Design of an Annular Disc-shaped Heat Pipe for Air-cooled Steam Condensers

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DESIGN OF AN ANNULAR DISC-SHAPED HEAT PIPE FOR AIR-COOLED STEAM CONDENSERS

by

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ABSTRACT

Limitations on water utilization are turning into an expanding issue for the power and electricity generation industry. As a contribution to the solution of water consumption problems, utility companies are shifting toward using air-cooled condensers (ACC) in replace to the typical water-cooling methods of once-through cooling and the surface condenser/wet-cooling tower combination. Although the ACC is a dry cooling method, the industry is quite hesitant to switch over to ACC mainly for three reasons: (a) lower power output, (b) higher capital cost, and (c) larger physical footprint. All these drawbacks are because of the high overall thermal resistance of condensing steam to the ambient air compared to condensing it to water.

In this study, detailed mathematical equations were derived to model the heat transfer process through the fined tubes of the ACC. The total thermal resistance model was analyzed and investigated theoretically. The model was used to identify the design components with the most significant effect on the overall thermal resistance of the ACC system.

This study proposed a feasible cooling system based on heat pipe technology, using a novel disc-shaped heat pipe design. The solution addresses the three problems highlighted in using the air-cooled condensers in steam powerplant condensers. The analysis covered design and manufacturing considerations, in addition to the thermal
performance and the limitations of the proposed annular disc-shaped heat pipe. The proposed annular disc-shaped heat pipe was investigated using three analysis techniques. The first is a theoretical investigation of the heat transfer limitations of the proposed annular disc-shaped heat pipe. This analysis was used to predict the capillary and boiling thermal limitations of the proposed heat pipe design. Secondly, an annular disc-shaped heat pipe was designed and built for the experimental investigation using de-ionized water as the working fluid. The results obtained by the parametric analysis were used as the input for the experimental design. Third, A detailed mathematical set of equations was derived to model the heat pipe thermal resistance.

The experimental setup was validated by comparing the results to well-referenced experimental results of similar disc-shaped heat pipe with different evaporator configurations. The experimental results were compared to the thermal resistance model developed in this study. The results showed a starting regime of the heat pipe, where the thermal resistance is decreasing until it reaches a steady performance before it starts to increase again when it reaches the heat transfer limits. The experimental results showed a good agreement with the model prediction in the steady-state regime for heat inputs over 300 w. The data identified two thermal performance regimes of the heat pipe, a single-phase, and a two-phase regime. The second regime starts when the vapor region reaches the isothermal state.
The heat pipe was tested with a maximum total heat input of 600 W, using three types of wicks. The study presented the effect of the wick parameters such as porosity, permeability, mesh number, and wire diameters on the performance. The impact of the filling ratio was investigated by testing the heat pipe performance with 30 ml, 60 ml, 90 ml, and 120 ml filling amount.

The study showed high promising results for the proposed solution. It established the theoretical and experimental validation of the system by validating the concept of the annular disc-shaped heat pipe. Further investigation can be conducted starting from the experimental results.
DEDICATED TO MY FAMILY
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### LIST OF NOMENCLATURE

- **A** = Cross-sectional area (m²)
- **C_p** = Specific heat (KJ/Kg.K)
- **D** = Diameter (m)
- **D_h** = Hydraulic diameter (m)
- **h_f** = Latent heat (J/kg)
- **h** = Heat Transfer Coefficient (W/m²-K)
- **K** = Permeability (m²)
- **k** = Thermal conductivity (W/m-K)
- **m** = Mass flow rate (kg/s)
- **N** = Mesh number (in⁻¹)
- **Nu** = Nusselt number
- **p_o** = Total pressure (Pa)
- **p** = Pressure (Pa)
- **Pr** = Prandtl Number
- **R** = Resistance (Ω)
- **R** = Thermal resistance (k/w)
- **Re** = Reynolds number
- **R_{air}** = Gas constant of air
\( r \) = Radius (m), radial coordinate.

\( T \) = Temperature (°C)

\( t \) = Thickness (m)

\( U \) = Velocity (m/s)

SD = Standard deviation

SEM = Standard error from mean

Subscripts

\( c, \text{cond} \) = Condenser

\( e, \text{evap} \) = Evaporator

\( \text{eff} \) = Effective

\( f \) = Fin

\( l \) = liquid

\( s \) = Solid

\( v \) = vapor

\( w \) = wick

Greek Symbols

\( \eta \) = Thermal efficiency
\( \varepsilon = \) Thermal effectiveness
\( \varepsilon = \) Porosity
\( \sigma = \) Fluid surface tension
\( \rho = \) Density (kg/m\(^3\))
\( \mu = \) Dynamic Viscosity
\( \nu = \) Kinematic Viscosity
\( \theta = \) Tangential coordinate

**Acronyms**

AACC= A-frame air cooled condenser
AC = Air cooled
ADHP= Annular disc-shaped heat pipe.
CFD = Computational Fluid Dynamics
CHAPTER 1: INTRODUCTION

Background

Limitations on water utilization are turning into an expanding issue for the power and electricity generation industry. The water withdrawal and consumption in the power generation industry and its environmental effect has been widely studied in the literature [1-11]. The U.S Geological Survey (USGS) [2] has reported that 41% (540 Mm3/day) of the U.S. total freshwater withdrawals are used in electricity production. Most of the water in thermoelectric production is used in once-through coolers, where the water is returned to its source. The energy production accounts for about 3.3% (12.5 Mm3/day) of total water consumption in the United States [9].

All the power plants that are based on the Rankine cycle, either in simple or combined cycles, require heat rejection to the environment to condense the turbine exhaust steam[12]. The heat rejection in modern combined-cycle ranges from 40% of the fuel energy and up to 70% for a traditional Rankine-cycle [13]. Typically, there are three major condensation systems used for this heat rejection:

- Once-through water cooling-steam surface condensers.
- Combined systems of a wet-cooling tower and steam surface condensers.
- Air-cooled condensers systems (Dry-cooled condensers).
The once-through cooling systems are still utilized in over 30% of today’s fleet of power plants. In once-through cooling systems, the water from an environmental source such as near-by lake or river is cycled through the condenser of the turbine exhaust steam. The water is then returned to the same source warmer by about 20°F [1]. In between the commercially used cooling methods, this method is considered the most inexpensive. However, the resulting increase in temperature of the cycled water creates an environmental concern. Another disadvantage of using this method is the cost of maintenance to clean the tubes of fouling to maintain its performance.

In the combined surface condenser with a cooling tower system, the turbine exhaust steam condenses on the outside surface of a bundle of tubes by extracting the heat to the cold water circulated inside the tubes. After receiving the rejected heat from the condensed steam, the circulated water is piped to the top of a cooling tower. And it flows downward through fill material that breaks the water up into droplets or spread it out into a thin film to maximize the surface area exposed to the cooling air, which is drafted through the tower by natural convection or large fans. The cooling method in these towers is considered wet cooling, where its cooling capacity is limited by the cooling air wet-bulb temperature (TWB). This type of cooling system can maintain consistent exhaust pressure for the turbine. However, this type of cooling system consumes a large amount of water. About 80 to 90% of the rejected heat is released to the
atmosphere through the evaporation of water. Thus, this system requires a large amount of make-up water to keep the condenser working on the required performance [1]. In this type of cooling system, about 0.47 gallons (1.8 L) of water are consumed per kilowatt-hour of electricity consumed at the point of end-user[10].

Although the once-through cooling systems withdraw up to 100 times more water per unit of electric production than the cooling towers technologies, the cooling tower’s systems consume at least twice as much water as once-through cooling systems [7].

**Air Cooled Systems (Dry Cooling)**

Every year more restrictions on utilization of surface or groundwater are added and imposed by communities around the country, which affect the cost of electricity production. As a contribution to the solution of water consumption problems, utility companies are shifting toward using air-cooled condensers (ACC) in replace to the typical water-cooling methods of once-through cooling and the surface condenser/wet-cooling tower combination. Although the ACC is a dry cooling method, the industry is quite hesitant to switch over to it, that’s mainly due to three reasons:

(a) lower power output

(b) higher initial cost.

(c) larger physical footprint.
All these drawbacks of the air-cooled condenser are because of the high overall thermal resistance of condensing steam to the ambient air compared to condensing it to water, due to the poor thermal conductivity of air[14].

In the dry cooling systems, the heat extracted from the exhausted steam is ultimately rejected to the ambient air-driven across finned-tube heat exchangers. This process typically is done by using one of two main types of dry systems: direct or indirect dry cooling. In the indirect dry cooling systems, the steam condenses in a surface condenser, like the once-through and recirculating systems process. However, the heated cooling water is then cooled down by circulating it in an air-cooled heat exchanger. In the direct system, the steam is condensed in the air-cooled condenser by directing the exhaust steam directly into the ACC. In these two dry systems, the air is driven by mechanical or natural draft units. All the steam dry cooling systems in the United States are using the direct system with a mechanical-draft ACC [15].

In contrast to the steam surface condensers, the exhaust steam in the air-cooled condenser (ACC), condenses inside an of arrayed cells of finned tubes by extracting the heat to the continual external airflow outside the tubes. Thus, the primary heat transfer mode in the process is by convection without any water consumption as there is no evaporation process; that is why it is referred to as Dry cooling. Because of that, the condensing steam temperature is theoretically limited by the external air dry-bulb
temperature (TDB). As the dry-bulb temperature always equals or higher than the wet-bulb temperature, the dry-cooling methods generally provide higher exhaust pressure for the turbine compared to the water-based systems. The higher steam condensing temperature penalizes the cycle efficiency, particularly on hot days with low humidity as the condenser increases the steam temperature to be able to extract the required amount of heat to hot ambient temperature.

Figure 1 depicts the T-S diagram of an ideal Rankine cycle, typically the heat transferred to the water in the boiler is represented by the area under process curve 2-3, and the heat rejected in the condenser is represented by the area under the process curve 4-1. The difference between these two areas sums up the total net-work produced in the cycle. The cycle work increases, which is the integral of T.ds around the cycle, by decreasing the exit pressure from P4 to P4’. As a result of that, the mean temperatures at which the heat is absorbed and rejected both decreases, the most substantial change is in the mean temperature of the heat rejection, which increases the cycle thermal efficiency.
Figure 1 T-S diagram of the ideal Rankine cycle [16]

Figure 2 shows a typical characteristic curve of a steam turbine. From the curve, the power generated from the turbine decreases with the increase in turbine exhaust temperature. However, the higher the exhausted temperature, the higher the heat rejection required. That is due to the reduction in the turbine efficiency with increasing the turbine back pressure and condensing temperature. In general, the condenser must be designed according to the turbine power and the ambient conditions to have enough cooling capacity under all operating conditions. When ambient conditions reduce the efficiency of the condenser, the steam temperature must increase to reject the needed heat; and less power is generated [7].
For a 500 MW steam power plant, reducing the steam condensing temperature by 15°C from 50 °C to 35 °C would give an estimation of 5% more power production, that’s equal to about $11M more annual income [17].

The air-cooled condenser (ACC) systems consist of many air-cooled heat exchangers arranged in an A-frame configuration, as illustrated in Figure 3. The turbine exhaust steam is ducted through a distribution manifold located above a row of finned tubes. A typical full-scale ACC consists of several such rows referred to as “streets” or “lanes.” Each row consists of three to six primary condenser cells connected in series with secondary reflux (dephlegmator) condenser, as shown in Figure 4.
In the A-frame ACC system, the finned tubes are typically about 30–40 ft long and grouped in 8 ft bundles across. Typically, one cell has plan dimensions of 40 ft x 40 ft, which contains around five bundles on each face, with a total of 10 bundles per cell. Many different geometries of finned-tube geometries have been used over the years, such as circular tubes with wrapped, round fins, and elliptical tubes with plate fins. Finned tubes are usually arranged in two to four rows in a staggered array. However, the most recent ACC designs have used elongated flow tubes separated by plate fins, referred to as a single row condenser (SRC) [15].

As an example, a 500-MW combined-cycle plant, typically have 30–40 cells to condense an approximate of 125–150 kg/s of steam. Cells are usually arranged in a (5 x 6), (8 x 5), or two (4 x 5) layouts. That makes the total footprint of the ACC about 200 ft x
250 ft. adding to that bracing and vertical steel columns to support the cells and fans. The fan deck is usually 60–80 ft above ground level, makes the large steam manifold at the top of the cells at about 100–120 ft above grade.

The NSF/EPRI Joint Solicitation Informational Webcast conducted, 2013, reported the average cost of a cell of 12x12 m2 footprint size, to be about $1.5 Million/ACC cell.

Figure 4: ACC street with 6 Cells [20].
**Heat Pipes**

A heat pipe is a closed two-phase heat transfer device that can passively transfer a large amount of heat energy over a small temperature difference. The heat pipe in its conventional design was introduced by Gaugler in 1944, when he introduced the use of the wick structure, giving the heat pipe the ability to work against gravity. However, the full potential of the heat pipe design was not realized by the scientific community until it was highlighted by Grover research in 1964, where he showed that the heat pipe has a higher thermal conductivity than any known metal. [21].

The heat pipe is a sealed device that contains both the liquid and vapor phases of a working fluid. The device is heated from one end, called the evaporator, and cooled from the other end, known as the condenser side. Heat is applied at the so-called evaporator section, and the phase change occurs, the vapor flows through the heat pipe due to the pressure difference generated by the evaporation process, the heat is then extracted at the other end, where the vapor condenses back to the liquid phase, the so-called condenser section, The liquid then flows back to the evaporator section under the influence of gravity or by a capillary structure passively completing the heat transfer cycle.
Figure 5: Conventional heat pipe schematic [22]

Figure 5 shows a schematic of a conventional heat pipe showing its operational concept of heat and working fluid circulation. The heat pipe has been used in many different configurations, with single or multiple heat sources or sinks with or without adiabatic sections. A significant advantage feature of the heat pipe is its flexibility to be designed according to the specific application [23].

Heat pipes have been used in many variant applications. However, it is well used in space applications heating or cooling in vehicles, electronics cooling, and fuel cells. The heat pipe low maintenance cost and its ability to efficiently transmit heat over low driving temperatures make it an attractive choice for Multi-cascade machines [23-26]. And its ability to transfer high heat flux rates makes it preferable for electronic cooling where the footprint size is challenging [14, 27]
Heat pipes can be classified into different categories based on their geometries, applications. The two primary ways to classify the heat pipes are based on either its working fluid operating temperature, or the control mechanism of driving the liquid between the condenser and evaporator (wick, gravitational, centrifugal, electrostatic, or osmotic) [23]. Different applications have different operating conditions and different temperature range [28]. Therefore, choosing a suitable working fluid is crucial. The working fluid selection must consider the operating temperature and pressure, and the compatibility between the working fluid and the heat pipe container and wick materials.

The working temperature range for the commonly used working fluids in heat pipe applications are presented in Figure 6. Heat pipes can be classified into four different types based on its operating temperature, Table 1 summarize this classification. [28]

For the proposed application in the steam condenser for steam power plants, several working fluid options are suitable for the working temperature range, such as methanol, ammonia, acetone, and water. For its availability, well-established properties, and compatibility with most of the casing materials, the de-ionized water was selected for this study as the working fluid.
### Table 1: Classifications of heat pipes by operating temperature [28]

<table>
<thead>
<tr>
<th>Type</th>
<th>Temp. Range</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>High Temperature</td>
<td>&gt;700 K</td>
<td>Using liquid metals, Potassium, Sodium, and silver</td>
</tr>
<tr>
<td>medium Temperature</td>
<td>550-700 K</td>
<td>organic fluids such as Naphthalene and Biphenyl</td>
</tr>
<tr>
<td>Room Temperature</td>
<td>200-550 K</td>
<td>typically, methanol, ethanol, ammonia, acetone, and water</td>
</tr>
<tr>
<td>Cryogenic (Low Temperature)</td>
<td>1-200 K</td>
<td>fluids such as helium, argon, neon, nitrogen, and oxygen</td>
</tr>
</tbody>
</table>

**Figure 6:** Operating temperature range of conventional working fluids [28]
CHAPTER 2: OBJECTIVES & MOTIVATION

Objectives of the Present Study

Reflecting upon the review of literature from Chapter 1, water consumption in power generation is a high raised concerned in the industry. The commercially available type of air-cooled condensers for steam turbine applications addresses this problem and provides a low water consumption solution. However, this solution penalizes the performance of the steam turbine cycle due to its relatively low heat transfer efficiency. This study aims to introduce a novel solution to this problem using a novel air-cooled condenser, employing the heat pipe technology to increase the thermal of the steam condensers. In general, the objective of this study is to identify the current challenges that constrain the thermal performance of the air-cooled condenser and to introduce the concept of the proposed heat pipe condenser while addressing all the design and performance challenges that affect the proposed solution. The deliverables and measures of study in the effort to achieve these objectives are outlined below:

1. Thermal analysis of the state-of-art air-cooled condenser
   a. Thermal resistance analysis of the fin-tube structure.
   b. Correlations of the heat transfer coefficients of the air and steam sides.
c. Predicting the condenser’s pressure drop and heat transfer performance.

d. Defining the main effective parameters on the condenser performance.

2. Conceptual design of the annular disc-shaped heat pipe.

3. An analysis study of an annular disk-shaped heat pipe.
   a. Thermal resistance analysis and 1D modeling.
   b. Define the heat transfer limitations of the design.
   c. Numerical modeling of the heat pipe process.

4. Experimental design and validation of the derived model.

5. Compare the proposed heat exchanger performance to the current state-of-art steam turbine air-cooled condenser.
   b. Reduction of the turbine backpressure of the cycle.

**Novelty and Intellectual Contribution**

Following are aspects of the various ways the current study is unique and novel:

- Air-cooled condenser analysis considering both air and steam sides.

- Novel Annular Disc-Shaped Heat Pipe design (ADHP).

- Thermal resistance analysis of ADHP.
- Modeling heat transfer limitations of ADHP.

- Experimental study of the ADHP under steady and transient states.

The proposed Annular disc-shaped heat pipe has a novel design that has not been discussed in the literature before. The working concept of it can be thought of as a combination between the flat or disc-shaped heat pipe and the concentric heat pipes designs. The evaporator has the same concept of the typical annular heat pipe with the heat transmit radially. The disc-shaped heat pipe has been introduced and analyzed by Vafai [29]. However, this study proposes a new design for the evaporator side. The new design makes it suitable for the proposed application of steam condenser.

This study provides a detailed analysis of a general model to predict the annular disc-shaped heat pipe thermal performance with the consideration of the design shape, heat pipe material, working fluid types, and condenser cooling method.

The study provides an extensive experimental investigation for the novel proposed heat pipe that provides a base for further investigation and design optimization in the effort to achieve the comprehensive solution to one of the well stated technical challenges in power generation.
CHAPTER 3: ANALYSIS OF AN ACC UNIT

ACC Thermal Resistance Analysis

This chapter presents the analysis of the performance of a typical air-cooled condenser unit (ACC). The analysis considers the most recent type of finned tubes used in ACC, starting with the determination of its heat transfer features. Figure 7 depicts the main geometrical parameters of a flattened finned-tube, which is the most recent design used in ACC [30]. The analysis described in this study was presented by the author in previous work [31].

Figure 7: Geometry of plate-fin heat exchanger with flattened channels [30].
In the Rankin cycle, the turbine backpressure depends on the condenser temperature, which is controlled by the overall conductance between the steam side and the cooling air passing over the fined tubes. This analysis provides a comprehensive methodology for predicting the overall heat transfer coefficient for the fin-and-tube heat exchanger. The outside heat transfer coefficient is obtained by a series of correlations for laminar/turbulent single-phase flow in between the plate fins. On the other side, the two-phase condensation region is also predicted by correlations for steam condensation inside the tubes. The two sides, external single-phase and internal single-phase or two-phase flows, were combined to predict the condenser temperature and heat transfer performance.

A sketch of a cross-sectional view of a finned-tube is illustrated in Figure 8; the red arrows indicate the heat transfer path between the steam inside the tube and the external cooling air. Figure 9 illustrates the thermal resistance network for the cross-section under investigation.
Figure 8: Heat Transfer path in a finned tube

Figure 9: Equivalent thermal resistance circuit of a finned tube
the fin tube total thermal resistance is obtained from:

\[ R_{total} = \frac{1}{U_i A_i} = \frac{(T_s - T_a)}{Q} \]  

(1)

\[ \frac{1}{U_i A_i} = \frac{1}{h_c A_{int}} + R_w + \left( h_B A_B + \frac{1}{\frac{R''}{A_{cont}} + \frac{1}{\eta_F h_F A_F}} \right)^{-1} \]  

(2)

\[ \frac{1}{U_i A_i} = \frac{1}{h_c A_c} + \frac{t_w}{k A_c} + \frac{1}{\eta_0 h_a A_{tot}} \]  

(3)

Where \( R'' \) is the contact resistance, \( R_w \) is the tube wall resistance, and \( \eta_0 \) is the overall surface efficiency calculated by:

\[ \eta_0 = 1 - \frac{A_f}{A_{tot}} \left( 1 - \frac{\eta_f}{C_1} \right) \]  

(4)

\[ C_1 = 1 + \eta_f h_a A_f \left( \frac{R''}{A_{cont}} \right) \]  

(5)

\( A_{int} \) = tube surface area, \( A_B \) = un-finned surface area

\( A_i \) = tube area, \( A_{cont} \) = fin-tube contact area

\( A_{tot} \) = air-side total area, \( R_w \)=tube wall resistance

\( R' \) = contact thermal resistance, \( h_a \) = air side heat transfer coefficient

\( \eta_f \)=fin efficiency, \( \eta_0 \) = air-side overall efficiency

\( h_c \)=steam side heat transfer coefficient.

Typically, the conductance resistance in the fin is much smaller than the conviction resistance. Hence, equation 3 can be simplified as:
\[
\frac{t_w}{k_p A_c} \ll \frac{1}{h_c A_c}, \quad \text{and} \quad h_B \frac{A_B}{A_f} \ll \frac{1}{R'' \frac{A_f}{A_{cont}} + \frac{1}{\eta_F h_F}}
\]  
(6)

The overall heat transfer coefficient can be written as:

\[
\frac{1}{U_i} = \frac{1}{h_c} + R'' \frac{S}{t_f} + \frac{S}{\eta_F h_F (2L_f + t_f)}
\]  
(7)

The analytical solution for a rectangular fin efficiency is [32]:

\[
\eta_F = \frac{\tanh(mL_e)}{mL_e}
\]  
(8)

Where,

\[
m = \sqrt{\frac{2h_f}{k_f t_f}}, \quad \text{and} \quad L_c = L_f + \frac{t_f}{2}
\]  
(9)

From the general derived analysis of the finned tube heat exchanger, the total thermal resistance strongly depends on the condensing heat transfer coefficient, the fin-tube contact resistance, and the fin thermal efficiency. The recent manufacturing technology dramatically reduced contact resistance. However, the use of these technologies is still costly to be implemented commercially for mass production and requires a lot of effort and accuracy [33-35].

Some methods investigated in the literature, such as internal tabulators and internal fins, showed an enhancement of the condensing heat transfer rate [36-40]. However, it is commercially challenging to use such methods due to the small internal
diameters and the very long tubes, which make the process economically not efficient. The fin thermal resistance has been intensively studied in the literature to optimize the geometry, size, and spacing [41-44]. Which makes the current implemented fin-tube configuration, is the optimal design considering performance, coast, and manufacturing. Further studies suggested using guided solid and slotted plates to enhance the external heat transfer coefficient between the fins [45-48]

The proposed solution in this study considers the three components of the thermal resistance and provide an improvement of the overall thermal performance of the condenser by implementing the heat pipes technology in the power plant condensers.

**Heat Transfer Coefficients Correlations**

The heat transfer coefficient in the external airflow between the fins is modeled as concurrently developing flow throw a channel. Following the same analysis presented in our previous work [31], using the Nusselt number expression proposed by Sparrow [49] and presented by Teertstra et al. [50], the developing flow region is expressed as:

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

(10)

The developing region dimensionless thermal entrance length \( (L_{th}^*) \) is given by:
The heat transfer coefficient in the fully developed flow region between the flat passages depends on the fin’s geometry and design. The Nusselt number is obtained using the expression developed by Shah and Sekulic [51]:

\[ Nu_{fd} = 7.541(1 - 2.61 \left( \frac{b}{H} \right) + 4.97 \left( \frac{b}{H} \right)^2 - 5.119 \left( \frac{b}{H} \right)^3 + 2.702 \left( \frac{b}{H} \right)^4 - 0.548 \left( \frac{b}{H} \right)^5 \] (12)

Equations (10) and (12) represent the developing and fully developed regions, respectively. The two expressions are combined using the addition of asymptotes method to obtain the overall Nusslet number of the flow over the fins [52].

\[ Nu_a = \left[ (Nu_{dev})^n + (Nu_{fd})^n \right]^{1/n} \] (13)

n is the blending parameter that controls the behavior of the model in the transition region. The value for the blending parameter used in this model (n=3), as determined by Teertstra et al. [50]. Thus, the heat transfer coefficient can be evaluated from

\[ h_a = \frac{Nu_a k_f}{D_h} \] (14)

The accuracy and applicability of Equation (14) were experimentally validated in [53].

Kroger [54] derived an analytical solution for the condensing heat transfer coefficient in inclined finned flattened tubes
\[ h_c = 0.9245 \left[ \frac{k^3 \rho^2 L_t g \cos \theta \ h_{fg}}{\mu m_{a1} C_p \left( T_v - T_{ai} \right) \left[ 1 - \exp \left(-\frac{UA_c}{m_a C_p} \right) \right]} \right]^{0.333} \] (15)

Where \( m_{a1} \) is the mass flow rate of air flowing on one side of the finned tube, \( h_{fg} \) is the latent heat of the steam. The overall heat transfer coefficient is approximated by the effective airside heat transfer coefficient based on the condensation surface area of the flattened tube, ignoring the film resistance.

Kroger’s analysis applies to that part of the tube where vapor velocity and the shear stress on the condensate film are negligible. The shear stress on the condensate film strongly affects the development of the film region, especially at the inlet region to the relatively high velocity. However, gravity control becomes more critical further from the inlet, and the above correlation gives a good approximate condensation heat transfer coefficient for long inclined tubes.

**Condenser Steam-Side Temperature**

The steam condensing side of the air-cooled condenser has a high effect on predicting the condenser performance. However, most of the current models in the literature seem to be based solely on airside analysis, disregarding the steam-side resistance for quantifying condenser performance could lead to erroneous results, in particular when presented in terms of plant output.
In this study, a detailed ACC model is presented to evaluate the condenser temperature and pressure. The model considers the airside thermal resistance and the steam-side one.

In the derived model, isothermal heat rejection is assumed inside the steam condenser, and neglecting the sensible heat rejection. Additionally, the model does not consider any undesirable air-leakages or tube fouling in the condenser.

Upon these assumptions, the total heat transfer inside the condenser is given by:

\[ \dot{Q} = \dot{m}_s h_{fg} \]  

All the energy rejected from the steam must be transferred to the air crossing the heat exchanger, to satisfy the energy balance on the air-cooled condenser under the stated assumptions,

\[ \dot{Q} = \dot{m}_a C_p a (T_{ao} - T_{ai}) \]  

Where \( m_s \) and \( m_a \) are the condensed steam and air mass flow rates respectively, \( h_{fg} \) is the steam’s latent heat, \( T_{ao} \) and \( T_{ai} \) is the outlet and inlet temperatures of the air. From the equations (16) and (17) the air temperature at the exchanger exit can be expressed as:

\[ T_{ao} = \frac{\dot{m}_s h_{fg} + \dot{m}_a C_p a T_{ai}}{\dot{m}_a C_p a} \]  

\( h_{fg} \) is steam’s latent heat of vaporization, which is a function of the steam temperature.

\[ h_{fg} = f(T_s) \]
However, in an ACC, the latent heat rejected during steam condensation is extracted by the convection heat transfer mode, which is given in the following equation:

$$\dot{q} = hA\Delta T = \varepsilon hA(T_s - T_{ai}) \tag{20}$$

The energy balance on the condenser yields:

$$\dot{m}_s h_{fg} = \varepsilon U A(T_s - T_{ai}) \tag{21}$$

And then the steam temperature can be expressed as:

$$T_s = \frac{\dot{m}_s h_{fg} + U A T_{ai}}{U A} \tag{22}$$

Or

$$T_s = \frac{\dot{m}_s f(T_s) + h A T_{ai}}{h A} \tag{23}$$

Where $U A$ is the overall heat transfer coefficient between the air and steam. Then from the definition of the heat exchanger effectiveness:

$$\varepsilon = \frac{\dot{m}_a C_p_a(T_{ao} - T_{ai})}{\dot{m}_a C_p_a(T_s - T_{ai})} = 1 - \exp(-NTU) \tag{24}$$

$NTU$ is the number of transfer units defined as:

$$NTU = \frac{U_i A_i}{C_p_a \dot{m}_a} \tag{25}$$

Where, $U_i$ is the total thermal resistance between the steam and air, found by equation (7).

In equations (23) and (24), the only unknown is the air temperature at the outlet. The cooler area is fixed. The ambient temperature is known from site conditions; the mass
flow rate of air and heat transfer coefficient are known from air-side analysis, and the steam-side heat transfer rate is known from the steam condensing analysis. Then from equations (18) and (24) the steam temperature can be presented as:

\[ T_s = \frac{\dot{m}_s f(T_s) + \dot{m}_a C_p T_{ai}}{\frac{\dot{m}_a C_p}{1 - \exp(-NTU)} + T_{ai}} \]  

(26)

By calculating the condenser steam temperature, the condenser pressure can be determined from the steam temperature by applying the stated assumption of isothermal heat transfer under saturation conditions.

Equation (26) shows the high dependency of the steam condenser temperature on the air side heat transfer and the efficiency of the heat transfer mechanism on it, typically the fin effectiveness. This study is introducing a novel condenser design using the two-phase heat transfer mechanism by using the heat pipe as an alternative to the typical metal fin.
CHAPTER 4: PARAMETRIC ANALYSIS OF ANNULAR DISC-SHAPED HEAT PIPE

In this chapter, the design of an annular disc-shaped heat pipe (ADHP) model is presented. The general design was introduced in our previous work [31]. In this study, a detailed analytical study is performed on the designed heat pipe. A one-dimensional thermal resistance model is derived for investigating the proposed design. In the second part, the heat transfer limitations of the ADHP are analytically investigated.

Although heat pipes are highly effective thermal conductors, they have heat transfer limitations. These limitations depend on various design and application parameters such as the working temperature, the required heat flux, the working fluid, the type of wick structure. The primary heat transfer limitations are the capillary limit, which mainly defined by the wick structure and its capability to overcome the pressure drop over the heat pipe. The maximum heat flux defines the boiling limit that the evaporator can transfer before the nucleation boiling start to appear on its surface. The sonic limit that can prevent the vapor from reaching the condenser. The entrainment limit may occur if the condensed drops block the vapor flow. And the frozen start-up limit defined by the minimum heat flux required to start the two-phase transition in the heat pipe [23]. Defining al these limits gives what so-called the heat pipe performance map, Figure 10 shows a sample heat pipe map, it defines the maximum heat transfer a heat
pipe could achieve as a function of its working temperature [22]. In general, in the process of designing a heat pipe, the working point of the application has to be identified, and the design should be made to have this working point within the heat transfer limits map.

![Heat Pipe Performance Map](image)

*Figure 10: Typical heat pipe performance map [22]*

The working fluid and its intrinsic properties strongly influence many of the heat pipe’s heat transfer limitations. Thus, selecting the proper working fluid type is crucial, considering the application operating temperature range, the suitability of it with the wick and solid container materials, and also the availability and financial cost. To avoid
the need for extreme vacuum pressure that requires high vacuum equipment and introduce the risk of high stresses, it is a common practice to select the working fluid to have its saturation pressure at the required working temperature between 0.1 and 20 bar[23]. Since the operating temperature of the heat pipe under investigation in the scope of this work is 40-60 °C, and considering its low cost, and the compatibility with the copper material; water, with a useful range of [30°C – 200°C], is an obvious choice and was therefore selected for this work.

The focus of the mathematical modeling of the current investigation is primarily on the effective thermal conductivity and temperature drop of heat pipes. The effective thermal conductivity is directly related to the heat transfer capability of the heat pipes. Though the characteristic limitations of heat pipes are of great importance, the resulting thermal conductivity of the heat pipes takes precedence. Only after the mathematical modeling is accomplished, the limitations are looked at, specifically the capillary and boiling limit.

The solution proposed in this study, using an annular disc-shaped heat pipe condenser, was introduced in our previous work [31]. Figure 11 depicts the proposed condenser design. On this design, the turbine exhaust steam extracts the heat energy to the atmosphere air through a series of annular disc-shaped heat pipes staked on the steam
pipe outer surface. In this configuration, the evaporator section is the inner wall of the heat pipe, and the condenser section is the outer surface of the heat pipe.

![Diagram](image-url)

**Figure 11: Proposed Air-cooled condenser design using annular disc-shaped heat pipes**

Figure 12 shows the cross-section view of the proposed heat pipe design. The heat energy extracted from the exhaust steam through the tube wall vaporizes the water in the evaporator wick region. The generated vapor pressure drives the vapor radially to the colder region where it condenses on the top and bottom surface of the disc, releasing its latent heat to the heat pipe container solid wall. The heat is then conducted through the condenser top and bottom walls to the external cooling air passing in between the staked heat pipes. The condensed liquid in the top and bottom wick radially flows back to the
evaporator side by the act of the wick structure capillary force to passively close the operation cycle. The gravity forces drive some of the condensed liquid from the top wick to fall into the bottom side wick. However, the total flow of the liquid should all go back to the center evaporator. The evaporator surface is also surrounded by a wick structure to ensure the distribution of the condensed liquid and to use the evaporator surface effectively.

The design also proposing enhancement of the external air-side heat transfer by adding dimples features to the outside surface, the dimples increase the convection heat transfer area between the heat pipe and the external airflow, which increase the convection heat transfer rate.

The proposed material for the heat pipe external wall is plastic for the advantage of weight, cost, manufacturing, and maintenance. Even though plastic materials have a relatively weak conductance, the external wall has a small thickness, which makes the wall thermal resistance relatively very small in this case.
To increase the condensation heat transfer rate on the steam side, the study proposing using a super-hydrophobic coating on the inner side of the turbine exhaust tube. The hydrophobic coating prevents the tube’s internal surface from getting wet, which changes the condensation mode from film to dropwise condensation. This increases the heat transfer coefficient by order of magnitude, which significantly reduces the inner tube condensation thermal resistance in comparison to the other thermal resistances in the thermal path; therefore, reliable correlations for the condensation process are not needed [55].

As discussed in Chapter 3, the thermal performance of the heat exchanger is strongly affected by the contact (welding) resistance between the fins and the steam tube.
external wall. Mainly due to its relatively high thermal resistance and the high cost of advanced technologies. The proposed design of the heat pipe heat exchanger, as illustrated in Figure 11, eliminates the contact resistance—the annular disc-shaped heat pipes made of molded plastic in long arrays. The heat exchanger is assembled by sliding these arrays over the steam tubes. The temperature deviation along the steam tube creates a pressure difference between the staked heat pipes, for that, O-rings are used to isolate the heat pipes to prevent the working fluid from moving in between the ADHP. This study considers this conceptual design. However, a more detailed study is suggested for manufacturing considerations such as the selection of container material, wick structure type, and working fluid. And to consider the assembly of the heat exchanger as a complete unit. This study provides comprehensive data for the conceptual design validation that can be used for heat pipe optimization study.

**Heat Pipe Thermal Resistance Analysis**

The thermal resistance model of the disc-shaped heat pipe is developed to predict the thermal performance of the proposed design. From the literature, the most meaningful metric quantifying the heat pipe performance is the effective thermal resistance or effective thermal conductivity. The presented analysis considers all the major thermal resistances along the path of the heat transfer in the heat pipe. The
performance of a heat pipe operating under the maximum overall heat transfer rate can be characterized by the total thermal resistance [24, 56-59]. Hence, the heat pipe overall heat transfer rate can be defined as:

\[
Q = \frac{\Delta T}{R_{\text{tot}}} \tag{27}
\]

Where \(\Delta T\) is the overall temperature difference between evaporator, turbine exhaust steam in this application, and the external cooling air that is passing over the condenser surface, and \(R_{\text{tot}}\) represents the total thermal resistance between the two temperatures.

Figure 13 illustrates the thermal resistance network for the annular disc-shaped heat pipe proposed in this study. The figure shows a cross-section view of half of the disc-shape. The relative sizes of the heat pipe regions in the schematic are adjusted to present it clearly. In this model, the conduction heat transfer in the radial direction through the solid container and the wick structure, \(R_{10}\) and \(R_{11}\), respectively, are neglected. This assumption is reasonable in our design due to the Viton insulation placed between the evaporator and condenser. And considering the small contact area between the condenser and evaporator wick.
Figure 13: Annular disc-shaped heat pipe thermal resistance map.

The description of each thermal resistance is listed in Table 2, along with its comparative value for typical heat pipe [60]. The overall thermal resistance is obtained from:

\[
R_{tot} = \sum R_i 
\]  

(28)

\(R_i\) is the thermal resistance of the heat pipe components and calculated using equations (29) to (39).

- Inner tube condensing resistance (evaporator heat source)

\[
R_1 = \frac{1}{h_e A_e} 
\]  

(29)
Where $h$ is the convection heat transfer coefficient of the evaporator, and $A$ is the surface area. The evaporator heat source could be in any form of heat input. For the proposed annular disc-shaped heat pipe, the evaporator heat source represents the steam condensing on the inner side of the steam tube.

*Table 2: Comparative values for heat pipe thermal resistances*

<table>
<thead>
<tr>
<th>Thermal Resistance</th>
<th>Description</th>
<th>Comparative values [°C/W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>R1 and R9</td>
<td>Convection between heat source/sink and heat pipe wall</td>
<td>$10^{-3}$ to $10^{+1}$</td>
</tr>
<tr>
<td>R2 and R8</td>
<td>Evaporator and condenser wall resistance (normal to the surface)</td>
<td>$10^{-1}$</td>
</tr>
<tr>
<td>R3 and R7</td>
<td>Evaporator and condenser wick resistance (normal to the surface)</td>
<td>$10^{-1}$</td>
</tr>
<tr>
<td>R4 and R6</td>
<td>Resistance of the liquid-vapor interface.</td>
<td>$10^{5}$</td>
</tr>
<tr>
<td>R5</td>
<td>Radial resistance of the vapor</td>
<td>$10^{4}$</td>
</tr>
<tr>
<td>R10</td>
<td>Wick thermal resistance in the radial direction</td>
<td>$10^{2}$</td>
</tr>
<tr>
<td>R11</td>
<td>Wall thermal resistance in the radial direction</td>
<td>$10^{4}$</td>
</tr>
</tbody>
</table>
• Tube solid-wall resistance

\[ R_2 = \frac{\ln(\frac{r_w}{r_e})}{2\pi t_{evap} k_s} \]  

(30)

With \( t_{evap} = 2t_v + 2t_w \) is the total height of the evaporator surface exposed to the wick. And \( k_s \) is the solid’s thermal conductivity.

• Evaporator saturated-wick resistance

\[ R_3 = \frac{\ln(\frac{r_v}{r_w})}{2\pi t_{wick} k_{eff}} \]  

Where \( t_{wick} \) is the height of the evaporator wick (\( t_{wick} = t_{evap} \)). And The effective thermal conductivity of the saturated wick, \( k_{eff} \), depends on the wick type. For the wire screen mesh, the expression derived by Rayleigh [61] is used, equation (32):  

\[ k_{eff} = \frac{k_l[k_l + k_s - (1 - \varepsilon)(k_l - k_s)]}{k_l + k_s + (1 - \varepsilon)(k_l - k_s)} \]  

(32)

Where \( k_l \) and \( k_s \) are the liquid and solid thermal conductivity, respectively. \( \varepsilon \) is the porosity of the wick structure and is calculated by

\[ \varepsilon = 1 - \frac{\pi SN d_w}{4} \]  

(33)

\( S \) is the crimping factor (\( S = 1.05 \)) presented by Chi [62]. \( N \) is the mesh number, and \( d_w \) is the wire diameter of the screen mesh.
And for the metal fiber wick using the expression derived by Mantle and Chang [63] equation (34)

\[ K_{eff} = K_s \left[ 1 + \frac{\varepsilon}{(1 - \varepsilon)/m + [k_s/(k_l - k_s)]} \right] \]  

(34)

\[ m = \left[ 1.2 - 29 \left( \frac{d}{l} \right) (0.81 - \varepsilon)^2 + 1.09 - 2.5 \left( \frac{d}{l} \right) \right] \]  

(35)

Up to our knowledge, there is no reported expression to estimate the porosity of fiber metal wick; however, there are some reported experimental values [63]. In this study, the value reported by the manufacturer is used.

- Vapor resistance

\[ R_s = \frac{T_v \Delta P_v}{RQ \rho_v} \]  

(36)

The vapor resistance is dependent on the vapor pressure drop. And as the heat pipe is working just below the limits; The vapor pressure drop in the heat pipe under investigation is in the range of 1%. Hence the vapor resistance was neglected in this analysis [64, 65]

- Condenser saturated-wick resistance

\[ R_{7B} = R_{7T} = \frac{t_w}{A_c k_{eff}} \]  

(37)

In Figure 13, the thermal resistances 6 to 9 are split for top and bottom surfaces. \( A_c \) is the condenser top or bottom surface area.
• Condenser solid-wall resistance

\[ R_{8B} = R_{8T} = \frac{t_s}{A_c k_s} \]  

(38)

• Condenser-ambient resistance

\[ R_{9B} = R_{9T} = \frac{1}{2h_\infty A_c} \]  

(39)

The external heat transfer coefficient is a function of the cooling medium and the condenser shape. Equation (13) is applicable, assuming the proposed disc-shaped heat pipes are aligned linearly over a steam tube. However, for our investigation, the heat transfer coefficient was obtained using a three-dimensional model of the designed heat pipe inside the wind Tunnel, the heat pipe design and wind tunnel are described in detail in the Experimental Setup chapter. The problem was simulated using the commercial CFD software STAR CCM+. The condenser boundary was model with total heat input equal to the heat input used in the experimental procedure. Figure 14 & Figure 15 show the condenser external face velocity and heat transfer coefficient under 600 w total heat input and 42 m/s inlet velocity. With the assumption of symmetry conditions on the top and bottom condenser surfaces.

There is a stagnation region at the beginning of the heat pipe wall due to the large thickness of the heat pipe’s sidewall (1.5 in) that facing the airflow. The spots of
separation and recirculation at the front side of the heat pipe is very noticeable. The large insulation cylinders used on the top and bottom caused a high circulation behind it. The effect is also apparent by looking at the heat transfer coefficient distribution over the surface. However, by analyzing the affected regions in Figure 15, region (1) is the area occupied by the insulated sidewall with 1.5 in thickness, and no heat transfer is expected to be transferred from this region. Region (2) is on the backside of the 2 in insulation cylinders. And it has a thickness less than the diameter of the cylinders. This penalty is accepted, as it also increases the heat transfer coefficient on the area further away from the evaporator centerpiece. Finally, the calculated average heat transfer coefficient values over the condenser surface are presented for a range of total heat input in Figure 16.

Figure 14: CFD velocity distribution over the annular-disc shaped heat pipe
Figure 15: CFD - Heat transfer coefficient over the condenser face

Figure 16: Calculated average heat transfer coefficient over the condenser external surface

Note that the resistance values of top and bottom condenser sides are equal under the assumption of symmetric conditions, which means having an equal temperature of
the top and bottom surfaces. By comparing the values in Table 2, the heat conduction between the solids of the evaporator and condenser was neglected \((R_{11} = 0)\), same assumption valid for the conduction between the evaporator and condenser wicks \((R_{10} = 0)\).

Applying the simplification assumptions stated; using equations \((29) - (39)\) back in equation \((28)\) the total thermal resistance of the pipe can be written as:

\[
R_{tot} = R_1 + R_2 + R_3 + R_{cond} + R_9
\]  
\((40)\)

\(R_{cond}\) is the total resistance of the condenser wick and wall on the top and bottom sides. 

\[
\frac{1}{R_{cond}} = \frac{1}{R_{7T} + R_{8T}} + \frac{1}{R_{7B} + R_{8B}}
\]  
\((41)\)

With the assumption of symmetry conditions on the top and bottom sides:

\[
R_{cond} = \frac{R_{7T} + R_{8T}}{2}
\]  
\((42)\)

The theoretical overall thermal resistance can be calculated by substituting equations \((30 - (31))\ & (35 - (39))\ into equation \((40)\)

in our experimental work, presented in Chapter 5, the temperature measured inside the centerpiece evaporator cylinder, so \(R_1\) is not included in the analysis. And to investigate the influence of the external heat transfer rate, the heat pipe effective thermal
resistance is identified by excluding $R_1$ and $R_9$ from equation (40). As illustrated in Figure 17.

![Diagram of heat transfer](image)

*Figure 17: Effective thermal resistance of the ADHP*

**Heat Transfer Limitations**

Although heat pipes devices are known for being very efficient heat transfer conductors, its performance and operation constrained with several heat transfer limitations [23]. The heat pipe limitations depend on various parameters, such as size and shape of the heat pipe components, working fluid type, material and parameters of wick type, and working fluid [55]. Physical phenomena that may affect the heat pipe’s heat transfer performance include capillary forces, choked flow, initial boiling, and interfacial shear [23, 66].
Capillary Limitation

The heat pipe performance is solidly dependent on its design shape, wick and case material, and the type of working fluid. However, its operation is governed by the capillary pressure difference across the liquid-vapor interfaces in the evaporator and condenser [24]. The capillary limit is one of the significant parameters that affect heat pipe performance. Usually, it is a significant heat transfer limiting factor, especially in low-temperature working heat pipes [26]. The stable working fluid circulation in a heat pipe is achieved through the capillary pressure head developed by the wick structure. For properly and steady operation of a heat pipe, the net capillary pressure head must be higher than the summation of all the pressure losses occurring on the liquid and vapor flow paths in the wick and vapor regions. The heat pipe pressure balance is expressed as

\[(ΔP_c)_{max} ≥ ΔP_v + ΔP_l + ΔP_{e,δ} + ΔP_{c,δ} + ΔP_g\] (43)

Where

\[(ΔP_c)_{max}\] = Maximum capillary pressure differences the wick structure can generate.

\[ΔP_v\] = Total vapor pressure drop occurs in the vapor region

\[ΔP_l\] = Total liquid pressure drop occurs in the vapor region
\[ \Delta P_{e,\delta} = \text{The pressure drop across the phase change in the evaporator} \]

\[ \Delta P_{c,\delta} = \text{The pressure drop across the phase change in the condenser} \]

\[ \Delta P_g = \text{Gravitational forces.} \]

Figure 18 shows a schematic of the vapor and liquid pressures along the heat pipes. The effect of the gravity forces’ presence is very noticeable in the liquid pressure. The gravity pressure may have a positive or a negative sign, depending on the heat pipe orientation and the evaporator’s location as above or below the condenser, and it can be expressed as

\[ \Delta P_g = \rho_l g L_t \sin \phi \]  \hspace{1cm} (44)

Where \( \phi \) is the heat pipe inclination angle from the horizontal axis; thus, this pressure drop is negligible when using the heat pipe in a horizontal orientation. The vapor pressure drop in the condenser and evaporator arises from the friction and inertia forces, which provide some pressure recovery in the condenser as the mass gradually reduce through condensation.
The maximum capillary pressure is derived from the Young-Laplace equation, which is considered the fundamental equation for capillary pressure \[\Delta P_{c,\text{max}} = \frac{2\sigma}{r_{\text{eff}}}\] (45)

Where \(r_{\text{eff}}\) is the effective capillary radius. Faghri [23] has proposed empirical expressions for the wick effective radius and permeability. The expressions for the wick types used in this study are listed in Table 3. It must be mentioned here that it is recommended to obtain the values for the wick properties experimentally. However, that type of experiment is out of this study scope.
**Table 3: Expressions for wick effective capillary radius and permeability**

<table>
<thead>
<tr>
<th>Screen type</th>
<th>Capillary radius</th>
<th>Permeability</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wire screen (Type 1)</td>
<td>( \frac{1}{N} ) or ( \frac{d + w}{2} )</td>
<td>( \frac{d^2 \varepsilon^3}{122(1 - \varepsilon)^2} )</td>
<td>N = mesh number</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>d = wire diameter</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>w = opening size</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>\varepsilon = porosity</td>
</tr>
<tr>
<td>Fiber metal (Type 2)</td>
<td>( \frac{d}{2(1 - \varepsilon)} )</td>
<td>( \frac{A(X^2 - 1)}{X^2 + 1} )</td>
<td>d = fiber radius</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>A = 6 \times 10^{-10} \text{ m}^2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>( X = 1 + \frac{Bd^2 \varepsilon^3}{(1 - \varepsilon)^2} )</td>
<td>B = 3.3 \times 10^7 \text{ m}^2</td>
</tr>
</tbody>
</table>

As been discussed, the capillary limit is constrained by the maximum pumping capability of the wick structure. For the heat pipe to run under steady-state, it requires a continuous flow of the condensed flow from the condenser region to the evaporator. At a specific limit, the evaporation rate exceeds the amount of mass flow return, which causes an increase in the evaporator region and the dry-out of some spots. This limit strongly depends on the characteristics of the used wick. i.e., type, porosity, material, and permeability.

Figure 19 shows the variation of the maximum capillary pressure, permeability, and effective thermal conductivity with the mesh number of a copper screen wick.
Assuming working fluid at 30°C, such as $\sigma = 7.12 \times 10^{-3}$ N/m, a list of the calculated values is presented in Table 4.

<table>
<thead>
<tr>
<th>Mesh number [in]</th>
<th>Porosity []</th>
<th>k_eff [W/m.k]</th>
<th>Permeability [m²]</th>
<th>R_eff [m]</th>
<th>Max capillary [Pa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>7.94E-01</td>
<td>9.41E-01</td>
<td>3.89E-08</td>
<td>1.27E-03</td>
<td>1.13E+02</td>
</tr>
<tr>
<td>16</td>
<td>7.62E-01</td>
<td>1.00E+00</td>
<td>1.35E-08</td>
<td>7.94E-04</td>
<td>1.81E+02</td>
</tr>
<tr>
<td>20</td>
<td>7.36E-01</td>
<td>1.06E+00</td>
<td>7.75E-09</td>
<td>6.35E-04</td>
<td>2.27E+02</td>
</tr>
<tr>
<td>22</td>
<td>7.28E-01</td>
<td>1.08E+00</td>
<td>6.20E-09</td>
<td>5.77E-04</td>
<td>2.49E+02</td>
</tr>
<tr>
<td>30</td>
<td>7.03E-01</td>
<td>1.14E+00</td>
<td>3.00E-09</td>
<td>4.23E-04</td>
<td>3.40E+02</td>
</tr>
<tr>
<td>40</td>
<td>6.70E-01</td>
<td>1.23E+00</td>
<td>1.46E-09</td>
<td>3.18E-04</td>
<td>4.54E+02</td>
</tr>
<tr>
<td>50</td>
<td>6.29E-01</td>
<td>1.35E+00</td>
<td>7.74E-10</td>
<td>2.54E-04</td>
<td>5.67E+02</td>
</tr>
<tr>
<td>60</td>
<td>6.29E-01</td>
<td>1.35E+00</td>
<td>5.37E-10</td>
<td>2.12E-04</td>
<td>6.80E+02</td>
</tr>
<tr>
<td>80</td>
<td>6.37E-01</td>
<td>1.32E+00</td>
<td>3.14E-10</td>
<td>1.59E-04</td>
<td>9.07E+02</td>
</tr>
<tr>
<td>100</td>
<td>6.29E-01</td>
<td>1.35E+00</td>
<td>1.93E-10</td>
<td>1.27E-04</td>
<td>1.13E+03</td>
</tr>
<tr>
<td>145</td>
<td>7.37E-01</td>
<td>1.06E+00</td>
<td>1.48E-10</td>
<td>8.76E-05</td>
<td>1.64E+03</td>
</tr>
</tbody>
</table>
Figure 19: Variation of Capillary maximum pressure, thermal conductivity, porosity, and permeability vs. the wick mesh number.

The permeability is a property of the wick material that measures its ability to transmit the liquid under an applied pressure difference. The liquid flow in the thin wick region is related to the viscous frictional forces and inertial forces. However, in the operation of low-temperature heat pipes, the viscous forces are the dominant term, which is inversely related to the wick permeability. Thus, the higher the permeability, the lower the pressure drop in the liquid flow, and the flow can be simulated by the Darcy low [62]

\[
\frac{dP_l}{dr} = \frac{\mu_l}{K} \bar{u}_l = \frac{\mu_l \dot{m}_l}{\rho_l AK}
\]  

(46)
In the low-temperature heat pipe operation, the mass flow rate can be related to the amount of heat input by the vapor’s latent heat at the heat pipe working temperature [62].

\[ \dot{m} = \frac{Q}{h_{fg}} \quad (47) \]

On the other hand, having high permeability requires a small capillary effective radius, as can be seen from expressions in Table 3, which means that increasing the wick permeability must be paid by having a reduction in the wick maximum capillary pressure.

Figure 20 shows the variation of the permeability, thermal conductivity, and maximum capillary with the change in the porosity of a copper wire screen mesh. The values calculated a screen mesh with \( d = 5.59 \times 10^{-3} \) mm, using water at 30°C as a working fluid. The calculated values are presented in
Table 5: Variation of screen wick properties with porosity

<table>
<thead>
<tr>
<th>Porosity</th>
<th>K_eff [w/m.K]</th>
<th>Permeability [m2]</th>
<th>Max capillary [Pa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05</td>
<td>22.80</td>
<td>3.54E-15</td>
<td>5.87E+03</td>
</tr>
<tr>
<td>0.10</td>
<td>11.44</td>
<td>3.16E-14</td>
<td>5.56E+03</td>
</tr>
<tr>
<td>0.20</td>
<td>5.50</td>
<td>3.20E-13</td>
<td>4.94E+03</td>
</tr>
<tr>
<td>0.30</td>
<td>3.48</td>
<td>1.41E-12</td>
<td>4.33E+03</td>
</tr>
<tr>
<td>0.40</td>
<td>2.47</td>
<td>4.55E-12</td>
<td>3.71E+03</td>
</tr>
<tr>
<td>0.44</td>
<td>2.19</td>
<td>6.95E-12</td>
<td>3.46E+03</td>
</tr>
<tr>
<td>0.60</td>
<td>1.44</td>
<td>3.46E-11</td>
<td>2.47E+03</td>
</tr>
<tr>
<td>0.70</td>
<td>1.15</td>
<td>9.75E-11</td>
<td>1.85E+03</td>
</tr>
<tr>
<td>0.80</td>
<td>0.93</td>
<td>3.28E-10</td>
<td>1.24E+03</td>
</tr>
<tr>
<td>0.90</td>
<td>0.76</td>
<td>1.87E-09</td>
<td>6.18E+02</td>
</tr>
</tbody>
</table>

Figure 20: variation of permeability, thermal conductivity, and maximum capillary pressure with the porosity of metal screen mesh.
In conclusion, it is an optimization process to select the most fitted wick type and parameters for the heat pipe design and application. In the proposed annular disc-shaped heat pipe design, the condenser cooling surface is relatively large; and the thermal conductivity was not an as significant factor in the selection process. And the large vapor region size gave us the flexibility of using multi-layers of the screen to increase the liquid flow area without affecting the vapor flow cross-section area. That decreased the viscous forces acting on the liquid flow without the compromise in the effective capillary radius of the wick.

The vapor pressure strongly depends on the vapor region design, and its effect on the total pressure drop is a case to case decision. Ababneh [67] has presented a detailed model to evaluate the effect of the vapor pressure drop over the total capillary pressure in a flat heat pipe. His analysis considered four possible models of the pressure drop, considering all possible acting forces on the model, including frictional, inertial, interface forces. Applying his model using the designed heat pipe parameters under investigation; the vapor and inertial forces can be neglected in the model of liquid flow pressure drop and the overall heat pipe pressure drop for the capillary limit calculations.
Boiling Limit

The boiling limit is known as the heat flux limit, as it defines the maximum heat flux that can be transferred through the evaporator wall. At an adequate heat flux input to the evaporator wall, the wall temperature becomes excessively high, and the nucleate boiling starts on the liquid in the evaporator wick. The vapor bubbles that form on the wall get trapped in the wick and prevent the liquid from flowing back to wet the evaporator wall, which causes hot spots on the evaporator wall and starts to dry out the evaporator wick [24]. The boiling limit is defined by the orientation and design of the evaporator. For conventional cylindrical heat pipe, where the heat input is on the radial direction normal to the evaporator wall, the boiling limit is the radial heat flux limitation compared to the axial heat flux limitation for other heat pipe limits [23]. However, the design of the annular disc-shaped heat pipe under investigation in this study makes the evaporator heat flux in the same direction as the vapor and liquid flow in the radial direction, as shown in Figure 12.

The boiling or heat flux limit determination is based on nucleate boiling theory. It is comprised of two separate phenomena, bubble formation, and the subsequent growth or collapse of the bubbles [24]. The formation of bubbles is governed by the temperature difference between the evaporator wall and the working fluid, and the number and size
of the formed nucleate sites [23, 24]. Equation (48) defines this critical temperature difference, in terms of the maximum heat flux.

\[ Q_{\text{max}} = \left( \frac{K_{\text{eff}}}{T_{\text{wick}}} \right) \Delta T_{\text{cr}} \]  

(48)

\( K_{\text{eff}} \) is the effective thermal conductivity of the saturated wick, defined by equation (32). And \( \Delta T_{\text{cr}} \) is the critical temperature difference, superheat temperature, defined by equation (49) [56].

\[ \Delta T_{\text{cr}} = \frac{T_{\text{sat}}}{h_f g \rho_v} \left( \frac{2\sigma}{r_n} - \Delta P_{c,m} \right) \]  

(49)

\( T_{\text{sat}} \) is the saturation temperature of the working fluid. \( \sigma \) is the liquid surface tension. \( \Delta P_{c,m} \) is the maximum capillary pressure difference, which is a function of both the fluid surface tension and the critical wick radius. And \( r_n \) is the critical nucleation site radius, which is assumed to be from 2.54x10^{-5} \text{ m} \) to 2.54x10^{-7} \text{ m} [68]. The critical temperature difference represents the temperature drop across the wick region to maintain the nucleation bubbles of radius \( r_n \) at the wall-wick interface.

The behavior of the established vapor bubble on the evaporator surface is dependent upon the working liquid temperature and the pressure difference across the liquid-vapor interface caused by the vapor pressure and liquid surface tension. The heat flux beyond which the bubble growth may happen is developed by using the Clausius-
Clapeyron equation to relate the pressure and temperature over the liquid-vapor interface, to perform the pressure balance on the vapor bubble [62]. The maximum heat limit for the annular disc-shaped heat pipe under investigation can be determined using equation (50)

\[
Q_{\text{max}} = \frac{2\pi K_{\text{eff}} h_{\text{evap}} T_v}{h_f \rho_v \ln \left( \frac{T_{\text{evap}}}{T_{\text{wick}}} \right)} \left( \frac{2\sigma}{r_n} - \Delta P_{c,m} \right)
\]

(50)

As shown in equation (50), the boiling limit of the heat pipe is highly dependent on the design shape of the heat pipe, wick material, and the vapor temperature, also known as the working temperature, which is strongly affected by the condenser side cooling capacity. In this study, the condenser was cooled by forced convection in the air tunnel, which simulates the real conditions of the proposed heat pipe application in air-cooled condensers. The experimental setup is explained in Chapter 5. This setup limits the capability to investigate the boiling limit by having a relatively small heat transfer coefficient on the condenser side. Typically, to measure the boiling limit, one needs to increase the heat input to the evaporator while keeping the vapor temperature constant by increasing the cooling capacity in the condenser side, by increasing the cooling medium flowrate. Figure 21 illustrates the effect of the nucleate radius value on the variation of boiling limit, represented by the maximum heat input, with the vapor temperature. The values were calculated using equation (50) for the annular disc-shaped
heat pipe design presented in Figure 13 using the parameters listed in Table 6, with thermal properties calculated at the vapor temperature.

![Figure 21: effect of vapor temperature and nucleation radius on the boiling limit](image)

**Table 6: Parameter values for the boiling limit.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_{eff}$</td>
<td>[W/cm. K]</td>
<td>2.100 E-2</td>
</tr>
<tr>
<td>$h_{evap}$</td>
<td>[cm]</td>
<td>3.175</td>
</tr>
<tr>
<td>$r_{vapor}$</td>
<td>[cm]</td>
<td>3.226</td>
</tr>
<tr>
<td>$r_{wick}$</td>
<td>[cm]</td>
<td>3.175</td>
</tr>
<tr>
<td>Mesh # (N)</td>
<td>[cm$^{-1}$]</td>
<td>57.1</td>
</tr>
</tbody>
</table>
CHAPTER 5: EXPERIMENTAL SETUP

an experimental study was designed, constructed, and conducted to investigate the characteristic thermal performance of the designed annular disk-shaped heat pipe. This chapter presents the process and stages of the experimental study through all the designing, building, and conducting phases. The main aim of the study to reduce the air-side thermal resistance of the steam condenser; the experiment is designed to measure the total thermal resistance of the heat pipe under various configurations. However, due to the cost and time required for building the heat pipes, a benchmark case is designed and conducted. The results of the experimental data are used to validate the theoretical model that is used for further investigation of the ADHP. The main design drive of the heat pipe shape is its proposed application that requires it to extract the heat from the outer surface of the steam tube. A circular disc-shaped is selected considering its low-cost manufacturing and processing. However, the idea of the annular heat pipe could be in different shapes, such as elliptic or more flattened shapes, to adapt to the steam condenser tubes. The heat pipe was extensively tested under transient and steady-state conditions. The heat pipe was air-cooled using a variable-speed blower that gave the flexibility to test the heat pipe under different airspeed conditions.
Annular Disc Heat Pipe Design

The disc-shaped heat pipe was designed for the steam condenser application. In the experimental study, the heat transfer from the steam side was simulated by adding a heat source to the disc’s inner surface. The heat pipe was air-cooled to test the design under similar real application conditions. However, this cooling method came in a compromise of the total heat transfer amount through the condenser. Hence, a large condenser surface area required to achieve considerable total heat transfer. The wind tunnel size also constrained the design of the condenser size. This study did not cover the optimization of the heat pipe surface. However, the results obtained establishes an adequate base for an optimization study for future work and development.

Heat Pipe Design Considerations

Considering the proposed application purpose of the disc-shaped heat pipe, it did not feature any adiabatic section. In the meaning of heat pipe adiabatic section purpose. The heat pipe was designed in the CATER lab, considering the below main features to allow for variation and freedom in the experiment matrix based on results analysis:

1- Flexibility of vacuuming the system to any absolute pressure.
2- Flexibility of charging the system initially and changing the charge ratio.
3- Simple assembly procedure.

4- The capability of using different types and configurations of wick structures.

5- The ability to conduct the experiment with or without the adiabatic section.

6- The ability to measure the transient internal vapor temperature and pressure.

The ability to run the heat pipe under different initial vacuum pressure gave us an understanding of the effect of the existence of non-condensable gases on the heat pipe performance. The second feature was considered to study the effect of the working fluid charging amount on the heat pipe performance and to identify the range of the optimum filling ratio. The wick material and structure have a significant effect on the heat pipe performance and heat transfer limitations, so it was essential to consider the simplicity of opening and re-assembling the heat pipe with the ability to change the wick type or thickness.

**Annular Disc-Shaped Heat Pipe Dimensions**

Figure 22 shows a cutaway view of the annular disc-shaped heat pipe. The heat pipe consists of two 110 copper plates. The plate’s dimensions are 1/8 in thickness, 2 in inner diameter, and 10.4 in outer diameter, as shown in Figure 23. The plates are sandwiched over a 1.25” thick aluminum tube with 9 in and 10.5 in inner and outer
diameters, respectively, as shown in Figure 24. The aluminum tube (sidewall) surface was anodized with a black hard-coat finish to eliminate any chemical reaction with the working fluid (water). Considering the thickness and surface coating of the sidewall, the heat transfer on the sidewall was neglected. The top and bottom plates were mounted to the sidewall with 24 of 1/4 -20 size bolts on each side, to minimize the heat transfer through the bolts Teflon washers were used to seal and lock the bolts to the copper surface. The surface connection between the condenser plates and the sidewall was sealed by a 2-271 O-ring placed in a groove located between the bolts and the tube inner wall edge.

![Figure 22: Cutaway view of the annular disc-shaped pipe](image-url)
Figure 23: Condenser copper plate dimensions

Figure 24: Sidewall of the annular disc-shaped heat pipe
A 1.5 in height and 2.5 in diameter cylindrical 110 copper block was also sandwiched between the two copper discs. The radial outer surface of the cylindrical center copper block was considered as the evaporator section, and the outside surface of two copper disc plates are the condenser surface. The two condenser copper plates, aluminum sidewall, and center copper piece were aligned to be concentric. Four 150 w cartridge heaters with 0.5 in diameter and 1 in length were inserted in the copper centerpiece in symmetrical position patterned around the perimeter of 1.26 in circle, as shown in Figure 25. Two 2 in thick Rohacell cylinders were used to insulate the top and bottom surface of the evaporator.

![Figure 25: Dimensions of the center copper cylinder (evaporator)](image)

To ensure the alignment of the condenser plates, evaporator block, and the sidewall tube, the sidewall outside diameter was 0.1 in bigger than the condenser top and
bottom plates. A tip of 0.05 in thickness of the aluminum sidewall outer diameter was extracted vertically for 0.125 in on both sides (up and down), to lock the condenser copper plates in concentric location mechanically. The condenser plates lock into the evaporator copper block over 0.25 extended step with 1/8 in depth, as shown in Figure 26. Two Viton gaskets were placed between the evaporator and condenser plates to seal the connection in between and to increase the thermal contact resistance to minimize the conduction heat transfer between the evaporator and condenser. The size of the gaskets used was 2 in, 2.5 in, and 0.031 in inner diameter, outer diameters, and thickness, respectively. Figure 27 shows the built annular disc-shaped heat pipe without the copper top plate to show the internal components of the assembly.

![Figure 26: Mechanical lock assembly of the heat pipe components](image-url)
Three different types of wick material were tested, two copper screen mesh wicks with 145 and 200 mesh numbers, and an AISI 316L stainless steel fiber structure wick. The wick material was wrapped around the evaporator block and held in place using two thin copper wires located at approximately 0.5 in from the evaporator top and bottom sides. Six acrylic posts (0.5 in x 0.5 in) were used to hold the layers of wick against the inner faces of the top and bottom condenser copper plates. The specifications of the used wick types are listed in Table 7. The effective thermal conductivity of the wick was
calculated based on the working fluid of water. The built heat pipe dimensions and materials are presented in Figure 28, Figure 29, and

### Table 8

<table>
<thead>
<tr>
<th></th>
<th>Wick #1</th>
<th>Wick #2</th>
<th>Wick #3</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Type</strong></td>
<td>Screen</td>
<td>Screen</td>
<td>Fiber</td>
</tr>
<tr>
<td><strong>Material</strong></td>
<td>Copper</td>
<td>Copper</td>
<td>SS 316</td>
</tr>
<tr>
<td><strong>Mesh number [in⁻¹] (m)</strong></td>
<td>145 (3683)</td>
<td>200 (5080)</td>
<td>-</td>
</tr>
<tr>
<td><strong>Porosity</strong></td>
<td>46%</td>
<td>35%</td>
<td>60%</td>
</tr>
<tr>
<td><strong>Permeability [m²]</strong></td>
<td>8.54x10⁻⁸</td>
<td>2.15x10⁻⁸</td>
<td>2x10⁻⁸</td>
</tr>
<tr>
<td><strong>Layer thickness [m]</strong></td>
<td>0.11x10⁻³</td>
<td>0.1x10⁻³</td>
<td>0.1x10⁻³</td>
</tr>
<tr>
<td><strong>Effective conductivity [W/m/K]</strong></td>
<td>2.07</td>
<td>2.9</td>
<td>0.46</td>
</tr>
</tbody>
</table>
Figure 28: Built heat pipe CAD

Figure 29: Cross-section A-A from Figure 28
Table 8: Material of the built ADHP

<table>
<thead>
<tr>
<th>Part</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporator</td>
<td>110 Cooper</td>
</tr>
<tr>
<td>Condenser</td>
<td>110 Copper</td>
</tr>
<tr>
<td>Sidewall</td>
<td>Anodized Aluminum (black hard coat surface)</td>
</tr>
<tr>
<td>Insulation</td>
<td>Rohacell</td>
</tr>
<tr>
<td>Working Fluid</td>
<td>De-Ionized Water</td>
</tr>
</tbody>
</table>

**Experimental Measurements**

The most common and well-referenced method to investigate the heat pipe thermal performance is by identifying the heat pipe effective thermal resistance $R_{\text{eff}}$ or the effective thermal conductance. Also, it is very informative to investigate the heat pipe maximum heat transfer limit $Q_{\text{max}}$ [23-26, 58, 59, 62, 65, 69-71]. The heat pipe thermal performance was investigated under steady and transient conditions. A data acquisition system was used to record all the readings of the thermocouples, pressure transducer, and the heaters voltages. The data acquisition system consisted of a FLUKE 2688A data logging system and three FLUKE 2680 precision analog input modules to continuously read the TCs, heaters, and pressure transducer. Each module has a capacity of 20
channels. To capture the transient heating process during the test, the DAQ was set to read 1 s\(^{-1}\) frequency. A sketch of the experimental setup is presented in Figure 37. Where \((\theta = 0)\) is the radial axis in parallel with airflow. And \((h)\) is the vertical height.

**Temperatures**

The internal evaporator temperature was recorded using ten T-type thermocouples distributed along the perimeter of 1.88 in diameter circle. Figure 30 shows the detailed location of the evaporator thermocouples. The thermocouples inserted in 1/16 in diameter holes. The 10 thermocouples were placed at different depths to check the heat distribution in the evaporator. Eight thermocouples were placed at 0.75 in depth, one at 0.5 in depth (TC# 2), and one at 1 in depth (TC# 10). Placing the eight thermocouples at 0.75 in depth makes it at the center height of the vapor region between the condenser plates. The thermocouples were placed inside the copper block to avoid disturbing the vapor flow inside the heat pipe by routing the wires in the vapor region. The distance between the evaporator surface and the location of the thermocouples was considered in the calculations assuming one-dimensional heat conduction in the radial direction by neglecting the axial heat transfer considering the insulation on the top and bottom sides and the high heat transfer rate at the evaporator surface. The orientation of
the evaporator was locked in position, making TC #1 at the center of the evaporator section facing the cooling wind direction inside the wind tunnel.

![Figure 30: Location of the evaporator thermocouples](image)

A total of 28 T-type thermocouples were used to measure the temperature variation on the condenser outer surface. The thermocouples were distributed along the radial axis over the top and bottom copper plates of the condenser. The locations of the temperature thermocouples on one of the condenser’s surfaces are depicted in Figure 31. As illustrated, the temperature variation was reordered at two angular axes locations, parallel ($\theta = 0$) and perpendicular ($\theta = 90^\circ$) to the outside wind tunnel cooling airflow.
A T-type thermocouple probe was used to monitor the vapor temperature inside the heat pipe. The thermocouple was inserted from the side wall at \((\theta = 0)\) through a \(\frac{1}{4}\) in NPT tap, as shown in Figure 32. The exact location of the sidewall tap is illustrated in section A-A in Figure 24. The vapor thermocouple position was fixed at approximately the center point of the vapor region \((r = 3\text{ in and } z = 0.75\text{ in})\), at this location, the vapor thermocouple is parallel to the condenser surface thermocouple #5 in Figure 31. The
change in the working fluid temperature over the vapor region is negligible, and the vapor region can be considered isothermal [72-74]. To validate this assumption and to study the vapor temperature variation along the radial axis, one case was repeated with fixing all the conditions and changing the vapor thermocouple locations (R= 1.5 in, R= 2.5 in, and R= 3.5 in). The results showed a maximum difference of ±1.3°C during the steady run, which falls within the thermocouple accuracy. Hence the isothermal vapor region assumption is valid in our case.

The thermocouples were calibrated against an RTD thermocouple with ±0.1°C accuracy. A separate fitting curve was generated for each thermocouple using the readings over 4 different temperatures in the range of 5°C – 160°C. In the calibration process, the maximum allowed temperature difference to the RTD was 1.5°C. The thermocouples that had higher error were replaced with new ones.

Finally, two T-type thermocouples were used to measure the air temperature at the inlet of the wind tunnel.

**Pressure**

The absolute pressure inside the heat pipe was monitored using a pressure transducer of Omega PX209-015A5V powered by a 5V VDC power supply. The transducer has a voltage reading accuracy of 0.25% (Linearity, Hysteresis, and
Repeatability). The Omega transducer was calibrated against an MKS transducer with 0.25% accuracy, to ensure the accuracy of the vapor pressure application. The calibration was done by measuring the pressure in a vapor chamber under vacuum was and increasing the pressure by heating the water inside the chamber. The measured data were fitted within 0.9% of the fitted linear equation. The calibration process described in more detail in [75, 76] The pressure transducer was inserted through the heat pipe side wall across from the vapor temperature thermocouple at $\theta = 180^\circ$. The location of the vapor pressure transducer is illustrated in Figure 32

Figure 32: location of vapor temperature and pressure measurements
Table 9: Pressure transducer calibration measured values

<table>
<thead>
<tr>
<th>Omega [Volt]</th>
<th>MKS [kPa]</th>
<th>Omega [KPa]</th>
<th>Error [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.178</td>
<td>2.863</td>
<td>2.846</td>
<td>0.569</td>
</tr>
<tr>
<td>0.178</td>
<td>2.840</td>
<td>2.865</td>
<td>0.873</td>
</tr>
<tr>
<td>0.216</td>
<td>3.610</td>
<td>3.634</td>
<td>0.654</td>
</tr>
<tr>
<td>0.228</td>
<td>3.884</td>
<td>3.890</td>
<td>0.149</td>
</tr>
<tr>
<td>0.247</td>
<td>4.281</td>
<td>4.278</td>
<td>0.070</td>
</tr>
<tr>
<td>0.285</td>
<td>5.072</td>
<td>5.063</td>
<td>0.161</td>
</tr>
<tr>
<td>0.301</td>
<td>5.403</td>
<td>5.396</td>
<td>0.128</td>
</tr>
<tr>
<td>0.347</td>
<td>6.347</td>
<td>6.345</td>
<td>0.018</td>
</tr>
<tr>
<td>0.436</td>
<td>8.185</td>
<td>8.185</td>
<td>0.003</td>
</tr>
<tr>
<td>0.456</td>
<td>8.608</td>
<td>8.605</td>
<td>0.032</td>
</tr>
<tr>
<td>0.475</td>
<td>9.014</td>
<td>9.011</td>
<td>0.037</td>
</tr>
<tr>
<td>0.563</td>
<td>10.811</td>
<td>10.818</td>
<td>0.061</td>
</tr>
</tbody>
</table>

Figure 33: Pressure transducer calibration curve vs. MKS transducer
Heaters

Four 1.0 in long AC insertion heaters were used to generate the required power as the heat source for the evaporator section. The specifications of the used heaters are listed in Table 10. Important to mention that the heater resistance listed in the table is the value listed in the heater specification. However, we have measured the exact resistance of each heater during the experiments, and the exact value was within ±1.5 Ω of the manufacture specification. The heaters were controlled using a 120 v, 20 amp VARIAC. And each heater was connected to a rheostat for fine-tuning of voltage. The maximum total heat input \( (Q_{total}) \) achieved in this test was 600 w. where heat input from each heater \( (Q_i) \) was calculated using the measured voltage \( (V_i) \) and resistance \( (R_i) \).

\[
Q_i = \frac{V_i^2}{R_i} \quad (51)
\]

\[
Q_{total} = \sum Q_i \quad (52)
\]
Table 10: Insertion heaters specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>Watt [w]</td>
<td>150</td>
</tr>
<tr>
<td>Watt density</td>
<td>Watt [w/in²]</td>
<td>226</td>
</tr>
<tr>
<td>Voltage (AC)</td>
<td>Volt [V]</td>
<td>120</td>
</tr>
<tr>
<td>Resistance</td>
<td>Ohm [Ω]</td>
<td>96</td>
</tr>
<tr>
<td>Heating element diameter</td>
<td>Inch [in]</td>
<td>0.495</td>
</tr>
<tr>
<td>Heating element length</td>
<td>Inch [in]</td>
<td>1.0</td>
</tr>
<tr>
<td>heated length</td>
<td>Inch [in]</td>
<td>0.5</td>
</tr>
</tbody>
</table>

Experimental Heat Loss

The experimental heat losses are identified as the amount of heat that does not transfer radially from the evaporator through the vapor. The heat losses were measured by measuring the heat transfer through the insulation cylinders. A total of four T-type thermocouples were used to measure the temperature in the top and bottom insulation Rohacel cylinders. Two 1/16 in diameter and 1 in depth holes were drilled at the radial out surface of each cylinder. By that, the thermocouple locations were along the center
axis of the insulation cylinders, as illustrated in Figure 34. The heat loss in the system was calculated as:

\[ Q_{loss} \% = \frac{Q_{loss}}{Q_{in}} \times 100 \]  

(53)

\[ Q_{loss} = \frac{K_{ins}A_{ins}\Delta T}{L} \]  

(54)

Where \( Q_{loss} \) and \( Q_{in} \) are the heat loss and total heat input, respectively, \( K_{ins} \) is the thermal conductivity of the Rohacel (= 0.03 w/m.k), \( A_{ins} \) is the circular cross-sectional area, and \( L \) is the vertical distance between the thermocouples. The heat losses in the system were calculated using the measured insulation temperatures in equations (53) and (54), the maximum calculated heat losses were less than 0.2% of total heat input. Thus, the heat loss to the ambient was negligible compared to the total heat input, especially at power input over 300 w, as the loss percentage of 0.1%. Figure 35 shows the calculated heat loss percentage and temperature difference in the insulation for one experimental case with heat input up to 700 w.
Figure 34: location of temperature measurement in the insulation cylinders.

Figure 35: Variation of heat loss percentage with total heat input
**Vacuum and Charging Station**

The vacuum and charging connections were completed at separate locations using the taps on the sidewall located at $\theta = 90^\circ$ and $\theta = 270^\circ$, respectively. Figure 36 shows the schematic of the charging station. The heat pipe was vacuumed using Hitachi direct drive rotary vacuum pump type 160VP with a minimum absolute pressure of $10^{-1}$ Pa. A 1/4 NPT compact backflow-prevention valve was used to connect the heat pipe to the vacuum pump with a pressure limit of 5000 psi. The simple-type charging station, Figure 36, was used to charge the working fluid into the heat pipe, the method has a direct connection between the working fluid container and the heat pipe [23, 24, 67, 70]. A Swagelok gate valve was used in between the working fluid (de-ionized water) container (100 ml graded beaker) and the heat pipe, to control the filling process of the heat pipe. The beaker was mounted 5 in vertically over the heat pipe filling point. By gradually opening the valve after vacuuming the heat pipe, the working fluid flows into the heat pipe by the pressure difference force and by the gravity force as the fluid container placed above the heat pipe.
The system was first vacuumed until the pressure inside does not change for 30 min. After turning off the pump, the gate valve to the filling container was opened and closed quickly to allow a minimal amount of water into the heat pipe, typically a small enough amount that it evaporates. Then the vacuum pump started again to pull out the entrained water along with the non-condensable gases and air remaining in the system. This procedure was repeated for few times to remove the air and non-condensable gases until a stable pressure inside the heat pipe was achieved.

*Figure 36: schematic of the vacuuming and charging station*
Test Section

The heat pipe was tested inside the vertical center of a wind tunnel with 6.5 in x 13 in the cross-sectional area. The flow is guided through a flow conditioning section with 5 layers of fine mesh screen in the front, to make sure that the airflow enters the test section was as uniform as possible.

Figure 37: CAD of the wind tunnel and test section

The boundary condition of the airflow at the test section inlet was defined by measuring the dynamic pressure using pressure pitot over the flow cross-section area at
20cm from the entry of the rectangle test section, as shown in Figure 37. The dynamic pressure data were used to calculate the mass flow rate and average velocity of the flow. First, the velocity was obtained from the measured dynamic pressure, \( \Delta P \), by:

\[
U_i = \sqrt{\frac{2\Delta p}{\rho}}
\]  
(55)

Where \( \rho \) is the air density calculated by the ideal gas law at the measured temperature assuming constant temperature over the cross-sectional area. And the mass flow rate through each cell was calculated by:

\[
\dot{m}_i = \rho U_i A_i
\]  
(56)

The sum of all the cells calculated the total air mass flow rate,

\[
\dot{m}_t = \sum \dot{m}_i
\]  
(57)

Then, the average velocity upstream was calculated from:

\[
\bar{U} = \frac{\dot{m}_{\text{total}}}{\rho * A_{\text{total}}}
\]  
(58)

The measured values and conditions are described in Appendix A. with the fan running at 1428 rpm, the average velocity calculated to be 41.96 m/s.

**Test Procedure**

In general, the heat pipe performance was tested under different configurations of changing wick type, filling ratio, condenser size, and heat input. To identify the effective
thermal conductivity or effective thermal resistance of the designed heat pipe. The general procedure for all test cases was as follow:

1. Vacuum the heat pipe.

2. Open the filling gate valve and fill the system to the required charge amount.

3. Start the wind tunnel and wait for a steady flow to be achieved.

4. Start the DAQ system.

5. Start the heaters with a total power of 40w.

6. Monitor the average temperature of the evaporator, condenser, and vapor to reach steady. (1 min with less than 0.1 temperature variation).

7. Increase the total power by 40 w. and repeat step #6.

8. Repeat step #7 till reaching the maximum total power from the heaters (600 w) or the maximum heat pipe heat limit.

9. Turn off the heaters and monitor all temperatures to reach the ambient temperature.

Data Reduction

All the temperature, pressure, and voltage readings were continuously recorded for the experimental case through all the test steps. The measured data were used to
evaluate the equivalent thermal resistance and effective thermal conductivity of the heat pipe as:

\[ R_{HP} = \frac{T_e - T_c}{Q_{in}} \]  \hspace{1cm} (59)

Where \( R_{HP} \) is the total heat pipe equivalent of thermal resistance, considering the temperature difference between the condenser and evaporator surfaces. \( T_e \) and \( T_c \) are the average evaporator and condenser temperatures, respectively.

\[ T_e = \frac{\sum T_i}{N} \]  \hspace{1cm} (60)

Where \( T_i \) is the temperature reading of thermocouples distributed inside the evaporator copper block, the top and bottom condenser temperatures were measured on two axes (\( \theta = 0 \), and \( \theta=90^\circ \)) with 5 thermocouples on each line on the top plate and 7 thermocouples on each line for the bottom plate, gives a total of four condenser average temperatures (\( T_{t,0}, T_{t,90}, T_{b,0}, T_{b,90} \)).

\[ T_{j,\theta} = \frac{\sum_i T_{j,\theta,i}}{N} \]  \hspace{1cm} (61)

Where \( j=t \) or \( b \), for top and bottom, respectively. \( N \) is the number of used thermocouples along the axis. The total average of each plate is referred to as \( T_t \) and \( T_b \) for the top and bottom condenser side, respectively. And \( T_c \) is the average of the 28 thermocouples distributed on the two sides of the condenser.
The total thermal resistance of the system considering the external condenser conviction resistance is calculated as:

\[ R_{HPT} = \frac{T_e - T_\infty}{Q_{in}} \]  

\textbf{Uncertainty Analysis}

For proper understanding and interpretation of the experimental data, a proper uncertainty analysis is required to increase the reliability of the experimental results. The uncertainty analysis consists of two types of error, random and systematic errors. The systematic error comes from the instruments used in the experiment, and usually comes due to the wrong calibration or zeroing of the device. And typically, it is repeated and carried over for each repetition of the experiment. Unlike the random error, that tends to be normally distributed around a mean. Random errors come from the randomness of the experiment, human error, or the accuracy of the used devices, which is usually provided by the device manufacturer. The random error can be reduced to a certain confidence level by the repetitive test [77] For this study, a confident interval of 95% is used.

The uncertainty associated with the temperature reading using thermocouples and DAQ equipment is ± 0.5 K. The uncertainty related to the power supplied to the
heaters is approximately ± 5%. Pressure measurement readings have an uncertainty of 4%. Using the methodology proposed by Moffat [78], the error calculated using the model of equations (63 - 66) accounts for the systematic and random errors. A detailed explanation of the model is described by Otto [79].

\[
X_i = X_{i\text{(measured)}} + \delta X_i \tag{63}
\]

\[
R = (X_1, X_2, ..., X_n) \tag{64}
\]

\[
\delta R_{X_i} = \frac{\partial R}{\partial X_i} \delta X_i \tag{65}
\]

\[
\delta R = \sqrt{\sum_{i=1}^{N} \left( \frac{\partial R}{\partial X_i} \delta X_i \right)^2} = \sqrt{\left( \frac{\partial R}{\partial X_1} \delta X_1 \right)^2 + \left( \frac{\partial R}{\partial X_2} \delta X_2 \right)^2 + \cdots + \left( \frac{\partial R}{\partial X_N} \delta X_N \right)^2} \tag{66}
\]

Where \(X_i\) is the mean of a measurable property, and \(\delta X_i\) is the uncertainty associated with the measurement. \(R\) is a result of data processing or a function that depends on several measurements. So, equation (63) shows the dependency of the resulting uncertainty on the error of each measurement used in its calculation. And the partial derivative in equation (65) represents the sensitivity of the result relative to each measurement. Finally, the total uncertainty of a result is calculated by the root sum square of all contributing measurements as shown in equation (66).
The main processed results in this study are power input as a function of measured voltage and electrical resistance, the average temperature of evaporator and condenser, the condenser heat transfer coefficient, and the effective thermal resistance of the heat pipe as a function of heat input and temperature difference.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power Input</td>
<td>±1.8 %</td>
</tr>
<tr>
<td>Temperatures</td>
<td>±2 °C</td>
</tr>
<tr>
<td>Te</td>
<td>±0.7 °C ~ ±1 °C</td>
</tr>
<tr>
<td>Te</td>
<td>±0.41 °C ~ ±1 °C</td>
</tr>
<tr>
<td>dT</td>
<td>±0.8 °C ~ ±1.4 °C</td>
</tr>
<tr>
<td>R</td>
<td>± 1.82 % dT(min)</td>
</tr>
<tr>
<td></td>
<td>± 1.80 % dT (max)</td>
</tr>
</tbody>
</table>
CHAPTER 6: RESULTS

Experimental Results

The heat pipe was tested under different configurations. The performance of the heat pipe was recorded in transient and steady-state for each total power input starting from 40 w to the maximum of 600 w, with 40 w steps. Each configuration was tested under different filling ratios of 60 ml, 90 ml, and 120 ml, referred to as A, B, and C, respectively. The experimental cases are listed in Table 12. The specifications of the wick types are listed in Table 7. Also, to understand the heat pipe response time during startup heating and cooling down, it was tested under particular total heat input for 15 min of a continuous run before cooling down back to the starting conditions of temperature and pressure.

Table 12: configuration of the experimental cases.

<table>
<thead>
<tr>
<th>Case #</th>
<th>HP container</th>
<th>Wick type</th>
<th>No. of top Wick layers</th>
<th>No. of bottom Wick layers</th>
<th>Filling Amount</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Copper</td>
<td>-</td>
<td>0</td>
<td>0</td>
<td>Empty</td>
</tr>
<tr>
<td>1</td>
<td>Acrylic</td>
<td>1</td>
<td>0</td>
<td>5</td>
<td>B</td>
</tr>
<tr>
<td>2</td>
<td>Copper</td>
<td>1</td>
<td>5</td>
<td>5</td>
<td>A, B, C,</td>
</tr>
<tr>
<td>3</td>
<td>Copper</td>
<td>2</td>
<td>0</td>
<td>5</td>
<td>A, B, C,</td>
</tr>
<tr>
<td>4</td>
<td>Copper</td>
<td>3</td>
<td>0</td>
<td>5</td>
<td>A, B, C,</td>
</tr>
</tbody>
</table>

Symbols

A = 60 ml, B = 90 ml, C = 120 ml

Wick types

Type 1 = copper screen mesh number 145
Type 2 = copper screen mesh number 200
Type 3 = stainless steel fiber wire
Repeatedly Test

Following the uncertainty discussion in Chapter 5, the repeatedly of the testing reduces the random error percentage in the measurements. Few experimental cases were repeated to achieve a reliable level of confidence in the measuring system. As a sample of the repeatedly comparison test, Figure 38 compares the transient average evaporator temperature for Case 2C (Table 12) configuration over the run of 15 min at two total power inputs of 280 w and 600 w. the figure shows a very good repeatedly of the measurements. Figure 39 shows the absolute pressure transient reading inside the annular disc-shaped heat pipe for the same case 2C at 600 w.

![Figure 38: Transient average evaporator temperature for 120 ml water filling with 5 layers of 145 copper screen mesh (Case 2C)](image)
The two figures show the response of the heat pipe temperatures and pressure to the heating processes during the startup and cooling down. During the startup heating process, the temperature response is faster than the pressure. The temperature takes about 3 min to reach the steady-state while the pressure took about 7 min to reach the flat, steady-state reading.

Figure 39: Transient absolute vapor pressure for 120 ml water filling with 5 layers of 145 copper screen mesh at 600 w (Case 2C)

To validate the repeatedly of the experimental setup, Case 2B was repeated for three runs. With disassembling and cleaning the heat pipe for each pipe run before assembling and charging it again to mitigate any systematic or human error during the
assembly. The variation of the vapor temperature and average evaporator temperature with the heat input are presented in Figure 40 and Figure 41, respectively. The deviation between the three runs was less than 8%.

Figure 40: Repeatedly test - variation of vapor temperature with heat input for 90ml water with 5 layers of 145 copper screen mesh (Case 2B).
Figure 41: Repeatedly test - variation of average evaporator temperature with heat input for 90ml water with with 5 layers of 145 copper screen mesh (Case 2B).

**Model and Experimental Validation**

To validate the experimental setup and to assure the validity of the data obtained in this study, the results must be compared to a previously established experimental data. In this study, the data were compared to the experimental results of North and Avedisian [80]. Their data were used by Zhu and Vafai [74] to validate their analytical model of an asymmetrical disc-shaped heat pipe. North has presented two experimental results for two different heat pipe designs: air-cooled manifold heat pipe and water-cooled disc-shaped heat pipe. The results showed typical behavior of the heat pipe performance. The
analysis presented in the study by North suggested that the performance curve (evaporator heat flux variation with its temperature) is split into two regions: a single-phase regime and a boiling or tow-phase regime. The single-phase regime is the starting region, where conduction is the primary dominating heat transfer mode. The second regime is where the evaporation dominates the heat dissipation process, and that is validated by the increase in the heat transfer rate with temperature variation\[80\]. This performance analysis applies to any heat pipe shape\[80\]. For this study, we have used the performance curve of the disc-shaped heat pipe presented by North for the comparison; it was selected because of the similarity in the mass and heat dissipation flow direction.

Figure 42 shows the performance curve for Case 2C and Case 3C, along with the data presented by North for 116 ml methanol filling. The tested copper disc-shaped heat pipe by North has an outer diameter of 14.9 cm and 2.54 cm thickness. The heat pipe has a complete disc-shape with a plain solid bottom and top plates. The evaporator section is part of the bottom plate with 7.62 cm. Which is different than the proposed design in this study, in which the evaporator section is the concentric part of the disc. However, as the primary heat transfer mode in heat pipes is the phase change and vapor flow, the two heat pipes should have similar performance curves. Thus, the comparison provides meaningful information for the validation purpose of the experimental setup. Another
The main difference between the two experiments is that North’s heat pipe was water-cooled comparing to the air-cooled heat pipe in this study.

Figure 42: Experimental validation of the current study with North [80].

The two cases that are presented in the comparison have similar performance curves with two identified regimes. The single-phase region of the current study (A_c) has lower heat dissipation per temperature increase comparing to the same region of North (A_N). This deviation is expected as the conduction is the primary mode in this region. In the current design, the evaporator is insulated from the condenser by Viton gasket, which limits the heat transfer and causes the temperature increase in the evaporator. It is also noted that the boiling region in North (B_N) starts at lower temperatures comparing to the
same region in the current study ($B_c$). The differences in the starting pressure explain this behavior; the lower the starting pressure, the lower the phase change starts. The heat dissipation in $B_N$ region is also higher than $B_c$. Many factors affect this slope, but the main factor would be the external condenser heat transfer coefficient. The results reported by North in [81] for the air-cooled manifold heat pipe showed that the higher airflow rate yielded lower temperatures and higher heat fluxes than the lower flow rate for the same heat pipe. And that explains the shifting of the $B_c$ slope trend line to left from $B_N$. North’s heat pipe used a 3 mm thick sintered powder copper wick on each of the disc’s inner sides. With an effective pore radius of 36 $\mu$m, 50% porosity, and $2.1 \times 10^{-11}$ m$^2$ compared to the wick properties used in this study listed in Chapter 5. However, these parameters are expected to affect the maximum heat transfer and capabilities of the heat pipe more than the steady performance. Figure 43 depicts the performance curve of the empty case in this study (Case 0 in Table 12) comparing to the empty case presented in North’s study. The curves’ linear slopes are almost identical to the slope of the single-phase region of each study, which supports the explanation provided above of the dominating of conduction mode during the starting regime.
Figure 43: Performance curve of the empty heat pipe for the current study and North [1]

In the second part of the experimental validation, the experimental condenser temperature measured at 16 points on the surface was compared to an analytical solution. The experimental case used for this comparison is the empty heat pipe case (no working fluid charged). As the heat pipe was vacuumed, the heat transfer through the evaporator surface is neglected. And as the evaporator top and bottom sides are insulated, the dominant heat transfer path was the conduction through the top and bottom copper plates. Either top or bottom plate can be modeled as an annular disc-shaped fin with a thickness equal to the condenser plate, with constant heat flux at the inner radius of the
fin, assuming the symmetry of the two plates. Figure 44 shows a schematic of the cross-section of the analyzed condenser plate.

![Figure 44: schematic of the fin analysis - top condenser cross-section](image)

The heat flux at the fin inner circle is equal to half the measured actual total evaporator heat input after excluding the losses and divided by the fin cross-sectional area at the inner radius. The fin geometry is a circular disc with an inner and outer radius as of the condenser plate dimensions illustrated in Figure 23. The governing equation for the radial fin is obtained by applying energy balance on the fin:

\[
q(2\pi r)(t) - q(2\pi)(r + \Delta r)(t) - h_\infty (2\pi r\Delta r)(T - T_\infty) = 0
\]  
(67)

Rearranging equation (67) by dividing it by \(2\pi\Delta r\), letting \(\Delta r \to 0\), and substituting Fourier’s law

\[
\frac{d}{dr} \left( r \frac{dT}{dr} \right) - \frac{h_\infty r}{tk}(T - T_\infty) = 0
\]  
(68)
Introducing:

$$\beta^2 = \frac{h_\infty}{tk}, \ Z = \beta r, \ \theta = \frac{T - T_\infty}{T_0 - T_\infty}$$

Where $T_0$ is the fin temperature at the inner surface. Then equation (68) can be written as

$$z^2 \frac{d^2 \theta}{dz^2} + z \frac{d\theta}{dz} - \beta^2 z^2 \theta = 0 \quad (69)$$

Equation (69) is a modified Bessel’s equation of zero-order, and the solution for it is

$$\theta = C_1 I_0(z) + C_2 K_0(z) \quad (70)$$

Neglecting the heat transfer through the fin tip, the boundary conditions of the fin are

$$r = r_{in}: \quad \frac{dT}{dr} = \frac{-q_{in}}{k} \quad \text{or} \quad z_{in} = r_{in}\beta: \quad \frac{d\theta}{dz} = \frac{-q_{in}}{k\beta(T_0 - T)} \quad (71)$$

$$r = r_{out}: \quad \frac{dT}{dr} = 0 \quad \text{or} \quad z_{out} = r_{out}\beta: \quad \frac{d\theta}{dz} = 0 \quad (72)$$

Solving equation (70) for $C_1$ and $C_2$ using the boundary condition in equations (71) & (72)

$$C_1 = \frac{SK_1(z_{out})}{F(z_{in}, z_{out})}, \quad C_2 = \frac{SI_1(z_{out})}{F(z_{in}, z_{out})} \quad (73)$$

Where

$$S = \frac{-q_{in}}{k\beta(T_0 - T)}, \quad F(z_{in}, z_{out}) = I_0(z_{in})K_1(z_{out}) - I_1(z_{out})K_0(z_{in}) \quad (74)$$

The analytical solution is compared to the experimental results of the evacuated run case. Where the external heat transfer coefficient was obtained from the measured average condenser temperature and heat input at each heat flux step. The condenser
temperature was measured at 16 locations on two radial axes at 0° and 90° from the cooling air direction.

![Diagram showing condenser temperature distribution at the parallel and perpendicular radial axes – analytical vs. experimental (Case 0).](image)

**Figure 45:** Condenser temperature distribution at the parallel and perpendicular radial axes – analytical vs. experimental (Case 0).

The analytical model of the annular disc fin was solved using the commercial software EES. The fin dimensions used are 1 in inner radius, 4.5 in outer radius, and 0.125 in fin thickness. Figure 45 shows the comparison between the analytical and experimental condenser temperature distribution at the two radial axes. The comparison was made for all the 15 total heat input values from 40 w to 600 w with 40 w steps. However, only three power steps presented in the figure as the results showed a similar trend for all
configurations. For further validation, the analytical solution was compared to the average condenser temperature (average of the 2 radial axes at each radius location); the results are presented in Figure 46.

![Radial condenser temperature distribution](image)

**Figure 46: Radial condenser temperature distribution of empty heat pipe – analytical vs. average experimental (Case 0).**

The analytical model successfully predicted the average condenser temperature within an error of ±2°, which is within temperature measurement uncertainty. However, the temperatures at $r = 3.75$ in on both axes were out of the range. The recorded temperature was almost constant at the initial condition, after a couple of tests, this was noticed, and the diagnosis showed that the two connections through the DAQ were loose, which gave unreliable readings for some cases.
The model was used to validate the thermal resistance model derived in Chapter 4 by obtaining the thermal resistance of the heat pipe under different filling ratios. The heat pipe thermal resistance in our study is strongly affected by external convection resistance.

Figure 47 shows the experimental and theoretical thermal resistance values of the heat pipe; the external convection resistance is excluded from the $R_{hp}$ to highlight the effect of the external heat transfer coefficient. The experimental $R_{hp}$ obtained using the average measured temperatures on the evaporator and condenser. The one-phase heat transfer regime, previously discussed in Figure 42, is apparent in the presented values. At low heat input, the thermal resistance is relatively high, which is explained by the fact that the only heat transfer path at this stage is the conduction between the evaporator and condenser, as the two-phase heat transfer process would not be started at this stage. In the heat pipe design under investigation, the evaporator is insulated from the condenser using the Viton gasket in between, and that leaves the contact area is only the 1/8 in thickness at the inner radius of the condenser plates. The developed model assumes isothermal two-phase inside the heat pipe. Hence, the developed model does not predict this region and highly underpredict the actual thermal resistance. The starting regime can be reduced by decreasing the starting absolute pressure inside and eliminate the existence.
of the non-condensable gases inside. [24]. The same curves are presented for the case of 90 ml filling of stainless-steel wick type in Figure 48.

**Figure 47:** Heat pipe effective and total thermal resistance for 120 ml charged heat pipe with N=200 copper wick

**Figure 48:** Heat pipe effective and total thermal resistance for 90 ml charged heat pipe with stainless steel wick
For further validation of the thermal resistance model, it was applied to the disk-shaped heat pipe studied by North [80]. Using North’s published results, the total thermal resistance was calculated based on the temperature difference between the cooling water average and the evaporator center point temperature. Value of 5000W/m. K was used for the external heat transfer coefficient. The use of water cooling reduced the total thermal resistance, as illustrated in Figure 49. The dimensions of the North’s heat pipe are described in detail in [81]. The model was used to calculate a single thermal resistance value, due to the lack of the varying parameters. However, the predicted value is within 5% of the obtained experimental value in the two phase regime. Finally, the results presented in this analysis solidified the level of confidence in the experimental and analytical results obtained in this study. They provided the required validation of the experimental design.
Figure 49: Thermal resistance model validation vs current experimental and North [81].

The design configuration of North’s heat pipe has the evaporator and part of the condenser acting on the bottom plate; that radically reduced the conduction thermal resistance. That explains the relatively small resistance in the one-phase regime comparing to the design proposed in this study, as can be seen in Figure 49. This result also agrees with the conduction performance curves obtained from the empty evacuated cases for both designs presented in Figure 43. A schematic of the bottom plate in North’s design is presented in Figure 50.
Figure 50: schematic of the fin analysis for North’s design - bottom condenser plate

Vapor Temperature

Following the discussion in Chapter 5, the assumption of isothermal vapor along the heat pipe radial axis is will stated in the literature [29, 82]. The vapor temperature was measured at three different locations along the radial axis to get a clear understanding of the variation on the system, as illustrated in Figure 51.

Figure 51: Temperature locations in the vapor region

The vapor temperature measured at the center height of the vapor camper, at three locations along the radial axis $r = 1.25$ in (location 1), 2.5 in (location 1), and 3.5 in (location
1). with \( r \) is the radius measured from the center of the disc heat pipe; that makes the first point at 0.25 in from the evaporator surface. The measurement is taken for the Case 6D with 600 w total heat input. Figure 52 shows the full recorded temperatures of the three locations. The same data presented over 1 min average in Figure 53.

*Figure 52: Transient vapor pressure variation over the radial axis*
The measured vapor temperatures show a good alignment between locations two and three during the run period. The measured value at location 1 is higher by 3% of the location 2 temperature in the startup and steady regions. The difference grows up to 5% at the beginning of the cooling down period, enclosed in red dashed lines in Figure 53, before it matches the other temperatures again at the close to flattening trend at the end of the cooling-down period. This error is higher than the calibrated thermocouple’s uncertainty of ±0.5. At location 1, the thermocouple is very close to the evaporator surface, that area is expected to be the driest region in the heat pipe, as not much condensing happens at the condenser plate close to the heat pipe center.
On the other hand, the thermocouples in locations 2 and 3 are in the active condenser region, and it may be affected by the condensed water drops that could fall from the top plate by the gravity forces. Figure 54 shows the top view case #1B, and the dry region can be noticed around the evaporator centerpiece. This region is not symmetrical around the center, and that’s due to the non-uniform cooling flow. It is also, noticeable that there is more condensation accumulating on the working fluid charging side of the heat pipe.

Figure 54: Top view of the heat pipe with Acrylic top plate Case 1B
To investigate the effect of the existence of non-condensable gases inside the heat pipe; the same test was repeated with different initial pressure at 1.5 kPa and 6.3 kPa by vacuuming the heat pipe and then open the air valve carefully to just increase the pressure. Worth mentioning that this process had to be repeated several times to achieve a successful case, as it was hard to control the amount of air that enters the heat pipe once the valve is slightly open. The results of the cases are compared to the theoretical saturation pressure at the measured pressure and presented in

![Figure 55: effect of non-condensable gases on the vapor region (run 1 @1.5 KPa, run 2 @6.3 KPa initial pressure)](image-url)
The evaporator temperature is very critical in defining the annular disc-shaped heat pipe under investigation. In the proposed design and application, it represents the temperature of the steam condenser inner wall, which directly defines the condensing temperature and pressure for the steam turbine. The evaporator temperature was measured at 10 locations, and the exact locations are presented in Figure 30. Thermocouples # 1, 2, and 10 in Figure 30 are used to evaluate the thermal distribution over the evaporator vertical axis. The three thermocouples are distributed at \((Z/L = 1/2, 1/3, 2/3)\) at \(\theta = 0, 10^\circ,\) and \(350^\circ,\) respectively, where \(Z\) is measured from the top evaporator surface and \(L = 1.25\) in the total height of the cooper evaporator.

*Figure 56: evaporator axial temperature distribution for case 4C*
The average temperatures of the three locations for Case 6C heat pipe are plotted against the total heat input in Figure 56. The error bars are generated from the average of the three temperatures at each heat input. It is clear from the presented data the isothermal behavior of the evaporator during the runs. However, the temperature error reaches around 1.5°C at the high-power input of more than 600 w. this increase in the temperature deviation indicates the increase in the axial conduction inside the evaporator, which is a strong indication of reaching close to the heat pipe’s heat transfer limit.

The 8 circumferential thermocouples (#1 and 4-10) are distributed at \( \theta = 0, \pi/4, \pi/2, 3\pi/4, \pi, 5\pi/4, 3\pi/2, 7\pi/4 \), respectively. The circumferential temperature distribution for the same case used in the axial temperature distribution evaluation is plotted in Figure 57.
The evaporator has an explicit isothermal behavior at power input below 300 w. However, the temperature deviation starts after that. The maximum recorded difference is ±2.5 °C. This error is due to the non-uniform cooling on the condenser surface, as it was discussed in Chapter 5. The thick sidewall and insulation cylinders cause recirculation on the surface, which creates a discrepancy in the amount of cooling between the front and back sides of the heat pipes. This explanation is justified by comparing the temperatures of the front side ($\theta = 0$, $\pi/4$, and $7\pi/4$) to the backside ($\theta = 3\pi/4$, $\pi$, and $5\pi/4$).

**Condenser Temperature**

To have the full characterization of a heat pipe thermal performance under different heat loads and to accurately test its heat transfer limitation, it is recommended...
to be tested with the capability of controlling the heat pipe working (vapor) temperature [70]. Such a test could be achieved by employing active high capacity liquid cooling methods, i.e., water or dry ice cooling. However, such working conditions does not imply the real application of the proposed design under investigation. As explained, the condenser temperature was measured on the top and bottom plates along two radial axes at $\theta = 0$ and $\pi/2$. The temperature distribution over the top and bottom condenser for the entire range of heat inputs for Case # 2D is presented in Figure 58 to Figure 61. As expected, the temperature variation starts to take flat shape the farther we move away from the evaporator. It is a justified assumption to consider the uniform temperature for the condenser plates up to a total power of 500 w. however, at higher power values, the temperature difference over the plate axis increase and reaches about 6°C. The maximum variation in the temperature happens over the top plate axis at $\theta = 0$. 
Figure 58: Radial temperature distribution of the bottom plate at $\theta = 0$, (Case 2C)

Figure 59: Radial temperature distribution of the bottom plate at $\theta = \pi/2$ (Case 2C)
Figure 60: Top plate radial temperature distribution at $\theta = 0$ (Case 2C)

Figure 61: Top plate radial temperature distribution at $\theta = \pi/2$ (Case 2C)
In general, it is noticed that the variation is higher on the axes that in line with the flow (θ = 0) comparing to the crossflow axes (θ = π/2). Figure 62 shows the average temperatures of each axis and the total average over the top and bottom plates along with the total average temperatures over the condenser considering all the surface area. The Up-and-Down method described by Dixon [83] was adopted in this study to evaluate the assumption of using the total average temperature for the condenser. The standard error of the mean for the average temperature was calculated using equation (75)

\[ SEM = \frac{SD}{\sqrt{n}} \]  

(75)

Where SD is the standard deviation of the samples, and n is the number of samples. In our experiment, n = 24 as we have 24 thermocouples distributed over the condenser plates. The calculated error for each input power is listed in Table 13. The maximum calculated at the maximum power equal to 0.96

Figure 62: Surface average temperature of the top and bottom condenser plates vs. the total heat input (Case 2C)
Table 13: Standard error of the condenser mean temperature

<table>
<thead>
<tr>
<th>Q [w]</th>
<th>Mean [°C]</th>
<th>SD</th>
<th>SEM</th>
</tr>
</thead>
<tbody>
<tr>
<td>40 W</td>
<td>19.65954</td>
<td>0.467101</td>
<td>0.208894</td>
</tr>
<tr>
<td>80 W</td>
<td>20.60837</td>
<td>0.67418</td>
<td>0.301503</td>
</tr>
<tr>
<td>120 W</td>
<td>21.65522</td>
<td>0.814823</td>
<td>0.3644</td>
</tr>
<tr>
<td>160 W</td>
<td>22.46974</td>
<td>0.868578</td>
<td>0.38844</td>
</tr>
<tr>
<td>200 W</td>
<td>23.53013</td>
<td>0.919198</td>
<td>0.411078</td>
</tr>
<tr>
<td>240 W</td>
<td>24.57882</td>
<td>0.999848</td>
<td>0.447146</td>
</tr>
<tr>
<td>280 W</td>
<td>25.50234</td>
<td>1.128819</td>
<td>0.504823</td>
</tr>
<tr>
<td>320 W</td>
<td>26.41906</td>
<td>1.274685</td>
<td>0.570056</td>
</tr>
<tr>
<td>360 W</td>
<td>27.08954</td>
<td>1.345409</td>
<td>0.601685</td>
</tr>
<tr>
<td>400 W</td>
<td>28.02874</td>
<td>1.400084</td>
<td>0.626137</td>
</tr>
<tr>
<td>440 W</td>
<td>29.24575</td>
<td>1.554618</td>
<td>0.695246</td>
</tr>
<tr>
<td>480 W</td>
<td>30.10759</td>
<td>1.684263</td>
<td>0.753225</td>
</tr>
<tr>
<td>520 W</td>
<td>30.83722</td>
<td>1.77339</td>
<td>0.793084</td>
</tr>
<tr>
<td>560 W</td>
<td>31.71099</td>
<td>1.859096</td>
<td>0.831413</td>
</tr>
<tr>
<td>600 W</td>
<td>32.39506</td>
<td>1.982404</td>
<td>0.886558</td>
</tr>
<tr>
<td>640 W</td>
<td>33.35027</td>
<td>2.036062</td>
<td>0.910554</td>
</tr>
<tr>
<td>680 W</td>
<td>34.39917</td>
<td>2.085069</td>
<td>0.932471</td>
</tr>
<tr>
<td>700 W</td>
<td>34.9005</td>
<td>2.146479</td>
<td>0.959935</td>
</tr>
</tbody>
</table>
Thermal Resistance

The effective thermal resistance of the heat pipe is the characteristic parameter for the heat pipe. Figure 63 presents the effective thermal resistance of the heat pipe measured for different wick types at the same filling ratio.

![Graph showing thermal resistance vs power for different wick types.]

*Figure 63: variation of heat pipe effective thermal resistance with wick type*

As expected, the stainless-steel wick has the highest thermal resistance, and the wick with a smaller mesh number has the smallest effective thermal resistance. The results show that the higher the total power, the lower resistance of the heat pipe. And that’s due to the response time of the heat pipe to have the full steady operation, by
thoroughly having wetted evaporator without having an excessive amount of water in
the condenser, which adds more resistance to the system.

The effect of the filling ratio on the heat pipe effective thermal resistance is illustrated in Figure 64. By comparing the three different filling ratios, Cases 3A (60 ml), 3B (90 ml), and 3C (120 ml); the small filling ratio has a higher thermal conductivity, and this increase is more evident at high heat fluxes. However, excisive filling (120 ml) also penalizes the thermal resistance at small heat fluxes. These results suggest that the optimum filling amount for the heat pipe depends on its application and the amount of heat flux applied to it.

![Graph showing ADHP effective thermal resistance at different filling ratios.](image)

*Figure 64: ADHP effective thermal resistance at different filling ratios.*
Figure 63 shows the variation of the ADHP thermal resistance with the change in the heat input. The figure shows the results for Case 1B (90 ml) and 1C (120 ml), as discussed before, the optimum filling ratio for the heat pipe depends on the heat input operating range of the application.

![Graph showing ADHP thermal resistance variation with heat input (case 1C and 1B)]

*Figure 65: ADHP thermal resistance variation with heat input. (case 1C and 1B)*
CHAPTER 7: CONCLUSION AND FUTURE WORK

A novel air-cooled heat exchanger design has been proposed as an alternative solution for the steam condenser in steam power plants. The proposed condenser design implies the high thermal conductive heat pipe technology to reduce the total thermal resistance between the internal steam and external cooling air. A novel annular disc-shaped heat pipe has been designed to fit the application over steam tubes in the condenser.

A detailed experimental setup has been designed to test the transient and steady performance of the proposed heat pipe. The thermal performance and heat transfer limitations of the heat pipe has been tested and compared to the theoretical model. The heat pipe has been tested under different configurations of wick materials and working fluid charging ratios. The experimental results showed a high impact of the external cooling heat transfer coefficient on the heat pipe thermal resistance. And the filling ratio has some influence on the heat pipe thermal performance, especially at low power input conditions. The low filling ratios also have a longer startup response time. However, excessive water in the heat pipe adds resistance to the condenser side, which also penalizes the overall thermal performance. The experimental results suggest that the optimal filling ratio for the designed heat pipe falls in the range of 7% to 10% of the total heat pipe enclosed volume.
The heat transfer limits of the system have been addressed. A detailed derivation of the boiling and capillary limits of the system have been presented. The experimental results showed that heat transfer limits are profoundly affected by the existence of non-condensable gases in the system. With 120 ml filling charge and 1.5 kPa initial vacuum pressure, no dry-out indication noticed for up to 700 w of heat input. However, an indication of dry-out was noticed above 400 w for the same configuration with 6.3 kPa initial vacuum pressure.

A 3D numerical model has been derived in the study, Appendix A. The model considers the heat transfer and hydrodynamics aspects of three regions of the heat pipe, vapor, saturated wick, and solid regions. The model assumed only single-phase flow in the vapor and wick regions. The phase change during condensation and evaporation is modeled at the wick-vapor interface. The mass change in each phase was incorporated as boundary conditions for each region. Heat source and sink were added to the wick region energy equation to account for the latent heat of evaporation and condensation. The inertial and viscous momentums in the liquid flow in the wick were considered by adding momentum sources to the momentum equation of the liquid region. A commercial software Star CCM+ was used to solve the model. The model showed a good agreement with experimental results with the advantage of simplicity and time saving compared to the modeling of the two-phase problem.
As a continuous work for this investigation, it is suggested to experimentally test different sizes of the heat pipe, considering the ration of the evaporator to the condenser surface ratio. The results of such an experiment would be needed to validate the model further.

Identifying the heat transfer limitation of the system requires the experimental setup to have the ability to control the condenser external heat transfer coefficient. Such an experiment could be achieved by using temperature-controlled water flow as a cooling medium, which would give the ability to increase the cooling capacity as increasing the heat input to the evaporator and maintain the working vapor temperature constant. The heat pipe limit could be identified by the heat input at where the evaporator temperature starts to increase while the condenser temperature remains constant rapidly.

The developed CFD solution does not consider the effect of the condenser external cooling. The experimental results showed the importance of the external heat transfer coefficient and its effect on thermal performance. The model could be improved to include condenser conditions. And to count for the non-symmetry conditions of the system.

The CFD model could be used to conduct a parametric optimization study, considering aspects of heat pipe dimensions, filling ratio, wick types, casing materials.
APPENDIX A: NUMERICAL ANALYSIS OF CONVENTIONAL HEAT PIPE
NUMERICAL SETUP

Heat pipes employ phase change as the means to store and transfer heat energy. The operational characteristics of the heat pipes make it a highly effective passive device for transiting heat over small temperature difference [23]. Heat pipes have been utilized in a wide range of application mainly in space satellites, its durability, passively operation, and high conductivity, with the capability of design variation to fit custom designs. Indeed, the most common use of heat pipes is in electronics cooling. Its high heat transfer capability, and the ability to maintain constant evaporator temperature under a relatively large range of heat flux, makes it a favorable choice as heat sinks for electronic devices [24]. The growth in heat pipe usage in a variety of industrial applications raised an essential need for more understanding of the heat pipe thermal characteristics and performance limitations [59].

The numerical and analytical modeling of the heat pipes has progressed significantly in the last few decades[59]. However, the physics phenomena involved in the heat pipe operation make it challenging to have a comprehensive, detailed solution for the system. Table 14 summarize the heat pipe operation and the interactions between the heat pipe regions [23]. Many researchers have numerically modeled the heat pipe in steady, transient, and start-up states. Garimella and Sobhan [84] have presented a review of the recent advances in heat pipe modeling. The review is focused more on the non-
conventional heat pipes and its applications. Faghri [59] presented a comprehensive review of the state-of-art works in heat pipes, covering experimental and modeling advances in the field.

The thermal fluid phenomena in a heat pipe working process can be split into four categories: (1) heat conduction in the solid outer container, (2) liquid flow in the wick region, (3) vapor flow in the core region, and (4) the interaction between vapor and wick. The first two categories have been extensively studied, many detailed analytical and numerical models have been done. However, due to the complexity of the liquid flow in the porous region, it is difficult to obtain an exact solution for the liquid flow in the wick and the liquid-vapor interface region. That requires some practical information to be extracted experimentally then used in the numerical modeling [26].

![Figure 66: Schematic of the heat pipe and the coordinate system](image-url)
Table 14: Heat pipe operation’s flow chart and its interaction between different regions.

<table>
<thead>
<tr>
<th>Heat Source</th>
<th>Heat Sink</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Conduction in heat pipe container</td>
<td></td>
</tr>
<tr>
<td><strong>Variables</strong></td>
<td><strong>Phenomena</strong></td>
</tr>
<tr>
<td>Temperature</td>
<td>Conjugate Effects</td>
</tr>
<tr>
<td>heat Input</td>
<td>Boundary Conditions</td>
</tr>
<tr>
<td>Heat Output</td>
<td>Geometry Effect</td>
</tr>
</tbody>
</table>

| Liquid Flow and Heat Transfer in Wick Structure |
| **Variables** | **Phenomena** | **Governing Equations** |
| Temperature | Phase Change | Continuity Equation |
| Velocities | Porous Media Flow | Navier-Stokes Equations |
| Pressure Drop | Interfacial Shear Stress | Averaged Energy |
| Gravity | Liquid Entrainment | Equation |
| | Boiling Limitation |

| Liquid-Vapor Interface |
| **Variables** | **Phenomena** | **Governing Equations** |
| Contact Angles | Capillary Pressure | Interfacial Mass Balance |
| Interfacial Curvature | Disjoining Pressure | Interfacial Momentum |
| Interfacial Position | Evaporation | Balance |
| Interfacial Mass Fluxes | Condensation | |

| Vapor flow in heat pipe core |
| **Variables** | **Phenomena** | **Governing Equations** |
| Velocities | Compressibility Effects | Continuity Equation |
| Pressure Drop | Continuum Flow | Momentum Equations |
| Density | Mass Diffusion | Energy Equation |
| Temperature | Geometry Effects | Mass Diffusion Equation |
| Concentration | Sonic Limitation | Equation of State |
Numerical Model

The finite volume method is used to discretize and solve the governing equations of the fluid flow and heat transfer in the heat pipe. As shown in Figure 66, the cylindrical heat pipe structurally consists of three sections: evaporator, adiabatic, and condenser sections. However, for the mathematical modeling purpose, the heat pipe problem is divided into core region (vapor), porous wick region (liquid), and the solid container region. The governing equations for each region were derived along with its boundary conditions and the coupling equations of the interfaces. The following common assumptions are made in this simulation; these are usually made in analyzing heat pipes [29, 74, 84-95]:

1. Vapor and liquid flows are steady, laminar, and subsonic.
2. Vapor flow simulated as an ideal gas.
3. Incompressible, non-Darcian transport flow in the wick region. (Wang and Ching [91])
4. Axisymmetric around the heat pipe long axis.
5. Porous wick is always saturated with liquid-phase working fluid.
7. Single-phase (liquid) flow in the porous wick region.
8. Phase change processes (i.e., condensation and evaporation) occur at the wick-vapor interface.

9. Vapor and liquid suction and injection rates are uniform over the evaporator and condenser sections.

10. Radiative and gravitational effects are negligible.

Under the stated above assumptions, the 2D steady single-phase governing equations (i.e., continuity, momentum, and energy equations.) for the vapor, wick, and solid container regions are presented below.

**Vapor Region**

The continuity equation of the vapor flow in the core region is derived from the conservation of mass of the incompressible single-phase flow in the region and represented in equation (76). The vapor conservation of momentum in the axial and radial directions are presented by Navier-Stokes equations (77) and (78). The energy equation is derived by applying the conservation of energy to the vapor region, equation (79).

\[
\frac{\partial u}{\partial x} + \frac{1}{r} \frac{\partial (r v)}{\partial r} = 0 \tag{76}
\]

\[
\rho_v \left( u \frac{\partial u}{\partial r} + u \frac{\partial u}{\partial x} \right) = -\frac{\partial p}{\partial x} + \mu_v \left( \frac{\partial^2 u}{\partial x^2} + \frac{1}{r} \frac{\partial u}{\partial r} + \frac{\partial^2 u}{\partial r^2} \right) \tag{77}
\]
\[\rho_v \left( v \frac{\partial v}{\partial r} + u \frac{\partial v}{\partial x} \right) = - \frac{\partial p}{\partial r} + \mu_v \left( \frac{\partial^2 v}{\partial x^2} + \frac{1}{r} \frac{\partial v}{\partial r} + \frac{\partial^2 v}{\partial r^2} \right) - \frac{v}{r^2} \] (78)

\[\left( v \frac{\partial T}{\partial r} + u \frac{\partial T}{\partial x} \right) = \alpha_v \left( \frac{\partial^2 T}{\partial x^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial r^2} \right) \] (79)

**Porous-Wick Region**

The governing equations of 2D steady-state of the liquid flow in the porous wick region are presented by equations (80) to (83). The continuity equation of the liquid in the porous region has the same form as the vapor region, equation (80). However, the momentum equations in the porous region have extra momentum terms to present the inertial and viscous resistance of the porous wick region, equations (81) and (82). As mentioned above, the model considers the flow in the heat pipe regions as single phases. So to account for the phase change process in the heat pipe, the latent heat of evaporation and condensation processes is modeled by adding heat source and heat sink terms in the wick condenser and evaporator regions, respectively, equation (83).

\[\frac{\partial u}{\partial x} + \frac{1}{r} \frac{\partial (ru)}{\partial r} = 0 \] (80)

\[\rho_l \left( v \frac{\partial u}{\partial r} + u \frac{\partial u}{\partial x} \right) = - \frac{\partial p}{\partial x} + \mu_l \left( \frac{\partial^2 u}{\partial x^2} + \frac{1}{r} \frac{\partial u}{\partial r} + \frac{\partial^2 u}{\partial r^2} \right) - \frac{\mu_\varepsilon}{K} u - \frac{C_F}{\sqrt{K}} |u| u \] (81)

\[\rho_l \left( v \frac{\partial v}{\partial r} + u \frac{\partial v}{\partial x} \right) = - \frac{\partial p}{\partial r} + \mu_l \left( \frac{\partial^2 v}{\partial x^2} + \frac{1}{r} \frac{\partial v}{\partial r} + \frac{\partial^2 v}{\partial r^2} \right) - \frac{\mu_\varepsilon}{K} v - \frac{C_F}{\sqrt{K}} |v| v \] (82)
\[
\left( v \frac{\partial T}{\partial r} + u \frac{\partial T}{\partial x} \right) = \alpha_{eff} \left( \frac{\partial^2 T}{\partial x^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial r^2} \right) + S_i \tag{83}
\]

**Solid Container**

The heat transfer process in the heat pipe solid container is described by pure conduction, as represented in equation (84)

\[
\left( \frac{\partial^2 T_{wa}}{\partial x^2} + \frac{1}{r} \frac{\partial T_{wa}}{\partial r} + \frac{\partial^2 T_{wa}}{\partial r^2} \right) = 0 \tag{84}
\]

**CFD Model Validation**

For the validation of the presented numerical model, the numerical simulations are performed for a 1000 mm long cylindrical heat pipe with an outer diameter, wall thickness, and wick thickness dimensions of 25.4 mm, 0.85 mm, and 0.356 mm, respectively. The heat pipe axial sections have the dimensions of evaporator endcap 10 mm, evaporator 64 mm, an adiabatic section of 606 mm, and condenser section of 296 mm, and condenser endcap of 24 mm. The model results of the outside copper wall are compared to the experimental results measured by Faghri [96] and the 3D numerical results presented by Pooyoo [90]. Figure 67. Shows the comparison between the measured and simulated results, a good agreement can be concluded from the
comparison. Ignoring the temperatures of the evaporator and condenser endcaps, the maximum deviation between the simulated and measured data is 0.65°C.

Figure 67: CFD temperature distribution validation
Table 15: Wind-Tunnel measurements

<table>
<thead>
<tr>
<th></th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
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<td>in</td>
<td>7.00</td>
</tr>
<tr>
<td>Width</td>
<td>in</td>
<td>13.00</td>
</tr>
<tr>
<td>Average temperature</td>
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<tr>
<td>kinematic viscosity</td>
<td>m²/s</td>
<td>$1.534 \times 10^{-5}$</td>
</tr>
<tr>
<td>Total mass flow rate</td>
<td>m³/s</td>
<td>2.77</td>
</tr>
<tr>
<td>Average velocity</td>
<td>m/s</td>
<td>41.4</td>
</tr>
<tr>
<td>Fan speed</td>
<td>rpm</td>
<td>1428</td>
</tr>
<tr>
<td>Reynolds number (Re)</td>
<td></td>
<td>$6.2 \times 10^5$</td>
</tr>
</tbody>
</table>

Figure 68: Wind-Tunnel air-flow measurement grid
Figure 69: Wind-tunnel local traversed mass flow

Figure 70: Wind-tunnel local traversed mass flow
LIST OF REFERENCES


