of condensate, $\dot{m}_c$, times the enthalpy of vaporization, $h_{fg}$, as follows;

$$Q_{rej} = \dot{m}_c h_{fg} \quad (5.8)$$

Equation 5.8 yields the isothermal heat rejection and is, inherently, only applicable under constant-temperature conditions. Negligible sensible cooling of the steam or condensate was maintained throughout testing to ensure this equation remained valid. With the mass flow rate of air through the condenser established in Section 5.2.2, the thermodynamic-maximum heat transfer rate can be evaluated as;

$$\dot{Q}_{max} = \dot{m}_a C_p (T_s - T_\infty) \quad (5.9)$$

where $T_s$ and $T_\infty$ is the measured steam and ambient air temperature, respectively. The effectiveness of a cross-flow heat exchanger, $\varepsilon$, is defined as the ratio of actual heat rejected to the maximum heat rejection possible, which is expressed in equation 5.10. Additionally, in heat exchanger theory, effectiveness can also be defined by the number of transfer units ($NTU$). Both terms are equated in equation 5.10 as follows;

$$\varepsilon = \frac{\dot{Q}_{rej}}{\dot{Q}_{max}} = \frac{\dot{m}_c h_{fg}}{\dot{m}_a C_p (T_s - T_\infty)} = 1 - \exp\left(-\frac{NTU}{U_a A_a/C_{min}}\right) \quad (5.10)$$

where $A_a$ is the air-side heat transfer area fixed by the design calculations as described in Section 5.1. $C_{min}$ is the minimum capacity rate, given by the product of the mass flow of air and specific heat capacity. Therefore, the only unknown in equation 5.10 is the overall heat transfer coefficient based on the air-side - $U_a$, determining which is the primary objective of the measurement procedure described thus far. $U_a$ can be solved for by rearranging equation 5.10, where the overall conductance, $U_a A_a$, also equates to the thermal resistance model given in equation 5.11;

$$\frac{1}{U_a A_a} = \frac{1}{\eta_{o,a} A_a h_a} + \ln\left(\frac{d_e}{d_i}\right) + \frac{1}{A_s h_c} \quad (5.11)$$

Equation 5.11 represents the various thermal resistances encountered, in series, in an ACC. Sequentially, as heat flows from the steam core to the bulk ambient, there is resistance to flow inside the tube (steam-side resistance), resistance through the tube material (wall resistance) and, finally, resistance through the air-side boundary layer (air-side resistance). However, as this specific test is arranged to purposely minimise the steam-side resistance, it can be assumed negligible. In addition, the
wall resistance is calculated to be negligible also. Thus, these terms cancel out of equation 5.11, reducing it to the following form:

$$\frac{1}{U_a A_a} = \frac{1}{\eta_{o,a} A_a h_a}$$  \hspace{1cm} (5.12)

Therefore, the overall air-side heat transfer coefficient based on the air-side, $U_a$, is defined here as the air-side heat transfer coefficient, $h_a$, lumped with the air-side surface efficiency, $\eta_{o,a}$. This product term is extracted from equation 5.12 as:

$$U_a = h_a \cdot \eta_{o,a}$$  \hspace{1cm} (5.13)

Ultimately, the air-side heat transfer coefficient can be determined from equation 5.13, with the air-side surface effectiveness calculated from the work of Gardner [128, 150]. The air-side surface effectiveness is dominated by the fin effectiveness, which accounts for temperature gradients across the extended fin surface. The relevant equations to solve these terms were presented in Chapter 3, Section 3.2. Finally, the air-side heat transfer coefficient can be expressed non-dimensionally, in the form of the Nusselt number defined in equation 5.14:

$$Nu_a = \frac{h_a d_e}{k_a}$$  \hspace{1cm} (5.14)

5.2.3.3 Results

Air-side thermal characteristics, such as overall heat transfer coefficient, film heat transfer coefficient, and film thermal resistance, were evaluated for the range of air flow rates established in Section 5.2.2. Figure 5.11 presents the overall air-side heat transfer coefficient results, where the data presented is from a series of identical tests - to satisfy experimental reproducibility.

![Figure 5.11: Variation in overall air-side heat transfer coefficient with air mass flow rate](image-url)
One of the most obvious features of figure 5.11 is that there appears to be some scatter present between the data from repeat tests. The mean of the data is overlaid on the plot, which illustrates the deviation between the data points. However, examination of the data reveals the scatter to be relatively small, emphasised by a maximum standard deviation from the mean of approximately 0.794 W/m²K. This is certainly within the bounds of experimental uncertainty. The most likely source of the scatter was erroneous measurement of the mass flow of condensate, the effects of which propagated through the data reduction. The trends in the data are explained with reference to figure 5.12, which presents the air-side heat transfer coefficient as a function of air mass flow rate. The only difference between figure 5.11 and figure 5.12 is that the magnitude of the data presented in figure 5.11 is smaller due to the surface effectiveness included in the $U_a$ term.

![Figure 5.12: Variation in air-side heat transfer coefficient with air mass flow rate](image)

With respect to figure 5.12, as expected, an increase in air mass flow rate brings about an increase in heat transfer coefficient. An increase in mass flow rate increases the mean air velocity, which alters and disrupts the boundary layer surrounding the finned tube. On a global scale the increase in velocity tends to reduce the boundary layer thickness and, as a consequence, the associated film resistance decreases, with the net result being an increase in heat transfer coefficient. Furthermore, the increase in air velocity augments the internal thermal energy transport throughout the boundary layer. This is manifested by an increase in the advection component of convection.

It was paramount that the air-side characteristics be established to a high degree of certainty, as any errors could be manifested and exacerbated when analysing the condensing characteristics. Therefore, the measurements described thus far in Sections 5.2.2 and 5.2.3 were carried out numerous times, for repeatability purposes, and to compensate for any errors in equipment, measurement technique, etc. In this regard, the air-side aerodynamic and thermal characteristics, which were established
as part of this experimental program, are mean data, averaged-out from the entire
data population obtained from numerous replicate tests. Figure 5.13 and table 5.3
present a summary of the mean air-side characteristics. These parameters established
the air-side boundary conditions, representative of the full-scale MACC, under which
the condensing flow was analysed.

Figure 5.13: Graphical summary of the mean air-side thermal characteristics

Table 5.3: Summary of aerodynamic and thermal characteristics on the air-side of the
reduced-scale ACC

<table>
<thead>
<tr>
<th>( \omega ) (rpm)</th>
<th>( \dot{m}_a ) (kg/s)</th>
<th>( \text{Re}_a ) (-)</th>
<th>( U_a ) (W/m²K)</th>
<th>( h_a ) (W/m²K)</th>
<th>( (R_{th})_a ) (K/W)</th>
<th>( \text{Nu}_a ) (-)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1965</td>
<td>0.074</td>
<td>5498</td>
<td>26.27</td>
<td>27.64</td>
<td>0.0219</td>
<td>33.38</td>
</tr>
<tr>
<td>2302</td>
<td>0.091</td>
<td>6780</td>
<td>27.57</td>
<td>29.07</td>
<td>0.0209</td>
<td>35.10</td>
</tr>
<tr>
<td>2627</td>
<td>0.108</td>
<td>8020</td>
<td>28.66</td>
<td>30.33</td>
<td>0.0201</td>
<td>36.62</td>
</tr>
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<td>2935</td>
<td>0.123</td>
<td>9191</td>
<td>30.04</td>
<td>31.84</td>
<td>0.0192</td>
<td>38.44</td>
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<td>3230</td>
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<td>32.87</td>
<td>0.0186</td>
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<td>3504</td>
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<td>34.11</td>
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</tr>
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<td>0.171</td>
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<td>42.69</td>
</tr>
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<td>4013</td>
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<td>13719</td>
<td>34.99</td>
<td>37.47</td>
<td>0.0165</td>
<td>45.23</td>
</tr>
</tbody>
</table>
5.3 CONDENSATE-SIDE CHARACTERISATION

Once the air-side characteristics were firmly established, it was possible to proceed with condensate-side characterisation. As one of the main objectives of this thesis was to enhance understanding of two-phase condensing flows of steam, the condensate-side characterisation of the reduced-scale ACC consisted of a significant body of experimentation. The characteristics are mainly quantified by the hydrodynamic and thermal properties of the condensing flow. These were examined over a range of parameters applicable to Rankine cycle-based ACCs. This section outlines the experimental methodologies employed in this study, which includes details on the experimental facility, measurement procedure, and data analyses.

The hydrodynamic characteristics of the two-phase condensing flow were quantified in terms of pressure drop measurements, where the objective was to measure the fluidic losses under realistic conditions, examine the effect of various parameters on these losses, and ascertain the quantitative contributions to the losses. The thermal characteristics were evaluated through condensation heat transfer measurements, where quantifying the magnitude of heat transfer, and thermal resistance, for a range of condensing flows was the primary objective. In addition, the measurements were supplemented by two-phase flow regime identification efforts.

In a similar manner to condensation studies by [24, 90, 94, 151, 152], complete condensation occurred in all cases. As noted by Bae et al. [94], condensing along the entire tube length more closely approximates the actual operating conditions in condensers. This contrasts with partial condensation, where an average vapour quality is imposed and a very small variation in quality subsequently occurs through the condenser. A clear limitation of this approach is that it does not reflect the conditions in application and, as such, complete condensation was favoured.

5.3.1 Condensation test facility

The main component of the condensation facility was the reduced-scale, cross-flow ACC described thus far. Supplementing this was a custom-built steam and condensate loop, into which the heat exchanger was integrated as shown in figure 5.14. This steam and condensate loop provided steam, in a slightly superheated state, to the condenser and facilitated the removal of subcooled condensate. Steam was supplied from a boiler with a maximum power rating of 2250 W - according to the product data sheet. The boiler was specified to provide steam at a maximum flow rate of 1 g/s. However, an additional two identical boilers were arranged in parallel with the original to provide higher mass flow rates, if necessary.

Similar to the full-scale test facility, the initial condition of the steam exiting the boiler was considered too “wet”. For suitable measurement conditions, a single-phase flow of dry, or slightly superheated, steam at the condenser inlet was desired. This was not the case initially, with a steam quality of \( x \approx 0.82 \). Therefore, to improve
5.3 CONDENSATE-SIDE CHARACTERISATION

Volume within dashed region is under vacuum during condensation tests.

Figure 5.14: Overview of steam and condensate loop with reduced-scale ACC installed.
the dryness fraction of the steam, a separator was installed downstream of the boiler, and upstream of the condenser, as shown in figure 5.14. The addition of the separator removed the moisture and liquid droplets from the flow, resulting in an improved dryness fraction of $x \approx 0.97$. In both instances, the steam quality was inferred by diverting the steam from the inlet of the reduced-scale ACC into an adiabatic container of water. The increase in the mass of water as the steam condensed was logged, as was the increase in the temperature of the water, all the while ensuring that the temperature remained below the boiling point. The steam quality was thus inferred by applying equation 5.15:

$$x = \frac{\dot{Q}}{\dot{m} h_{fg}} - \frac{h_f}{h_{fg}}$$

(5.15)

where $\dot{Q}$ is the heat transferred to the water, $\dot{m}$ is the change in mass of the water over the time period, $h_f$ is the enthalpy of formation, and $h_{fg}$ is the enthalpy of vaporization.

Downstream of the separator, a series of ball valves were installed to throttle and control the flow rate entering the condenser. The combined effect of the steam separator and throttling across the partially-open valves imparted a degree of superheat to the steam entering the condenser. Generally, this degree of superheat remained within $1 \sim 6 \text{ K} > T_s$, which was inferred by comparing the measured pressure and temperature directly upstream of the condenser inlet. The temperature at the condenser inlet was measured using a K-type thermocouple, identical to those described in Chapter 4, Section 4.4, and calibrated in the same manner. The pressure at the inlet was measured with a 0 - 0.35 bar absolute pressure transducer with an accuracy of ±0.08% of FSD. This transducer was identical to those installed on the full-scale MACC and, accordingly, were also isolated from the system by a valve - to avoid exceeding the recommended operating range.

Connecting the condenser to the stationary steam and condensate loop was flexible hosing. This hosing was connected to the inlet and exit of the condenser and enabled inclination of the condenser, relative to the horizontal position shown in figure 5.14. At the condenser exit, another K-type thermocouple and 0 - 0.35 bar absolute pressure transducer pair were installed to evaluate the flow conditions leaving the condenser tube. The measured temperature at this point, generally, indicated subcooled condensate. The extent of subcooling varied between $5 \sim 18 \text{ K} < T_s$. Ensuring slightly superheated steam at the condenser inlet and slightly subcooled condensate at the exit implied that a quality variation of $1 \leq x \leq 0$ occurred through the condenser, signifying complete condensation.

At the end of the steam and condensate line was the condensate tank, within which the condensate was collected. The time period for which the condensate accumulated was measured, for a given test, after which the tank was disconnected from the line. Thereafter, the tank was weighed using a high-accuracy balance, with a readability resolution of 0.2 g. The known mass of the empty tank was subtracted
from the measured mass to determine the accumulated mass of condensate which, using equation 5.16, was used to calculate the mass flow rate.

\[
\dot{m}_c = \frac{m_c}{t} \tag{5.16}
\]

In line with the full-scale measurements, the condensate-side characterisation experiments were carried-out at sub-atmospheric pressure. Similar to the reasoning given for the full-scale characterisation, vacuum conditions were necessary to ensure that conditions were representative of those in an operational ACC. However, one of the key objectives of the reduced-scale approach was to achieve vacuum conditions without any air leaks occurring, thus facilitating pure-vapour, or single component, condensation. This condition was not possible on the full-scale MACC, as air leaks were an unavoidable feature of the design. As described in Chapter 4, a dephlegmator-vacuum pump system was installed to achieve the desired sub-atmospheric operating pressure. However, despite this, air leakage remained and air was continuously present in the MACC, albeit for a short period of time before it was evacuated by the vacuum pump. Nevertheless, the presence of air during condensation meant that pure-vapour condensation could not be attained at the full-scale.

To explicitly quantify the effects of the condensate layer on heat transfer and, hence, condenser performance at sub-atmospheric conditions, the presence of air must be avoided. Initially, the test facility shown in figure 5.14b was fabricated from standard industrial plumbing fittings. A series of vacuum tests confirmed that these fittings would not be capable of sustaining the desired sub-atmospheric pressures. This is evidenced by the plot of vacuum decay in figure 5.15.

![Figure 5.15: Plot of vacuum decay measured with initial industrial fittings](image)

The data presented in figure 5.15 was measured by condensing a fixed quantity of steam in the region highlighted by the dashed box in figure 5.14a, and monitoring the resulting pressure. With steam flowing through the condenser, the valve downstream of the separator and the valve on the condensate tank exit were closed, thus stopping
the steam flow and creating a closed system. The fans were set to maximum speed and the fixed quantity of steam condensed, creating the initial vacuum seen in figure 5.15. However, after a period of approximately 2400 seconds, it can be seen that the pressure began to increase due to contribution of the partial pressure of air, which was leaking into the system through unsuitable fittings. Thus, single component condensation could not be achieved.

To overcome this issue, and to forgo the introduction of a dephlegmator-vacuum pump system, a new steam and condensate loop was fabricated from specialist vacuum-grade fittings, provided by Swagelok. These fittings were specified to withstand the temperature and pressure range they were expected to be subjected to during testing. Another series of vacuum tests, performed with the newly-designed facility, confirmed that the fittings were capable of maintaining a vacuum. The results from one such typical test are given in figure 5.16.

![Graph A](image1.png)

(a) Log of steady-state vacuum

![Graph B](image2.png)

(b) Steady-state vacuum with x and y-axes scaled

Figure 5.16: Plot of steady-state vacuum achieved with vacuum-grade fittings
As can be seen in Figure 5.16, the steady-state pressure achieved with the vacuum-grade fittings allowed for condensation testing to proceed, confident that the system was free of any significant quantities of air. Ultimately, this yielded pure-vapour condensation.

5.3.2 Parametric analysis

A key objective of this study was to relate the findings in the reduced-scale ACC to the full-scale MACC. One reason for this was that any findings could lead to enhanced understanding of the condensate-side phenomena in the MACC, and in ACCs in general. Ultimately, this could contribute to improved condenser performance. Additionally, acquired data could potentially be used in thermodynamic models to predict condenser performance, with the confidence that it was obtained under realistic conditions. Therefore, it was necessary that the test conditions be representative of those in the full-scale MACC.

As outlined in Chapter 4, Section 4.3, the MACC test conditions were based on operational ACC data, combined with the nominal characteristics from a 50 MW steam turbine. An important feature of the test conditions was operating at sub-atmospheric pressure where, at saturation conditions, the vapour temperature is coupled to the pressure. As the vapour temperatures were accordingly low, the vapour density was relatively small in comparison to operating at atmospheric pressures, or above. The net effect of this was that large vapour velocities were present during testing. This presented a unique opportunity to examine steam condensation at the low pressures, low temperatures, and subsequent large vapour velocities which are encountered in typical Rankine cycle applications. Indeed, the author could only find one other similar study at such conditions, that of Berrichon et al. [22] who, for a given flow rate, observed much larger vapour velocities at low pressures. This characteristic is summarised in Table 5.4, which presents a comparison between the thermophysical properties of steam/water at sub-atmospheric pressure and those at atmospheric pressure, where all properties are taken from Rogers and Mayhew [13].

Table 5.4: Effect of low pressure on steam/water thermophysical properties for a mass flux of $G = 5 \text{ kg/s m}^2$

<table>
<thead>
<tr>
<th>Parameter</th>
<th>$P_s$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>100,000 Pa</td>
</tr>
<tr>
<td>$\rho_v$ (kg/m$^3$)</td>
<td>0.5982</td>
</tr>
<tr>
<td>$\rho_l$ (kg/m$^3$)</td>
<td>958.35</td>
</tr>
<tr>
<td>$\mu_v$ (kg/s.m)</td>
<td>$12.27 \times 10^{-6}$</td>
</tr>
<tr>
<td>$\mu_l$ (kg/s.m)</td>
<td>$281 \times 10^{-6}$</td>
</tr>
<tr>
<td>$u_v$ (m/s)</td>
<td>8</td>
</tr>
</tbody>
</table>
One of the most important parameters to consider for comparative purposes is the Reynolds number, in this case defined as the vapour-only Reynolds number. Section 4.3 in Chapter 4 addresses and summarises the range of vapour-only Reynolds numbers that were examined on the full-scale MACC - $5145 \geq \text{Re}_v \geq 1893$. Applicability of those Reynolds numbers here is limited to the annular-finned geometry MACCs as the reduced-scale tube geometry was almost identical to those. The range of Reynolds numbers examined on the MACC were based around the nominal steam flow rate exiting a 50 MW steam turbine, which would be located upstream of the MACC in a thermoelectric power plant. It follows that a similar range of Reynolds numbers were investigated on the reduced-scale tube. As stated in Section 5.3.1, the maximum flow rate from the boiler in the reduced-scale facility was approximately 1 g/s. This could be varied by altering valve positions and/or adding another boiler in parallel. The range of flow rates examined in this study and their corresponding mass fluxes, vapour-only Reynolds numbers, and liquid-only Reynolds numbers are summarised in table 5.5.

<table>
<thead>
<tr>
<th>$\dot{m}_c$ (g/s)</th>
<th>G (kg/s.m²)</th>
<th>Re$_v$</th>
<th>Re$_l$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.94</td>
<td>1.87</td>
<td>4185</td>
<td>118</td>
</tr>
<tr>
<td>0.81</td>
<td>1.61</td>
<td>3678</td>
<td>93</td>
</tr>
<tr>
<td>0.65</td>
<td>1.29</td>
<td>3027</td>
<td>66</td>
</tr>
<tr>
<td>0.57</td>
<td>1.13</td>
<td>2688</td>
<td>55</td>
</tr>
<tr>
<td>0.42</td>
<td>0.84</td>
<td>2029</td>
<td>35</td>
</tr>
<tr>
<td>0.3</td>
<td>0.59</td>
<td>1478</td>
<td>22</td>
</tr>
</tbody>
</table>

5.3.3 Experimental arrangement

The experimental arrangement employed for pressure drop and condensation heat transfer measurements did not deviate significantly from the test facility described in Section 5.3.1, and shown in figure 5.14. However, specific alterations were made to the experimental set-up for each measurement case, which are described here. This was necessary to ensure satisfactory conditions for each measurement case and allow for certain variables to be measured unobtrusively.

5.3.3.1 Single-phase pressure drop measurements

In contrast to the more rigid MACC arrangement, the reduced-scale ACC experimental set-up was quite flexible, in that various components could be interchanged and modified at will. Hence, it was possible to carry-out experiments that were simply not possible on the full-scale. A prime example of this was the ability to measure the single-phase pressure drop of each individual phase. Quantifying the individual
pressure losses not only improved understanding of their contribution to the overall two-phase pressure drop, but also allowed the data to be expressed in dimensionless form. Measuring the pressure drop associated with the liquid and vapour phases meant that the test facility and arrangement presented thus far was modified as follows.

Common to both the single-phase liquid and single-phase vapour pressure drop experiments was the finned tube, and the location of the pressure transducers - which differed from that shown in figure 5.14b. As can be seen in figure 5.14, the inlet and exit pressure transducers were nominally located upstream and downstream, respectively, of the flexible hosing and were not mounted exactly at the tube. The pressure transducers were mounted in this manner during condensation heat transfer measurements to accommodate the flexible hosing. However, in such a case, it was probable that some fluidic losses would occur through the flexible hosing which would lead to erroneous pressure readings. Therefore, to ensure accurate measurement of the pressure at the condenser inlet and outlet, and thus a more accurate pressure drop measurement, the transducers were re-arranged and fitted directly to the condenser inlet and exit ports. This arrangement permitted exact measurement of the pressure drop through the condenser tube only. The flexible hosing was removed along with sensors which were mounted inside the tube to measure temperature. In this configuration the tube was fixed at a inclination angle of $\approx 10^\circ$.

For the vapour-phase pressure drop measurements, the experimental set-up was, in fact, quite similar to that given in figure 5.14. The only additional feature was a second finned tube installed downstream of, and in series with, the primary tube. This second tube was employed as an after-condenser, where the uncondensed single-phase vapour leaving the upstream tube was condensed by the bank of cooling fans - which were mounted onto the second tube. As only the single-phase pressure loss associated with the vapour phase was desired, condensation only occurred in the after-condenser. The condensate, subsequently, flowed into the condensate tank to be collected and weighed. Figure 5.17 gives an overview of this arrangement.

Figure 5.17: Experimental arrangement for single-phase vapour pressure drop testing
It is important to note that the vapour phase pressure drop measurements were acquired at sub-atmospheric conditions, as the two-phase pressure drop measurements were also carried-out under vacuum. It was necessary that the absolute pressures be reflective of those encountered during two-phase testing as the vapour velocity is much larger at low pressures and, hence, will contribute to increased fluidic losses. The vacuum was created in the same manner as described in Section 5.3.1, for monitoring the vacuum decay. Thus, for a range of steam mass flow rates generated by the boiler, which were controlled by throttling the inlet valve, the absolute pressure at the inlet and outlet of the finned tube was measured. A given flow rate of steam was subsequently condensed in the after-condenser where the mass flow of condensate was inferred by weighing the mass collected over a known time period. Ultimately, the single phase friction factor was determined by equation 5.17;

\[ f = \frac{\Delta P_m D^2 \rho}{L \mu G^2} \]  

(5.17)

The experimental arrangement for the the single-phase liquid pressure drop measurements differed insofar that the measurements did not need to be acquired under vacuum. As the thermophysical properties of water do not vary with temperature to the same extent as the vapour phase does, testing at atmospheric pressure was deemed appropriate. Accordingly, the set-up for the liquid-phase pressure drop measurements was more straightforward, as illustrated in figure 5.18;

![Experimental arrangement for single-phase liquid pressure drop testing](image)

Figure 5.18: Experimental arrangement for single-phase liquid pressure drop testing

The gear pump shown in figure 5.18 generated, and recorded, each mass flow rate required. The water was then pumped through the finned tube, with the inlet and outlet pressures recorded for each flow rate and was, subsequently, collected in a large tank. The pump was connected to this reservoir which re-circulated the water again. The single-phase friction factor was again determined using equation 5.17.

5.3.3.2 Two-phase pressure drop measurements

The experimental set-up for conducting two-phase pressure drop measurements was almost identical to that illustrated in figure 5.17, expect for the omission of the second
finned tube after-condenser. Accordingly, the fans were removed from the second finned tube and mounted on the original tube. As the objective was to quantify the two-phase condensing pressure drop, complete condensation occurred in the reduced-scale lab ACC. The location of the pressure transducers was identical to the location indicated by figure 5.17, in that they were mounted directly at the tube inlet, and outlet. Similarly, the tube was inclined at a fixed angle of approximately 10° to allow condensate drainage.

In a similar manner to the MACC, the pressure transducers on the lab ACC were arranged in pairs consisting of an atmospheric and vacuum pressure transducer. The specialist vacuum transducers were isolated from the system by a valve and were only engaged once the corresponding atmospheric pressure transducer indicated an appropriate sub-atmospheric pressure. The transducers were wired in the same manner as those on the full-scale MACC. All relevant details on the pressure transducers are given in Chapter 4, Section 4.4.

5.3.3.3 Condensation heat transfer measurements

A typical set of condensation measurements were carried-out with the set-up as depicted in figure 5.14. This meant that the pressure transducers were restored to their original positions, upstream and downstream of the flexible hosing - which was omnipresent for heat transfer measurements. In order to characterise the condensation process inside the tube, measurement of the temperature difference from the steam core to the tube wall was necessary. Therefore, the tube was instrumented with a series of thermistors, with one set used to measure steam temperature and a corresponding set assigned to measure wall temperature.

As detailed in Section 5.1, the heat exchanger tube initially had no finned surface on the exterior, simply consisting of a bare aluminium tube. Before the fins were applied to the tube, five collinear sensor sites, spaced evenly at specific increments along the tube axial length, were machined into the tube outer wall. The tube wall was approximately 3.3 mm thick and each site was machined to a depth of approximately 2 mm into the wall - taking care not to protrude through the wall. The thermistors were embedded in the sites using thermal paste and, subsequently, the fins were applied. The objective was to mount the thermistors as close as was practically possible to the inside wall so as to get an accurate reading of the temperature at that point. Although the sensors were not mounted exactly at the inside wall, the small radial depth of approximately 1.3 mm, between the sensor site and inner wall, implied that any thermal resistance and, hence, temperature difference could be assumed minimal. The aluminium tube was also chosen for this very reason, as the high thermal conductivity of aluminium (k \( \approx \) 200 W/m\(^2\)K) further minimised the thermal resistance which could, otherwise, lead to erroneous temperature measurements. Using Fourier’s law of heat conduction, the potential temperature difference, due to the 1.3 mm wall thickness and with \( Q = 2250 \) W, was approximated as 0.2 K. In addition, the thermal resistance due to this infinitesimal thickness was calcu-
lated to be approximately $0.75 \times 10^{-4}$ K/W. Based on these values, no significant appreciable temperature discrepancy can, therefore, be expected.

To measure the steam core temperature, a corresponding set of five thermistors were mounted inside the tube, at specific axial intervals along the centreline. These sites coincided with those machined into the tube wall, thus providing a platform to measure the temperature difference from the steam core to the tube wall. Each corresponding set of thermistors constituted a thermistor pair, culminating with five pairs along the tube length, as shown in figure 5.19. The thermistors inside the tube were mounted in custom-made polycarbonate inserts which ensured the thermistor probe remained along the centreline. As the tube was under vacuum, the thermistor lead wires were soldered to cables, inside the tube, which emanated from a vacuum-pressure feedthrough with a hermetic seal to prevent air leakage into the system. The location of this feedthrough is indicated in figure 5.14a

![Cross-Sectional View](image)

Figure 5.19: Location and arrangement of thermistors in instrumented tube, with angular tube rotation to alter measurement platform highlighted

The final measurement arrangement consisted of five thermistor pairs, which allowed for the condensate-side temperature difference along the axial length to be determined. In addition, the temperature of the cooling air entering the heat exchanger was measured with an array of NTC thermistors similar to those embedded in the tube wall. These thermistors were located upstream of the heat exchanger inlet plane and, thus, measured the bulk ambient air temperature.

As shown in figure 5.19, to enhance measurement resolution on the condensing-side it was possible to rotate the tube around its centreline axis. As a consequence, the collinear set of wall thermistors rotated relative to the position of the steam core thermistors - which remained fixed along the axial centreline. This methodology
permitted measurement of the temperature difference at various locations around the tube circumference. Prior to a set of measurements, the tube was rotated by loosening the inlet and outlet connections and, subsequently, setting the rotation angle. Nine rotation angles were chosen about the r-plane shown in figure 5.19. As shown in the results section in Chapter 7, measurements exhibited symmetry about this axis. Hence, rotation of the tube was limited to $0^\circ \leq \varphi \leq 180^\circ$.

The tube could also be inclined at a number of orientations relative to the horizontal setting shown in figure 5.14. The flexible hosing connecting the heat exchanger to the steam and condensate line permitted this where, at the start of a given test, the condenser inclination angle was set to a specific position. Five inclination angles were investigated in this study varying from $\theta = 0^\circ$ (horizontal) to $\theta \approx 35^\circ$. Due to practical restraints of the test facility, inclination angles greater than $35^\circ$ were not possible. An schematic illustrating the angle of inclination is given in figure 5.20.

![Figure 5.20: Reduced-scale ACC inclination angle](image)

5.3.3.4 Measurement architecture for condensation measurements

To characterise the condensation process, robust, reliable, and highly-accurate temperature sensors were needed. Negative temperature coefficient (NTC) thermistors were chosen as the measurement platform. Thermistors, in general, are characterised by their highly non-linear resistance-temperature response profile, expressed as an R-T curve. In the case of NTC thermistors, the electrical resistance decreases in response to increasing temperature. The set of thermistors mounted inside the tube were specialist thermistors, hermetically sealed for use in harsh environments. On occasions, high temperature, high pressure steam would be flowing through the tube - which qualified as a harsh environment. Couple this with the fact that standard thermistors are not particularly suitable in the presence of moisture (wet steam/condensate), meant that specialist thermistors were necessary. More standardised, unshielded thermistors were used in the tube wall sites, and in the bulk ambient air, as they would not be exposed to steam.

Thermistors were chosen over thermocouples due to their high sensitivity, high accuracy, and fast response time. The NTC thermistors had a nominal resistance
of 10 kΩ at 25 °C and an accuracy of ±0.1 °C at 0 - 70 °C. Before mounting the thermistors, they were calibrated to confirm the validity of the R-T characteristics provided by the manufacturer’s data sheet. Details of this calibration are given in Appendix B. Figure 5.21 presents the resulting R-T curve, based on data procured from the manufacturer, and also as calculated from the Steinhart-Hart [153] equation - solved through the calibration presented in Appendix B.

![Figure 5.21: Resistance-temperature response characteristics of thermistors](image)

An important feature of the R-T curve given in figure 5.21 is the non-linearity exhibited, which is especially pronounced in the temperature range of 0 - 60 °C, within which most of the condensation measurements were expected to occur. The thermistor response to changes in this temperature range is highly sensitive, thus providing good resolution to measure small temperature differences.

Due to the nature of thermistor designs they suffer for a phenomenon commonly known as Joule heating, or “self heating”, whereby the temperature of the thermistor is elevated above the temperature of the measurement medium. This occurs as a result of resistive heating, as the electrical current passes through the resistor material. Therefore, some power must be dissipated in the sensor during measurement. To ensure this is within an acceptable limit and that it does not jeopardise the measurement, the Joule heating must be quantified. This was done through use of a simple voltage divider circuit, with the thermistor integrated with a 10 kΩ resistor. For an excitation voltage range of 1 - 20 V, the power dissipated was calculated using equation 5.18:

$$P = \frac{V_T^2}{R_x}$$  \hspace{1cm} (5.18)

where $V_T$ is the absolute voltage measured between the thermistor and ground. $R_x$ is the thermistor resistance, from which the temperature was determined. An example of data obtained from one such Joule heating test is given in figure 5.22.
The power dissipated from/heat generated within the thermistor is shown to vary in a linear fashion. The thermal dissipation constant is the amount of power required to raise the temperature of the thermistor by 1 °C, under certain ambient conditions. This was determined to be $\approx 4.54 \text{ mW/°C}$, from the reciprocal of the slope of the data given in figure 5.22. However, this could be viewed as a worse-case scenario as this is the thermal dissipation constant in still air. In application, the thermistors will be mounted inside the tube and in the wall, where the bulk convection of the fluid and thermal conductivity will lead to higher heat removal rates. Nevertheless, the supply voltage was carefully selected as 1.5 V to minimise any thermal intrusion on the measurements. This resulted in a maximum power dissipation in the thermistor of $\approx 0.07 \text{ mW}$, and temperature rise due to internal heating of $\approx 0.03 \text{ °C}$.

Each thermistor was integrated as a comparator, in parallel with a reference voltage divider, forming a Wheatstone bridge as shown in figure 5.23. The resistance from the thermistor was expressed as a differential voltage measurement, as given by equation 5.19.

$$V_{out} = V_{ex}\left(\frac{R_2}{R_1 + R_2} - \frac{R_x}{R_3 + R_x}\right) \quad (5.19)$$

where $V_{out}$ is the output voltage, $V_{ex}$ is the excitation supply voltage, and $R_x$ is the thermistor resistance. A Wheatstone bridge measurement circuit was used for a number of reasons. Firstly, the output voltage signal, as shown in equation 5.19, is the result of ratiometric measurement which improves upon the thermistor’s natural sensitivity. Wheatstone bridge circuitry also has the additional benefits of noise reduction, and alleviates Joule heating with each additional comparator - with the current being divided equally among these voltage dividers in parallel.

The output voltages were acquired by a NI-9205 DAQ module which was interfaced with LabVIEW 2011 to record the data. The NI-9205 module is a 16-bit resolution...
5.3 Condensate-side characterisation

Figure 5.23: Wheatstone bridge measurement circuit with thermistor $R_x$ forming one arm

Card containing 32 channels, into which the thermistors were wired, in a reference single-ended (RSE) manner. Signal scanning, sampling, amplification, and conditioning occurred within the DAQ module.

5.3.4 Experimental procedure

Unlike the air-side measurements, described in Section 5.2, all condensate-side measurements were carried out with steam at sub-atmospheric (vacuum) conditions. This was to ensure that test conditions were representative of ACC operational conditions, as certain steam properties can vary dramatically with temperature and pressure. Therefore, at the start of any test, a steady-state vacuum was firstly established, followed by the subsequent data acquisition. For the most part, the methodology was identical for both the pressure drop and heat transfer measurements, with the exception of additional parameters investigated in the heat transfer measurements.

5.3.4.1 System preparation

1. The test facility was arranged as shown in figure 5.14, or as described in Section 5.3.3.2, for the heat transfer or pressure drop measurements, respectively.

2. Condenser inclination angle was set.$^1$

3. Condenser rotation angle was set.$^2$

4. At start-up, the boiler was engaged to produce steam. All the valves in the line were fully open and the valve on the condensate tank was also open to atmosphere.

5. Steam flow was commenced by releasing it from the boiler chamber. Steam at approximately 1.3 bar absolute and 110 °C was allowed to flow through the

---

1 Step was only applicable to heat transfer measurements
2 Step was only applicable to heat transfer measurements
steam line and into the condenser. The steam subsequently flowed into the condensate tank and, ultimately, was vented to atmosphere. This procedure is known as “purging” and serves to remove any residual air pockets, or other non-condensables, from the system. Note: all fans were switched off during this step.

6. After a period of 5-10 minutes of steam flow it was assumed all non-condensables were removed, at which point an isothermal heat exchanger surface should exist. This was confirmed using an IR camera.

7. Once isothermal conditions were confirmed, the valve directly upstream of the condenser inlet and the valve on the condensate tank were closed, thereby creating a closed system containing a fixed quantity of steam.

8. The bank of fans were set to maximum speed ($\approx 4000$ rpm). The reduction in volume as steam collapses to form condensate creates the vacuum. Eventually the saturation temperature reaches the ambient air temperature and accordingly, the saturation pressure reduces to it’s equilibrium value. After a period of approximately 20 - 30 minutes, steady-state vacuum was attained.

9. Steps 1 - 8 were repeated at the start-up of each data acquisition test.

5.3.4.2 **Data acquisition**

1. Once steady-state vacuum was established, steam flow was recommenced by opening the inlet valve upstream of the condenser. This was set to a specific position to set the rate of flow.

2. With the flow rate set, the fan speed was reduced incrementally from the maximum of $\approx 4000$ rpm, for the range of fan speeds examined in Section 5.2.

3. At each fan speed, the temperatures at the condenser inlet and outlet, the core temperatures, and the wall temperatures were all recorded using LabVIEW 2011. Simultaneously, the pressures at the condenser inlet and outlet were recorded.

4. The time period for which the condensate collected was recorded using a stopwatch. The tank was subsequently separated from the system and placed on a scales where the mass of condensate was measured. The mass flow rate was thus inferred as described in Section 5.3.1.

5. Steps 1 - 4 were repeated for the desired range of steam/condensate flow rates.

6. Step 5 was repeated for the range of angular rotations by changing Step 3 in the system preparation.\(^3\)

---

3 Step was only applicable to heat transfer measurements
7. Step 6 was repeated for the range of condenser inclinations by changing Step 2 in the system preparation, thus completing the test matrix shown in table 5.6.4

Table 5.6: Complete experimental test matrix for condensation heat transfer measurements

<table>
<thead>
<tr>
<th>$\theta$ (°)</th>
<th>$\varphi$ (°)</th>
<th>$\dot{m}_c$ (g/s)</th>
<th>$\omega$ (rpm)</th>
</tr>
</thead>
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5.3.5 Data Reduction

The section outlines the methodologies and analyses used to extract useful data from the raw measurements and, subsequently, present this data in a meaningful manner. The data reduction for the condensation measurements is subdivided into a local analysis, and a global analysis.

5.3.5.1 Pressure drop

The two-phase condensing pressure drop measurements were analysed in the same manner as those on the full-scale, as described in detail in Chapter 4, Section 4.6.

5.3.5.2 Local heat transfer analysis

The energy rejected as steam condenses flows through a number of transfer media in series. Each heat transfer medium acts as a barrier to heat flow and is, therefore, considered a thermal resistance. Analysis of steady-state, one dimensional, radial heat flow from the steam core to the ambient air is shown in figure 5.24. The analysis presented in figure 5.24 only considers radial heat flow, where axial heat flow is neglected. However, as a result of the heat transfer process, temperature gradients in the fluid flow direction are expected to exist in the condensate film and tube wall. This would result in heat conduction in both the film and wall, which may have implications on a radial heat transfer analysis. However, according to Shah and

4 Step was only applicable to heat transfer measurements
5.3 Condensate-side Characterisation

Sekulić [19], longitudinal heat conduction in the condensate and wall can be assumed negligible for the following conditions:

\[ Pe > 10 \quad \& \quad x^* \geq 0.0005 \quad (5.20) \]

where \( Pe \) is the Péclet number, which is defined as the ratio of the rate of advection (rate of thermal energy transported to the fluid) to the rate of diffusion (thermal energy conducted axially within the fluid). \( x^* \) is the dimensionless axial distance. Both terms are defined in equation 5.21:

\[
Pe = \frac{\rho \bar{u} C_p D}{k} = \frac{\bar{u} D}{\alpha} = Pr \cdot Re \quad x^* = \frac{x}{D \cdot Pr \cdot Re} \quad (5.21)
\]

For all investigated experimental conditions, equation 5.20 was satisfied and, thus, only radial heat flow was considered in the analysis.

As can be seen in Figure 5.24, the local radial heat flow from the steam core to the bulk air consists of the following processes: convection through the condensate film to the tube wall, conduction through said wall and fins, and subsequent convection from the wall and fins to the bulk air, through the boundary layer film. Assuming all the heat flows one dimensionally in the radial direction, the local heat transfer for a single angular position can be expressed as:

\[
d\dot{Q}_i = h_{c,i} dA_c (T_{s,i} - T_{w,i}) = -k_w d\theta dL \left( \frac{T_{w,i} - T_{w,0,i}}{\ln (r_o/r_i)} \right) = U_{a,i} dA_a (T_{w,0,i} - T_\infty) \quad (5.22)
\]
where $dA_c$ is the elemental heat transfer area on the condensing-side (i.e. inside the tube) and is given as $r_d \theta dL$. A similar term is used for the elemental tube wall heat transfer area. Finally, the elemental heat transfer area on the air-side (i.e. on the tube exterior) is given by $dN_f(r_f^2 - r_e^2)\theta$, where $N_f$ is governed by the elemental length, $dL$, in the following relationship:

$$dN_f = \frac{dL}{S_f}$$

(5.23)

$S_f$ in equation 5.23 is constant as the fin spacing. Equation 5.23 can be recognised as a re-arrangement of equation 5.1, which was previously presented in Section 5.1. As there is no appreciable thermal resistance and, hence, temperature difference between the measured steam core temperature, $T_s$, and the temperature at the condensate film interface, $T_{cf}$, these two terms are assumed equal. Thus, $T_s$ can be used in the condensing convection component in equation 5.22.

In general, most analyses assume that the wall thermal resistance is negligible. This is justified based on the small wall thickness, combined with the high thermal conductivity of the wall material as discussed previously in Section 5.3.3.3. If this is assumed to be the case, the inner wall temperature, $T_{wi}$, is said to be equivalent to the outer wall temperature, $T_{wo}$, and, therefore, no appreciable temperature gradient exists in the wall. This idealisation reduces the inner and outer wall temperatures to a single value - $T_w$, which the wall thermistors in this study are said to measure. Under the assumptions laid forth, the heat flow on the air-side can be calculated using equation 5.24:

$$d\dot{Q}_{a,i} = U_{a,i}dA_a(T_{w,i} - T_\infty)$$

(5.24)

where $U_{a,i}$ is the local, overall air-side heat transfer coefficient, which is equivalent to the global, overall heat transfer coefficient based on the air-side established in Section 5.2. The condensing heat transfer coefficient, $h_c$, can, thus, be calculated using equation 5.25:

$$h_{c,i} = \frac{d\dot{Q}_i}{dA_s(T_{s,i} - T_{w,i})}$$

(5.25)

where $d\dot{Q}_i$ is the heat transfer rate given by equation 5.24. In dimensionless form, the heat transfer coefficient is expressed as the Nusselt number as follows:

$$Nu_{c,i} = \frac{h_{c,i}D}{k_l}$$

(5.26)

The analysis described thus far, and illustrated in figure 5.24, is for that of a single tube angular position, $\varphi$. The same analysis is applicable to any angular tube rotation given in table 5.6. Figure 5.25 presents the same thermal resistance network for a number of arbitrary angular tube rotations.
As the tube is rotated about its axis, the wall thermistors rotate accordingly. As seen in figure 5.25, the result of this is that the thermal resistance network is shifted from its original position to that given by the position of the wall thermistors, which are in unison with the angular tube position. It is important to note that even though the thermal resistance networks depicted in figure 5.25 are in parallel, they are not quantified simultaneously. The thermal resistance networks were measured and analysed individually, for each specific angular position.

The location of the wake region behind the tube is also illustrated in figure 5.25. This has not been measured, nor validated, but is an assumed representation, based on the work of Hu and Jacobi [154], to demonstrate the air flow field characteristics around the tube as it was rotated. For each tube rotation, the fan bank was rotated in unison with the tube, resulting in the air flow being aligned parallel with the angular position of the tube and wall thermistors. This resulted in the wake region being consistently located downstream of the thermal resistance network being analysed, as shown in figure 5.25. The significance of this approach was that the structure of the air flow field in the vicinity of the wall measurement region, essentially, remained relatively constant regardless of tube rotation angle. This was deemed necessary to compensate for the fact that the air-side thermal measurements, described in Section 5.2, are bulk measurements - as opposed to local.

Measuring the local air-side heat transfer coefficient around the perimeter would have been extremely difficult and outside the scope of this work. Even prominent studies, such as [154, 155], on local air-side convective behaviour around an annular-finenned tube adopted a mass transfer methodology. In these studies, the authors found that it was practically impossible to directly measure the thermal characteristics. The justification was that any attempt to measure temperature in such localised regions would have disrupted the flow field and, hence, altered the heat transfer behaviour. Hu and Jacobi [154] experimentally measured the variation in local Sherwood number around the tube and found a variation ranging from 25 - 40%. This certainly
implies that the local heat transfer coefficient will vary around the tube circumference. However, by at least maintaining the same flow structure at any given angular position, the heat transfer coefficient is expected to stay relatively constant. Due to a lack of any analytical solutions or empirical data, the bulk coefficients which were measured, were adopted in the local analysis. Consequently, these values were used in the data reduction equation 5.24 to calculate the local heat transfer rate.

The result of measuring the temperature difference from the steam core, $T_s$, to the wall, $T_w$, along the length of the tube, for a range of angular rotations, led to the development of a measurement grid - shown in figure 5.26.

![Figure 5.26: Local measurement grid](image)

This grid is comprised of an array of local data points, where each given point is a temperature difference measurement. The y-axis is the angular rotation and the x-axis is the axial location of the thermistor sites. By applying the local data reduction methodology outlined in this section, a series of contour plots of the various condensing quantities could be generated for presenting the results. As shown in the results chapter, these contour plots could be interpreted to visualise the condensing flow.

5.3.5.3 **Global heat transfer analysis**

The condensing heat transfer characteristics outlined thus far can also be expressed in terms of the bulk properties. Although local quantities enhance understanding of fluid and heat transfer transport mechanisms, the bulk properties are often more useful for correlating data and as input variables in models. From the same set of condensation measurements from which the local properties were evaluated, the global properties can be determined with reference to figure 5.27.
Assuming steady-state, application of the 1st law of thermodynamics on the condensing-side yields the enthalpy rate equation for isothermal heat rejection from the steam as follows:

$$Q_{\text{rej}} = \dot{m}_c h_{fg} \quad (5.27)$$

where $\dot{m}_c$ is the mass flow of condensate. For all tests reported in this thesis, full condensation occurred such that all the vapour was converted into condensate. Thus, $\dot{m}_c$ is equivalent to the vapour flow rate, $\dot{m}_v$. As full condensation occurs, an energy balance on the condenser imposes that the heat rejected by the condensing steam is absorbed by the air flow, in accordance with equation 5.28:

$$Q_{\text{gain}} = \dot{m}_a C_p (T_{\text{out}} - T_\infty) \quad (5.28)$$

where $\dot{m}_a$ is the mass flow rate of air, established in Section 5.2. $T_{\text{out}}$ is the air temperature at the outlet of the heat exchanger. This is calculated from equation 5.29, in which the heat exchanger effectiveness, $\varepsilon$, was also established in Section 5.2.

$$T_{\text{out}} = \varepsilon (\bar{T}_w - T_\infty) + T_\infty \quad (5.29)$$

An energy balance on the condenser equating the loss of enthalpy of the steam to the gain in enthalpy of the air is expressed as:

$$\dot{Q} = \dot{m}_c h_{fg} = \dot{m}_a C_p (T_{\text{out}} - T_\infty) \quad (5.30)$$

The final equation for determining the bulk heat flow is the heat transfer rate equation, defined by equation 5.31:

$$\dot{Q}_{\text{trans}} = U_a A_a (\bar{T}_w - T_\infty) \quad (5.31)$$
where $U_a$ is the overall air-side heat transfer coefficient. $\bar{T}_w$ and $T_\infty$ are the axially-averaged and circumferentially-averaged wall temperature, and ambient temperature, respectively. Equation 5.31 can be equated with either of the energy balance equations 5.27 or 5.28 to determine a heat-energy balance. Figure 5.27 illustrates such a heat-energy balance from experimentation. This is based on the bulk heat rejection as determined using equation 5.27 and the global heat transfer rate, evaluated with equation 5.31, for the range of steam/condensate flow rates employed.

![Figure 5.28: Heat-energy balance from condensation measurements for range of steam/condensate flow rates investigated](image)

Ultimately, the global condensing coefficient is calculated, from the bulk heat transfer given by equation 5.27, 5.28, or 5.31, as follows:

$$\bar{h}_c = \frac{Q}{A_s (\bar{T}_s - \bar{T}_w)}$$  \hspace{1cm} (5.32)

where $A_s$ is the heat transfer area inside the tube. $\bar{T}_s$ is the mean, axially-averaged steam core temperature. Expressed non-dimensionally as Nusselt number, equation 5.32 takes the form:

$$\bar{N}u_c = \frac{\bar{h}_c D}{\bar{k}_f}$$  \hspace{1cm} (5.33)

Finally, the total and condensate-side thermal resistance can be defined as:

$$(R_{th})_{tot} = \frac{\bar{T}_s - T_\infty}{\dot{Q}} \quad (R_{th})_c = \frac{\bar{T}_s - \bar{T}_w}{\dot{Q}}$$  \hspace{1cm} (5.34)

### 5.4 UNCERTAINTY ANALYSIS

An uncertainty analysis conforming to the same method outlined in Chapter 4, Section 4.7 was undertaken to determine the uncertainty associated with the reduced-
scale ACC measurements. Maintaining consistency with the uncertainties reported in Chapter 4, the uncertainty of the primary variables is presented in table 5.7, whilst the uncertainty of the calculated parameters is given in table 5.8.

Table 5.7: Uncertainties associated with the reduced-scale ACC primary variables

<table>
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<th>Variable</th>
<th>Description</th>
<th>Units</th>
<th>Uncertainty</th>
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<tbody>
<tr>
<td>$D$</td>
<td>Tube diameter</td>
<td>m</td>
<td>1%</td>
</tr>
<tr>
<td>$\dot{m}_a$</td>
<td>Air mass flow rate</td>
<td>kg/s</td>
<td>3.4%</td>
</tr>
<tr>
<td>$\dot{m}_c$</td>
<td>Condensate mass flow rate</td>
<td>kg/s</td>
<td>2.2%</td>
</tr>
<tr>
<td>$T_s$, $T_w$, $T_\infty$</td>
<td>Temperature measurements</td>
<td>K</td>
<td>0.5%</td>
</tr>
<tr>
<td>$\Delta P$</td>
<td>Pressure drop</td>
<td>Pa</td>
<td>1.8%</td>
</tr>
</tbody>
</table>

Table 5.8: Uncertainties associated with the reduced-scale ACC calculated parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Units</th>
<th>Uncertainty</th>
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<tbody>
<tr>
<td>$G$</td>
<td>Mass flux (velocity)</td>
<td>kg/sm$^2$</td>
<td>5.5%</td>
</tr>
<tr>
<td>$h_c$</td>
<td>Condensing coefficient</td>
<td>W/m$^2$K</td>
<td>6.3%</td>
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<tr>
<td>$Nu_a$</td>
<td>Air-side Nusselt number</td>
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<td>9.1%</td>
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<tr>
<td>$Nu_c$</td>
<td>Condensing Nusselt number</td>
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<td>7.9%</td>
</tr>
<tr>
<td>$\dot{Q}$</td>
<td>Heat transfer rate</td>
<td>W</td>
<td>4.1%</td>
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<tr>
<td>$Re_a$</td>
<td>Airflow Reynolds number</td>
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<td>5.8%</td>
</tr>
<tr>
<td>$Re_v$</td>
<td>Vapour Reynolds number</td>
<td>-</td>
<td>6.6%</td>
</tr>
<tr>
<td>$(R_{th})_a$</td>
<td>Air-side thermal resistance</td>
<td>K/W</td>
<td>8.7%</td>
</tr>
<tr>
<td>$(R_{th})_c$</td>
<td>Condensate-side thermal resistance</td>
<td>K/W</td>
<td>7.2%</td>
</tr>
<tr>
<td>$\Delta P_f$</td>
<td>Frictional pressure drop</td>
<td>Pa</td>
<td>6%</td>
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<tr>
<td>$\Delta P_{mom}$</td>
<td>Momentum pressure recovery</td>
<td>Pa</td>
<td>7.7%</td>
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It should be noted that, to preserve the clarity and coherency of the results presented in Chapters 6 and 7, the corresponding uncertainties which have been established in Chapter 4 and this chapter, respectively, are purposely omitted from the presentation of the results. As such, experimental uncertainty bands are not included in the plots presented in this thesis.

5.5 summary

A single annular-finned circular tube, which is almost an exact replica of those which constitute the heat exchanger in two of the full-scale MACC modules, was presented as the candidate geometry for experimentation in this chapter. In order to reproduce the experimental conditions from the full-scale, and simulate operational ACC conditions in general, air-cooling was employed. This chapter described the test facilities, methods, and data analyses used to characterise the air-side boundary conditions.
of the reduced-scale ACC. Both the aerodynamic and thermal boundary conditions quantified were fully representative of those attained on the full-scale. Following this, the experimentation relating to the condensate-side characterisation was outlined. A detailed description of the test facility, instrumentation, and methodologies to obtain both global and local condensation heat transfer measurements was presented. Novel features of the research, such as steam condensation at vacuum conditions and local heat transfer measurements, were highlighted throughout this chapter. The majority of the findings from this experimentation are given in Chapter 7.
This chapter presents hydrodynamic and thermal measurements from a series of full-scale MACC prototype designs. These measurements quantified the condenser performance and the condensate-side characteristics under typical Rankine cycle operating conditions. Measurements of condenser temperature and pressure, as a function of fan speed, established the qualitative and quantitative relationships between fan speed and condenser operating performance. Such relationships described the extent to which the MACC could control and manipulate the exit conditions of a steam turbine in a Rankine cycle thermoelectric power plant. Ultimately, the relationships lead to the development of fan algorithms, and were incorporated into thermodynamic predictive models.

Condensate-side characteristics presented herein are mainly expressed in terms of the two-phase condensing pressure drop, and condensate-side thermal resistance. Both parameters were evaluated for a range of MACC prototypes, each consisting of a unique heat exchanger geometry. Variables which were investigated throughout the condensate-side characterisation program, and which are reported in this chapter, include air mass flow rate (fan speed), steam/condensate mass flow rate, condenser inclination angle, and size of vacuum pump. Some of these variables were only examined for certain MACC modules, but the conclusions were generalised to be applicable to all cases.

6.1 PERFORMANCE CHARACTERISTICS

The thermal and hydraulic performance of an ACC is generally expressed with reference to the air-side. The prevalent metrics for quantification are the overall heat transfer coefficient, $U$, or thermal resistance, $R_{th}$, and pressure drop, $\Delta P$. Separate air-side characterisation measurements were carried-out to establish the air-side surface heat transfer and flow friction characteristics. These measurements, known as surface basic characteristics, are reported in Moore et al. [124, 123] and essentially solved the heat exchanger rating problem. However, to close the rating problem, the terminal values of steam temperature, $T_s$, and pressure, $P_s$, were needed. These measurements are presented herein, thereby quantifying the condensate-side performance.

6.1.1 Preliminary six row circular tube condenser measurements

The first set of condensate-side measurements were acquired from a pre-existing six row prototype module. In its initial state the module was not fully instrumented and,
consequently, only absolute pressure measurements were obtained in accordance with the method described in Chapter 4, Section 4.5. Nevertheless, these measurements provided the initial insight into condenser performance, and established a measurement platform with which to broaden and refine moving forward. For a range of condensate flow rates, the absolute pressure was measured as fan speed was varied, with the results presented in figure 6.1:

![Figure 6.1: Variation in measured pressure as a function of fan speed for a range of condensate flow rates for the six row circular tube condenser](image)

The pressure measurements given in figure 6.1 exhibit some important novel features and characteristics associated with the MACC concept. Of particular significance is the confirmation that, at a given mass flow rate of steam/condensate, varying the fan speed alters the condenser temperature which, accordingly, leads to the change in pressure seen in figure 6.1. For all flow rates examined, increasing fan speed results in a reduction in pressure. Conversely, reducing fan speed leads to an increase in pressure. These relationships are indicative of the capability of the MACC to control and manipulate the pressure at the outlet of a steam turbine in a thermoelectric power plant.

Regarding the magnitude and shape of the pressure curves given in figure 6.1, it is apparent that the variation in measured pressure is much more pronounced at the higher flow rates. For example, comparing one of the larger flow rates of 0.18 kg/s with the lowest flow rate of 0.10 kg/s, it is clear that the higher flow rate results in much greater pressures in the MACC. This is to be expected and is consistent with observations in the literature. However, as fan speed increases, the subsequent reduction in pressure at 0.18 kg/s is much more dramatic than the corresponding decrease in pressure at 0.10 kg/s. This is emphasised by examining the variation in pressure over a common fan speed range where, at 0.18 kg/s, increasing fan speed from \( \approx 570 \text{ rpm} \) to \( \approx 1000 \text{ rpm} \) reduces the pressure by approximately 40%. This contrasts with the decrease in pressure observed at 0.10 kg/s where only a 10%
reduction in pressure occurs across the same range of fan speeds. In general, based on the evidence in figure 6.1, it appears that the lower flow rates seem to be largely insensitive to changes in fan speed. The significance of this and the other trends in the data presented will become apparent when explained in detail with the proceeding results, where temperature measurements compliment the pressure data.

6.1.2 *Four row circular tube condenser measurements*

With the initial measurements on the pre-existing six row MACC module established, the condensate-side characterisation progressed to the second iteration - a four row module. Based on air-side predictions, the four-row design was shown to outperform the original six-row condenser. Thus, the design was considered as a candidate for commercial deployment and, consequently, further experimentation was deemed necessary. This module was fully instrumented meaning that temperature data was recorded in conjunction with pressure data. The observations from the preliminary testing provided a basis on which these subsequent tests were selected.

6.1.2.1 *Vacuum pump assessment*

One of the conclusions from the preliminary measurements was that condenser pressure appeared insensitive to changes in fan speed at low condensate mass flow rates. It was speculated that this could be attributable to the inability of the vacuum pump to adequately displace the suspected greater quantities of air leaking into the MACC at such conditions. In fact, increased air ingress was actually confirmed, in accordance with Dalton’s law, whereby the presence of air in the MACC was manifested by a partial pressure. A larger partial pressure subsequently resulted from more air in the system. This phenomenon is explicitly quantified in figure 6.2.

![Graphs showing partial pressure variation with fan speed and pressure differential](image)

Figure 6.2: Variation in partial pressure of air in the four row circular tube MACC

The partial pressure of air given in figure 6.2 was inferred by subtracting the vapour saturation pressure (evaluated from the vapour saturation temperature) from the total measured pressure. As can be seen, this partial pressure increases quite
dramatically with both fan speed and pressure differential between the MACC and atmosphere. In figure 6.2a, it can be seen that the partial pressure increases as condensate flow rate is reduced, for a given fan speed, and also as fan speed is increased - at a given condensate flow rate. In both these cases, the primary factor causing the increase in partial pressure is the reduction in condenser pressure, which equates to an increase in pressure differential. Indeed, when presented as a function of pressure differential, this becomes more obvious. Figure 6.2b illustrates an almost exponential increase in the partial pressure of air with increasing pressure differential. This implies that much more air is leaking into the MACC at the larger pressure differentials present at the lower condensate flow rates and higher fan speeds. Hence, it follows that this pressure differential must be principally responsible for promoting air leakage into the MACC. Although the data presented in figure 6.2 is for the four row circular tube condenser, the same qualitative relationships and conclusions apply to the other MACC prototypes.

The ability of a given vacuum pump to displace a particular flow rate is dependent on the pressure rise across the pump. The smaller, original pump installed on the six row condenser (henceforth referred to as Pump A) permitted a given flow rate of air to be evacuated at a particular pressure rise - which was the difference between the air partial pressure and the atmospheric pressure. However, the same flow rate of air could be displaced across a larger pressure rise with a larger capacity vacuum pump. The net effect of this is that the partial pressure of air must decrease - to satisfy a larger pressure rise. Hence, it was suggested that Pump A could only exhaust the air leakages at relatively small pressure rises, thus implying that the resulting large partial pressure of air remaining in the system offset the expected reduction in steam pressure. Ultimately, this was responsible for the apparent insensitivity in measured pressure observed, at low flow rates and high fan speeds, in figure 6.1. An example demonstrating this phenomenon is shown in figure 6.3a.

Similar to figure 6.2, the data presented in figure 6.3a is from the four row condenser as the six row design was not fully instrumented and, thus, the necessary saturation pressure could not be determined. Irrespective of the lack of six row data, the qualitative trends exhibited in figure 6.3a should be indicative of those on the six row MACC with Pump A installed. It is clear that the contribution of the partial pressure of air is quite significant to the overall, measured pressure. In fact, at maximum fan speed, which results in the greatest pressure differential, the partial pressure of air accounts for approximately 30% of the measured pressure. This partial pressure is also seen to be a strong function of fan speed, serving to negate the inferred reduction in steam pressure, $P_s$, thereby resulting in the relatively stagnant measured pressure characteristics seen. As the vacuum pump could only maintain a certain air removal rate at a specific pressure rise, increases in fan speed resulted in a negligible reduction in pressure being observed. Therefore, a larger capacity vacuum pump (referred to as Pump B) was installed, and the measurements were repeated to investigate the hypothesis that the vacuum pump was responsible for the pressure
insensitivity. These measurements are shown for the same condensate flow rate in figure 6.3b, with the results from other flow rates given in figures 6.3c and 6.3d.

Figure 6.3: Comparison between the variation in pressure contributions with fan speed for Pump A (a) and Pump B (b) - (d), for the four row circular tube condenser

Comparing figure 6.3a and, specifically, figure 6.3b, it is clear that significant differences exist between the pressure measured with Pump A and that measured with Pump B installed. For the same condensate mass flow rate of 0.10 kg/s, the pressure achieved with Pump B is consistently, and substantially, lower across the same range of fan speeds. This is primarily due to the reduced partial pressure of air, which Pump B can adequately displace at larger pressure rises. Furthermore, it can be seen in all the measurements associated with Pump B (figures 6.3b - 6.3d) that the rate of pressure change as a function of fan speed is much greater than that with Pump A. The implication of this on a thermoelectric power plant is that the degree of control over condenser pressure and, hence, turbine gross output is greater if a large vacuum pump is used to displace air leaks. Hence, the sizing of the vacuum pump can have a significant influence on plant performance. The culmination of this investigation was that for all subsequent measurements presented in this thesis, Pump B was employed. All details relating to the vacuum pumps are given in Chapter 4, Section 4.2.
6.1.2.2  *Four row condenser performance characteristics*

The absolute pressure measurements from the six row module were repeated for the four row condenser, with the results presented in figure 6.4 in terms of air mass flow rate. Each air flow rate corresponds to a specific fan speed employed during experimentation. As the four row module was fully instrumented, temperature measurements were also acquired, as given by figure 6.5. The range of steam/condensate flow rates examined were reflective of those expected in typical MACC applications.

![Figure 6.4](image1)

**Figure 6.4:** Variation in measured and predicted pressure with air mass flow rate for a range of condensate flow rates for the four row circular tube condenser

![Figure 6.5](image2)

**Figure 6.5:** Variation in measured and predicted temperature with air mass flow rate for a range of condensate flow rates for the four row circular tube condenser

As was expected, the pressure measurements from the four row condenser shown in figure 6.4 exhibit the same general trends as those observed in the six row meas-
urements. Specifically, increasing the mass flow rate of air results in a reduction in pressure, increasing the steam/condensate flow rate leads to increased pressure, and vice-versa. The most notable difference between the measurements presented in figure 6.4 and those in figure 6.1 is the lower pressures which were achieved in the four row measurements. As a result of the lower flow rates employed (0.06 kg/s and 0.08 kg/s), the pressure will, inherently, be lower but 0.10 kg/s is common to both data sets and is much lower and sensitive to changes in air flow rate for the four row case. As discussed heretofore, this is attributable to the larger vacuum pump with the ability to continually displace the air leaks at lower pressures.

The temperature measurements in figure 6.5 illustrate the same overall characteristics as those established in the pressure measurements. However, as the steam temperature and pressure are interdependent, examining the temperature measurements provides an explanation for the shape of the curves. As air flow rate increases, the thermal resistance of the air film on the tube exterior decreases and, consequently, the air-side heat transfer coefficient increases. This leads to a reduction in condenser temperature and, hence, condenser pressure. Therefore, it is the variation in temperature which actually causes the variation in pressure. Furthermore, as the air flow rate is infinitely increased, the condenser temperature asymptotically approaches the ambient air temperature - which is the thermodynamic limit of the heat sink. This can be seen in figure 6.5 where the slope of any given temperature curve has a more pronounced gradient at low air flow rates, with the gradient saturating towards the higher air flow rates as the condenser temperature nears the ambient temperature. Consequently, the pressure must also approach the saturation pressure for the given ambient temperature.

The theoretical variation of pressure, and temperature, with air flow rate is also given in figures 6.4 and 6.5, respectively. This theory is based on an air-side-only approach, as described in Chapter 3, Section 3.2. The implications of this approach is discussed in detail in the forthcoming single row measurements. Nevertheless, it is important to note at this point that the theory, essentially, confirmed and validated the qualitative relationships established from the measurements. Ultimately, this lent a degree of confidence to the experimentation.

The test matrix for the four row design also consisted of investigating the effects of condenser inclination. However, as varying this parameters did not lead to an appreciable change in the measured temperature, or pressure, its inclusion as temperature, or pressure, plot was deemed to be unnecessary. The effects of condenser inclination angle are better quantified in terms of thermal resistance, measurements of which are presented in Section 6.3.

6.1.3 Single row rectangular tube condenser measurements

In comparison to the four row test matrix, the measurement program for the single row, rectangular tube design was abridged by eliminating vacuum pump size and
condenser inclination as variables. Hence, evaluating the performance of the single row condenser simply consisted of measuring the temperature and pressure as function of air flow rate, for a range of representative condensate flow rates, with Pump B installed and at the nominal inclination angle of \( \sim 40^\circ \). These results are given in figures 6.6 and 6.7, where the predicted response is also given.

![Figure 6.6: Variation in measured and predicted pressure as a function of air mass flow rate for a range of condensate flow rates for the single row rectangular tube condenser](image)

![Figure 6.7: Variation in measured and predicted temperature as a function of air mass flow rate for a range of condensate flow rates for the single row rectangular tube condenser](image)

It is evident from figures 6.6 and 6.7 that the relationship between pressure and air flow rate, and temperature and air flow rate, exhibits the same characteristics as were identified in the previous condenser designs. These characteristics and
the reasons for them have been discussed in detail hitherto. Overlaying the measured data is the response as predicted using the effectiveness-NTU model outlined in Chapter 3, Section 3.2. As alluded to in the previous section, this model uses air-side theory only to determine an idealised condensate-side response, and has been validated through comparison with specific air-side measurements. In modelling the idealised response, the condensate-side resistance is purposely neglected - as described in Chapter 3, Section 3.2. Therefore, differences will, inherently, exist between the measured condensate-side data and that as predicted from the air-side-only approach. However, it is noticeable that the idealised variation predicts the same trends as those observed in the measurements, with insensitivity at high air flow rates being a particular feature.

As discussed in Section 6.1.2, insensitivity at high air flow rates is mainly attributed to the condenser temperature asymptotically approaching the ambient temperature. However, the fact that insensitivity is a feature of the data generated by the air-side model suggests that the aerodynamic boundary conditions also have an influence. In general, an increase in Nusselt number occurs as Reynolds number increases - due to the longer entry lengths that transpire and the thinner boundary layers associated with the high Reynolds number laminar flow. The subsequent velocities in the fin channels are larger in this developing region. Hence, an increase in air flow rate (Reynolds number) leads to an increase in air-side heat transfer coefficient (Nusselt number) which, ultimately, decreases condenser temperature. However, as shown by Sparrow [127], and Sparrow and Gregg [55], the rate of increase in Nusselt number diminishes at higher Reynolds numbers. This is manifested in the relationship between air flow rate and temperature seen in figure 6.7. Although air-side heat transfer coefficient increases as air flow rate increases, the rate of increase is less at the higher air flow rates. As a result, the rate of temperature decrease reduces. This air-side characteristic is, thus, also partially responsible for the insensitivity observed in the measurements.

For all condensate flow rates considered, the predicted variation in pressure and temperature is consistently lower than the corresponding measured values. The difference is more pronounced for the pressure values due to the non-linear relationship between saturation temperature and pressure. Indeed, focussing on the variation of pressure with air flow rate given in figure 6.6, the difference between the measurements and the modelling approach varies from as low as 3% to as large as 33%, with an average difference of about 24%. Such large discrepancies highlight the limitations associated with attempting to predict condensate-side performance using air-side theory only. By disregarding the steam-side thermal resistance, as was also done in modelling approaches by Hongbin and Ling [51], Pieve and Salvadori [48], and Martín [46], the model consistently under-predicts the actual, measured pressure and temperature. This suggests that a relatively large thermal gradient exists on the condensate-side, from the steam core to the condenser surface which, contrary to current methodology, should not be assumed negligible.
6.2 PRESSURE DROP MEASUREMENTS

In conjunction with temperature and absolute pressure measurements, steam condensation (i.e. two-phase) pressure drop measurements were also acquired from each of the MACC prototypes. However, as the circular tube and rectangular tube geometries were markedly different, the flow conditions through each tube were, inherently, dissimilar. As a consequence, the pressure drop measurements are presented here separately - one set encompassing both the six and four row circular tube measurements, and another separate set for the rectangular tube condenser. In addition to the raw measurements, reduced data, which quantifies the various components constituting the two-phase condensing pressure losses in the circular tube condensers, are presented. The frictional component of this data is compared to a number of relevant and widely-cited two-phase frictional pressure drop correlations for round tubes, given in Chapter 2, with a view to establishing the most applicable model.

6.2.1 Six row and four row circular tube condenser measurements

The pressure drop from the inlet to the outlet of the tube bundle was measured for a range of air flow rates, and steam/condensate flow rates. These measurements provided two distinct relationships - specifically the variation of pressure drop with air flow rate, and the variation of pressure drop with steam/condensate flow rate. As excessive pressure losses on the condensing-side of an ACC can be detrimental to performance, establishing these quantitative relationships was seen as an important part of the experimental program - to ensure that pressure losses were within an acceptable limit.

6.2.1.1 Relationship between air flow rate and pressure drop

Figure 6.8 presents the variation in measured pressure drop, with air flow rate, for the range of condensate flow rates investigated on the six row circular tube condenser. For clarity and conciseness, only the six row measurements are presented here as including the four row measurements on the same plot would render it difficult to distinguish between the data sets. In any case, the four row measurements depict the same qualitative relationships as those shown in figure 6.8 and, as such, it was unnecessary to include them here. Instead, the four row measurements form part of the data set for expressing the relationship between pressure drop and steam/condensate flow rate, given later in this section.

Figure 6.8 illustrates some important features of the air flow rate-pressure drop relationship. Across all the condensate flow rates investigated, there is a noticeable trend of increasing pressure drop with air flow rate. This trend occurs due to the variation in the air-side characteristics which, ultimately, alters the two-phase fluid dynamic behaviour inside the tubes. As shown previously in figures 6.5 and 6.7, increasing air flow rate causes a reduction in the vapour temperature. This temperature
6.2 Pressure Drop Measurements

Figure 6.8: Measured pressure drop as a function of air mass flow rate for a range of condensate flow rates for the six row circular tube condenser.

Reduction is accompanied by a decrease in vapour density, which must be offset by an increase in the vapour velocity - to maintain continuity. This phenomenon is illustrated most clearly by figure 6.9, which presents the inferred inlet vapour velocity as a function of air flow rate.

Figure 6.9: Inferred inlet vapour velocity as a function of air mass flow rate for a range of condensate flow rates for the six row circular tube condenser.

Figure 6.9 illustrates a clear trend of vapour velocity increasing with both condensate mass flow rate, and air mass flow rate. Intuitively, an increase in velocity was expected with an increase in condensate flow rate. However, it was somewhat surprising to note the effect of varying air flow rate where, for any given condensate flow rate, an increase in air flow resulted in an increase in vapour velocity. This phenomenon is much more apparent at the higher condensate flow rates, with greater
increases in velocity observed as air flow was increased. Hence, the relationship is consistent with the response to changes in air flow rate seen in the absolute pressure measurements presented earlier. As large variations in the absolute pressure and, hence, temperature are not observed at the low condensate flow rates, the vapour density remains relatively constant. The net effect of this is that the resulting variation in vapour velocity is much smaller than that at the high condensate flow rates. Although the vapour velocity expressed in figure 6.9 is the velocity at the inlet to the tubes, the increase brought about by increasing air flow should still remain valid for the velocity at any axial point along the tubes. This is despite the fact that vapour velocity actually decreases through the tubes, as the steam condenses. It is well recognised in the literature that vapour velocity is one of the primary contributors to frictional and shearing effects. Ultimately, these effects are manifested by the pressure losses shown in figure 6.8, with the increase in losses as a function of air flow rate being a direct result of the phenomenon exhibited in figure 6.9.

Another characteristic of the air flow rate-pressure drop relationship shown in figure 6.8 is that the variation in pressure drop at the higher condensate flow rates is more pronounced than that at the lower flow rates. For example, the pressure drop at 0.16 kg/s increases by approximately 45% as air flow rate is increased from ~10.4 kg/s to ~18.9 kg/s. This contrasts with the increase in pressure drop observed at a condensate flow rate of 0.10 kg/s, which is only about 25%, across the same range of air flow rates. Again, this trend is consistent with the absolute pressure measurements presented in figures 6.6 and 6.4, and the inferred vapour velocities given in figure 6.9 - which are principally responsible for the pressure losses. At low condensate flow rates, the frictional effects are not exacerbated to the same extent seen at the higher flow rates, ultimately, leading to the variation in pressure drop trends illustrated in figure 6.8. The magnitude of the measured pressure drop data presented thus far is explained in detail in the following section.

6.2.1.2 Relationship between condensate flow rate and pressure drop

In this section, the steam/condensate flow rates referred to thus far are expressed non-dimensionally, in terms of the vapour Reynolds number defined in Chapter 2. In addition, the data from the six row condenser is supplemented by pressure drop measurements from the four row design. This extra data is included to better quantify the pressure drop-condensate flow rate relationship, and exhibits the same qualitative relationship with air flow rate as the data for the six row design given in the previous section. Figure 6.10 presents these pressure drop measurements for the range of vapour Reynolds numbers associated with both circular tube condenser designs. The measurements are from a nominal air flow rate of approximately 16.5 kg/s, which was common to all data sets. Presenting the pressure drop measurements in this manner explicitly quantified the relationship with the vapour Reynolds number only. However, it is suffice to say that the pressure drop measurements acquired at other air flow rates exhibiting the same qualitative trends.
6.2 Pressure Drop Measurements

Figure 6.10 explicitly illustrates the increase in condensation pressure drop, through the tube bundle, that results from an increase in the vapour Reynolds number. The most notable feature of figure 6.10, however, is the magnitude of the measured pressure loss - which is quite small. Indeed, the measured pressure losses for the range of vapour Reynolds numbers examined are only approximately 5 - 10% of some reported losses for comparative two-phase evaporating flows. However, it should be noted that there is very little data available in the literature for comparison with condensing flows of steam, particularly at typical Rankine cycle conditions. Almost all studies in the literature are based on boiling and/or evaporating flows. The reason, however, for the almost negligible measured pressure loss was found to be due to momentum recovery in the flow. This pressure recovery is expressed as a negative pressure loss, as shown in figure 6.11.

As the temperatures investigated in this study were relatively low (30 - 60 °C), the density of the vapour at the inlet to the tubes was quite small, with a large vapour velocity compensating for this. In contrast, the density of the condensate present at the exit of the tubes was four orders of magnitude greater than that of the vapour and, hence, for a given condensate flow rate, the velocity of the condensate was significantly smaller. The relative change in density results in fluidic deceleration through the tubes and, consequently, kinetic energy is recovered. This kinetic energy is recovered in the form of pressure energy, in a phenomenon known as momentum recovery - illustrated in figure 6.11. It is suggested that the vacuum conditions, under which the measurements were acquired, were largely responsible for the magnitude of momentum recovery observed.

Although the high velocity vapour flow contributes to momentum recovery, associated frictional effects are also part of the overall pressure drop mechanism. This is particularly valid at vacuum conditions where, for a given steam/condensate flow
rate, the vapour velocity is amplified over that at atmospheric conditions. This high velocity vapour flow is accompanied by various two-phase flow phenomena such as interfacial shearing, deformation at the interface, and liquid-phase frictional effects. Ultimately, these effects are quantified by two-phase frictional losses, calculated from the data reduction in Chapter 4, Section 4.6, with the results presented in figure 6.12.

It is clear from figure 6.12 that the frictional pressure losses incurred as the steam condenses through the MACC tube bundle are quite significant. As expected, these losses increase with vapour Reynolds number, and are due to two main contributing factors, i.e. the frictional effect of each individual phase on the tube wall. Shearing between the tube wall and the high velocity vapour phase results in energy dissipa-
tion, with the same shearing mechanism responsible for pressure losses in the liquid phase. Similar to the measured pressure and inferred momentum recovery, the frictional pressure losses are a strong function of vapour Reynolds number. However, as shown by the actual measured pressure drop in figure 6.10, the effect of the frictional losses, from a practical perspective, are minimal. This is due to the fact that the frictional losses are largely offset by the momentum recovery. This effect is demonstrated most clearly in figure 6.13.

![Figure 6.13](image_url)

**Figure 6.13:** Constituent components of condensing two-phase pressure drop as a function of vapour Reynolds number for the circular tube condensers. The data marker style follows those presented previously; black - 6 row, grey - 4 row.

The most obvious characteristic of figure 6.13 is the symmetry of the data owing to the comparable magnitudes of the frictional pressure drop and momentum recovery. The magnitude of the measured pressure drop is almost negligible in comparison to the other components as it varies from 2.5 - 6% of the frictional component across the range of vapour Reynolds numbers examined herein. Based on the trends in figure 6.13, it could be suggested that pressure losses on the condensing-side in ACCs are not an issue, as the magnitude of the overall losses are quite insignificant due to momentum recovery in the flow negating any frictional losses. However, in implying such a generalisation it would be remiss without stressing that the type of trend illustrated in figure 6.13 may not be prevalent in all ACCs.

Due to a lack of data, it is not known if momentum recovery is a feature of all condensing flows and if it offsets the frictional losses to such an extent as seen in this study. For instance, the data presented herein were obtained under vacuum conditions which, as outlined earlier, contribute to increased vapour velocities for a given flow rate. It is not known if the momentum recovery is as prominent in ACCs operating at higher pressures where, for the same condensate flow rates as those examined here, the vapour velocities would not be as large. Thus, the magnitude of momentum recovery may not be as pronounced. In such cases, the frictional pressure
drop could potentially be an issue. As it is traditionally the most contentious term to predict, a comparative assessment between the inferred frictional data and a range of two-phase frictional pressure drop correlations from the literature is presented, with the goal of establishing the most accurate predictive method.

6.2.1.3 Comparison with circular tube correlations

Figure 6.14 presents the experimental frictional data plotted with the predicted frictional pressure drop from the correlations of Lockhart and Martinelli [4], Friedel [5], Grönnerud [6], and Müller-Steinhagen and Heck [7].

![Figure 6.14: Frictional pressure drop, inferred from measurements, as a function of vapour Reynolds number plotted with the pressure drop predicted from Lockhart and Martinelli [4], Friedel [5], Grönnerud [6], and Müller-Steinhagen and Heck [7]](image)

It is immediately apparent from figure 6.14 that the experimentally-derived frictional data correlates best with the method of Lockhart and Martinelli and, to a lesser extent, Müller-Steinhagen and Heck. The methods of Friedel and, in particular, that of Grönnerud grossly over-predict the inferred frictional losses. This is emphasised by the differences between the experimental and predicted pressure drops, ranging from 39 - 51% for the Friedel model and 64 - 69% for the method of Grönnerud, over the range of vapour Reynolds numbers examined in this study. This was somewhat unexpected given that Ould Didi et al. [112] found that the method of Grönnerud and Friedel were the 2nd and 3rd best predictive methods, respectively, when compared with frictional data for refrigerants in round tubes. However, the caveat to that is that the experimental data in that particular study was for an evaporating flow with mass fluxes in the range of 100 - 500 kg/s.m², which were much larger than the range studied here. Therefore, the conclusions from that study may not be applicable here. Another reason for the discrepancy between the experimental data and the Friedel prediction, in particular, could be due to the highly-empirical nature of the Friedel correlation, which was constructed from a large data bank of approxim-
ately 25,000 points. Data representing the conditions investigated in this thesis may not have been included amongst those 25,000 points. Furthermore, Friedel himself found that the standard deviation of the data relative to his correlation to be about 30%, highlighting the inconsistency [156].

There is much better agreement between the inferred frictional pressure drop data and that given by the models of Lockhart and Martinelli, and Müller-Steinhagen and Heck. The differences between the experimental data and the models vary from 9 - 26% and 17 - 39% using the Lockhart and Martinelli, and Müller-Steinhagen and Heck methods, respectively. The mean deviation between the data and the Lockhart and Martinelli model was found to be 18%, whilst it was substantially larger for the method of Müller-Steinhagen and Heck at 31.5%. These deviations are illustrated in figures 6.15a and 6.15b, where the inferred frictional data is compared to the data predicted by the models. To highlight the accuracy/inaccuracy of the models, error bands representing ±20% deviations from parity are included in the plots.

![Figure 6.15](image)

Figure 6.15: Comparison between experimentally-derived frictional pressure drop data and predicted data from Lockhart and Martinelli [4] and Müller-Steinhagen and Heck [7]

Examining figures 6.15a and 6.15b, it is clear that the Lockhart and Martinelli model is in much better agreement than Müller-Steinhagen and Heck. In both cases, some scatter in the data is evident. However, in the case of Lockhart and Martinelli, for the most part the data remains within the ±20% error bands. The majority of the data falls outside of these error bands in the case of Müller-Steinhagen and Heck, which also consistently over-predicts the magnitude of the frictional pressure drop. Conversely, the method of Lockhart and Martinelli under-predicts the measured frictional losses, albeit with greater accuracy than any other method.

The capacity of the Lockhart and Martinelli model to predict the experimental data is a testament to it’s flexibility. It was originally developed to account for the interfacial pressure drop associated with the presence of two distinct and separate fluids, using data from isothermal liquid-gas fluid combinations in circular tubes. Lockhart and Martinelli correlated this data by introducing an empirical parameter,
\( \phi \) - defined in Chapter 2. This provided a solution that covers most types of two-phase flow regimes and, consequently, the Lockhart and Martinelli model has seen extensive use in literature focused on two-phase frictional pressure losses. Furthermore, it has been shown to yield reasonably accurate results for a wide variety of two-phase flow circumstances in round tubes, despite the fact the model does not distinguish between the different physics associated with specific flow regimes. In two of the only studies on investigating the pressure drop associated with steam condensation, Wang and Zhao [108] and Wang et al. [109] found the Lockhart and Martinelli correlation to be the most accurate method of predicting the two-phase frictional pressure drop in plate heat exchangers.

Given the database from which the Lockhart and Martinelli correlation was developed, it is expected to yield best results at low pressure adiabatic flows or boiling flows at low heat fluxes. In this respect, the agreement observed between the experimental diabatic condensing flow data and the model is somewhat surprising. However, the physics between a boiling flow and one of a condensing flow are not too dissimilar. Both involve phase change, where there is an exchange of mass, momentum, and energy. Both normally occur under isothermal conditions, with similar magnitudes of heat fluxes involved. In addition, the data with which Lockhart and Martinelli built their correlation upon was from liquid-gas combinations, including water-air. Thermophysical properties of air such as viscosity and density have an order of magnitude similarity to water vapour and, as such, the agreement between the model and data illustrated in figure 6.15a is more understandable.

Slight deviations between the data and model can be attributed to experimental error and, most likely, some maldistribution of the steam flow through the tube bundle. Indeed this could be the biggest source of error as it was assumed in the calculation methodology for the models that a uniform flow rate existed through all tubes in the bundle. In reality, this may not have always been the case, where some tubes could have had a higher flow rate than others, hence lending to larger losses.

6.2.1.4 Pressure drop predictions

As shown previously in figure 6.13, for the conditions examined in this thesis, the measured frictional pressure losses were largely offset by momentum pressure recovery. Hence, the actual measured pressure drop was minimal. However, it was unknown if this was generally the case, or merely a characteristic which manifested itself at the specific test conditions examined herein. To determine the effect of momentum recovery at other conditions, the frictional losses and momentum recovery were calculated, and presented as a ratio, for a range of geometrical parameters outside of those from which the measurements were acquired. Namely, these parameters were tube diameter and tube length. The frictional pressure drop was predicted using the method of Lockhart and Martinelli, previously established as the most accurate approach for steam condensation in circular tubes. The momentum recovery was calculated using equation 4.13 in Chapter 4. Figure 6.16 presents the findings.
The calculations presented in figure 6.16 are for a mass flux of 5 kg/s.m$^2$ and a temperature of 313 K (40 °C) - which was reflective of the experimental range. Only results from one mass flux value are presented as the calculations showed that, for a given geometry, the ratio of frictional loss to momentum recovery was almost always uniform across the range of mass fluxes examined. This was due to the fact that the mass flux term is implicit in both the frictional and momentum equations. It is clear from figure 6.16, however, that the pressure drop ratio is a strong function of both the tube diameter and tube length. This ratio is seen to be almost unity for large diameters and short lengths, meaning that the frictional losses are completely negated by the magnitude of momentum recovery. This was the case for the circular tube MACC geometry, with $D = 0.0275$ m and $L_t = 2.15$ m. However, an increase in tube length, or decrease in tube diameter, results in an increase in the ratio where excessive frictional losses are not offset to the same extent. This is particularly apparent for the limiting cases examined here where, at $D = 0.01$ m and $L_t = 10$ m, the frictional losses are approximately eleven times larger than the magnitude of momentum recovery. The net effect of this is that appreciable measurable fluidic losses will arise which will adversely effect condenser performance. Therefore, as a general guideline to ensure adequate condenser performance, it is recommended that long tubes with small diameters be avoided.

Ultimately, figure 6.16 illustrates a characteristic which was originally suspected - that frictional pressure losses are not always dampened by momentum recovery in the condensing flow. In this regard, it was fortuitous and somewhat coincidental that the MACC measurements were acquired at conditions which allowed this trait to be exhibited. Nevertheless, the measurements provided the platform to investigate this characteristic. It was earlier suggested that the sub-atmospheric conditions
from which the measurements were taken were responsible for the large level of momentum recovery in the flow. This was investigated and, indeed, confirmed in figure 6.17, where the ratio of frictional pressure drop to momentum pressure recovery is presented as a function of steam temperature for a range of diameters and lengths.

![Response surface for steam temperature and tube diameter](image)

(a) Variation with tube diameter for a nominal tube length of 2 m

![Response surface for steam temperature and tube length](image)

(b) Variation with tube length for a nominal tube diameter of 0.03 m

Figure 6.17: Response surfaces illustrating the variation in the ratio of frictional pressure drop to momentum pressure recovery as a function of steam temperature

Similar to figure 6.16, the calculations presented in figure 6.17 are based on a mass flux of 5 kg/s.m$^2$. For a set tube diameter and/or tube length, the calculated ratio of frictional pressure drop to momentum pressure recovery was independent of mass flux, in that it remained relatively constant for the range investigated. However, of
greater significance is that the trends illustrated in figure 6.17a and figure 6.17b offer compelling evidence that momentum recovery is a function of steam temperature. Indeed, as suspected, momentum recovery is augmented by operating at low pressures and, hence, low temperatures. As discussed hitherto with respect to figure 6.11 in Section 6.2.1.2, this phenomenon occurs as a consequence of the kinetic energy associated with the large vapour velocity being recovered as the flow condenses to form a relatively slow moving liquid. At lower temperatures, the vapour velocity is greater and, hence, the magnitude of recovery is larger. Therefore, for a given flow rate, as temperature is reduced the momentum recovery increases at a greater rate than the frictional pressure drop increases - thus leading to the characteristic shown in figure 6.17. Conversely, at higher temperatures the vapour velocity is smaller than that at lower temperatures, for a given flow rate, and the effect of momentum recovery is not as great. In these cases, the frictional pressure drop dominates over the momentum recovery, and will be appreciable in application.

Figures 6.17a and 6.17b are presented for the case of a single tube length and tube diameter, respectively. As the primary objective was to examine the effect of varying steam temperature on momentum recovery, it was necessary to constrain one geometrical parameter in each case due to there being more than three variables considered. The response surface plot is only possible if the remaining variables not represented are set to a constant value [157]. In each case, the excluded variables were set to dimensions corresponding approximately with the circular tube MACC geometry. The figures exhibit the same qualitative characteristics as those illustrated in figure 6.16, namely that as tube diameter decreases and/or tube length is increased, the frictional losses incurred exceed the momentum recovery, regardless of steam temperature. Nevertheless, it can be concluded that operating at low pressures and low temperatures is not only beneficial from a thermodynamic perspective, but also tends to mitigate against excess fluidic losses in circular tube condensers.

6.2.2 Single row rectangular tube condenser measurements

Similar to the circular tube measurements presented thus far, the two-phase condensing pressure drop through the rectangular tube bundle was measured over a range of air flow rates and condensate flow rates. However, as the magnitude of the measured pressure drop was shown previously to be relatively inconsequential, the measurements for the rectangular tube condenser simply consisted of ensuring that this magnitude was somewhat similar, and did not have an adverse effect on condenser performance. In this regard, a systematic investigation, similar to that presented for the circular tube measurements, was not undertaken. Figure 6.18a presents the measured condensing pressure drop, at a nominal air flow rate of ~ 16.5 kg/s, for the range of vapour flow rates examined on the rectangular tube condenser. This is accompanied by figure 6.18b, which presents those measured losses as a ratio of the absolute measured pressure in the condenser, $P^*$ - the pressure ratio.
6.3 Heat Exchanger Thermal Resistance

The final set of measurements on the condensate-side characteristics of the various MACC designs are given in terms of thermal resistance. In the case of an ACC, there are generally two main thermal resistances acting in series. The air-side resistance is
referred to as the controlling, or dominant, resistance as it usually represents more than 80% of the total thermal resistance [19]. The remaining resistance is attributed to the condensate-side, where the wall resistance is usually considered negligible. Therefore, thermal resistance measurements presented henceforth will be referred to in terms of air-side, and condensate-side. These measurements are presented for the four row circular tube and, subsequently, single row rectangular tube designs.

6.3.1 Air-side thermal resistance

The air-side thermal resistance is that resistance arising from the film of air on the condenser exterior, coupled with the conductive resistance through the fins. As a result of the unfavourable thermophysical properties of air, the associated thermal resistance is quite high, relative to other media such as water. However, the magnitude of this thermal resistance can be reduced by inducing forced convection conditions - through the use of a fan. A specific set of air-side experiments, similar to those described in Chapter 5, Section 5.2, were carried-out to quantify the effect of variable fan speed on the forced convection thermal characteristics of the various condenser designs. Accordingly, the air-side thermal resistance of the circular tube, and rectangular tube, condensers was also established, with these results presented in figure 6.19. These measurements are also reported in [124, 123], in terms of Nusselt number. As these experiments formed part of a separate study, the experimental details and methodology are not reported in this thesis, suffice to say that the experimental details are very similar to those described in Chapter 5, Section 5.2.

![Figure 6.19: Variation in air-side thermal resistance with air mass flow rate for the four row circular tube (CT) and the single row rectangular tube (RT) condenser. Also plotted is the predicted variation from the ε-NTU model given in Chapter 3.](image)

Figure 6.19 clearly illustrates the effect of varying fan speed (presented here as air mass flow rate) on the air-side thermal resistance. Applicable to both condenser cases
is the decrease in thermal resistance as air flow rate is increased. Essentially, this occurs due to the increased bulk motion of the fluid, which augments the advection component of convection. Furthermore, diffusion is also promoted due to localised effects at the boundary layer. Individually, or combined, these effects are ultimately responsible for the characteristic seen in figure 6.19. The manner in which this occurs for each heat exchanger type, however, is substantially different.

For the four row circular tube condenser, the staggered tubular arrangement causes interruptions in the air flow, whereby the boundary layer growth is abruptly halted, separates, and re-attaches as the flow progresses downstream through successive tube rows. The net effect of this is thinner boundary layers which, ultimately, lead to a decrease in thermal resistance. Alternatively, the aerodynamics of the single row plate-finned design are characterised by the more structured air flow through the rectangular-shaped fin channels. This can be classified as internal flow - as neighbouring boundary layers interact. These boundary layers grow in the air flow direction, thus reducing the effective flow area for inviscid flow. By continuity, the core inviscid flow must accelerate and a favourable pressure gradient is established. The net result of this is an increase in skin friction and heat transfer coefficient. This effect is exacerbated as air flow rate is increased, resulting in a decrease in thermal resistance such as that seen in figure 6.19.

An interesting feature of the results in figure 6.19 is the difference in air-side thermal resistance for each condenser type. Based on figure 6.19, it is clear that the plate-finned, rectangular tube design outperforms the annular-finned, circular tube design. Across the range of air flow rates examined here, an average ~ 15% improvement in air-side thermal performance was noted. As the heat transfer area is practically identical for each condenser, (circular tube condenser ~ 497 m$^2$, rectangular tube condenser ~ 499 m$^2$), the difference in thermal resistance is attributed to the drastically different air flow conditions associated with each condenser. Stafford [158] reported on the benefit of flow alignment in finned heat sinks and observed an improvement in thermal performance as a result. Although no direct comparison was made with annular-finned heat sinks, such findings imply that the flow alignment that occurs through the rectangular fin channels in the plate-finned design could be the reason for the improved thermal resistance. Furthermore, there are certain undesirable features of the air flow through the annular-finned condenser which could be causing the degradation in thermal performance. It is known that as a boundary layer re-attaches to a fin, separation bubbles or recirculation zones can form. Within these zones, relatively slow-moving fluid flows in a large eddy pattern, with the fin surface in contact with such zones being subjected to lower heat transfer coefficients as a consequence of the low fluidic velocities and thermal isolation. Therefore, the air flow characteristics through the multi-row, staggered tube bank are likely the reason for the increased thermal resistance seen in figure 6.19.

Also plotted in figure 6.19 is the predicted variation in air-side thermal resistance given by the $\epsilon$-NTU model from Chapter 3, Section 3.2. The difference in thermal
resistance between the designs is also an implicit feature of the air-side model, which supports the argument put forth previously. The predicted variation in thermal resistance agrees quite well with the measured data with an average discrepancy of 4.7% and 6.2% for the annular-finned and plate-finned designs, respectively. This demonstrates the accuracy in using this method, and the associated air-side theory, to predict the condenser air-side performance.

6.3.2 Condensate-side thermal resistance

The thermal resistance arising from all fluidic phenomena inside the tubes is termed the condensate-side, or steam-side, thermal resistance. This accounts for the resistance due to the presence of air, the formation of condensate, tube fouling, and conductivity through the tube wall. As described in the experimentation section in Chapter 4, it was not possible to explicitly measure the condensate-side resistance, nor any of the individual components of which it is comprised of. Instead, the condensate-side resistance was inferred from measurements of total thermal resistance, from which the established air-side resistance was subtracted - with the difference being the resistance inside the tubes. This section presents the total resistance measurements, the respective air-side resistances from figure 6.19, and the inferred condensate-side resistance, for a range of condensate mass flow rates and air mass flow rates, for both condenser types.

6.3.2.1 Four row circular tube condenser thermal resistance

Figure 6.20 presents the variation in the total, air-side, and condensate-side thermal resistance across the range of air flow rates examined for the four row annular-finned, circular tube condenser design. Each series of bar plots is presented for a specific vapour Reynolds number of 2280, 3050, and 3800, each of which equates to a given condensate mass flow rate during experimentation.

The thermal resistance measurements presented in figure 6.20 illustrate a number of interesting features. Some of these were expected, such as the fact that the total, overall resistance to heat flow reduces as air flow rate increases. This trend is common across all vapour Reynolds numbers investigated herein, irrespective of the variation in magnitude of the total thermal resistance - which is seen to decrease with increasing vapour Reynolds number. The air-side thermal resistance, as shown previously in figure 6.19, also decreases with increasing air flow rate, but is obviously uniform across the vapour Reynolds number range. This is expected, as the air-side thermal resistance is a function of the aerodynamic conditions and thermophysical properties of the air flow and, thus, should inherently be independent of the condensate-side conditions.

It is apparent from figure 6.20 that quite an appreciable difference in magnitude exists between the total and air-side thermal resistance measurements. This is a direct consequence of the influence of the condensate-side thermal resistance which,
Figure 6.20: Variation in thermal resistances with air mass flow rate for a range of vapour Reynolds numbers for the four row circular tube condenser
in some cases, is quite significant. At low vapour Reynolds numbers, in particular, this resistance is an appreciable contributor to the overall, total resistance. For example, at \( \text{Re}_v = 2280 \), the condensate-side resistance varies from as low as 11% to as large as 37% of the total resistance, with an average contribution of 26% across the entire air flow rate range. This is considerably larger than the 10 - 15% contribution suggested by Shah and Sekulić [19] for liquid-gas heat exchangers. Furthermore, it can be seen from figure 6.20 that no single value of condensate-side thermal resistance, that could be calculated from some appropriate correlation for example, would satisfactorily predict the measured values. This is due to the introduction of fan speed (analogous to air flow rate) as a variable, with figure 6.20 establishing that the condensate-side thermal resistance is a function of air flow rate.

A notable trend across all vapour Reynolds numbers examined, but which is most evident in the \( \text{Re}_v = 2280 \) case, is that the largest condensate-side thermal resistances tend to occur at the highest air flow rates. Generally, the condensate-side resistance arises as a result of liquid condensate accumulating inside the tubes, where the formation of a liquid film impedes heat transfer. In this regard, the fact that the resistance appears to be a function of air flow rate was somewhat surprising. However, examination of the data provided some insight and explanation. As described previously in Section 6.1.2, an increase in air flow rate leads to a reduction in condenser pressure which, conversely, causes an increase in the pressure differential between the condenser and the surrounding atmosphere. Air ingress was shown to be a function of this pressure differential, where an increase in mass flow rate of air leaking into the system occurs as the pressure differential instigating this flow increases. Numerous studies [159, 160, 92, 21] have shown that steam condensation is substantially inhibited when in the presence of a non-condensable gas such as air. The air mixes with the steam and, as the steam condenses, a film of air remains behind which acts as another barrier to heat flow. It is suggested that this phenomenon is exacerbated by the increased air ingress as air mass flow rate is increased and that, ultimately, it is responsible for condensate-side thermal resistance increasing with air mass flow rate. This relationship is explicitly illustrated in figure 6.21.

Another trend to note from figure 6.20 and, particularly, figure 6.21 is that the condensate-side thermal resistance decreases as vapour Reynolds number is increased. This was expected and is in-line with findings reported in the literature. The largest thermal resistance was observed at \( \text{Re}_v = 2280 \), where it had an average contribution of 26% of the total resistance, across the range of air flow rates. As the vapour Reynolds number was increased to 3050, this contribution reduced to a 17% average contribution, with the condensate-side thermal resistance forming 10% of the total resistance at \( \text{Re}_v = 3800 \). As described in numerous studies in the literature, the increase in vapour velocity associated with an increased Reynolds number augments heat transfer and, hence, reduces thermal resistance. This occurs due to two main reasons. Firstly, axial shear stresses and lateral Reynolds stresses are induced, both of which tend to deform the liquid interface and remove layers of liquid from the
6.3 HEAT EXCHANGER THERMAL RESISTANCE

Figure 6.21: Variation in condensate-side thermal resistance with air mass flow rate for a range of vapour Reynolds numbers for the four row circular tube condenser

film. This liquid usually becomes entrained in the vapour core flow but the initial disturbance of the film results in an increase in convective heat transfer. In terms of two-phase flow morphology, this is seen as a progression from a relatively stable flow regime such as stratified flow to stratified-wavy or annular flow, as described in Chapter 2, Section 2.2. It is widely recognised that higher heat transfer coefficients and, thus, smaller thermal resistances are associated with such flow regimes. In addition to all this, the increase in bulk vapour velocity tends to force the condensate film towards the tube exit, thus thinning the film and reducing the thermal resistance associated with it. The combination of these effects are manifested in the results presented in figure 6.21.

6.3.2.2 Effect of condenser inclination angle

The thermal resistance measurements presented thus far were all acquired from the four row circular tube condenser at the nominal condenser inclination angle of ~ 40°. However, as described in Chapter 4, it was possible to rotate and, hence, incline the MACC module at a range of angles relative to the horizontal. The purpose of this was to investigate any potential improvement in condenser performance and, if so, establish an optimum angle for operation. With the condenser in a horizontal position, a number of adverse effects were initially observed, the most prominent of which was the inhibition of condensate drainage. This was manifested by erroneous mass flow measurements, pressure surges, and water hammer. Therefore, to alleviate these issues, the condenser was inclined and set to a nominal angle of ~ 40°. This angle was not established by means of any robust analysis or experimental optimisation, but was simply chosen to facilitate measurements. All measurements described thus far were acquired at this inclination angle. Due to measurement program constraints, it was not possible to investigate a range of inclination angles in a systematic manner.
However, to investigate if any effects materialised, the angle was increased to ~ 70° and the measurements presented thus far were repeated, with the results given in figure 6.22 in comparison with the nominal thermal resistances from figure 6.21.

![Graph](image_url)

Figure 6.22: Comparison between the condensate-side thermal resistance, as a function of air mass flow rate and vapour Reynolds number, at two different inclination angles on the four row condenser

The results in figure 6.22 demonstrate that increasing the condenser inclination angle from the nominal ~ 40° has a negligible effect on the resulting condensate-side thermal resistance. This is evidenced by the overall agreement between the two sets of measurements, where the slight differences are again most likely attributable to experimental error. There appears to be a general trend of marginally lower thermal resistances for the inclination angle of ~ 70°, particularly at Re_{v} = 3800. However, it is difficult to say with any certainty if this was definitely a consequence of increasing the angle, or simply a result of variance in one of the test parameters. Furthermore, no significant decrease in thermal resistance was observed at the lower condensate flow rates, where it would be expected to occur due to condensate drainage via gravity being the dominant mechanism at such conditions. Although only one extra angle was examined here, it is expected that additional inclination angles greater than 40° would not bring about any appreciable improvement. It is possible that some optimum angle, from a condensate-side perspective, exists between the adverse horizontal position and 40°. Unfortunately, this was not determined in this study, but as current A-frame ACCs are inclined somewhere between 50° - 70°, the criteria for selecting this could, in fact, be based on air-side effects, or structural standards.

**6.3.2.3 Single row rectangular tube condenser thermal resistance**

The complete set of thermal resistance measurements previously presented in figure 6.20 are presented here for the single row plate-finned, rectangular tube condenser.
design. The variation in total, air-side, and condensate-side thermal resistance is given as a function of air flow rate and vapour Reynolds number in figure 6.23.

The trends in the data presented in figure 6.23 are identical to those observed in, and described for, the circular tube measurements. Specifically, total resistance decreases with both an increase in air flow rate and vapour Reynolds number. Air-side resistance decreases with increasing air flow rate, but is independent of vapour Reynolds number. Condensate-side resistance decreases with an increase in vapour Reynolds number but, somewhat counter-intuitively, increases as air flow rate is increased. It is also clear that the air-side resistance is the dominant resistance - as is generally assumed in a steam-air heat exchanger. Such an assumption usually implies that the condensate-side thermal resistance is minimal, or negligible, in an ACC. However, as shown in figure 6.23, this is not always the case. For the vapour Reynolds numbers examined herein, particularly at the lower Reynolds numbers examined, the resistance on the condensing-side is certainly appreciable and a contributing factor to the overall resistance of the heat exchanger. The magnitudes of these measured condensate-side resistances are given more clearly in figure 6.24a.

As stated previously, the condensate-side resistance in single component condensation arises from the formation of a liquid film as the vapour condenses. Progressively more liquid condensate accumulates as more vapour condenses and, as a result, the thickness of the condensate layer increases. This thermal medium, through which heat must flow, constitutes the resistance. Due to the thermophysical properties of
condensate \((k_{\text{water}} \approx 0.6 \ \text{W/mK}, \ \text{Pr}_{\text{water}} \approx 7)\), the thermal resistance will, inherently, be less than that of air. Although the magnitudes shown in figure 6.24a are quite small, the contribution of the condensate-side thermal resistance to the total resistance is clearly not insignificant, as evidenced by figure 6.24b which presents the condensate-side resistance as a percentage of the total resistance. Quantitatively, the contribution of the average condensate-side resistance to the average total thermal resistance varies from 22% at \(\text{Re}_v = 2650\) to 17% at \(\text{Re}_v = 3540\) to 13% at \(\text{Re}_v = 4420\), with the remaining resistance attributed to the air-side in each case.

As seen in figure 6.23, the magnitude of condensate-side resistance has a clear and appreciable effect on the total heat exchanger resistance, serving to plateau the resistance instead of it decreasing in a manner similar to the air-side resistance. Ultimately, this curtails condenser performance, with the benefit of increasing air flow rate not being fully realised as the temperatures achieved in the condenser will not be as low as desired. Clearly, this is due to the increase in condensate-side resistance with air flow rate, as seen most explicitly in figure 6.24a. It was proposed hitherto, when discussing the circular tube thermal resistance measurements, that this characteristic was a consequence of greater quantities of air leaking into the condenser at the larger pressure differentials brought about by increasing air mass flow rate. The presence of significant quantities of air in the condenser meant that air-vapour condensation, as opposed to pure-vapour condensation, was the characteristic transfer process occurring in the condenser. It is generally accepted in the literature that any quantity of air present during condensation impedes heat transfer and, as such, pure-vapour condensation is the preferential mode of condensation if at all possible.

It is not known if the quantity of air leaking into the system during experimentation is representative of typical air ingress in operational ACCs. Nearly all Rankine cycle ACCs operate under vacuum and air leaks inevitably occur as a result. Due to a
lack of available data, however, typical air leakage rates have not been quantified nor published in the general literature. Nevertheless, the results presented in this chapter are meaningful in that they clearly illustrate the adverse effect of air leakage on the thermal performance. Therefore, air leakage should be minimised to ensure optimal condenser and, hence, plant performance.

6.3.2.4 Summary

To summarise the findings on the condensate-side resistance, the measurements from both condenser designs presented in this section are combined here. Although the same approximate steam/condensate flow rates were employed for both condensers, the heat sink layout was different in each condenser. This meant that the resulting Reynolds numbers were not identical. However, this provided a range of Reynolds numbers with which to examine the condensate-side thermal resistance. Figure 6.25 presents the thermal resistance measurements, taken from both condensers, as a function vapour Reynolds number.

Figure 6.25: Condensate-side thermal resistance as a function vapour Reynolds number, combined from the four row circular tube condenser and single row rectangular tube condenser

The data presented in figure 6.25 are an attempt to explicitly illustrate the relationship between condensate-side thermal resistance and vapour Reynolds number. Hence, these measurements are presented for a single air mass flow rate of ~ 18kg/s. Although condensate-side resistance was shown in Section 6.3 to vary with air flow rate, data from other flow rates were omitted here as they demonstrated the same qualitative relationships as that in figure 6.25. It can be clearly seen that figure 6.25 depicts a general trend of condensate-side thermal resistance decreasing as vapour Reynolds number increases. A power-law function, derived from regression analysis, is plotted through the data to quantify the overall trend.
There are a number of interesting features and inferences from the relationship depicted by the power-law function. Outside of the actual, measured data, it can be seen that the magnitude of thermal resistance is potentially quite large at low vapour Reynolds numbers. This magnitude then drastically decreases as Reynolds number increases - as shown by the measured data. The trend then somewhat saturates around Re\(_v\) \(\approx\) 4500. Although data outside this range was not investigated, it is probable from extrapolating the relationship that further increases in Reynolds number from \(~\) 4500 would only bring about a minimal reduction in condensate-side thermal resistance.

The thermal resistance measured at Re\(_v\) \(\approx\) 4500, constitutes approximately 13% of the total resistance. In their seminal heat exchanger characterisation studies, Kays and London [116], assumed a “condensate-film heat transfer resistance” of less than 10%. However, such a conservative estimate is only applicable to their specific methodology, where excess steam at high vapour velocities is forced through the condenser tubes in a concerted effort to purposely minimise the condensate-side resistance. It is interesting to note, however, that increasing the vapour Reynolds number from those examined here would probably reduce the condensate-side resistance to magnitudes approaching those assumed by Kays and London. For more representative steam/condensate flow rates and full condensation, the condensate-side thermal resistance will be larger than that assumed by Kays and London. This is indicated by Shah and Sekulić [19] who stated that the air-side resistance represents more than 80% of the total resistance, therefore implying that the remaining \(~\) 20% is due to the condensate-side. Over the range of vapour Reynolds number investigated herein, the average contribution of the condensate-side thermal resistance to the total varied from 10% - 26%, thus somewhat vindicating Shah and Sekulić’s approach.

6.4 summary

In accordance with the methodologies described in Chapter 4, this chapter presented the findings from a series of hydrodynamic and thermal characterisation tests on a full-scale air-cooled condenser. Condenser operating performance was firstly quantified, where it was seen that varying fan speed leads to a reduction in temperature and pressure in the condenser. This ability to manipulate and control condenser operating conditions is beneficial from a thermodynamic perspective - as shown in Chapter 8. However, as shown in this chapter, there are factors such as vacuum pump performance which need to be considered in order to ensure condenser performance is not undermined or restricted. In this study, it was found that increasing vacuum pump capacity resulted in lower pressures achieved in the condenser, and greater control over these pressures when varying fan speed was subsequently noted.

Pressure drop measurements acquired from the MACC illustrated a number of interesting characteristics, none more so than momentum recovery in the flow. It was seen that the measured losses were quite small, in the range of \(\sim\) 120 - 250 Pa.
6.4 SUMMARY

Through analysis of the data, it was seen that the reason for this was momentum recovery offsetting the frictional losses. This appears to be unique to condensing flows, but the fact that it almost completely offsets the frictional losses was found to be somewhat coincidental. A parametric investigation lead to the conclusion that momentum recovery does not always offset the frictional losses, but merely was the case under the experimental conditions and condenser geometry examined in this study. It was shown that for condensers with round tubes in excess of ~ 4 m, the frictional losses incurred will exceed the magnitude of momentum recovery.

Thermal characterisation was quantified in terms of heat exchanger thermal resistance. It was found that there are two main factors which influence the condensate-side thermal resistance. Firstly, it was seen that increasing air flow rate leads to an increase in condensate-side resistance. This was attributed to the greater quantities of air leaking into the condenser as air mass flow rate was increased. It is well known in the literature that air acts as a deterrent to heat transfer and, as such, efforts should be made to mitigate against the tendency for air ingress to occur. Secondly, condensate-side thermal resistance was found to decrease with increasing vapour Reynolds number. When expressed as a ratio of the total thermal resistance, the condensate-side contribution varied from approximately 26% to 10%, across the vapour Reynolds number range.
Presented in this chapter are measurements which quantify the hydrodynamic and thermal characteristics of condensing flows of single-component, low pressure steam in an air-cooled annular-finned round tube. This geometry constituted the reduced-scale ACC, described in Chapter 5, the measurements from which were carried-out over a range of experimental test conditions applicable to full-scale, Rankine cycle-based ACCs. A significant feature of the reduced-scale ACC was that condensation in the absence of substantial quantities of air was achieved and, thus, all measurements reported herein are for pure-vapour condensation.

As this chapter consists of a significant body of work, the results are presented in a structured format which reflects the evolution of the experimental program. Similar to the full-scale MACC results in Chapter 6, the initial set of measurements were dedicated to establishing the condenser performance by investigating the variation in condenser temperature, and pressure, with air flow rate and steam flow rate. These measurements were succeeded by an investigation of the prevailing two-phase flow regime, which encompassed flow regime mapping and a novel experimental technique to establish the liquid-vapour flow structure in a non-invasive manner. Establishing the dominant flow regime provided the platform to quantify the two-phase condensing pressure drop and condensation heat transfer. Condensation heat transfer measurements were the primary focus of the work and are presented through local and global measurements. Detailed local measurements of the condensation heat transfer along the axial length of the tube, and around the tube circumference, provided insight into the condensation process not seen in literature to-date. Culminating this chapter are thermal resistance measurements. As heat exchanger thermal resistance is a ubiquitous metric in quantifying heat exchanger thermal performance, the measurements presented herein can be incorporated into thermodynamic predictive tools and ACC design criteria.

### 7.1 Performance Characteristics

In a similar manner to the full-scale MACC results presented in Chapter 6, the condensate-side performance of the reduced-scale ACC can be expressed in terms of the terminal values of temperature and pressure - $T_s$ and $P_s$, respectively. In heat exchanger design and/or characterisation, the terminal temperature and pressure of the condenser are determined if the heat exchanger geometry, size, and surface characteristics are known. As described in Chapter 5, the heat exchanger geometry is fixed - to replicate a single tube in the full-scale MACC, and the surface characteristics
of air-side pressure drop and heat transfer coefficient were established in that same chapter. Therefore, to close the rating problem, the terminal values were evaluated through experimental measurements. The absolute steam pressure and temperature results from this are presented as a function of air mass flow rate (fan speed), for a range of condensate mass flow rates, in figures 7.1 and 7.2, respectively.

![Figure 7.1: Variation in steam pressure with air mass flow rate for a range of condensate mass flow rates](image)

![Figure 7.2: Variation in steam temperature with air mass flow rate for a range of condensate mass flow rates](image)

The trends exhibited by the data in figures 7.1 and 7.2 are largely identical to the trends observed in the full-scale MACC performance characteristics. Namely, it can be seen that an increase in air mass flow rate leads to a reduction in steam temperature and pressure, and vice-versa. The variation in steam temperature (and steam pressure) is more pronounced at the higher condensate flow rates than that
seen at the lower flow rates - at which changes in air flow rate appear to have a somewhat negligible effect. Again, this trend is consistent with the full-scale MACC measurements where, in Chapter 6, the apparent insensitivity to changes in fan speed (presented here as air mass flow rate) at the lower condensate flow rates was attributed to the steam temperature being closer to the ambient air temperature. In general, across all condensate flow rates, it can be seen in figure 7.2 that the steam temperature asymptotically approaches the ambient air temperature as air flow rate is increased. However, the asymptotic approach is much more understated at the lower condensate flow rates, due to the simple fact that they are inherently closer to the ambient temperature.

Although the qualitative relationships depicted in figures 7.1 and 7.2 mirror those seen in the full-scale MACC measurements, there are differences quantitatively. As outlined in Chapters 4 and 5, the range of Reynolds numbers examined on the lab-scale ACC and MACC, respectively, were relatively similar. As such, the steam/condensate flow rates per tube were closely matched. In one instance in particular, almost the exact same flow rate per tube was investigated on both condensers. The measured variation in temperature at this flow rate of 0.65 g/s is presented, as a function of air-side heat transfer coefficient, in figure 7.3 for each respective case.

![Figure 7.3: Comparison between the variation in temperature as a function of air-side heat transfer coefficient achieved on the lab-scale ACC and full-scale MACC](image)

The most obvious feature of figure 7.3 is the difference in magnitude between the two sets of data. It is clear that, over the same common range of air-side heat transfer coefficients, the temperatures measured in the reduced-scale ACC are larger than those achieved in the MACC, at the same approximate steam flow rate. As the air-side heat transfer coefficients are similar, the increased steam temperatures in the reduced-scale ACC are a consequence of the smaller heat transfer area. The reduced-scale tube is almost identical to the full-scale tubes, apart from the fact that it is approximately half as long. Therefore, to satisfy the same thermal load, at the
same steam flow rate and air-side heat transfer coefficient, the temperature difference must increase. Essentially, the smaller heat transfer area necessitates the increase in steam temperate seen in figure 7.3. In general, for steam flow rates common to both condensers, the steam temperatures achieved in the reduced-scale ACC were consistently higher than those from the MACC. The relationship between steam temperature and steam flow rate in the reduced-scale ACC is explicitly illustrated in figure 7.4 for a range of air flow rates.

![Figure 7.4](image_url)

**Figure 7.4:** Measured steam temperature as a function of condensate mass flow rate for the range of air mass flow rates investigated.

Figure 7.4 clearly shows the increase in steam temperature which arises from an increase in steam/condensate mass flow rate. Although intuitive to expect an increase in system temperature and pressure with an increase in mass flow rate, the results depicted in figure 7.4 are, nevertheless, consistent with those presented in literature. As observed in similar complete condensation experiments by Oh and Revankar [151], and Henderson et al. [152], the condensate mass flow rate is simply a function of the inlet steam flow rate, which, in turn, directly determines the pressure and, consequently, temperature in the system. The authors duly noted that the system temperature must increase if the steam flow rate increases - if all the steam is to be condensed. This complete condensation is expressed in the form of latent heat rejection, as a function of condensate flow rate, in figure 7.5.

The heat transfer rate given in figure 7.5 is the isothermal (latent) heat rejected to the surroundings from the condenser as the vapour forms condensate. This is given by the product of the mass flow of condensate times the enthalpy of vaporization. Inherently, as more thermal mass is let into the system with an increase in flow rate, more energy will be rejected from the system.

Throughout the experimental program, it was seen that varying the air-side thermal boundary conditions - through varying the fan speed and, hence, the air flow rate - altered the fluidic properties inside the tube. As shown in figure 7.6, the most prom-
The prominent effect of increasing air flow rate was to cause an increase in vapour velocity. In fact, this phenomenon exacerbated the already large vapour velocities present during testing - a result of operating at low pressures and low temperatures. As noted in Chapter 5, the operating conditions examined in this thesis lead to larger vapour velocities over those which would occur at the same steam/condensate flow rate at atmospheric conditions. Ultimately, as shown in figure 7.6, this characteristic was augmented by increasing air flow rate.

Figure 7.6 presents temperature data at a condensate flow rate of 0.65 g/s, replotted from figure 7.2. Only one data set was included for clarity purposes, and to simply convey the general qualitative relationship between variable air-side boundary conditions and vapour velocity. Nevertheless, the trend depicted in figure 7.6 was
7.2 Two-phase flow regime identification

Identifying the prevailing two-phase liquid-vapour flow morphology provides a qualitative insight into the fluid dynamics and lends supporting evidence to the quantitative hydrodynamic and thermal measurements - presented in this chapter at a later stage. As the condensing pressure loss and heat transfer are intimately linked to the fluidic arrangement, establishing the prevailing flow topology is considered a starting point. This section presents the results from two separate approaches to establishing the two-phase flow regime. Firstly, flow pattern maps, which were originally introduced in Chapter 2, are presented here with the experimental data plotted. This approach is frequently adopted to identify the flow regime in the absence of experimental means. Following this, results from a novel, non-invasive, non-visual experimental measurement technique, described in Chapter 5, for determining the overriding two-phase flow structure are presented.

7.2.1 Flow regime mapping

At present, the standard theoretical approach for identifying the two-phase flow regime of a particular fluid/pair of fluids is through the use of flow pattern maps. These maps are an attempt, on a two-dimensional graph, to separate the space into individual regions - each corresponding to a specific flow pattern. The regions are segregated and defined by transition boundaries, which are curves plotted on the map given by empirical or analytical equations. The transition mechanisms differ considerably, however, depending on whether the tube is horizontally or vertically aligned [161]. As this study is only concerned with horizontal and slightly inclined flows,
all maps relating to vertical tube configurations have been discounted. Furthermore, many of the flow regime maps in the literature for gas-liquid flows in horizontal and slightly inclined tubes [58, 59, 2, 162] are used for predicting the flow morphology of adiabatic flows, as opposed to diabatic flows. Quibén and Thome [163] and Cheng et al. [164] both highlighted that extrapolating such maps to diabatic flow cases may not always produce reliable results. It is for this reason that only one such map is included here - that of Taitel and Dukler [2], which is one of the more popular and frequently-cited adiabatic maps in literature relating to two-phase flows. Figure 7.7 presents a sample range of experimental data points plotted on the Taitel and Dukler flow pattern map.

![Flow regime map of Taitel and Dukler](image)

Figure 7.7: Flow regime map of Taitel and Dukler [2] for horizontal co-current gas-liquid flow, with representative sample of experimental data range superimposed

The abscissa, ordinates, and transition boundaries in the Taitel and Dukler map are defined by the relevant equations given in Chapter 2, Section 2.2. The abscissa, X, is the Martinelli parameter which fixes the horizontal position on the map, regardless of the flow regime. However, the choice of ordinate is based on the specific transition being considered. $K_{TD}$ specifies the vertical position on the map for the stratified - stratified-wavy flow transition. $F_{TD}$ and $T_{TD}$ define the ordinate for the stratified-wavy - annular-dispersed and stratified-wavy - intermittent flow transition, and intermittent - bubbly flow transition, respectively.

In this study, the stratified - stratified-wavy flow transition is the applicable mechanism - emphasised by the location of the experimental data in figure 7.7. As can be seen, the majority of the sample experimental data range falls into the stratified-wavy regime, with some data points indicating full stratification. The ordered pairs are, therefore, defined by X and $K_{TD}$. The experimental data range plotted in figure 7.7 encompasses all measured flow rates, and covers a range of steam temperatures and steam qualities. In this regard, the sample data range is a good representation of the entire data population from experimentation. Based on the evidence of figure 7.7,
the dominant two-phase liquid-vapour flow morphology can, therefore, be expected to be stratified-wavy. This implies that a pool of liquid condensate resides in the bottom of the tube, whilst the usually well-defined interface between the stratified liquid phase and vapour phase is disturbed by interactions from the vapour flow. Physically, these interactions are manifested by waves on the liquid surface. Any disturbance at the liquid-vapour interface which results in the formation of a wave is known as Helmholtz unsteadiness [156]. Furthermore, substantial vapour-shear acting at the interface may also lead to the removal of liquid layers, and subsequent entrainment of liquid droplets in the vapour core flow.

The description of the flow topology heretofore is predicated on the fact that the map of Taitel and Dukler is applicable to the flow conditions encountered during experimentation and, thus, provides realistic flow predictions. Whist authors such as Breber et al. [60] found good agreement between the map and their experimental observations for condensing flows in horizontal tubes, others have noted issues when extrapolating to diabatic flows [165, 163, 164]. Chief amongst these issues is the fact that adiabatic flow maps do not account for the influence of heat flux. In the case of condensation, removal of heat results in both mass transfer (as condensate accumulates), and deceleration of the flow - as discussed in Chapter 6. Both of these fluidic phenomena have an effect on the transition between flow regimes and, hence, the sequence of flow regimes in a diabatic condensing flow.

Another considerable issue with adiabatic flow maps and extrapolating them to diabatic conditions is that the effect of heat transfer can lead to intrinsic ambiguity when actually classifying flow regimes. For example, in figure 7.7, the map of Taitel and Dukler predicts stratified-wavy and stratified flow for the conditions under which the experimental data were acquired. In reality, however, it is well known that condensation on the upper portion of the inside tube wall will result in that wall being covered with a thin film of liquid. Therefore, the condensing flow actually tends to assume a quasi-annular flow formation, even for circumstances that would produce stratified-wavy or stratified flow in an adiabatic system. This difficulty in basic flow regime classification undermines the methodology of flow regime mapping, and is potentially an avenue for further research efforts.

Flow regime maps for diabatic flows are much less common than those for adiabatic cases [164]. Specifically for condensation, maps have been proposed by [131, 61]. However, the most recent addition to the literature is the map of El Hajal et al.[3], which is essentially a modified version of the Kattan et al. [62] flow regime map for evaporating and boiling flows in horizontal tubes. Thus, the influence of heat flux is implicit in the map, with the similarities between evaporating and condensing flows being well established in the literature. This map is presented in coordinates of vapour quality (x - the abscissa) and mass flux (G - the ordinate). Plotting data on such a map, therefore, facilitates prediction of the evolution of the flow at fixed mass fluxes through the condenser tube. Results from plotting a sample range of experimental data on the map of El Hajal et al. are presented in figure 7.8.
Figure 7.8: Flow regime map of El Hajal et al. [3], evaluated for steam at $T_s = 70 \, ^\circ C$ and $G = 1.9 \, \text{kg/sm}^2$, in a horizontal tube of $D = 25.3 \, \text{mm}$

The flow map presented in figure 7.8 is specific to the experimental conditions for which it was generated. One sample set of experimental data at a mass flux of 1.9 kg/sm$^2$, and a mean steam temperature of 70 °C, was chosen to generate the map. Other data sets were excluded for reasons of clarity and cohesiveness. Plotting each data set would require a unique map for each, or at least a new set of transition boundaries on the existing map - which would have rendered the current map illegible. This is due to the fact that the equations which define the transition boundaries include thermophysical property terms - which are functions of the temperature, and terms dependent on the mass flux. These equations are presented in Chapter 2, Section 2.2. Nevertheless, the data set plotted in figure 7.8 is from one of the larger
mass fluxes investigated during experimentation, and clearly illustrates that the flow regime predicted by the map of El Hajal et al. is stratified flow.

The map of El Hajal et al. is analysed from right-to-left. In this manner, \( x = 1 \) corresponds to the condition at the tube inlet, whilst \( x = 0 \) corresponds to the condition at some point downstream of the tube inlet where all the vapour has been converted to liquid. In all measurements, full condensation occurred, resulting in a steam quality variation from inlet to outlet of \( 1 \leq x \leq 0 \). As can be seen in figure 7.8, this allows experimental data, at a given mass flux, to be projected along the entire x-axis of the El Hajal et al. map. In this case, the entire data range falls within the stratified flow regime. Stratified flow is characterised by a relatively thick liquid condensate pool in the bottom portion of the tube, separated from the vapour-phase by a well-defined, smooth interface. Based on the El Hajal et al. map, this type of flow topology is expected to exist along the entire length of the condenser tube. As the data plotted in figure 7.8 is extracted from the largest flow rate investigated, it is unlikely that, based on the prediction of El Hajal et al., stratified-wavy or annular flow will occur. However, stating this in a conclusive manner would be remiss of the fact that there are underlying issues associated with the map.

There are the obvious limitations relating to the range of applicability of the map - something which was acknowledged by the authors. According to El Hajal et al., the map is expected to be reliable and accurate over a range of parameters such as \( 16 \leq G \leq 1532 \text{ kg/sm}^2 \), \( 0.02 \leq P_r \leq 0.8 \), and \( 3.14 \leq D \leq 21.4 \text{ mm} \). The data examined by the author in this thesis is outside these specified ranges and, therefore, it is difficult to know with any great degree of certainty if the map can be relied upon to predict the actual flow pattern. Accentuating this is the fact that water/steam was never investigated as a working fluid, with the majority of the map evaluations being carried-out for various refrigerants. The issue with this is that most refrigerants have vapour densities of at least two - three orders of magnitude greater than the vapour density of water, at comparable temperatures [13]. As a consequence, for a given mass flow rate or mass flux, the vapour velocity for a water-steam flow will be much larger than the vapour velocity of a refrigerant. It is well established in the literature that the velocity of the vapour/gaseous phase in a two-phase flow has a significant influence on the flow distribution in a pipe. Hence, it is not unrealistic to expect an appreciable vapour velocity to alter the flow regime in practice from that predicted by a largely refrigerant-orientated map, such as that of El. Hajal et al. Applying this argument to the results in figure 7.8 implies that the flow may not be fully-stratified as the map indicates, but that the larger vapour velocity at the given mass flux could lead to a stratified-wavy flow structure, or even an annular geometry - particularly at high vapour qualities near the tube inlet.

Ultimately, it can be concluded from attempts at flow mapping, that there is a large element of ambiguity associated with the current approaches. A variety of flow maps are available in the literature, mostly based on empirical observations of specific fluids, at certain operating conditions which were predetermined to satisfy the
given author’s requirements. Included in this thesis, in figures 7.7 and 7.8, are two of the more popular and relevant maps. However, both maps predicted different flow regimes. It appears that the subjective nature of classifying flow pattern observations from one observer to another, and even the difference in opinion on actual flow pattern definitions can lead to predictions that are dependent on the map of choice. In short, there is no universal flow map. Exacerbating the problem is the reliance on refrigerants for generating the transition boundaries and coordinates for the current maps. As outlined in Chapter 2, Section 2.4, the properties of refrigerants are markedly different from water/steam and it is proposed that this can, ultimately, result in misleading flow regime predictions. As a consequence of the issues outlined, the flow pattern in the reduced-scale ACC tube was investigated using experimental means to provide empirical evidence for comparison with the flow mapping efforts.

7.2.2 Flow regime identification

The results from a novel, non-invasive, and non-visual experimental technique to identify the flow regime in a circular tube are presented in this section. This technique was described in Chapter 5, Section 5.3. Essentially, this method was predicated on the fact that the presence of a film of liquid at any point inside the tube is accompanied by a temperature drop from the steam core to the tube wall - on which the liquid film resides. A thicker film on the inner tube wall leads to a larger temperature drop than that given by a thinner film. A temperature drop through the liquid film has been recognised by many authors [81, 28, 156] as an intrinsic feature of condensation. By measuring the local temperature drop around the inner tube circumference, it is possible to infer the distribution of the liquid film inside the tube. This approach is based on the method originally pioneered by Rosson and Myers [23], who measured point values of condensing film coefficients in a horizontal pipe, although their analysis was limited to one axial plane. By adopting and expanding upon this method, the local temperature measurements should permit prevailing, and physically dissimilar, flow regimes such as annular flow and stratified flow to be distinguished from one another.

For a fixed flow rate through the tube, figure 7.9 presents a set of temperature difference \( (T_s - T_w) \) measurements which were obtained by a full rotation of the tube about its central axis. The graphic accompanying figure 7.9 illustrates that the rotation angle is taken from the top of the tube \( (\varphi = 0^\circ) \) to the bottom of the tube \( (\varphi = 180^\circ) \), before returning to the top again \( (\varphi = 360^\circ) \). All results presented in this section are for a horizontal tube.

The most noticeable feature of the measurements presented in figure 7.9 is the symmetry of the data. It can be seen that the data is axisymmetric about the radial r-plane centreline, i.e. \( 0^\circ \leq \varphi \leq 180^\circ \). The data plotted in figure 7.9 is taken from a number of identical experimental runs at the same mass flow rate, and from the same axial plane of the tube. The data is relatively consistent, displaying the same
overall trend. Namely, that as the rotation angle is increased the temperature difference increases, from the top to the bottom of the tube. It can be seen that the maximum temperature difference occurs in-and-around the bottom of the tube, with an increase in rotation angle from the bottom of the tube resulting in a decrease in the temperature difference.

The shape and magnitude of the temperature profile around the inside of the tube could be interpreted as an indication of the type of two-phase flow regime. Analysing the data in figure 7.9 from left-to-right, it can be seen that there is a relatively small temperature difference around the top of the tube. This temperature difference is the basis for condensation to occur, and remains relatively constant until an angle of approximately 115° - 120° is approached. Hereafter, the temperature difference increases quite dramatically - suggesting that the thickness of the liquid layer on the inside wall is increasing. The increase in temperature difference subsides and peaks between a rotation angle of about 160° - 180°. This implies the the majority of the liquid resides within this region. The dramatic increase in temperature difference from the top to the bottom of the tube is consistent with the measurements of Rosson and Myers [23], who observed an almost linear increase in temperature difference with rotation angle, from approximately 100°. Such measurements imply that the flow is assuming some quasi-annular flow. This is characterised by a thin film of liquid in the upper portion of the tube, with a thicker condensate pool residing in the bottom of the tube - as depicted in the graphic accompanying the plot in figure 7.9. In an adiabatic flow, this would most likely be classified as a stratified or, perhaps, a stratified-wavy flow. However, as the vapour is condensing in this case, a thin film of liquid is invariably formed which coats the entire perimeter of the inside tube wall.

Figure 7.9: Variation in measured temperature difference from steam core (\(\approx\) constant) to tube wall around the tube circumference at an axial plane \(x/L = 0.5\), with \(Re_v = 2690\). Graphic illustrates direction of tube rotation.
Therefore, this leads to a flow topology which resembles annular flow, albeit with a thicker film in the bottom portion of the tube.

It is important to note that the symmetry depicted in the data in figure 7.9 was also apparent at other axial locations, and at all other flow rates investigated, but such measurements are excluded here for the purposes of clarity - the axisymmetry is sufficiently illustrated by figure 7.9 alone. In addition, due to the axisymmetry, all subsequent measurements were abridged in that they were only acquired for a half-tube rotation, i.e. $0^\circ \leq \varphi \leq 180^\circ$. Figure 7.10 presents a set of such measurements, for each designated axial site along the length of the tube. For a set tube rotation, the temperature drop from the steam core to the wall was acquired simultaneously at each axial plane. The data is presented in the form of a polar plot - where the qualitative distribution rather than the actual quantitative distribution is the main concern here. Accompanying the polar plots in figure 7.10 is a schematic of the finned tube - to provide the reader with a sense of context in terms of measurement location.

The data presented in figure 7.10 illustrates a number of prominent characteristics, from which the flow topology can be inferred. Perhaps the most obvious characteristic is the multidimensional nature of the temperature difference measurements. An increase in temperature difference is generally seen to occur in two-dimensions - along the axial direction, and around the tube circumference from top to bottom. The increase in temperature difference along the axial length is simply an intrinsic feature of complete condensation. Inherently, as steam condenses along the length of the tube, the heat transfer rate decreases - due to the reduction in mass flow of vapour. This reduction in heat transfer is manifested by an increase in the temperature difference from the steam core to the inner tube wall. Therefore, for a given radial position, the temperature difference generally increases in the downstream direction from tube inlet to outlet. The polar plots in figure 7.10 are purposefully arranged so that each successive plot depicts the overall trend of temperature difference increasing for all points along the length of the tube.

It is also a feature of the data in figure 7.10 that the largest temperature differences are consistently located in the vicinity of the bottom of the tube. This is almost always the case, apart from the data presented in the polar plot at $x/L = 0.167$, at which point it is unlikely that an appreciably large condensate pool exists to cause an excessive temperature drop. However, the fact that the largest temperature differences exist around the bottom of the tube in all other cases offers compelling evidence that a substantial liquid pool resides in the bottom of the tube, since authors such as [23, 94] have recognised that the temperature drop through the condensate film is analogous to the film’s thickness. The temperature difference at the bottom also increases along the axial direction, suggesting that the liquid pool is growing in thickness along the tube length. Intuitively this characteristic might be expected but, nonetheless, confirms that as condensation progresses along the length of the tube, the resulting liquid condensate tends to accumulate and grow along the bottom portion of the tube as more mass is transferred. Again, this implies that a quasi-
Figure 7.10: Temperature difference from steam core to tube wall around the tube circumference at the axial planes investigated for $Re_v = 3025$. 

7.2 TWO-PHASE FLOW REGIME IDENTIFICATION
7.2 TWO-PHASE FLOW REGIME IDENTIFICATION

Annular, or stratified-wavy, flow regime is the dominant two-phase morphology for the conditions investigated in this thesis. It should be noted that figure 7.10 is based on data from the median flow rate of 0.65 g/s ($Re_v = 3025$), which was chosen to represent the entire range of flow rates. However, the characteristics illustrated and discussed hitherto are a feature of all flow rates investigated where, even though the quantitative distribution of temperature difference varies, the qualitative distribution remains largely similar to that shown in figure 7.10.

To supplement the qualitative distributions shown in figure 7.10, the same data is presented in a more quantitative format in the following figure 7.11. This data consists of temperature difference measurements around the circumference at each axial plane, and is accompanied by the same measurements for the range of specific vapour Reynolds numbers predetermined for the hydrodynamic and thermal measurements.

![Diagram](image1)

Figure 7.11: Variation in measured temperature difference from steam core to tube wall around the tube circumference for a range of vapour Reynolds numbers

It is clear that the measurements in figure 7.11 depict the same overall trends as those described thus far in figures 7.9 and 7.10. For each axial plane presented, the temperature difference from the steam core to the inner tube wall increases from the top to the bottom of the tube. This is broadly a feature of all flow rates examined, and can be interpreted as an increase in the condensate film thickness. This trend was
not, however, observed in the data obtained at \( x/L = 0.167 \) - which is excluded here. The reason for this is most likely due to large vapour velocities present at the tube inlet, coupled with the fact that little condensation would have occurred at that point. Therefore, the condensate layer present at \( x/L = 0.167 \) was probably minimal, with the associated temperature drop through the film being marginal. Nevertheless, the fact that figures 7.11a - 7.11d again illustrate that the largest temperature differences occur towards the bottom of the tube indicates that the liquid is migrating towards that point for all flow rates examined.

Figure 7.11 also demonstrates the effect of flow rate on the magnitude of temperature difference. In general, for all axial planes, an increase in flow rate appears to reduce the magnitude of temperature difference, for any given radial position. This is, perhaps, illustrated most clearly in figures 7.11a and 7.11b, where at angles greater than approximately 120° there is a clear and definitive trend. At these locations, it is suspected that liquid is accumulating and inhibiting heat transfer. The increase in vapour velocity which accompanies an increase in mass flow rate serves to “thin” the film of liquid, thereby augmenting heat transfer and reducing the temperature difference. At angles smaller than 120°, the effect of the condensate film is not as significant and, thus, the effect of flow rate on temperature difference is not seen to the same extent as around the bottom of the tube.

Increasing flow rate can also be seen to somewhat distort the profile by shifting the point at which the temperature difference begins to increase to the right. Again, this is seen most evidently in figures 7.11a and 7.11b, which represent the axial positions in the upstream section of the tube where the vapour velocity is greatest. These figures show that at the largest Reynolds numbers, the presence of the liquid film is not manifested by a larger temperature drop until some point further down around the tube circumference. For example, in figure 7.11a at \( \text{Re}_v = 2030 \), a dramatic increase in temperature difference occurs between 90° and 112°. However, in the same figure at \( \text{Re}_v = 4185 \), this increase never really materialises, suggesting that the influence of the condensate pool is negated by the larger vapour velocities - which tend to force the condensate towards the tube exit. Consequently, this reduces the tendency of the condensate to accumulate in the bottom of the tube, which would otherwise impede heat transfer. Ultimately, the effect of flow rate is to distort the temperature difference profile in the upstream regions, indicating that the two-phase flow regime there is also altered to approach a more symmetrical pattern.

Based on the results presented thus far in figures 7.9 - 7.11, it could be deduced that the two-phase flow morphology consists of a thin film of liquid around the majority of the inside perimeter of the tube, with a thicker liquid layer residing in the bottom. This can be thought of as quasi-annular, or annular-stratified flow - as coined by Carey [156]. Similar findings were reported by Guo et al. [166], who investigated, at flow rates similar to those examined in this thesis, the steam-liquid distribution during steam condensation via electrical capacitance tomography. Ultimately, these findings have implications on the results from the flow mapping in the previous sec-
tion. The discrepancies between the flow mapping predictions and the experimental observations highlight the subjective nature of classifying flow regimes. To an extent, the discrepancies also illustrate the inadequacies of the maps. Where the map of Taitel and Dukler, and El Hajal et al., predicted predominantly stratified-wavy and stratified flow, respectively, the experimental findings suggest that an annular film always surrounds the inner tube perimeter - as a result of the condensation heat transfer. It should be pointed out that the maps do predict the presence of a relatively thicker layer in the bottom of the tube. However, stratified and stratified-wavy flow, by definition, simply consist of a layer of liquid flowing along the bottom portion of the tube, with a vapour phase occupying the remaining volume. This traditional definition does not include a thin liquid layer around the vapour-phase perimeter. The maps abide by the traditional definition whereas a classification such as a quasi-annular flow, or a stratified-annular flow, may be more representative of the actual flow arrangement. The retort to this argument is such classifications only apply to condensing diabatic flows. Adiabatic flows will not consist of a condensing layer in any part of the tube and, hence, should be predicted satisfactorily by the current maps.

An issue with the proposed experimental technique for identifying the flow regime is that it only provides a means with which to infer the regime. It provides a platform to differentiate between very dissimilar flow patterns such as stratified flow and mist flow, for example. As with most data acquired over a period of time, the data presented thus far is time-averaged. The following data in figure 7.12 is a sample of instantaneous data, acquired over a thirty second time interval during experimentation. Examination of this data provides further insight into the flow regime.

The most noticeable feature of figure 7.12 is the progression of scatter in the data from the measurements taken around the top of the tube ($\varphi = 0^\circ - 90^\circ$) to those taken near the bottom ($\varphi = 135^\circ - 180^\circ$). It can be seen in figures 7.12a - 7.12c, which represent the upper portion of the tube, that the data is relatively smooth across the time period for both vapour Reynolds numbers examined. This implies that the condensing film is relatively constant at these measurement points. However, as shown in figure 7.12d in particular, measurements near the bottom of the tube exhibited significantly more scatter. These large fluctuations are attributed to the onset of waves in the liquid phase and, inherently, the effect of waves will only be appreciable towards the bottom of the tube where the liquid film resides. A similar phenomenon was also observed by Rosson and Myers [23] who found the effect of waves to be most pronounced at approximately $120^\circ$ from the top of the tube. Rosson and Myers presented some similar temperature measurements, and stated that the irrational variation in the temperature difference histories must be explicitly due to the action of waves moving up and around the inside tube perimeter in an intermittent fashion. The presence of a thicker film of liquid at the measurement point, albeit for a short period of time, will give rise to a larger temperature drop at that point. This is manifested in the measurements presented in figure 7.12d. Carey
Figure 7.12: Measured temperature difference history over a 30 second data acquisition period for the lowest (Re_v = 2030) and highest (Re_v = 4185) flow rates investigated (a) - (e). Variation in standard deviation from the mean temperature difference as a function of rotation angle (f). All measurements were acquired at x/L = 0.667
[156] identified strong vapour shear, associated with high velocity vapour flows, as the main cause of the surface waves.

Figure 7.12e presents the data from the extremity of $\varphi = 180^\circ$. This data is qualitatively similar to that presented in figure 7.12d, albeit with scatter that appears to be less pronounced. At $\varphi = 180^\circ$, the effects of waves will not be an issue but it is thought that the apparent scatter in the data at this point is due to shearing and the resultant deformation at the interface of the condensate layer. Such intermittent disturbances would most likely result in a fluctuation in condensate thickness, giving rise to the scatter illustrated. Ultimately, the scatter in the data is quantified by the standard deviation from the mean given in figure 7.12f, for both flow rates, as a function of the tube rotation angle. This plot contains data points determined from temperature difference histories at rotation angles which, to preserve clarity, were not included in figure 7.12. Nevertheless, this data compliments that already presented and, for both flow rates, it is clear that the greatest deviations occur near the bottom of the tube. In fact, there is a noticeable maximum in the standard deviation at $\varphi = 135^\circ$, which emphasises that waves are particularly a feature at this angular position in the tube. It is also interesting to note that the standard deviation is generally larger for the higher flow rate - possibly due to the higher vapour velocities and the subsequent disturbances of the condensate. Even though only two flow rates were presented here, they were deemed to be representative of the entire range as $Re_v = 2030$ and $Re_v = 4185$ was the smallest and largest flow rate investigated, respectively. Therefore, the wavy nature of the flow deduced from the data at those flow rates is expected to be a feature across all intermediary flow rates.

The instantaneous data presented in figure 7.12 and, in particular, figures 7.12d and 7.12e, appears to indicate a wavy element to the stratified-annular flow topology inferred from the previous time-averaged measurements. The presence of waviness in the flow would tend to be in agreement with the predictions of the Taitel and Dukler flow map, given in figure 7.7, which was shown to predict a stratified-wavy regime for adiabatic cases. For a diabatic condensing flow, it might be reasonable to assume that the traditional definition of a stratified-wavy flow for adiabatic conditions would be extended to include the annular condensing layer. If this were to be the case, the map of Taitel and Dukler would agree quite well with the flow regime (stratified-wavy with annular film ~ quasi-annular) identified in this section. However, the smooth-stratified prediction of the map of El Hajal et al., shown in figure 7.8, does not agree with the experimentally-deduced regime. According to that map, wavy flow is not encountered until much larger mass fluxes ($\approx 10 - 20$ kg/sm²) are considered. Therefore, it is suggested that a more suitable metric for presenting such a map might be the vapour velocity. It is widely acknowledged in the literature that vapour velocity is one of the primary factors for the formation and amplification of waves on the liquid surface. Basing the map on mass flux neglects to account for higher vapour velocities associated with fluids with much lower densities than those for
which the map was originally created. Ultimately, this can lead to misleading flow regime predictions.

To conclude the two-phase flow regime identification, it is difficult to quantitatively, or qualitatively, compare the results in this section with other findings - mainly due to a lack of similar studies. The vast majority of studies on two-phase flows in round tubes are for a vertical tube configuration. Vertical tubes are considerably easier to characterise due to nature of the set-up which naturally promotes an annular flow regime. Horizontal tube configurations are complicated by the influence of gravity and the tendency for stratification of the flow. Furthermore, almost all studies on horizontal, or slightly inclined, tubes employ some type of refrigerant as the working fluid [164]. Hence, the void in current literature is partly alleviated by the measurements presented in this section, where the prevailing flow regime can be thought of as stratified-wavy with an annular film. Although there is still some ambiguity in classifying this regime, the experimental methodology certainly allows for drastically dissimilar flow regimes to be differentiated and, ultimately, discounted. Table 7.1 presents a range of the most common flow regimes referred to in literature, and the likeliness of their presence at the experimental conditions described in this thesis - based on the measurements and interpretation of the data presented in this section.

Table 7.1: Two-phase flow regimes identified

<table>
<thead>
<tr>
<th>Flow regime</th>
<th>Present</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bubbly</td>
<td>✗</td>
</tr>
<tr>
<td>Plug</td>
<td>✗</td>
</tr>
<tr>
<td>Smooth-stratified</td>
<td>✗</td>
</tr>
<tr>
<td>Stratified-wavy</td>
<td>✓</td>
</tr>
<tr>
<td>Slug</td>
<td>✗</td>
</tr>
<tr>
<td>Annular</td>
<td>✓</td>
</tr>
<tr>
<td>Mist</td>
<td>✗</td>
</tr>
</tbody>
</table>

7.3 PRESSURE DROP MEASUREMENTS

The hydrodynamic characteristics of the two-phase condensing flow of steam are quantified by the pressure drop measurements. These measurements were acquired in accordance with the methodology outlined in Chapter 5, Section 5.3. Compared to the pressure drop measurements recorded from the full-scale MACC, the single ACC tube measurements were acquired in a more systematic and controlled manner. Hence, the measurement portfolio consisted of a larger test matrix. In addition to the two-phase pressure drop measurements of condensing steam, the program was extended to encompass single-phase pressure drop measurements of, firstly, vapour
and, subsequently, water. These measurements were carried-out to evaluate the contributions to the two-phase multiplier. The single-phase measurements are presented first, followed by the two-phase measurements. Succeeding this, the individual components which constitute the overall measured losses are evaluated and, ultimately, the frictional component is compared with correlations from the literature.

7.3.1 Single-phase pressure drop measurements

Generally, most hydrodynamic studies on two-phase flows are concerned with quantifying the two-phase frictional loss. If these losses are to be examined and compared to existing theory and/or empirical relationships, it is preferable for the single-phase contributions to be formally established. For non-complex geometries such as a round tube, laminar or turbulent flow theory exists to predict the fluidic losses associated with a single-phase fluid flow through the pipe. This theory was outlined in Chapter 2, Section 2.3. Figures 7.13a and 7.13b present results from single-phase measurements of vapour and water, respectively. Accompanying each is the corresponding theoretical solution.

Figure 7.13: Single-phase pressure drop measurements, presented in terms of friction factor, as a function of Reynolds number

Both figure 7.13a and 7.13b demonstrate that the measured single-phase friction factor, deduced from the pressure drop measurements, is in good agreement with the relevant theory for each respective case. For the single-phase vapour measurements shown in figure 7.13a, the Reynolds numbers examined were representative of those in the two-phase experiments. However, some of these Reynolds numbers are in the laminar regime or, at the very least, in the transitional regime. These data points can be seen to deviate from the theoretical turbulent flow solution in figure 7.13a. Nevertheless, the majority of the measurements are for a turbulent flow and, accordingly, are in good agreement with the turbulent flow theory. Ultimately, this
quantifies the contribution of the vapour phase to the two-phase frictional loss, whilst also validating the applicability of the theory.

During two-phase flow experiments, the liquid-phase was in the laminar regime - as outlined by the liquid Reynolds number range given in table 5.5 in Chapter 5. As such, the single-phase experiments were carried-out within this range. Although the liquid Reynolds numbers in practice were much lower than those presented in figure 7.13b, the single-phase measurements were acquired at the upper threshold of laminar flow. Doing so yielded more accurate measurements as the pressure drop was largest at such Reynolds numbers. As expected, good agreement was observed between the measured data and the theory.

7.3.2 Two-phase condensing pressure drop measurements

Similar to the full-scale MACC pressure drop measurements presented in Chapter 6, those acquired from the single tube were examined over a range of Reynolds numbers - defined here as vapour Reynolds number. In contrast to the MACC measurements, however, the single tube measurements were acquired at a single nominal fan speed. This approach allowed the pressure drop to be explicitly quantified as a function of steam flow rate alone, with the results presented as follows in figure 7.14.

![Figure 7.14](image-url)

- (a) Absolute pressure drop for single tube ACC
- (b) Comparison of pressure gradient

Figure 7.14: Measured two-phase condensing pressure drop (a), and pressure gradient (b), as a function of vapour Reynolds number

Figure 7.14a illustrates an expected trend - that of the overall, two-phase pressure drop increasing with Reynolds number. It can be seen that the magnitude of these losses are quite small, in the range of approximately 60 - 120 Pa over a vapour Reynolds number range of 1500 - 4500. Nevertheless, the measurements show a pronounced increase with Reynolds number, increasing by about 90% across the range. These measurements are both qualitatively, and quantitatively, similar to those obtained from the full-scale MACC, as evidenced by the pressure gradient measurements given in figure 7.14b. For comparative purposes, the pressure gradient is the relevant metric as the single tube and MACC tubes are different lengths. The absolute pressure
drop measurements from the MACC are, inherently, larger than those from the single tube - which is approximately half as long as the full-scale tubes. However, it can be seen that the pressure gradients compare quite well, with the single ACC tube measurements exhibiting slightly less scatter - perhaps a testament to the greater level of control in the experiments. Ultimately, the measurements show that, from a practical perspective, the condensing pressure loss is not an issue affecting condenser performance, under the range of Reynolds numbers examined herein. The reason for this was addressed in relation to the MACC measurements in Chapter 6, whereby the momentum recovery in the condensing flow offsets the two-phase frictional losses to the extent that the actual measured losses are quite small. This phenomenon is again illustrated in figure 7.15.

![Figure 7.15: Momentum recovery in the condensing flow (a), offsetting frictional losses (b), as a function of vapour Reynolds number](image)

In a similar manner to that observed for the full-scale pressure drop measurements in Chapter 6, the momentum recovery in the condensing flow is explicitly shown in figure 7.15 to offset the frictional losses. Although the inferred data is of the same approximate magnitude, it is quantitatively and qualitatively inverted - a phenomenon which appears to be a unique feature of condensing flows.

### 7.3.2.1 Two-phase frictional pressure drop analysis

Prediction of the two-phase frictional drop has, traditionally, been the most problematic aspect and, hence, main focus of hydrodynamic studies on two-phase flows. This is mainly due to the fact that the majority of studies on two-phase pressure losses are concerned with refrigerants in evaporating flows, where the frictional losses are not offset by momentum recovery. Furthermore, as shown by the pressure drop prediction results in Chapter 6, there are circumstances for condensing flows at which the frictional effects will not be offset by momentum recovery. Therefore, in such cases, the ability to predict the two-phase frictional pressure drop is of great importance as it ultimately allows the actual, measurable two-phase pressure drop to be evaluated. There are a number of empirical correlations available in the literature to predict
the frictional component. In the previous chapter, it was shown that the frictional pressure drop, deduced from the full-scale MACC measurements, agreed best with the method of Lockhart and Martinelli [8]. This was emphasised by an agreement of ±18%. As the single-tube measurements exhibited the same trends, and were similar in magnitude to the full-scale measurements, the Lockhart and Martinelli model was again employed to predict the frictional pressure drop. Figure 7.16 presents the frictional pressure drop data with the prediction of Lockhart and Martinelli.

Figure 7.16: Inferred frictional pressure drop as a function of vapour Reynolds number compared with the prediction of Lockhart and Martinelli [8]

From figure 7.16, the model of Lockhart and Martinelli can be seen to be in good agreement with the experimentally-derived frictional pressure drop data. There are some differences between the data and predictions, particularly towards the lower range of Reynolds numbers, but an average difference of approximately 16% is testament to the applicability of the Lockhart and Martinelli model to condensing flows. Indeed, in their seminal paper [8], the authors suggested that the correlation which they proposed could be utilised for the prediction of pressure drop during condensation when a step-wise solution is employed - as was done by the author. This applicability can then, perhaps, be best explained by the isothermal, two-phase, two-component, horizontal flow conditions under which Lockhart and Martinelli acquired their data. The test conditions from which the measured data in this thesis were obtained were not too dissimilar to those of Lockhart and Martinelli, notwithstanding the heat transfer. Nevertheless, the applicability of the Lockhart and Martinelli model to steam condensation in round tubes has not been verified by experimental means prior to the work of the author [167] in this thesis.

Lockhart and Martinelli originally presented their data by a series of empirically-based dimensionless curves. They correlated their data using, what is now referred to as, the Martinelli parameter - X. A series of empirical curves were then fairied through the data to provide a means with which to determine the two-phase multiplier - φ.
Both of these terms are defined in Chapter 2. A formal theoretical basis to calculate the two-phase multiplier was introduced by Chisholm [168], the details of which are also given in Chapter 2. Presenting the data on the curve of Lockhart and Martinelli, as shown in figure 7.17, provides greater insight into the contributing factors to the two-phase pressure drop.

Figure 7.17: Measured pressure drop data presented in terms of Martinelli parameter ($X$) and two-phase multiplier ($\phi$), compared with the solution of Lockhart and Martinelli [8].

Figure 7.17 presents the frictional pressure drop data in dimensionless form. The abscissa (Martinelli parameter) and ordinate (two-phase multiplier) are dependent on the flow conditions. The subscripts associated with the curve are indicative of this, where “lt” refers to laminar liquid flow and turbulent vapour flow - the conditions generally observed during experimentation. The Martinelli parameter is a ratio of the frictional pressure gradients for the liquid and vapour phases flowing alone in the pipe. The general position of the data along the x-axis demonstrates that the vapour frictional pressure gradient is one - two orders of magnitude greater than the corresponding liquid gradient. These gradients are calculated using the equations given in Chapter 2, Section 2.3, whilst incorporating the experimentally-measured friction factors from Section 7.3.1. The two-phase multiplier is a ratio of the two-phase frictional pressure drop, and single-phase pressure drop of either individual phase. Similarly, the position of the data along the y-axis illustrates that the experimentally-derived frictional pressure drop is one - two orders of magnitude larger than the single-phase liquid pressure drop - in this case.

Despite the fact that the data is slightly offset from the curve of Lockhart and Martinelli, it can be seen in figure 7.17 that the agreement is quite good. The Lockhart and Martinelli solution is defined by the theoretical Martinelli parameter, and two-phase multiplier expression formulated by Chisholm - which varies depending on the flow conditions. With a laminar liquid flow and turbulent vapour flow during ex-
perimentation, the constant $C$ in the Chisholm expression (equation 2.41 in Chapter 2) has a value of 12. Employing regression analysis to each experimental data point in figure 7.17 allows the corresponding value of $C$ to be obtained. The average value from the entire data range was found to be approximately 17. Compared to the values proposed by Chisholm, this lies between the value of 20 for turbulent-turbulent flow and 12 for laminar-turbulent flow. Incorporating this value of $C = 17$ into the Lockhart and Martinelli model and re-calculating the frictional pressure drop gives the results presented in figure 7.18.

![Figure 7.18: Comparison between experimentally-deduced frictional pressure drop and predicted frictional pressure drop of Lockhart and Martinelli [8]](image)

Incorporating $C = 17$ into the Lockhart and Martinelli model is shown to improve the agreement between the inferred data and the model - as can be seen be comparing figures 7.18a and 7.18b. An average difference of approximately 16% with $C = 12$ (figure 7.18a) is reduced to approximately 9% with $C = 17$ (figure 7.18b). In theory, the reason for this improvement is that increasing the value of $C$ increases the magnitude of the frictional pressure drop calculated using the Lockhart and Martinelli model. In essence, this shifts the curve closer to the experimental data originally presented in figure 7.16. Physically, however, the improvement can be explained by considering the nature of the liquid condensate flow. By conventional definition, it is a laminar flow under the conditions investigated during experimentation. Nevertheless, it could be argued that the large vapour velocities present, as a result of operating at low temperatures and pressures during condensation, accelerate the velocity of the liquid phase. Indeed, it is recognised in the literature that the vapour phase induces shearing forces on the liquid film which can deform the interface and/or force the film towards the tube exit [156]. If this is the case, the liquid Reynolds number could potentially be larger than that assumed, contributing a larger frictional pressure drop. Ultimately, it could be the case that increasing $C$ to 17 accounts for this in the Lockhart and Martinelli model. Therefore, the following correlation is proposed as a means of calculating the two-phase multiplier and, hence the frictional pressure drop, for condensing flows of steam in round, horizontal tubes.
\[ \phi_l = \left( 1 + \frac{17}{X} + \frac{1}{X} \right)^{1/2} \]  

(7.1)

7.4 CONDENSATION HEAT TRANSFER MEASUREMENTS

Condensation heat transfer measurements presented in this thesis are separated into those acquired on the local-scale, and those acquired on the global-scale. The local measurements provide a level of insight into the physical mechanisms of the condensing flows, which global measurements fail to capture. In this respect, the local measurements compliment those on the global-scale by providing underlying explanations to trends and characteristics. Global measurements, essentially, summarise local observations, and express the findings in a manner which permits comparison to relevant studies in the literature. In this section, the local measurements are presented first, followed by the global measurements. All measurements were acquired in accordance with the methodology described in Chapter 5, Section 5.3.

7.4.1 Local condensation heat transfer

Convective condensation measurements on the local-scale are limited in the literature, with few studies presenting detailed, local measurements. Of the studies that are available, the majority are for vertical tube configurations [24, 90], with only one such study available for a horizontal, or slightly inclined tube [23]. This thesis, and this section in particular, seeks to alleviate such deficiencies in the current literature by providing local steam condensation measurements. Local measurements, by their very definition, are expressed in terms of localised dimensions, with the measurements in this thesis presented as a function of tube length (non-dimensionalised as \( x/L \)), and circumferential position (non-dimensionalised as \( \varphi/\pi \)). Such local measurements were acquired over a range of parameters including vapour Reynolds numbers, air-side heat transfer coefficients, and tube inclination angles. However, in the interest of cohesiveness and conciseness, the measurements are presented here for a nominal fan speed (hence, nominal air-side heat transfer coefficient) and horizontal tube position. The effects of varying these parameters is addressed in the global results section, which follows this. The first set of local condensation measurements are presented in figure 7.19, in which the local circumferential condensing Nusselt number is given as a function of non-dimensional axial distance along the tube, for a range of vapour Reynolds numbers.

The measurements presented in figure 7.19 illustrate a number of important characteristics of convective condensing flows, perhaps none more so than the inherent degradation in heat transfer as condensation progresses along the length of the tube. This degradation, quantified here by a reduction in condensing Nusselt number, is shown by figures 7.19a - 7.19e to progress in the axial direction irrespective of circum-
Figure 7.19: Local circumferential condensing Nusselt number as a function of non-dimensional tube axial length, for a range of vapour Reynolds numbers (a) - (e). Circumferentially-averaged Nusselt number as a function of non-dimensional tube axial length (f)
ferential location. As alluded to in Section 7.2, in complete condensation studies such as this, deterioration in heat transfer along the tube length is a direct consequence of the vapour flow being continuously converted to a liquid flow. With vapour being progressively condensed in the mean flow direction, the energy remaining in the vapour flow, inherently, decreases - leading to lower heat transfer rates relative to those measured upstream in the tube. This decrease in heat transfer rate is manifested by a reduction in tube wall temperature, with the net effect being a reduction in condensing Nusselt number as seen in figure 7.19. Such variation of heat transfer in the axial direction is an intrinsically fundamental feature of complete condensation approaches. Similar findings have been reported by Goodykoontz and Dorsch [24, 90] and Bae and Maulbetsch [94], who locally examined complete steam condensation in a vertical tube, and complete condensation of refrigerants in horizontal tubes, respectively.

Regardless of vapour Reynolds number, it is clear that figures 7.19a - 7.19e share a number of common qualitative characteristics. These similarities are summarised in figure 7.19f, which presents the circumferentially-averaged Nusselt number along the axial distance. One of the most obvious features of the measurements presented in this figure, besides the ubiquitous decrease in Nusselt number in the axial direction, is that the largest Nusselt numbers are always present near the inlet to the tube, \( x/L = 0.167 \). This tends to occur due to two main reasons. Firstly, only a small quantity of vapour will have been condensed near the inlet to the tube, resulting in a relatively thin condensate film, which would otherwise impede heat transfer. Coupled with this are the presence of large vapour velocities, which are widely acknowledged to promote convective condensation heat transfer [57, 9, 22]. At the tube entrance, the specific volume of the vapour is considerably greater than that of the liquid-phase and, hence, the medium enters at a relatively large velocity. However, this velocity decreases in an almost linear manner, from inlet to outlet of the tube [28], which has implications on the shape and magnitude of the Nusselt number profiles in the axial direction.

It is a feature common to all measurements presented, and is perhaps illustrated most clearly in figure 7.19f, that the Nusselt numbers appear to converge as the tube exit is approached, \( x/L = 0.834 \). This is primarily a result of the majority of the vapour being condensed at this point and, inevitably, the heat transfer rate must decrease as a result. However, the slope and shape of the data for each individual vapour Reynolds number also gives an indication as to the progression of condensation through the tube. For example, in figure 7.19f, it can be seen that at \( Re_v = 2030 \) the slope of the line is relatively flat beyond \( x/L = 0.5 \). This contrasts with the slope of the lines at higher vapour Reynolds numbers where, at \( Re_v = 4185 \) for example, the slope is much more pronounced, and continues to decrease quite dramatically up until the last measurement site at \( x/L = 0.834 \). This disparity in slopes suggests that the majority of the vapour has condensed in the upstream portions of the tube at the lower Reynolds numbers, whilst the larger Reynolds numbers require more of the
tube length for full condensation. Presenting some sample measurements acquired from around the bottom of the tube circumference provides the best illustration of this where, in figure 7.20, the condensing Nusselt number is seen to approach the single-phase liquid Nusselt number as x/L approaches unity.

Figure 7.20: Local circumferential Nusselt number for a range of positions near the bottom of the tube as a function of non-dimensional tube axial length, coupled with the single-phase liquid Nusselt number solution

In both figure 7.20a, and figure 7.20b, the condensing Nusselt number is seen to asymptotically approach the single-phase liquid Nusselt number. However, the manner in which the single-phase Nusselt number is approached gives an insight into the condensing flow. In figure 7.20a, the condensing Nusselt number measurements at x/L = 0.334 and x/L = 0.5 appear much closer to the single-phase Nusselt number than the corresponding measurements at Re_v = 4185, in figure 7.20b. This suggests that more condensate is present at these points in the tube at the lower vapour Reynolds numbers. One reason for this is that more vapour has simply condensed at these points, which would be consistent with the argument put forth for explaining the characteristics of figure 7.19f. However, related to this is the influence, or otherwise, of the associated vapour velocities on the flow. If it is the case that lower Reynolds number flows fully condense earlier in the tube, then it follows that the vapour velocities in the downstream sections will be minimal - simply as a result of a negligible vapour-phase. Conversely, as higher Reynolds number flows require more of the tube to fully condense, the vapour velocities will be larger for a given axial location along the tube. Ultimately, the effect of increased vapour velocities will lead to larger condensing Nusselt numbers in the downstream locations, for the higher Reynolds number flows, as seen in figure 7.20b.

The increase in condensing Nusselt number with increasing vapour Reynolds number was seen, for any given axial or circumferential position, across figures 7.19a - 7.19e, and is also explicitly illustrated in figure 7.19f. Notwithstanding the large Nusselt numbers already present near the inlet to the tube, the influence of the Reynolds number was particularly apparent there, with the Nusselt numbers being augmented
7.4 Condensation Heat Transfer Measurements

to a large extent. This is primarily due to the large vapour velocities which accompany an increase in Reynolds number. Subsequently, these velocities have a continued influence further downstream along the length of the tube. The effect of vapour velocity is to increase the advection component of convection through induced shearing effects on the liquid-phase. Shear stresses imparted from the vapour-phase are nominally exerted on the surface of the condensate, i.e. at the liquid-vapour interface, and can be divided into lateral and axial stresses [57, 70, 156]. The lateral stresses tend to distribute the condensate film more evenly around the tube perimeter, in response to gravity which tends to stratify the flow. The primary axial stresses (vapour shear) tends to force the condensate towards the tube exit and also has the effect of “thinning” the liquid film through liquid shearing at the interface. This “thinning” is often manifested by entrainment of liquid in the vapour core flow. Ultimately, these effects combine to promote heat transfer. However, the influence of Reynolds number appears to diminish with axial distance. This can be seen in figure 7.19f, where at $x/L = 0.667$ and $x/L = 0.834$, the increases in Nusselt number with Reynolds number are considerably smaller than those upstream. Again, this is attributed to the much smaller vapour velocities present downstream in the tube, which do not augment heat transfer to the same extent as the upstream larger velocities. Nevertheless, vapour Reynolds number is still seen to improve heat transfer and, when expressed on a global-scale, this becomes more apparent. Such global measurements are presented in the section following this.

Further evidence for the favourable effect of vapour Reynolds number and, hence, vapour velocity on the local-scale is shown in a number of more subtle and nuanced attributes to the measurements discussed thus far. One such example is the apparent distortion in the Nusselt number profile as Reynolds number increases. In figures 7.19a and 7.19b, it can be seen that there is a clear division between the data measured around the top of the tube and that near the bottom of the tube. This division is first encountered at $x/L = 0.334$, and continues along the remainder of the tube. However, with increasing Reynolds number, this division in data appears to diminish, particularly at $x/L = 0.334$ and $x/L = 0.5$ - as seen in figure 7.19e. The measured Nusselt number at these points appears to be closer to those measured around the top of the tube, implying that the increased vapour velocity is either thinning, or distributing, the liquid pool at the bottom of the tube. Regardless of the actual mechanism, the net effect is an improvement in heat transfer.

It was a feature at almost every axial position that the Nusselt number decreased around the tube circumference, from top to bottom. As shown by some of the temperature difference measurements presented in Section 7.2 this was a direct result of the increase in temperature drop, from the steam core to the tube wall, through the condensate film. The following figures 7.21 and 7.22 explicitly illustrate the decrease in Nusselt number from the top to the bottom of the tube by presenting the qualitative and quantitative distribution in local Nusselt number as a function of non-dimensional angular position inside the tube.
Figure 7.21 and 7.22 present the variation in local circumferential Nusselt number for a range of vapour Reynolds numbers for all non-dimensional axial lengths examined. It is clear that, with the exception of x/L = 0.167, there is a tendency for the Nusselt number to decrease as the bottom of the tube is approached. At x/L = 0.167, which is the closest measurement site to the tube inlet, a definitive trend of decreasing Nusselt number around the circumference was not evident. As such, the data was more uniform in appearance. It was suggested that this was due to the lack of an appreciable condensate pool residing in the bottom of the tube, in conjunction with the large vapour velocities near the inlet. However, as can be seen by all measurements acquired at axial locations downstream of the inlet, the shape of the Nusselt number profiles certainly suggests the presence of a condensate pool around the bottom of the tube, which is inhibiting heat transfer.

For all measurements presented in figures 7.21c - 7.22f, there is a trend of condensing Nusselt number decreasing as the bottom of the tube is approached. This largely occurs regardless of vapour Reynolds number or axial position. However, it
Figure 7.22: Variation in local circumferential Nusselt number with non-dimensional angular position inside the tube for a range of vapour Reynolds numbers at axial locations $x/L = 0.5$, $x/L = 0.667$, and $x/L = 0.834$.
is clear that the extent of the decrease in Nusselt number varies with these parameters. As expected, the decrease is much more pronounced and obvious for the axial locations closer to the tube exit, for example $x/L = 0.834$. This is attributable to two factors. Firstly, the majority of the vapour will have been condensed as the tube exit is approached and, consequently, the resulting condensate will tend to accumulate in this region. The tendency of this condensate to reside in the bottom of the tube, however, is largely due to the reduction in vapour velocity towards the tube exit. In the absence of any significant inertial or shearing forces, gravitational forces will tend to dominate, leading to the apparent stratification of the condensate pool towards the tube exit. This contrasts with the Nusselt number profiles around the circumference in further upstream locations, in particular $x/L = 0.167$ and $x/L = 0.334$, where there was no discernible decrease in Nusselt number around the tube circumference. Hence, it could be assumed that the flow at this point resembled an annular-type topology. Intuitively, this would seem correct as large vapour velocities, such as those near the inlet, are recognised as one of the primary contributing factors to an annular profile [81, 156]. The measurements at $x/L = 0.334$ also appear to suggest that the presence of a condensate pool in the bottom of the tube is not as appreciable as further downstream. This is evidenced by the relatively flat data lines in figure 7.21d where, although the condensing Nusselt number certainly decreases around the circumference, the decrease is much more benign. For example at $Re_v = 3680$, there is an approximate 34% decrease in Nusselt number around the circumference at $x/L = 0.334$. This increases to approximately 70% at $x/L = 0.834$. Hence, it appears that the depth of the condensate pool is increasing along the bottom of the tube in the axial direction, and that this pool is explicitly accounting for a reduction in heat transfer. Indeed, such an explicit deterioration in heat transfer due to the condensate pool has been recognised by many researchers [57, 70, 94] as the primary disadvantage of filmwise condensation. Some researchers [70] even went as far to completely disregard the heat transfer through the film, assuming it was negligible in comparison to that in the upper regions.

Further evidence for the hypothesis that the condensate pool is progressively accumulating and, hence deterring, heat transfer is given by the point of abrupt transition in Nusselt number profiles around the circumference. In figure 7.21d at $Re_v = 4185$, a sharp, dramatic decrease in Nusselt number is not particularly apparent but does appear to occur around $\varphi/\pi \approx 0.75$. This indicates that the presence of the condensate pool is not really manifested by a deterioration in heat transfer until a angle of approximately 135° from the top is approached. This is in contrast with the same Reynolds number further downstream at $x/L = 0.834$ where, as seen in figure 7.22f, the decrease in Nusselt number occurs around $\varphi/\pi \approx 0.5$, which equates to 90° from the top of the tube. Similar trends are noted at the other Reynolds numbers examined.

The measurements presented in figure 7.21 are similar in approach to those originally presented by Rosson and Myers [23], who investigated point values of con-
densing film coefficients for methanol and acetone in a horizontal pipe. However, Rosson and Myers only examined the circumferential distribution in one axial plane, without ever extending their analysis to other axial locations. In this regard, the local measurements presented here can be viewed as an expansion of their original set of measurements. By examining both the circumferential and axial distribution of Nusselt number, the local measurements depicted in figures 7.19 and 7.21 demonstrate a degradation in condensation heat transfer in two-dimensions, due to two related mechanisms. The first set of local measurements quantified the reduction in heat transfer along the axial direction, which is an inherent feature of complete condensation due to the vapour flow progressively decreasing along the tube length. The second set of local measurements quantified the decrease in heat transfer, circumferentially, due to the formation and accumulation of condensate around the bottom of the tube. Figures 7.23 and 7.24 summarise the findings in graphical form.

The contour plots presented in figures 7.23 and 7.24 were developed in accordance with the technique described in Chapter 5, Section 5.3, by combining the local measurements from the axial and circumferential positions. The series of contour plots given in figure 7.23 are presented with an absolute scale - to illustrate and convey the quantitative and qualitative distribution of condensing Nusselt number at that particular vapour Reynolds number. Actual comparison between the results at the various Reynolds numbers is, subsequently, demonstrated by the same contour plots re-created in figure 7.24. These contour plots are presented with a common scale, encompassing the largest and smallest Nusselt numbers measured across all flow rates. In any case, the contour plots presented in figure 7.23 are a good graphical summary of the findings outlined thus far. Namely, the condensing Nusselt numbers are largest near the inlet to the tube and decrease progressively in the axial direction. In addition to the decrease in the axial direction, the condensation heat transfer rate was also found to deteriorate in the circumferential position, from the top to the bottom of the tube. This reduction in heat transfer was not apparent near the tube inlet and was only measurable, as an appreciable temperature difference, at downstream positions in the vicinity of the bottom of the tube. Thus, it was suggested that the condensate which was forming was accumulating around the bottom perimeter of the tube in downstream regions, thereby contributing a thermal resistance. As indicated by the contours in figure 7.23, the reduction in condensing Nusselt number in the axial direction is worse near the bottom of the tube than at the top.

The contour plots can also be interpreted to give an indication of the two-phase condensing flow morphology through the tube. If such an approach is adopted here, based on the qualitative distributions shown in figures 7.23a - 7.23e, an annular flow regime could be proposed to be present near the inlet to the tube, with the flow assuming a quasi-annular (annular with a stratified layer) in the downstream regions. This interpretation would be consistent with that deduced hitherto, from the temperature difference measurements in Section 7.2. The inference of annular flow can be justified by the approximate uniform distribution of Nusselt number around
Figure 7.23: Series of contour plots illustrating bi-directional variation of condensing Nusselt number for range of vapour Reynolds numbers. All contour plots are given with an absolute scale and original contour level.
7.4 CONDENSATION HEAT TRANSFER MEASUREMENTS

Figure 7.24: Contour plots re-created from figure 7.23 with common Nusselt number scale and increased contour level
the circumference in the upstream regions. In the downstream regions, however, the distribution of Nusselt number around the circumference is skewed, with an unfavourable gradient towards the bottom of the tube implying a deviation from the symmetrical annular flow regime to a more non-uniform stratified layer of liquid. Ultimately, it is probably the case that this approach and method of interpretation needs to be validated against experimentally-acquired images of the two-phase flow but, in its current form, local spatial measurements can provide a non-invasive, in-situ approach to identifying the two-phase flow regime.

It also appears, from interpretation of the contour plots, that there are localised regions of very low heat transfer. One particular example of this is in figure 7.23c, where it appears that a region along the bottom of the tube extending from approximately \( x/L = 0.58 \) to \( x/L = 0.834 \) is capable of transferring very little energy - indicated by a condensing Nusselt number of the order of 50. As given by the shade of the contours, larger Nusselt numbers are present directly above, and around, this region which seems to suggest that the overriding reason for the relatively negligible heat transfer is that the region is blanketed by the condensate pool. This is also evident towards the tube exit at the other flow rates and certainly suggests a rather large deterrent to heat transfer, in the form of a stratified layer.

In contrast to figure 7.23, the contour plots with relative scales presented in figure 7.24 allow for the effect of vapour Reynolds number to be explicitly demonstrated. It is immediately noticeable that the contour levels, particularly at the lower vapour Reynolds numbers, have disappeared and have been replaced by a more apparent uniform distribution of condensing Nusselt number. The reason for this is because the average, bulk condensing Nusselt number at the lower Reynolds numbers is smaller than those at the larger Reynolds numbers. In addition, the localised regions of large condensing Nusselt numbers, at \( \text{Re}_v = 3680 \) and \( \text{Re}_v = 4185 \) in particular, tend to skew the Nusselt number scale with is common to all contour plots. Nevertheless, the obvious conclusion to extract from figure 7.24 is that the condensing Nusselt number increases, for any given axial or radial location, with vapour Reynolds number. Even though they do not quantify the local highs or lows, this defining characteristic of convective condensation is also expressed in the global measurements, in the following section.

7.4.2 Global condensation heat transfer

Global condensation heat transfer measurements presented here quantify the overall findings from the local measurements in a format which allows for comparison with related studies in the literature. The studies referred to here have been described in detail in Chapter 2. Parameters which were investigated on the global-scale, and which are reported here, were also examined on the local-scale. Such parameters include steam flow rate (vapour Reynolds number), fan speed (air-flow Reynolds number) and tube inclination. However, the effects of these parameters on the condensation
heat transfer is better illustrated on the global-scale, with the local measurements presented thus far referred to, to explain the global results.

The first set of global measurements presented in figure 7.25 demonstrates a trend already seen and described in the local measurements - namely, an increase in Nusselt number with vapour Reynolds number. In this case each data point can be thought of as the circumferentially and axially averaged condensing Nusselt number from the series of contour plots shown in figure 7.24, previously. Supplementing the experimental measurements are a number of Nusselt number correlations from Chapter 2, with a view to evaluating the most applicable.

![Figure 7.25: Variation in global condensing Nusselt number with vapour Reynolds number compared with correlations from Shah [9], Ananiev et al. [10], Cavallini and Zecchin [11], and Dobson [12]](image)

One of the most noticeable features of figure 7.25 is that the measurements presented depict a strong dependency between condensing Nusselt number and vapour Reynolds number. The relationship is one of a large increase in Nusselt number with increasing vapour flow. Whilst expected, this is occurring despite the relatively modest increases in mass flux, of approximately 0.5 - 5 kg/s.m², examined in this study. At such low mass fluxes, studies such as [70, 62, 164, 76] all indicate that stratified flow will be the predominant flow regime expected to occur across the full range of vapour qualities, i.e. from tube inlet to outlet. For stratified flows it is, generally, accepted that the heat transfer is relatively insensitive to changes in flow rate, with the main factor affecting heat transfer being the condensate film thickness. However, it is debatable as to whether a stratified flow regime even exists at the conditions at which the measurements were acquired. A retort to the studies which indicate that stratified flow should exist is that the working fluid in each case is some type of refrigerant which, as described in Chapter 2, Section 2.4, have much higher vapour densities and, conversely, much lower vapour velocities than water vapour at comparable temperatures. Therefore, relying on a metric such as mass flux may be
misleading when determining the flow regime when vapour velocity is so widely acknowledged to be a major influence on flow morphology [81, 156, 74]. Furthermore, as shown in Section 7.2, interpretation of the data acquired in this study certainly suggests a flow regime more akin to annular, or at the very least wavy flow, than stratified. Notwithstanding the issue of extrapolation to refrigerant-based flow maps, the discrepancies between flow regimes is further compounded by a novel aspect of this study - the fact that all measurements were acquired at sub-atmospheric conditions. As described in Chapter 5, Section 5.3, low pressure operating conditions result in increased water vapour velocities. Hence, even though the range of mass fluxes investigated were quite low, the vapour velocities were, paradoxically, quite high.

Perhaps the single most compelling piece of evidence for the hypothesis that vapour velocities are a major influence in this study is given by the relative agreement between the correlations and measurements shown in figure 7.25. It can be seen that the majority of the correlations mimic the relatively large increase in condensing Nusselt number with increasing vapour Reynolds number, some to a larger extent, some less so. The correlations of Ananiev et al. [10] and Cavallini and Zecchin [11] are seen to agree best with the measurements, emphasised by average discrepancies of approximately 10% and 37%, respectively. Meanwhile, the popular correlation of Shah [9] is shown to severely under-predict the measurements, with an average difference of about 55%. Finally, Dobson’s correlation [74] is the least accurate here, over-predicting the measurements on average by 69%. Despite its continued extensive use in literature, Shah’s correlation was not in good agreement with the measurements here, perhaps a testament to the lack of experimental steam condensation data in the literature upon which the purely-empirical correlation was developed. Indeed, the database with which Shah built his correlation upon only included one data set for steam in horizontal tubes.

The relatively good agreement observed between the correlation of Ananiev et al. and the measurements was not overly surprising, given that Ananiev et al. correlated their own data from experiments of complete steam condensation in a horizontal tube. Even though they examined high temperature, high pressure steam, 189 - 310 °C and 12 - 99 bar, respectively, they encountered steam velocities very similar to those in this thesis, of the order 5 - 38 m/s. This lends further evidence to the fact that operating with relatively low flow rates at sub-atmospheric conditions is comparable, in terms of vapour velocity, to operating with high flow rates at high temperatures and pressures. To-date this has only been recognised by one other author in current literature [22]. Nevertheless, Ananiev et al. noted the effect of the vapour velocities they examined, whereby the large steam velocities in their experiments tended to “blow-off” the condensate which formed on the tube wall, resulting in a comparatively thin liquid film. Furthermore, Ananiev et al. recognised the importance of the relative densities, and their association with the respective phase velocities, by introducing a mean density term as part of their correlation.
As outlined in Chapter 5, and alluded to hitherto, the low operating pressures from which the measurements were acquired resulted in large vapour velocities in this study, ranging from approximately 5 - 40 m/s. These are assumed large, relative to velocities encountered in operational power plant condensers which, according to [169], are considered very large if greater than 10 m/s. Therefore, as described in relation to the local measurements in the previous section, it is hypothesised that these relatively large vapour velocities contribute to axial shearing at the liquid-vapour interface, leading to disturbances in the condensate-film surrounding the tube inner perimeter. Such large vapour velocities would also tend to force the condensate pool in the bottom of the tube to flow axially, towards the exit. This action prevents a stationary accumulation of condensate which would otherwise further impede heat flow. Combined, the resultant effect is an augmentation of condensation heat transfer as exhibited by the measurements in figure 7.25.

7.4.2.1 Effect of tube inclination on heat transfer

Despite the fact that the measurements presented thus far agree best with annular (shear-based) correlations, the local measurements given in Section 7.4.1, suggest that there was a tendency for accumulation of condensate towards the bottom of the tube. This is most likely an inevitable feature of complete condensing flows where, as the vapour velocity invariably decreases, the effect of the velocity to distribute the condensate evenly is negated. Hence, the condensate must accumulate towards the bottom. This contrasts with adiabatic flows, or partial condensation flows, where the velocity remains essentially constant. This accumulation was only a feature downstream in the tube, towards the exit but, nevertheless, was deemed worthy of investigation. One method of achieving this was to vary the angle of tube inclination and repeat the measurements. The results from this investigation are shown in figure 7.26, which presents the variation in condensing Nusselt number with vapour Reynolds number data from figure 7.25, for a range of inclination angles from the nominal horizontal position. Also presented in figure 7.26 are the two most favourable correlations evaluated from figure 7.25 - that of Ananiev et al.[10] and Cavallini and Zecchin[11], with both the correlation of Shah [9], and Dobson [74], discounted due to the disparity in predictions using their approaches.

An obvious feature of figure 7.26 is that there does appear to be an improvement in heat transfer for inclination angles greater than the nominal horizontal configuration ($\theta = 0^\circ$). However, this only really appears to be the case at the lower vapour Reynolds numbers, with the effect saturating at the higher Reynolds numbers examined here. In addition, it appears that the effect of increasing inclination angle diminishes with each further increase from the horizontal configuration. In this regard, heat transfer is augmented to the largest extent when the condenser inclination angle is increased from $0^\circ$ to $10^\circ$, with further enhancements at angles beyond $10^\circ$ being negligible, or at least indiscernible from the measurements. Nevertheless, the fact that the measurements depict an improvement in heat transfer from the hori-
horizontal position imply that the condensate layer is less of an inhibiting factor when the condenser tube is inclined.

For the horizontal configuration, the local measurements presented earlier in Section 7.4.1 certainly illustrated that there was an appreciable reduction in heat transfer from the top to the bottom of the tube, in downstream axial locations, with the deterioration in heat transfer getting progressively worse as the tube exit was approached. This was attributed to the condensate pool becoming relatively stagnant as axial shearing components associated with the vapour velocity diminished. Such a quiescent pool of liquid will invariably impede heat transfer. Therefore, an increase in condenser inclination, however slight, was expected to promote heat transfer in those localised regions through encouraging condensate drainage. This is illustrated in figure 7.26 where the largest improvement in condensing Nusselt number is seen to occur from the horizontal to $\theta = 10^\circ$. At the lowest vapour Reynolds number examined, this improvement was almost 25%, with an average increase of approximately 15% across the range of Reynolds numbers demonstrating the gains in terms of heat transfer which can be achieved by a relatively simple inclining of the condenser tube. A further increase in inclination from $\theta = 10^\circ$ to $\theta = 20^\circ$, could be said to improve Nusselt number, albeit by a very small amount of around 3%. Increases thereafter are negligible and, as such, the increase in condensing Nusselt number with inclination angle appears to saturate at about $\theta = 20^\circ$.

The improvement in heat transfer with inclination angle was manifested locally by increased condensing Nusselt numbers around the bottom of the tube, particularly in the downstream locations. The improvements were not seen elsewhere, such as around the top of the tube circumference, or near the inlet. Again, this offers evidence that a thicker pool of liquid does reside in the bottom of the tube towards the exit, and that this is responsible for inhibiting heat transfer for horizontal tube
configurations. As the vapour shear here is less prominent in downstream portions of the tube - compared to upstream locations near the inlet - the pool will tend to remain relatively stationary. Thus, an increase in inclination angle allows this localised pool to “run-off” and the impedance to heat flow will be negated to an extent. This tendency for the condensate to “run-off” will remain valid for the other increased inclination angles and, thus, it is responsible for the fact that no further gains are observed in the measurements in figure 7.26.

Due to practical limitations of the test facility, inclination angles greater than approximately \( \theta = 35^\circ \), were not possible. However, it is highly unlikely that any improvement in heat transfer would be observed at larger inclination angles than those examined here. It is more likely that an optimum angle exists between \( 0^\circ \) to \( 10^\circ \) as this is where the largest increase in condensing Nusselt number was seen. Once the tube was anyway inclined from the horizontal, the condensate pool was able to flow freely from the tube so it could be the case that an angle as small as \( 2^\circ \) - \( 5^\circ \) will satisfy this condition.

7.4.2.2 Effect of variable thermal boundary conditions on heat transfer

A novel feature of this study, the effect of which has not been examined on condensing flows, was the air-cooling solution and the ability to vary the air-side thermal boundary conditions, in particular. As shown in the full-scale MACC measurements in Chapter 6, the ability to vary fan speed results in the ability to control condenser temperature and pressure and has important implications on plant performance. However, the effect of varying fan speed and, consequently, the mass flow of air and air-side film coefficient (thermal boundary conditions) on condensing flows is not well understood, nor documented in the literature. Hence, it was decided to investigate the effect through measuring the condensing Nusselt number for the same range of vapour Reynolds numbers over a range of airflow Reynolds numbers evaluated in Chapter 5, Section 5.2. The outcome of one such investigation is presented in figure 7.27.

Regardless of airflow Reynolds number, the measurements presented in figure 7.27 exhibit the same overall trend as that observed in figures 7.25 and 7.26 - specifically an increase in condensing Nusselt number with increasing vapour Reynolds number. However, an additional noticeable feature of figure 7.27 is the influence of varying thermal boundary conditions, where the data appears to deviate. At large vapour Reynolds numbers, in particular, increasing the airflow results in an appreciable increase in condensing Nusselt number. This can be seen more clearly in figure 7.28, which presents the condensing Nusselt number from figure 7.27 as a function of airflow Reynolds number, for the same range of vapour Reynolds numbers.

At the largest vapour Reynolds number of 4185, it can be seen from figure 7.28 that there is an approximate 15% increase in condensing Nusselt number when the airflow Reynolds number is increased from 5498 to 13719. A similar increase in also seen at \( Re_v = 3680 \), where an approximate 17% improvement in condensing
Nusselt number is seen across the same airflow range. Although the extent of the improvement in Nusselt number diminishes at the lower vapour Reynolds numbers, it is still a feature. It is suggested that the reason for this characteristic is due to the relationship between fan speed and condenser temperature. As shown at the start of this chapter in figure 7.2, and also by the full-scale results in Chapter 6, Section 7.1, an increase in fan speed brings about a reduction in condenser temperature, for a given mass flow rate of steam/condensate. Furthermore, figure 7.6 at the start of this chapter demonstrated that the reduction in steam temperature is accompanied by an increase in vapour velocity, where it is likely that this increase in vapour velocity is ultimately responsible for the increase in condensing Nusselt numbers seen in figures 7.27 and 7.28. The increased velocities will have the same effect, albeit a smaller one, as increasing vapour Reynolds number. The velocities will impart axial shearing all around the tube circumference, in conjunction with shearing at the interface of the condensate pool in the bottom of the tube. The net effect of such actions is an increase in the convective component of condensation. Therefore, the ability to control the air-side thermal boundary conditions, through the use of variable speed fans, can serve to augment the heat transfer on the condensing-side. This is a finding not reported in literature to-date. However, it must be stressed that there is a significant penalty associated with increase fan speed, in the form of power consumption, which will more than likely offset any potential benefits on the condensing-side.

It can also be seen in figures 7.27 and 7.28 that the characteristic of increasing Nusselt number with increasing airflow is not particularly evident at the lower vapour Reynolds numbers examined. Again this is attributed to the fan speed - condenser temperature relationship. As shown in figure 7.2, and in the full-scale measurements in Chapter 6, Section 7.1, the rate of temperature reduction with increasing fan speed was much more pronounced at the higher flow rates. At the lower flow rates,
the reduction in temperature was relatively insensitive to fan speed, with the reasons for this described in detail in Chapter 6, Section 7.1. Therefore, it follows that the increase in vapour velocity, induced by increasing fan speed, will be smaller than the more dramatic increase seen at the higher flow rates. If the vapour velocity increase is not as apparent, the same increase in condensing Nusselt number will not occur.

7.4.2.3  Global condensation heat transfer summary

In sections 7.4.2.1 and 7.4.2.2, the measurements and associated discussions were each presented for the case where the parameter of interest was varied, with the remaining parameter being fixed at the nominal operating condition. In this regard, figure 7.26 presents the variation in condensing Nusselt number with vapour Reynolds number for a range of tube inclinations but for a nominal airflow rate. Figure 7.27 presents the same variation in condensing Nusselt number with vapour Reynolds number, but for a range airflow rates at the nominal horizontal tube configuration. However, it was the case during experimentation that for each tube inclination, the entire range of fan speeds were employed. Therefore, figure 7.29 is presented to summarise the findings already outlined hitherto, and also to illustrate the additional measurements from varying both tube inclination and airflow rate.

Figure 7.29 contains three individual 3-D response surfaces, each one presenting the variation in condensing Nusselt number over the entire range of airflow Reynolds numbers and condenser inclinations for a particular vapour Reynolds number. Only three response surfaces, each associated with a particular vapour Reynolds number, are presented to ensure the plot is legible and coherent. Irrespective of airflow rate and inclination angle, however, the main feature which is apparent in figure 7.29 is that the vapour Reynolds number is the main influence on condensing Nusselt number. The increase when going from the smallest Reynolds number of 2030 to the
Figure 7.29: Response surfaces illustrating the variation in condensing Nusselt number for a range of vapour Reynolds numbers, airflow Reynolds numbers, and tube inclinations

largest examined here, $Re_v = 4185$, is much greater than any improvement in Nusselt number achieved through increasing inclination angle or airflow rate. Regarding the influence of inclination angle, it can be seen that, similar to figure 7.26, this only really affects the Nusselt number at the lowest vapour Reynolds number. At $Re_v = 2030$ the increase in condensing Nusselt number with inclination angle occurs across the entire range of airflow Reynolds numbers. This is not seen to the same extent at $Re_v = 3025$, although the response surface at this point does depict the increase in condensing Nusselt number with airflow Reynolds number. The final response surface presented, for the case of $Re_v = 4185$, does not illustrate any significant response to inclination angle. However, the Nusselt number does increase with airflow Reynolds number. In all cases, there does appear to be some perturbations associated with the response surfaces. This can be attributed to the resolution of the measurements, where the spaces between the measured data points are interpolated to give the overall response. In addition to this, the fluctuating nature of the plot can be attributed to some scatter in the measured data.

7.5 HEAT EXCHANGER THERMAL RESISTANCE

In a similar manner to the full-scale MACC measurements in Chapter 6, heat exchanger thermal resistance measurements close out this chapter. The thermal resistances expressed in this section are the total, bulk thermal resistance, and its two constituent components - the air-side and condensate-side thermal resistances. An important objective of this study was to determine the relative magnitudes of the air-side and condensate-side thermal resistances for the specific test conditions
prescribed. Such measurements were already presented for the full-scale MACC in Chapter 6, albeit these measurements were acquired in an environment which wasn’t as controlled as that for the reduced-scale ACC. As such, the full-scale thermal resistance measurements were classified as air-vapour measurements, whereas the measurements presented herein can be regarded as pure-vapour. It should be noted that all measurements presented here were evaluated at the global-scale. Figure 7.30 gives an overview of the various thermal resistances, presented as a function of the airflow Reynolds number and vapour Reynolds number.

![Figure 7.30: Variation in thermal resistances with airflow Reynolds number and vapour Reynolds number](image)

It is clear from figure 7.30 that the air-side thermal resistance, evaluated and presented previously in Chapter 5, Section 5.2, is the dominant contributor to the overall thermal resistance. This is consistent with the findings from the full-scale measurements. The air-side thermal resistance is presented as a function of both airflow and vapour Reynolds number but, clearly, only varies with airflow as increasing air velocity increases the advection component of convection in the air film surrounding the tube. The net effect of this is a reduction in the thermal resistance provided by this film. It can also be seen that, due to it being the overriding resistance, the reduction in air-side thermal resistance with increasing airflow Reynolds number leads to a similar reduction in total thermal resistance. However, the total thermal resistance is offset from the air-side by the magnitude of the condensate-side resistance. This is seen to be small, relative to the air-side, but still contributes
to increase the total resistance by an appreciable factor. It is this condensate-side thermal resistance which is the primary focus here, and its contribution to the total heat exchanger thermal resistance, as there is very little in the current literature which quantifies this - particularly at conditions which pertain to operational ACCs.

In figure 7.30, the condensate-side thermal resistance is generally seen to decrease slightly with increasing airflow, but to a much larger extent with increasing vapour flow. Ultimately, this influences the total resistance in the same manner. For pure-vapour condensation, as was the case in the reduced-scale ACC, the resistance on the condensate-side is primarily comprised of the condensate film and pool which form in the tube as the vapour condenses. At the lowest vapour Reynolds number examined of $Re_v = 2030$, the condensate-side resistance can be seen to be a relatively large contributor to the overall resistance, ranging from approximately as large as 25% to as low as 20%, with an average contribution of about 24%. However, this contribution reduces quite drastically with increasing vapour flow rate. A relatively modest increase in vapour Reynolds number to 2690 reduces the contribution of the condensate-side resistance to an approximate 18% average. This average contribution continues to decrease to about 16%, 13%, and 11% for $Re_v = 3025$, 3680 & 4185, respectively. This contribution is explicitly quantified in figure 7.31.

![Figure 7.31: Variation in thermal resistance ratio as a function of vapour Reynolds number](image)

(a) For a range of airflow Reynolds numbers    (b) Average over the of airflow Reynolds numbers

Figure 7.31 clearly illustrates the reduction in the thermal resistance ratio, originating from the reduction in absolute condensate-side thermal resistance, as vapour Reynolds number increases. This was a feature regardless of the airflow Reynolds number examined. However, as shown by figure 7.31a, the magnitude of the thermal resistance ratio does vary depending on airflow Reynolds number, where at high air flows, the ratio is larger due to the more dramatic reduction in total thermal resistance with increasing air flow. Figure 7.31b summarises the average thermal resistance ratio across the range of airflow Reynolds numbers. It could be said that the average ratio of ~ 24% is quite large at $Re_v = 2030$. However, the measurements presented thus far may be thought of as the worst-case scenario insofar that they were all acquired at a horizontal tube configuration. As shown previously in Section 7.4, the
thermal performance of the reduced-scale ACC improved when it was inclined to any
generic angle from the horizontal. This same performance enhancement could also
be expected to be manifested in improved thermal resistance results. Therefore, the
variation in absolute condensate-side thermal resistance as a function of both airflow
Reynolds number and tube inclination angle is presented in figure 7.32, for the range
of vapour Reynolds numbers examined.

It is clear from figure 7.32 that the absolute worst condensate-side thermal resist-
ances occur when the tube is horizontal (inclined at 0°). This is particularly prevalent
at \( Re_v = 2030 \) and \( Re_v = 2690 \), which had an average contribution to the overall
resistance of about 22% and 18%, respectively, in figure 7.31. As can be seen in fig-
ure 7.32, the magnitude of the condensate-side resistance is much smaller at those
vapour flow rates when the tube is inclined. In relation to the local condensation
heat transfer measurements, this reduction in thermal resistance is attributed to the
promotion of condensate drainage from the bottom end of the tube, which would oth-
ervise remain relatively stationary. It can be seen that this phenomenon is not really
in evidence for the higher vapour Reynolds numbers, as it is assumed here that the
vapour shear is sufficient to induce drainage regardless of angle. In fact, figure 7.32
clearly illustrates that vapour Reynolds number is the dominant parameter in rela-
tion to the condensate-side thermal resistance. The reduction in thermal resistance
with increasing vapour Reynolds number is much greater than that brought about by
increasing airflow Reynolds number or increasing condenser inclination angle. Similar to that described for the condensation heat transfer measurements, the larger vapour velocities associated with an increase in vapour Reynolds number tend to augment heat transfer and, thereby, reduce thermal resistance by thinning the condensate film around the inside tube perimeter. This is in conjunction with the vapour velocity disturbing and deforming the condensate pool around the bottom of the tube. Hence, heat transfer is augmented all around the tube perimeter, with the thermal resistance undergoing a corresponding decrease as a result. Ultimately, these effects need to be quantified in context of the overall thermal resistance to determine the benefits of reducing condensate-side thermal resistance on condenser performance. Therefore, in a similar manner to figure 7.31, the contribution of the condensate-side thermal resistance is expressed as a ratio of the overall, total resistance of the reduced-scale ACC in figure 7.33, below.

![Figure 7.33: Variation in thermal resistance ratio with both airflow Reynolds number and tube inclination for a range of vapour Reynolds numbers](image)

The results depicted in figure 7.33 again illustrate that the dominant relationship of note is that of decreasing condensate-side thermal resistance and, hence, thermal resistance ratio with increasing vapour Reynolds number. The ratio is seen to decrease from local highs of ~ 26% at $Re_v = 2030$ to lows of ~ 10% at $Re_v = 4185$. It is important to note that, in contrast to the thermal resistance measurements from the full-scale MACC, the thermal resistance presented in figure 7.33 are purely as a result of pure-vapour condensation, with no air effects. From a design perspective, it
can be concluded that high flow rates would be advantageous due to the reduction in condensate-side thermal resistance observed herein. However, it must be stressed that the sub-atmospheric pressure conditions in this study certainly contributed to the magnitude of thermal resistance measured, where the large vapour velocities most likely resulted in the reduced values.

7.6 SUMMARY

This chapter presented the findings from the experimental investigation and approach described in Chapter 5, whereby the fluidic and thermal characteristics of condensing flows of steam at typical Rankine cycle operating conditions were examined over a range of parameters related to full-scale ACCs. This consisted of two-phase flow regime mapping, flow regime identification through a novel, non-invasive measurement technique, pressure drop measurements, local and global condensing heat transfer measurements, and heat exchanger thermal resistance measurements. It was found that there was an element of ambiguity associated with flow regime mapping, due in part to the subjective nature of classifying flow regimes and also due to uncertainty surrounding the applicability of the maps to the conditions investigated in this study. Through the spatial distribution of local temperature difference measurements, along the tube axial length and around the inside tube perimeter, the flow regime was inferred. This was seen to be annular nearest the tube inlet, with the flow deviating to stratified-wavy as the exit was approached. The manner of this transition varied depending on vapour Reynolds number.

Pressure drop measurements of the condensing flows illustrated the same overall characteristics as the measurements from the full-scale MACC, where the measured losses are quite small due to momentum recovery negating the frictional losses in the flow. However, as shown in Chapter 6, this is not always the case. Hence, a reliable method for calculating the frictional component of two-phase pressure drop, in particular, is necessary. In Chapter 6, it was found that the method of Lockhart and Martinelli compared best with the frictional data. Here the agreement between the Lockhart and Martinelli model was improved by incorporating a new constant, derived from regression analysis on the data, into the expression to calculate the two-phase multiplier. Ultimately, this improved the agreement from $\pm 18\%$ to $\pm 9\%$.

Local condensation heat transfer was seen to deteriorate in the axial direction, simply as a result of vapour being progressively condensed in that direction. Across all flow rates examined, local Nusselt numbers were seen to be largest nearest the tube inlet, progressively decreasing in the downstream direction and, ultimately, appearing to converge as the tube exit was approached. The manner of this reduction in heat transfer varied with flow rate where, in general, larger Nusselt numbers were associated with higher flow rates for any given axial location. Furthermore, the condensation heat transfer was found to decrease around the inner tube circumference, from the top to the bottom. This was attributed to the presence of a condensate pool.
residing in the bottom portion of the tube, which inevitably inhibited heat transfer. Nearest the inlet, this was not an issue however, but at downstream locations there was a definite and well-defined decrease in Nusselt number as the bottom of the tube was approached.

Global condensation heat transfer measurements summarised the local characteristics. The main, dominant relationship was a large increase in global Nusselt number with increasing vapour Reynolds number. This was compared to a number of established correlations, with the method of Ananiev et al. exhibiting the best agreement of ± 10%. The effect of tube inclination and variable air-side boundary conditions on heat transfer were also examined. It was found that inclining the tube from a horizontal configuration can bring about an increase in condensation heat transfer of approximately 25%, with the effect more apparent at lower vapour Reynolds numbers. Conversely, the effect of variable air-side boundary conditions was only manifested at the largest vapour Reynolds numbers investigated, where increases in condensation Nusselt number of approximately 17% were noted by increasing airflow Reynolds number. The same measurements were used to determine the condensate-side thermal resistance, the contribution of which to the total thermal resistance was shown to vary from as high as ~ 19% to a low of ~ 11% across the range of vapour Reynolds numbers examined.
To frame the measurements presented in this thesis in a more practical context, a thermodynamic model was developed - into which the thermal resistance measurements can be integrated. The output from this model are metrics, such as net plant output and efficiency loss, which offer a more insightful representation of condenser performance from an end-user perspective (i.e. a power plant operator). The thermodynamic model served the additional purpose of investigating the hypothesis that current modelling methodologies are fundamentally flawed by neglecting to account for the condensate-side thermal resistance. Results from current modelling approaches are, therefore, compared with the thermodynamic performance derived from experimental measurements to determine the extent, if any, of the discrepancies. To this end, the thermodynamic performance from an air-side only approach is presented first, followed by a comparison between the two approaches. All details related to the modelling approach are outlined in Chapter 3, Section 3.3.

8.1 AIR-SIDE APPROACH

It appears that current approaches to thermodynamic modelling of air-cooled Rankine cycle thermoelectric plants disregard the contribution of the condensate-side thermal resistance in the ACC. Many studies in the literature [51, 48, 46, 29] dismiss the condensate-side resistance, assuming it to have no real implications on the final outputs from the model. The results from this approach are presented here, where all calculations are based solely on air-side theory. This air-side theory was presented in Chapter 3, Section 3.1, and is incorporated into the model in conjunction with the ∼ 50 MW steam turbine characteristics given in figure 3.5b to, ultimately, give the results in figure 8.1. These results are presented to illustrate the thermodynamic performance advantages of the MACC. In terms of system-level performance, the key variables identified in relation to the MACC are the fan rotational speed, condenser size (equivalent to number of MACC modules), and dry bulb ambient air temperature. For the sample results presented in figure 8.1, two very different condenser sizes, of 260 modules and 980 modules, were examined across a range of representative ambient temperatures. The lower end of the ambient temperature range (10 °C) is representative of plant sites in nominally cold locations, whilst the upper end of 40 °C is typical of desert locations. A module size of 2 m x 2 m was adhered to in the thermodynamic model as it was consistent with the prototype module size characterised in Chapter 6.
Figure 8.1a presents the variation in condenser temperature which was calculated as air-side heat transfer coefficient (represented here as fan rotational speed) was varied. Essentially, these results illustrate the same qualitative trends and characteristics as the measurements from the full-scale prototype MACCs in Chapter 6, Section 7.1. The reasons for these trends and characteristics were explained in detail in that chapter. However, there are two additional features of note in figure 8.1a. Firstly, it can be seen that the effect of increasing ambient air temperature is to increase the condenser temperature proportionately. The implications of this on plant output are demonstrated in figures 8.1b and 8.1d. Additionally, the effect of increasing condenser size is clearly shown in figure 8.1a where, for a given fan speed, the temperature is consistently lower for a larger number of modules. This is simply attributed to the larger heat transfer area which, for a given fan speed, ultimately promotes lower temperatures in the condenser.

The implications of varying condenser conditions, in the manner seen in figure 8.1a, are clearly manifested in the gross output results in figure 8.1b. The variation
in gross output with fan speed depicted is a direct result of the variation in condenser temperature with fan speed. As described in relation to the MACC concept in Chapter 1, the ability to control and manipulate condenser temperature and pressure, through variable fan speed, allows for the steam turbine output be maximised. It can be clearly seen in figure 8.1b, particularly for the smaller condenser size of 260 modules, that increasing fan speed can dramatically increase steam turbine output by lowering the thermodynamic limit of the heat sink. The case for the larger number of modules is less sensitive to increasing fan speed as the temperatures are already quite low for such a large condenser. As increased ambient temperatures were shown to result in higher condenser temperatures, the gross output is, accordingly, lower for increased ambient values. Nevertheless, regardless of ambient temperature or condenser size, increasing fan speed serves to extract maximum power from the steam turbine, where infinitely increasing fan speed will result in progressively more gross power. However, such an approach overlooks the power consumed by the cooling fans which, as shown by figure 8.1c, can be quite significant at high rotational speeds. This is a key consideration in the MACC concept.

In accordance with the fan laws, the fan power consumption increases in a cubic manner with the rotational speed. Hence, the excessive consumption rates at high fan speeds. The power consumed by the fans is also greater for a larger condenser, simply due to the fact that there are more fans associated with the increased number of modules. The ultimate implication of fan power consumption on plant performance is exhibited by the net plant output ($P_{gr} - P_{ft}$) results in figure 8.1d. In essence, these results illustrate the principle technical benefit and defining characteristic of the MACC design - the ability to continually vary fan speed to achieve an optimum operating condition. The optimum operating condition is the point at which maximum net plant output occurs, at a given ambient temperature or condenser size, and is highlighted by the local maxima, or “peaks”, of the curves in figure 8.1d. The abscissa of the optimum operating point is the optimum fan rotational speed. Operating at fan speeds below this point has the effect of decreasing gross output by a greater magnitude than the decrease in fan power consumption. However, operating at fan speeds above the optimum is detrimental, as the increase in fan power consumption is greater than the corresponding increase in gross output. Hence, fan power offsets the gross output, and actually serves to reduce the net plant output. Therefore, it is clear that infinitely increasing fan speed in an effort to improve turbine output is not an effective, or advisable, strategy.

Figure 8.1d also illustrates that increasing condenser size leads to larger net plant output. This is a result of increased heat exchanger surface area which reduces the required air-side heat transfer coefficient, culminating in a reduction in fan power consumption. Accordingly, the optimum operating point is attained at lower fan speeds compared to those for the smaller condenser. However, the increased capital cost is a factor that must be considered with an increased number of modules. Land availability is another issue. Therefore, it is obviously not a feasible or realistic op-
tion to infinitely increase condenser size in an attempt to extract as much power as possible. Indeed, optimizing condenser size has been addressed in two separate investigations relating to this study by Poullikkas et al. [129, 130]. As with figures 8.1a and 8.1b, increasing ambient temperature has an adverse effect on net plant output. Irrespective of condenser size, increasing the ambient temperature has the effect of reducing net plant output. Increasing ambient temperature also results in an increase in the fan speed at which net plant output is maximised. Therefore, in an operational scenario, as ambient temperature changes, fan speed should be controlled to continuously maintain maximum plant output. This is a fundamental strategical improvement over current ACC operating schemes, and showcases the primary benefit of a MACC cooling system in a thermoelectric power plant. However, the underlying issue threatening to undermine the modelling is that all results presented in figure 8.1 are predicated on the assumption that the condensate-side resistance is minimal and, therefore, does not contribute in an appreciable manner. This issue is explicitly addressed next.

8.2 EXPERIMENTALLY-DERIVED APPROACH

In accordance with the quasi-modelling methodology described in Chapter 3, Section 3.3, temperature and thermal resistance measurements, acquired from a range of full-scale prototype MACC 2 m x 2 m modules, can be integrated into the thermodynamic model to predict the plant performance and, ultimately, compare and contrast results with the theoretical air-side approach. The experimentally-derived approach differs insofar that the condenser performance, in terms of temperature, pressure, and thermal resistance, was measured instead of relying on air-side theory to predict the same quantities. The input measurements of condenser temperature are presented in figure 8.2, accompanied by the corresponding calculated thermodynamic quantities of gross power and net power. In all cases, the results from the air-side theoretical approach are also presented.

The temperature measurements presented in figure 8.2a are the same measurements which were previously presented in figure 6.5. In that context, they were presented as part of the operational characteristics for the four-row circular tube MACC. Here they are presented as the preface to the thermodynamic model, where the variation in temperature with fan speed shown yields the experimentally-derived gross turbine output in figure 8.2b. Both figure 8.1a and 8.1b depict the same trends seen in their respective sub-figures in figure 8.1. However, of concern here is the difference in magnitude between the experimental approach and the air-side theory approach. This originates with the actual temperature measurements, which are shown to be consistently higher than the corresponding predicted values. Notwithstanding the trends with fan speed, which were explained in Chapter 6, Section 7.1, the reason the predicted temperature is consistently lower than the measurements is due to the fact that air-side theory just predicts the heat exchanger surface temperat-
8.2 Experimentally-Derived Approach

Figure 8.2: Variation in measured and predicted condenser performance, and related thermodynamic quantities, as function of fan speed over a range condenser sizes corresponding to the experimental steam flow rates investigated.

Accordingly, this surface will always be lower than the actual steam temperature inside the tubes - which is the quantity given by the measurements. This difference is seen across all steam/condensate flow rates investigated, but is noticeably largest at the smallest flow rate of 0.06 kg/s, where the average difference across the range of fan speeds is approximately 16%. It is quite clear from this that air-side theory alone is not capable of accurately predicting the condensate-side performance of an ACC. In reality, such an approach is fundamentally flawed as, in an operational scenario, an ACC will always be subjected to adverse condensate-side conditions such as air-leakage, subcooling, fouling, etc. None of the effects arising from such phenomena are accounted for in an air-side analysis which, whilst it may satisfactorily model ACC performance for an idealised scenario, certainly fails when additional variables are considered. Also, it is important to stress that the condensate-side phenomena encountered during experimentation are largely unavoidable and, as such, are simply
undesirable features of all commercially operating ACCs. In this regard, the temperature measurements presented in figure 8.2a are thoroughly representative of ACC characteristics expected in an operational power plant.

Each steam/condensate flow rate which was experimentally investigated on the single prototype module translated to a specific number of MACC modules in a power plant scenario, where the lowest flow rate examined equates the largest number of modules, and vice-versa. The gross power in figure 8.2b is expressed as a function of this condenser size and, similar to the characteristics seen in figure 8.1b, the gross output is seen to increase with fan speed. However, the manner of this increase is quite different whether the experimentally-derived, or air-side, approach is considered. It is especially noticeable that the gross output as derived from the experimental measurements is consistently lower than the same corresponding output from air-side theory. This is a direct consequence of the disparity in condenser performance shown in figure 8.2a, ultimately highlighting the importance of, firstly, the condenser operating conditions on plant performance and, secondly, the ability to reliably predict these conditions. However, the ultimate effect of this is manifested in the net plant output as shown in figure 8.2c.

The implications of adopting air-side theory in an attempt to predict condenser conditions, and the resultant under-prediction of those conditions, is shown to yield drastically different results than using actual measurements to predict plant performance. As the gross output predicted from air-side theory over-predicted the actual, expected output based on condenser measurements, it follows that the net output is also over-predicted. Generally, this over-prediction is largest for the greatest number of modules, decreasing as condenser size reduces - as summarised in figure 8.3.

Figure 8.3: Disparity between the net plant output, as given by the air-side approach and the experimental approach

Figure 8.3 clearly illustrates the disparity between the two approaches of modelling and, hence, predicting thermodynamic performance, with the disparity eman-
ating from the original difference between condenser performance shown in figure 8.2a. From this root cause, the differences propagate through the model, resulting in discrepancies in net output as large as 1.5% in some extreme cases. Taking the average across the range of fan speeds, however, the difference is approximately 1% for the case of 650 modules, decreasing to 0.64% for 500 modules, and 0.43% for 400 modules. This is in-line with the deviation between the measured temperatures and predicted equivalent in figure 8.2a, where the difference was largest at the lowest flow rate (largest amount of modules) and was seen to decrease with increasing flow rate (smaller number of modules). Nevertheless, differences as low as 0.43% could still be an issue if the model was extended to include techno-economic aspects, where the general over-prediction by air-side theory could result in misleading financial estimates.

A further issue with the air-side approach is that the optimum operating points evaluated do not correlate with those determined from the experimental measurements - as can be clearly seen in figure 8.2c. The optimum fan speeds from the air-side approach in figure 8.2c are 400 rpm, 500 rpm, and 620 rpm for 650, 500, and 400 modules, respectively. However, for the same condenser sizes, the optimum fan speeds based on the experimental approach were found to be 470 rpm, 580 rpm, and 690 rpm. This has implications on the fan algorithm, which will govern the operation of the MACC. If this is based on theory that is limited to the air-side only, then it is highly likely that the actual optimum operating points will not be attained during plant operation. Therefore, the maximum net plant output will not be achieved. Ultimately, this could undermine the principal advantage of the MACC and, hence, solely focusing on air-side performance, as appears to be the case in the literature relating to ACCs, could be a detrimental oversight. Again, it must be emphasised that the condensate-side measurements from the full-scale prototype MACC modules were acquired under typical Rankine cycle operating conditions of temperature, pressure, steam/condensate mass flow rate, and air mass flow rate. Adverse condensate-side phenomena, such as air leakage and subcooling, were encountered but, through correspondence with ACC manufacturers and users, it appears these are conditions to which every operational ACC is susceptible to. As such, the measurements inputted into the model and, ultimately, the experimentally-derived approach is probably more reflective of the thermodynamic performance of a plant with the MACC system installed.

8.2.1 Case study - Gemasolar

All results presented thus far were calculated from the nominal thermodynamic model, which is defined by the nominal ~ 50 MW steam turbine characteristics shown in figure 3.5b in Chapter 3. However, as shown in the accompanying figure 3.5a, the characteristics of another ~ 20 MW steam turbine were procured, which could also be integrated into the model. This turbine is currently operational in a
central tower type CSP plant - “Gemasolar” - in southern Spain. Presently, this plant is water-cooled. However, to determine the outcome if the plant was air-cooled, the thermodynamic performance of Gemasolar was modelled with the MACC system installed. This model differs from the one presented hitherto in that the steam turbine characteristics incorporated into the model are different, coupled with the fact that the condenser modelled in this case was the single row rectangular tube condenser. The experimental measurements from this design, which were integrated into the model, were presented in Chapter 6, Section 7.1. An important novel feature of this model, however, was a condenser pressure constraint of 0.06 bar a (6000 Pa). This was a specific request from the plant operator, who identified the practical limitations of the condenser air ejection system which could not operate at pressures lower than 0.06 bar a. Such practicalities of ACC performance are rarely, if ever, reported in literature and, as such, it was deemed worthy of investigation through the thermodynamic model. The results from this model are henceforth presented, starting with figure 8.4, which presents the condenser temperature based on experimental measurements, extrapolated to a range of ambient temperatures for two representative condenser sizes, for the unconstrained and constrained cases.

Figure 8.4: Variation in condenser temperature as a function of fan speed for 200 and 300 modules, across a range of ambient temperatures

The characteristics discussed previously in relation to the temperature measurements are also evident in figure 8.4. Namely, these are a reduction in temperature with fan speed, increase in temperature with ambient air temperature, and reduction in temperature with increasing condenser size (number of modules). However, these characteristics are not as prominent in figure 8.4b, which presents the same results as figure 8.4a but for the fact that the 0.06 bar a constraint is imposed. This 0.06 bar a constraint is equivalent to a temperature of approximately 36.2 °C (309 K). As can be seen in figure 8.4b, this constraint is seen to inhibit condenser performance at low ambient temperatures, where the benefit of increasing fan speed is negated. For 300 modules at an ambient temperature of 25 °C, for example, increasing fan speed from approximately 230 rpm to 345 rpm brings about a reduction in temperature which
approaches the 309 K restriction imposed. Hence, further reductions in temperature beyond this fan speed are not possible. At the worst cases (200 and 300 modules at 10 °C), the temperature appears as a flat line across the fan speed range, effectively rendering the advantages of variable speed fans redundant. As shown in figures 8.1 and 8.2, the condenser conditions have a large bearing on the overall plant performance, with the imposed constraint resulting in net plant output characteristics as shown in figure 8.5.

![Figure 8.5](image)

(a) Unconstrained case  (b) Constrained (0.06 bar a) case

Figure 8.5: Variation in net plant output as a function of fan speed for 200 and 300 modules, across a range of ambient temperatures

As can be seen by comparing figures 8.5a and 8.5b, the net plant output is not affected by the 0.06 bar a constraint at high ambient temperatures. However, at low ambient temperatures it can be seen to curb plant output. This is seen for both 200 and 300 modules at 10 °C, where the local maxima in the same curves presented in figure 8.5a are not attained. Furthermore, not only is there a deterioration in the magnitude of plant output (in some cases by ~ 0.7 MW), but the optimum operating point is altered. These issues are explicitly quantified in figure 8.6, by presenting the constrained and unconstrained optimum net plant output, and corresponding optimum fan speed, as a function of a range of representative ambient temperatures for both condenser sizes considered.

It is especially evident from figure 8.6a that condenser performance suffers from the imposed constraint at low ambient temperatures in particular. For 300 modules, the difference in optimum net plant output - as determined from the local maxima of the curves in figure 8.5 - between the unconstrained and constrained cases is approximately 0.65 MW, 0.53 MW, and 0.36 MW at 5 °C, 10 °C, and 15 °C, respectively. Although the losses are largely towards the lower ambient temperatures, which CSP plants will rarely be exposed to, conventional thermoelectric power plants could suffer similar performance losses - which is a concern that needs to be addressed. Figure 8.6b presents the variation in optimum fan speed with ambient temperature - essentially the fan control algorithm for the Gemasolar plant with the MACC system installed. Again, this illustrates the significant discrepancies which arise due to the
imposed constraint at low ambient temperatures. For 200 modules, the difference in fan speed varies from approximately 62% to 20% at 5 °C and 20 °C, respectively, before reaching parity at the higher ambient temperatures. The same approximate differences exist for the 300 module case. This suggests that neglecting to consider the influence of ACC auxiliary components could lead to an erroneous fan algorithm, where the response of the condenser fans to changes in ambient temperature will not result in the maximum net plant output expected.

Ultimately, regardless of application, figure 8.6 highlights the deterioration in ACC performance due, exclusively, to the incapacity of the air ejector system. This finding is somewhat analogous to basing the thermodynamic model solely on air-side theory, where misleading results could occur if basing the model on experimentally-derived measurements without considering all the constituent components. However, the caveat to that implication is that it is not know if all air ejector systems are sus-
ceptible to a 0.06 bar limitation. Indeed, this limitation was unbeknownst to the author prior to correspondence with the Gemasolar plant operators. If it is indeed the case, there is certainly scope for improvement in current air-ejector designs to alleviate this limitation, in an effort to realise maximum condenser performance and, ultimately, optimum plant output.
This thesis presented an experimental investigation into the thermal and fluidic characteristics of steam condensation in ACCs for Rankine cycle-based thermoelectric power plants. An experimental programme consisting of local, global, and full-scale measurements on circular tube condensers established the predominant two-phase flow regimes, pressure losses, and heat transfer of condensing flows of steam, at conditions typical of those encountered in application. The outcomes and findings from this thesis are summarised and concluded here. The conclusions are listed in a manner consistent with the main body of the thesis, in that the conclusions from the full-scale MACC are presented first, followed by those from the reduced-scale ACC. The conclusions are related to the initial research objectives presented in Chapter 1. Closing out the chapter are recommendations for future work in the area of ACCs. These recommendations list a number of research topics which were deemed to be outside the scope of this study and which, if completed, should address any shortcomings still existing in the current literature.

9.1 Conclusions

Whilst not conclusions specific to this work, there were a number of phenomena alluded to in this thesis which merit inclusion here as general closing remarks regarding ACCs. Firstly, it was established that air ingress is an undesirable, but unavoidable, feature of full-scale condensers operating at sub-atmospheric pressure conditions. In this regard, the air leakage which occurred during experimentation on the MACC was not just specific to this isolated case but is, in fact, common to all Rankine cycle-based condensers. As such, the defining heat transfer process in operational ACCs is air-vapour condensation, opposed to pure-vapour condensation. In addition, it was seen that multi-row condenser designs suffer from the row effect, whereby air becomes progressively warmer as it flows downstream through the tube bundle. From a condensate-side perspective, the net result of the row effect is vapour backflow, where non-condensed steam enters a common exit manifold and flows back-up certain tubes. Combined with the presence of air in the condenser, this leads to the formation of air pockets, or non-condensable zones. These zones reduce the effective heat transfer area of the condenser and can be detrimental to performance. It was seen that these adverse conditions can be eliminated through installing a dephlegmator (reflux condenser), which allows the desired condenser conditions to be achieved, ultimately providing the platform to acquire measurements from which the following conclusions were drawn from.
9.1.1 Full-scale MACC

- The dephlegmator vacuum pump was found to have a significant influence on condenser operating conditions, in that the capacity of the vacuum pump affected the absolute pressure which could be achieved in the condenser. It was seen that increasing vacuum pump capacity results in a smaller partial pressure of air in the condenser, ultimately leading to an absolute pressure closer to the saturation pressure.

- The qualitative and quantitative relationships between condenser temperature and pressure with fan rotational speed (air-side heat transfer coefficient) and steam flow rate were established through measurements, which encompassed the full range of relevant parameters, on a number of full-scale prototype MACC modules. The relationships are defined by a reduction in temperature and pressure with increasing fan speed, for a given mass flow rate of steam, and vice-versa. As fan speed increases, it was seen that the condenser temperature asymptotically approaches the ambient air temperature, with the response to variations in fan speed being more pronounced at higher steam flow rates. In addition, for a given set fan speed, an increase in temperature and pressure occurs as steam flow rate is increased.

- The two-phase condensing pressure drop measured through the various tube bundles in this study was found to be relatively minimal. The measured values varied from approximately 120 Pa - 280 Pa over a vapour Reynolds number range of 1890 - 5150. Relative to the corresponding absolute pressures, these losses never exceeded a pressure loss to absolute pressure ratio of 6% - satisfying the criterion of 10 - 15% recommended by Palen et al. [20].

  - Measured condensing pressure losses were small due to the effect of momentum recovery in the flow, which offset the incurred frictional losses. It was found that the momentum recovery and frictional pressure drop are comparable in magnitude, although qualitatively and quantitatively inverted. This appears to be a phenomenon unique to condensing flows, as the major of two-phase pressure drop studies in the literature are based on evaporating refrigerants, which report of momentum pressure losses.

  - The frictional pressure losses, inferred in this study, compared favourably with the predictions of the Lockhart and Martinelli [4] model, with a mean deviation of 18% emphasising this. Other well-cited correlations and models such as those of Friedel [5], Grönnerud [6], and Müller-Steinhagen and Heck [7] were also compared with the results, but were found to be less accurate.

  - Using the Lockhart and Martinelli model to predict the frictional pressure drop, an investigation into the ratio of frictional pressure drop to
momentum recovery, at conditions outside of those which were investigated experimentally in this study, was carried-out. In essence, this was to determine if momentum recovery in a condensing flow always offsets the frictional losses. This was found to be false where, in extremity, the frictional losses can be 12 times as large as the magnitude of momentum recovery. However, this was for an extreme case of $D = 0.01$ m and $L_t = 10$ m. It was shown that a threshold point around $D = 0.02$ m and $L_t = 4$ m exists, after which the frictional losses tend to exceed the momentum recovery. In this regard, it was somewhat coincidental that the pressure drop measurements in this study were acquired at conditions which allowed the trait of momentum recovery offsetting the frictional losses to be exhibited.

- ACC thermal resistances were quantified over a range of fan speeds and steam flow rates. It was shown that the total thermal resistance, $(R_{th})_a + (R_{th})_s$, decreased with increasing fan speed, as expected, but also decreased with increasing steam flow rate - due to the reduction in the magnitude of steam-side thermal resistance. This steam-side thermal resistance was seen to increase with fan speed, suggesting that the increased flow rates of air leaking into the condenser inhibited heat flow. The contribution of the steam-side resistance, to the total resistance, was found to vary from a worst-case of 33% to a low of 5%. The average contribution across the fan speed range was quantified as 26% at $Re_v = 2280$ to 13% at $Re_v = 4420$.

- The effect of system-level variables, such as fan orientation and condenser inclination, had a negligible impact on the steam-side thermal resistances measured. At the global-scale, the difference between the steam-side thermal resistance in a forced-draft flow and an induced-draft flow varied from approximately 9 - 18%, with no definite trends to account for these differences. Similarly, the difference in thermal resistance from a condenser inclination angle of $\sim 40^\circ$ to $\sim 70^\circ$ was minimal. However, adverse performance was noted for the condenser in a horizontal position but, once inclined at some generic angle to the horizontal, the issues dissipated.

9.1.2 Reduced-scale ACC

- In the absence of accurate empirical correlations (due to the ACC consisting of a single, individual tube) or reliable analytical solutions (due to the atypical channels formed by the annular fins around the circular tube), the air-side boundary conditions were established through specific experimental methods. The airflow Reynolds number, as provided by the bank of axial fans, varied from 5498 - 13719, resulting in an overall air-side heat transfer coefficient variation of approximately 26 - 35 W/m²K. Expressed non-dimensionally as Nusselt
number, this variation equates to 33 - 45. These boundary conditions were representative of those established on the full-scale MACC.

- Through a more isolated and controlled experimental environment, pure-vapour condensation - opposed to air-vapour condensation at the full-scale - was the defining mechanism in the reduced-scale ACC. From a thermal perspective, this meant that the measurements explicitly quantified the condensation heat transfer only, with no air effects to consider.

- The tube was instrumented and set-up in a manner which allowed for local, high resolution temperature measurements, not seen in current literature, to be acquired. The local regions of interest were the tube axial length, and tube circumference, over which heat transfer measurements were acquired through the instrumentation arrangement.

- A two-phase flow regime identification study was carried-out by plotting experimental data, which encompassed the full range of operating conditions, on some of the most widely-cited flow regime maps from the literature. The map of Taitel and Dukler [2] suggested that stratified-wavy flow was the predominant flow morphology during experimentation. However, the map of El Hajal et al. [3] illustrated a fully-stratified flow regime, from inlet to outlet of the tube. It was found that the applicability of either map to the experimental conditions examined in this study has not been verified in literature.

- Local temperature difference distribution measurements, along the axial length of the tube and around the inside circumference, were analysed to determine the spatial evolution of the condensing flow. It was generally seen that, nearest the tube inlet, the temperature difference profile around the circumference was quite uniform - suggesting that the condensate film surrounding the inner wall was quite uniform, and that annular flow was present at this point. However, for all other downstream axial locations there was an increase in the temperature difference around the tube circumference, from the top to the bottom of the tube. This distribution implies that condensate was accumulating in the bottom of the tube, offering evidence that the flow was deviating from an annular flow to a stratified flow as it progressed downstream.

  - The extent to which the temperature difference profile increased from top to bottom varied with axial location and vapour flow rate. For the lowest flow rate examined of $Re_v = 2030$, there was a clear and discernible increase in the magnitude of temperature difference from the top to the bottom, for all axial locations downstream of the immediate inlet. Inevitably, this temperature difference became progressively larger towards the exit. However, as flow rate was increased, the increase in temperature difference around the circumference was not as obvious, and tended to occur further downstream in the tube. This indicated a delay in the transition to
stratified flow, with the larger vapour velocities ensuring an annular-type regime for a greater portion of the tube length.

- Analysis of the temperature difference history at each circumferential measurement site, for the range of axial locations, suggested a wavy element to the flow. Temperature difference measurements around the top and bottom of the tube were relatively steady for a given acquisition window. However, at circumferential angles of $\varphi \approx 135^\circ - 160^\circ$, there was considerable scatter in the temperature measurements and, hence, the temperature differences. The scatter was attributed to the presence of waves, intermittently forming and receding in the vicinity of the inner wall measurement site. This suggested a stratified-wavy flow topology was present.

- Condensation pressure drop measurements exhibited the same overall characteristics as those acquired on the full-scale. Again the method of Lockhart and Martinelli [4] was shown to agree best with the experimentally-derived frictional data, with a mean deviation of $\pm 18\%$. However, through regression analysis on the experimental data, agreement was improved to $\pm 9\%$ by increasing the Chisholm [168] constant $C$ to 17. Ultimately, an improved correlation with this constant incorporated was presented for calculating the two-phase multiplier and, hence, the frictional pressure drop for condensing flows of steam in round horizontal tubes.

- Local condensation heat transfer measurements exhibited multi-dimensional characteristics. Firstly, it was found that the condensation heat transfer deteriorated in the axial direction. The largest condensing Nusselt numbers were consistently determined nearest the tube inlet, with the smallest being found closest to the exit. This is an inherent feature of complete condensation, and occurred across all steam flow rates investigated. However, the qualitative and quantitative variation along the tube length varied with vapour flow rate. For a given axial location, larger Nusselt numbers tended to occur at higher flow rates, and vice-versa. Furthermore, the difference in the slopes of the data were an indication that the majority of the steam condensed in upstream locations in the tube at the low vapour flow rates, whilst the higher flow rates required more of the tube for full condensation.

- The multi-dimensional nature was evidenced by the variation in condensation heat transfer around the inner tube circumference where, generally, there was a reduction in Nusselt number from the top to the bottom of the tube. This reduction was attributed to the condensate layer which is known to impede heat transfer. This was a characteristic common to all axial locations, except for the site closest to the inlet - at which point no substantial deterioration in heat transfer occurred around the circumference. Generally, the decrease in Nusselt number around the circumference
9.1 Conclusions

got progressively worse as the tube exit was approached - an indication that the condensate was accumulating and, thus, offering a progressively larger thermal resistance towards the exit.

- Contour plots developed from a local measurement grid presented the spatial distribution of condensing Nusselt number around the inner tube circumference along the axial length. These plots summarised the local findings - largest Nusselt numbers at the inlet, which are also approximately uniform around the circumference, a deterioration of Nusselt number in the axial direction for any given circumferential position, and the reduction of Nusselt number in the axial direction is most prominent at the bottom of the tube.

- Global condensation measurements were presented as a function of vapour Reynolds number, for a range of tube inclinations and airflow Reynolds numbers (variable air-side thermal boundary conditions). It was seen that, in general, the condensing Nusselt number was a strong function of vapour Reynolds number. This is emphasised by an increase in Nusselt number of approximately 90 - 200, with an increase in Reynolds number from approximately 2000 to 4200. As shown by the slope of the measured data in figure 7.25, this was seen to be a large increase in Nusselt number over a relatively modest vapour Reynolds number range. This was attributed to the fact that the measurements were acquired at typical Rankine cycle ACC operating conditions. It was shown in this study that, for a given steam flow rate, operating at sub-atmospheric pressures result in much larger vapour velocities relative to those which would occur at atmospheric pressures. Increasing vapour velocity is one of the principal mechanisms for augmenting heat transfer.

  - A comparative assessment between the experimental measurements and a number of widely-cited empirical convective condensation correlations was carried-out to determine the most applicable. Notwithstanding the disparity between the individual correlations, the best agreement was observed with that of Ananiev et al. [10], with an average discrepancy of 10%. Other correlations such as the popular method of Shah [9] did not agree well.

  - The effect of tube inclination on condensing flows was investigated by examining a range of condenser inclination angles. In general, an increase in global Nusselt number was found to occur as the tube was inclined from the nominal horizontal configuration. This effect was most apparent for an inclination angle of 10° downwards from the horizontal, with further increases in inclination angle from 10° resulting in no additional gains. The effect of increasing inclination angle was greatest at $Re_v = 2030$, where the increase was approximately 25%. Locally, this increase was mainly seen along the bottom of the tube, as the otherwise accumulating
condensate drained from this region with greater ease. At higher values of $Re_v$, the increase in Nusselt number with increased tube inclination was not as prominent, suggesting there is a greater sensitivity to tube inclination at low vapour Reynolds number flows.

- Increasing air flow rate, through increasing fan speed, was seen to augment condensation heat transfer - a finding not seen in literature to-date. This was only seen to occur at the higher vapour Reynolds numbers of $Re_v = 4185$ and $Re_v = 3680$, which experienced increases in global Nusselt number of 15% and 17%, respectively, when the air flow rate was increased from the lowest value to the largest. The root cause of this was increased vapour velocities, which were shown to be a beneficial consequence of reducing steam temperature as fan speed was increased. The fact that this phenomenon was only seen to occur towards the higher steam flow rates is due to the temperature-fan speed relationship at these flow rates. A greater reduction in temperature with increasing fan speed was seen to occur at high steam flow rates, in comparison to low flow rates. Hence, it follows that there will be a larger increase in vapour velocity.

- Thermal resistance measurements largely exhibited the same characteristics as those on the full-scale. The total thermal resistance was shown to decrease with both increasing fan speed and steam mass flow rate, and the condensate-side thermal resistance decreased with increasing steam flow rate. However, contrary to the full-scale measurements, the condensate-side thermal resistance decreased with increasing fan speed - due to the larger vapour velocities induced. As there was no air leaking into the system to inhibit heat transfer in the reduced-scale ACC, this offered indirect evidence that increased air ingress must be the cause of the increased condensate-side thermal resistance in the full-scale MACC.

- The contribution of the condensate-side thermal resistance to the total thermal resistance was quantified over a range of vapour Reynolds numbers, airflow Reynolds numbers, and condenser inclinations. The dominant relationship was the reduction in the thermal resistance ratio with increasing vapour Reynolds number. When averaged across the airflow Reynolds number range and condenser inclination angles, the thermal resistance ratio was found to vary from ~ 19% at $Re_v = 2030$ to ~ 11% at $Re_v = 4185$. This explicitly confirms that, whilst certainly not insignificant, the condensate-side thermal resistance is minimal in comparison to the dominant air-side thermal resistance.
9.2 RECOMMENDATIONS FOR FUTURE WORK

The following is a list of recommendations for further research, based on observations and findings throughout the course of this study.

- The dephlegmator-vacuum pump system installed on the MACC in this study was done so in a somewhat arbitrary manner. There are some basic guidelines available, recommending the size of the dephlegmator heat transfer area relative to the primary condenser, but these are more a “rule of thumb”. In addition, there is nothing in the literature regarding sizing the vacuum pump/air ejector system based on air leakage rates, condenser size, etc. In industry, it seems to be standard practice to simply oversize the vacuum pump by some generic factor. As these are critical components of an ACC, it is recommended that a more systematic study is carried-out to optimise their design. This could prove beneficial for both industry, and also for researchers working on ACCs.

- Backflow and the formation of non-condensable zones were phenomena both encountered in this study. Basic guidelines were generated as to how to alleviate and eliminate the emergence of these issues. However, as the primary objective of this study was to characterise the condenser, it was simply seen as a necessary step to eliminate backflow without considering how various parameters influence the onset, or otherwise, of backflow. As such, a more robust systematic investigation of backflow in multi-row condensers is recommended. From a design perspective, the number of tube rows and tube length are key considerations and, perhaps, should be flexible. Amongst the variables which should be examined systematically are steam flow rate, air flow rate, and air flow distribution, i.e. uniform vs. non-uniform. Another parameter which may merit investigation is fan-heat exchanger spacing.

- The majority of recent ACCs being installed in thermoelectric power plants are comprised of a single row of rectangular-shaped tubes, such as those on the plate-finned MACC module. These tubes appear to be the current state-of-the-art. However, there is very little information relating to the condensate-side hydrodynamic and thermal performance of these tubes. Limited data, in the form of pressure drop and thermal resistance measurements, are presented in this thesis. However, as this study was more focused on circular tubes, it is recommended that a similar set of local and global measurements on a single air-cooled rectangular tube be carried-out to determine: the type of flow regimes, the qualitative and quantitative spatial distribution of heat transfer, and the effect of parameters such as tube inclination and fan speed on bulk performance.

- Although the predominant two-phase flow regimes for condensing flows of steam in circular tubes were identified in this study, there remains an element
9.2 RECOMMENDATIONS FOR FUTURE WORK

of ambiguity - due to the subjective nature of classifying flow regimes, and
the interpretation of the temperature difference measurements. As such, it is
recommended that the annular-finned circular tube condenser employed in this
study be re-designed to incorporate a series of viewing windows along the tube
length to actually view and capture images of the flow regime. These viewing
windows should be some transparent tubing, of the same approximate diameter,
and ideally should be integrated at the same axial locations from which the
local condensation heat transfer measurements presented in this thesis were
acquired. Images of the flow could be captured using a high-speed camera. In-
stantaneous images should provide insight into flow fluctuations, whether they
be random or systematic, whilst time-averaged images should compliment the
contour plots presented in this thesis.

– The same approach could be adopted for a single rectangular tube.

• Computational Fluid Dynamics (CFD) approaches to two-phase flows, in gen-
eral, are rare in the literature - perhaps a testament to the complex nature
of such flows. As such, there is scope for developing simplified numerical mod-
els for comparison with empirical results in the literature. In particular, the
local measurements presented in this thesis would allow for qualitative and
quantitative comparisons to be made with CFD models.


[58] O. Baker, Design of pipelines for the simultaneous flow of oil and gas, in: Fall Meeting of the Petroleum Branch of AIME, Society of Petroleum Engineers.


This appendix provides additional details on the full-scale MACC experimental test facility described in Chapter 4. This test facility, an overview of which is given in figure A.1, pre-existed the author’s involvement and, thus, was deemed unnecessary for inclusion in the main body of this thesis.

Steam was supplied by a diesel-fired shell boiler with a maximum capacity of 3500 kg/hr. This boiler generated steam at pressures varying between 5 and 7 bar, which is far greater than typical Rankine cycle ACC operating pressures. A steam supply line, connecting the boiler to the MACC inlet manifold, was therefore constructed to achieve pressure reduction and, ultimately, more realistic pressures at the MACC inlet. Pressure in the line was reduced in stages using a number of valves. As the pressure reduction at each stage was accompanied by a reduction in steam density, the cross sectional area of the piping downstream of each valve increased so that excessive steam velocities were avoided.

Due to the cyclic nature of the boiler-firing process, a pressure reducing valve was necessary to ensure a steady and constant pressure at the MACC inlet. This valve eliminated the pressure fluctuations arising from the boiler operating between 5 and 7 bar, and was installed in the supply line - downstream of the boiler. Consequently, a constant pressure was maintained downstream of the valve and remained at the set-point value regardless of the boiler operating pressure. In addition to the pressure reducing valve, a steam separator was also mounted on the line. This was located directly downstream of the boiler and upstream of the pressure reducing valve. The purpose of the steam separator was to improve the dryness fraction of the steam leaving the boiler and, hence, entering the MACC. Initial measurements established that the steam leaving the boiler was excessively “wet”, containing entrained water droplets. The separator removed these droplets and, thus, ensured slightly superheated steam at the MACC inlet.

Figure A.1: Overview of MACC steam supply line and ancillary components
INSTRUMENTATION CALIBRATION

This appendix is dedicated to providing supplementary information to the experimental Chapters 4 and 5. Specifically, the additional information presented here is related to the calibration of thermocouples and NTC thermistors.

B.1 THERMOCOUPLE CALIBRATION

Prior to installation on the MACC, all thermocouples were calibrated to an accuracy of ±0.1 K in a Lauda RM6 thermal bath. This was done by completely submerging the tips of the thermocouples in the baths water reservoir. A series of calibration experiments were then carried-out by varying the set-point temperature of the thermal bath from 20 °C (293.15 K) - 90 °C (363.15 K) in increments of 5 °C. For each temperature increment, the water temperature was allowed to reach thermal equilibrium (steady-state) at which point the thermocouple measurements were recorded. A recirculation pump was incorporated into the bath to ensure temperature uniformity throughout the water reservoir. Figure B.1 presents a set of sample calibration results for a single thermocouple.

![Figure B.1: Sample thermocouple calibration curve](image)

B.2 THERMistor CALIBRATION

Prior to mounting the NTC thermistors inside of, and in the wall of, the reduced-scale ACC tube they were calibrated to confirm the accuracy of the R-T characteristics
provided by the thermistor manufacturer. This was accomplished by establishing the constants in the Steinhart-Hart Steinhart and Hart [153] equation. The Steinhart-Hart equation relates the temperature of the thermistor to the electrical resistance - as shown by equation B.1. Once the constants have been established, equation B.1 can be used to predict the thermistor resistance for a range of temperature values.

\[
\frac{1}{T} = C_1 + C_2 \cdot \ln (R) + C_3 \cdot \ln (R)^3
\]  \hspace{1cm} (B.1)

where \(T\) is the temperature, in Kelvin, and \(R\) is the resistance in ohms. The unknowns in equation B.1, \(C_1\), \(C_2\), and \(C_3\), are constant for a given thermistor type, and must be determined through calibration. Equation B.1 can, therefore, be solved by obtaining three reference temperature and resistance values. These values were obtained from a series of calibration experiments, outlined as follows.

The thermistors were immersed in an insulated Lauda E100 thermal bath which provided a stable and adjustable temperature environment for the thermistor probe. The set-point temperature of the bath was varied in increments of 5 °C, whilst spanning a range of 20 - 80 °C. For each set-point temperature, the water temperature was allowed to reach thermal equilibrium, which was aided by a recirculation pump incorporated into the thermal bath - to ensure temperature uniformity throughout the water reservoir. The temperature of the bulk water was measured using a Fluke 1504 reference thermistor, which had an accuracy of ±0.005 °C, whilst the thermistor resistance was measured using a Fluke 117 multimeter with a resistive accuracy of ±0.9% of the reading. Three temperature measurements and three corresponding thermistor resistance measurements from the calibration experiment allowed for the constants in the expanded Steinhart-Hart equation B.2 to be solved as follows;

\[
\begin{bmatrix}
\frac{1}{T_1} \\
\frac{1}{T_2} \\
\frac{1}{T_3}
\end{bmatrix} = \begin{bmatrix}
C_1 + C_2 \cdot \ln R_1 + C_3 \cdot (\ln R_1)^3 \\
C_1 + C_2 \cdot \ln R_2 + C_3 \cdot (\ln R_2)^3 \\
C_1 + C_2 \cdot \ln R_3 + C_3 \cdot (\ln R_3)^3
\end{bmatrix} = \begin{bmatrix}
1 \ln R_1 \ (\ln R_1)^3 \\
1 \ln R_2 \ (\ln R_2)^3 \\
1 \ln R_3 \ (\ln R_3)^3
\end{bmatrix} \cdot \begin{bmatrix}
C_1 \\
C_2 \\
C_3
\end{bmatrix} \hspace{1cm} (B.2)
\]

Equation B.2 can be expressed as \(A = B \cdot X\), where matrix inversion allowed the constants \(C_1\), \(C_2\), and \(C_3\) to be calculated as 1.91E-03, 1.08E-04, and 5.38E-07, respectively. These values can be inputted into the original equation B.1 to determine the R-T characteristics for a range of temperatures. One such resulting R-T curve is given in figure 5.21 in Chapter 5.
MATLAB ALGORITHM

This appendix presents a sample MATLAB (version 7.8.0 (R2009a)) algorithm to calculate the thermodynamic performance of a Rankine cycle power plant for a range of parameters.

AIR-SIDE-ONLY THERMODYNAMIC MODEL

air_side_plant_modelling.m was the m-file employed to generate thermodynamic performance data based on air-side theory only. As explained in Chapter 3, the steam-side thermal resistance was purposely excluded to evaluate the “idealized” response based on an air-side analysis - as is the approach frequently employed in literature.

Listing C.1: air_side_plant_modelling.m

```matlab
%M-file models the effect of the MACC3 on plant performance for Torresols %Gemasolar CSP plant. Analysis is based solely on an AIR-SIDE approach!! %Variation in Tambient is also simulated.
clc;
clear all;

%%PROPERTIES%%
%Geometric properties of MACC3 plate-fin HEX design
FT = 0.00025; %Fin thickness
L = 0.2; %Fin length
b = 0.00235; %Fin spacing (pitch)
k_fin = 200; %Thermal conductivity of aluminium fin
Dh = 2*b; % Hydraulic diameter
NF = 850; %No of fins on ONE SIDE of a 2m tube (425 fins/m)
H = 0.0186; %Fin height
AF = L*(H+(b/2))*2*NF*2*35; %Finned area of ONE module
AT = AF+(b*L*NF*2*35); %Total heat transfer area of ONE module
FA = b*H; % Channel flow area
FFA = 2.01565; % Free flow area

%Properties of Air at INLET AMBIENT temperature of 20C
rho_i = 1.205; %Density of air at inlet
k_i = 0.0257; %Thermal conductivity
Pr = 0.713; %Prandtl number
Cp = 1005; %Specific heat capacity
nu_i = 15.11*(10^-6); %Kinematic viscosity
mu_i = rho_i*nu_i; %Dynamic viscosity
alpha_i = (k_i/(rho_i*Cp));
```
% Properties of Air at FILM BULK temperature
Ts = 100; % Steam saturation temp
Ta = 20; % Ambient air temp at inlet to channel
Tf = ((Ts+Ts)/2); % Film temp
rho_f = 1.067;
rho_m = ((rho_i+rho_f)/2); % Mean density
k_f = 0.0285;
u_f = 18.90*(10^-6);
mu_f = rho_f*nu_f;
alpha_f = (k_f/(rho_f*Cp));

%% AERODYNAMIC CHARACTERISTICS %%
% Import fan and HEX characteristic curves
vdot_fan = xlsread ('ebm_papst_MACC_fan_data','g3:g17'); % 2 fans in parallel
dp_fan = xlsread ('ebm_papst_MACC_fan_data','b3:b17');
N_arb = (200:20:1000); % Arbitrary range of fan speeds
N_fan = flipud(N_arb'); N = flipud(N_fan);
vdot_sys = xlsread ('MACC3_system_curve','b3:b24');
dp_sys = xlsread ('MACC3_system_curve','c3:c24'); % System resistance curve
p_fan = 2880; % Power of 1 fan at 1000rpm in W
p_fans = p_fan*2; % Power of 2 fans in W
Fan_p = ones(length(N),1);
% Fan curve - HEX curve intersections
addpath('C:\Users\alan.odonovan\Documents\MATLAB\MACC Models');
for i = 1:length(N_fan);
vdot = vdot_fan*(N_fan(i)/N_fan(1));
dp = dp_fan*((N_fan(i)/N_fan(1))^2);
Fan_p_a = p_fans*((N_fan(i)/N_fan(1))^3);
Fan_p(i,1) = Fan_p_a;
[x{i},y{i}] = intersections(vdot,dp,vdot_sys,dp_sys);
figure(1); plot(vdot,dp); hold on; hold all;
plot(vdot_sys,dp_sys,'k--');
plot(x{i},y{i},'ro')
xlabel ('Volumetric Flow Rate (m^3/hr)'); ylabel ('Pressure (Pa)');
end
% Postprocessing results
X = cell2mat(x');
mdot_HEX = ((flipud(X))./3600).*rho_i; % HEX mass flow rate in kg/s
U_ch = ((flipud(X))./3600)./FFA; % Channel velocity in m/s
Re_Dh = U_ch.*Dh/nu_i; % Reynolds number based on hydraulic diameter
Y = cell2mat(y');
Dp_HEX = flipud(Y);

%% THERMAL CHARACTERISTICS %%
Lstar = (L./Dh).^((1./(Pr*Re_Dh)))); % Inverse Graetz number
Nu_fd_ITD = (1./(4*Lstar)); % Fully developed Nusselt number for ITD
Nu_fd_BULK = (6.51*{(1.0+(2.61*(b/H))+(4.97*(b/H)^2))-5.19*(b/H)^3}+(2.77*(b/H)^4)-(0.548*(b/H)^5))); % Fully developed Nusselt number for BULK
Nu_dev = ((0.664./(Lstar.*0.5.*Pr^((1/6))))); % Developing Nusselt number
Nu_ITD = (Nu_fd_ITD.^(-3)+Nu_dev.^(-3)).^(1/3); % n=3 is the blending parameter
\[
\text{h.ITD} = \text{Nu.ITD} \cdot \frac{k_i}{Dh}; \quad \% \text{Based on wall to inlet temperature difference (ITD)}
\]

\[
\text{Nu.BULK} = (\text{Nu_fd.BULK}^1.5 + \text{Nu.dev}^1.5)^{1/1.5};
\]

\[
\text{h.BULK} = \text{Nu.BULK} \cdot \frac{k_f}{Dh}; \quad \% \text{Based on bulk fluid to wall temperature (LMTD)}
\]

\[
m = \sqrt{\left(2 \cdot \text{h.BULK}/(k_fin \cdot FT)\right)}; \quad m_l = m \cdot H;
\]

\[
\text{fin.eff} = \left(\frac{\text{tanh}(m_l)}{m_l}\right); \quad \% \text{Fin effectiveness}
\]

\[
\text{surf.eff} = 1 - \left(\frac{\text{AF}}{\text{AT}}\right) \cdot (1 - \text{fin.eff}); \quad \% \text{Surface effectiveness}
\]

\[
\% \text{CONDENSER CONDITIONS} \%
\]

\[
\text{Tamb} = [5; 10; 15; 20; 25; 30; 35; 40]; \quad \% \text{Set range of ambient temperatures}
\]

\[
\text{N.modules} = [200; 400; 600; 800; 1000];
\]

\[
\% \text{Number of MACC modules ctrl} = \text{length(Tamb)} \cdot \text{length(N.modules)}; \quad \% \text{Loop and matrix controller}
\]

\[
\text{A.\_cond} = \text{AT} \cdot \text{N.modules}; \quad \% \text{TOTAL condenser heat transfer area}
\]

\[
\text{mdot.tot} = \text{zeros(length(h.BULK))}; \quad \% \text{TOTAL flow rate through all modules}
\]

\[
\text{for aa} = 1: \text{length(N.modules)};
\quad \text{mdot.tot}_a = \text{N.modules}(aa,1) \cdot \text{mdot_HEX};
\quad \text{mdot.tot}(:,aa) = \text{mdot.tot}_a;
\quad \text{end}
\]

\[
\text{Ts} = \text{zeros(length(h.BULK));} \quad \% \text{Save variables from k loop which are written over on each j loop}
\]

\[
\text{Ps} = \text{zeros(length(h.BULK));} \quad \% \text{Save variables from each j loop}
\]

\[
\text{Ps.BULK} = \text{zeros(length(h.BULK))}; \quad \% \text{Save variables from each i loop into cell which is controlled by N.modules}
\]

\[
\text{Ps.amb} = \text{cell}(1, \text{length(Tamb)}); \quad \% \text{Save variables from each i loop into cell which is controlled by N.modules}
\]

\[
\text{for i} = 1: \text{length(N.modules)};
\quad \text{for j} = 1: \text{length(Tamb)};
\quad \text{for k} = 1: \text{length(h.BULK)};
\quad \text{f} = \left(1 - \exp\left(\left(1 - \text{surf.eff}(k,1) \cdot \text{h.BULK}(k,1) \cdot (\text{AT} \cdot \text{N.modules}(i,1))\right)/\left(\text{mdot.tot}(k,i) \cdot \text{Cp}\right)\right)\right)\cdots
\quad \left(\text{mdot.tot}(k,i) \cdot \text{Cp} \cdot (\text{Ts} - \text{Tamb}(j,1))\right) - \left(787.560742787318 \cdot \text{Ts}^2\right)
\quad - \left(3063.42586429436 \cdot \text{Ps}\right)\cdots
\quad + 26625272.03826; \quad \% \text{GEMASOLAR CF in Watts}
\quad \text{Ts}_a = \text{fzero}(\text{f},55); \quad \% \text{Saving variables from each k loop iteration}
\quad \text{Tk}_a = \text{Tk} + 273.15; \quad \% \text{Convert degreesC to Kelvin}
\quad \text{P}_a = \exp(18.79 - (0.0075 \cdot \text{Tk} + 5965.6/\text{Tk}) - (5965.6/\text{Tk})) ; \quad \% \text{Condenser pressure in bar}
\quad \text{Ts}(k,:) = \text{Ts}_a; \quad \% \text{Saving variables from each k loop iteration}
\quad \text{Ps}(k,:) = \text{P}_a;
\quad \text{end}
\quad \text{Ts.BULK}(:) = \text{Ts}; \quad \% \text{Save variables from each j iteration}
\quad \text{Ps.BULK}(:) = \text{Ps};
\quad \text{end}
\quad \text{Ts.amb}(:,i) = \text{Ts.BULK}(:,1: \text{length(Tamb)}); \quad \% \text{Save variables from i iteration}
\quad \text{Ps.amb}(:,i) = \text{Ps.BULK}(:,1: \text{length(Tamb)});
\quad \text{end}
\quad \text{Ts.pp} = \text{cell2mat(Ts.amb)};
Ps_pp = cell2mat(Ps_amb);

%%POWER-PLANT ANALYSIS%%
Fan_p_ud = flipud(Fan_p);
Fan_p_pp = ones(length(h_BULK):length(N_modules)); %TOTAL fan power consumption in power plant
for bb = 1:length(N_modules);
    Fan_p_pp_a = (Fan_p_ud*N_modules(bb,1)/1000000); %Fan power in MW
    Fan_p_pp(:,bb) = Fan_p_pp_a;
end
ST_P = zeros(length(h_BULK):ctrl); %Gross power from turbine
Net_P = zeros(length(h_BULK):ctrl); %Net power
Eff = zeros(length(h_BULK):ctrl); %Plant efficiency loss
for iii = 1:ctrl;
    ST_P_a = (-0.000775968979484065.*(Ts_pp(:,iii).^2))+(0.00296829217553583.*Ts_pp(:,iii))...+20.029570457444; %GEMASOLAR ST 2nd order polynomial curve-fit MW
    ST_P(:,iii) = ST_P_a;
end
for cc = 1:8;
    Net_P_a = ST_P(:,cc) - Fan_p_pp(:,1);
    Net_P(:,cc) = Net_P_a;
    Net_P_a = ST_P(:,cc+8) - Fan_p_pp(:,2);
    Net_P(:,cc+8) = Net_P_a;
    Net_P_a = ST_P(:,cc+16) - Fan_p_pp(:,3);
    Net_P(:,cc+16) = Net_P_a;
    Net_P_a = ST_P(:,cc+24) - Fan_p_pp(:,4);
    Net_P(:,cc+24) = Net_P_a;
    Net_P_a = ST_P(:,cc+32) - Fan_p_pp(:,5);
    Net_P(:,cc+32) = Net_P_a;
end