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In Tube Evaporation of a Low GWP Zeotropic Mixture for Navy Chiller Applications

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ABSTRACT

This work analyzes the evaporative heat transfer characteristics of the low global warming potential (GWP) zeotropic refrigerant mixture, R471A, in a horizontal smooth tube. R471A has recently been suggested as a replacement for conventional HFC refrigerants in Naval cooling applications due to its low GWP and comparable thermophysical properties. However, phase change of zeotropic mixtures introduces mass transfer resistance, which can reduce heat transfer performance and make predictions complicated. Currently, there is limited experimental data for zeotropic mixtures in evaporating flow, motivating the need for this study. To determine the most accurate modeling method for zeotropic mixtures, the results from this work were compared with predictions from established heat transfer correlations from literature. Correlations developed by Gungor-Winterton, Guo, Mishra, Thome-GW87, and Zhang were chosen, with Gungor-Winterton's predicting the most accurately with a Mean Absolute Percentage Error (MAPE) of 27.30%.

Experiments were conducted in a 4.62 mm inner diameter and 6.35 mm outer diameter copper tube for dew point temperatures of 22°C and 33°C and mass fluxes of $300\frac{kg}{m^2s}$ and $500\frac{kg}{m^2s}$. The refrigerant entered the test section as a subcooled liquid and exited as a superheated vapor, with counterflow ethylene-glycol/water in the surrounding annulus with an inner diameter of 14.73 mm and an outer diameter of 21.59 mm. Measured temperatures, pressures and flow rates were used to calculate heat transfer coefficients along the length of the tube during evaporation.

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NOMENCLATURE

Symbol	Description	Unit
B	Scaling factor	-
Bo	Boiling number	-
D	Tube diameter	m
F	Enhancement factor	-
F_c	Thome's mixture correction factor	-
Fr	Froude number	-
G	Mass flux	$\frac{kg}{m^2 s}$
h_{fg}	Latent heat of vaporization	$\frac{J}{kg}$
k	Thermal conductivity	$\frac{W}{mK}$
M	Molecular weight	$\frac{kg}{mol}$
P	Pressure	kPa
Pr	Prandtl number	-
P_r	Reduced pressure	-
q	Heat flux	$\frac{W}{m^2}$
Re	Reynold's number	-
S	Suppression factor	-

T	Temperature	K
T^*	Zhang's ratio of temperatures	-
x	Quality	-
X_{tt}	Martinelli number	-
α	Heat transfer coefficient	$\frac{W}{m^2K}$
β	Mass transfer	$\frac{m}{s}$
μ	Dynamic viscosity	Pa s

Subscript	**Description**
b	Bubble point
cb	Convective boiling
crit	Critical
d	Dew point
id	Ideal
l	Liquid
lo	Liquid only
mix	Mixture
nb	Nucleate boiling
sat	Saturation

v	Vapor

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Chapter 1 Introduction

Motivation

Refrigerants are fluids that evaporate and condense in vapor compression based heating and cooling systems including air conditioners, refrigerators, heat pumps, industrial chillers, and mobile climate control systems. When vapor compression systems were first introduced, so-called natural refrigerants such as carbon dioxide, ammonia, ether and hydrocarbons were used. In the 1940s and 1950s, a new class of fluorinated refrigerants, chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs), were developed that could operate with increased efficiency and safety. However, it soon became apparent that these fluids had a significant ozone depletion potential (ODP), so a global phasedown of CFCs and HCFCs was initiated by the 1987 Montreal Protocol [1].

A new category of fluorinated refrigerants, hydrofluorocarbons (HFCs) were heralded for their negligible impact to the depletion of the ozone layer but are now diminished for their high global warming potential (GWP). Examples of common HFCs include R134a, a once universal refrigerant in everything from automative air conditioners to refrigerators and large centrifugal chillers. However, the 2016 Kigali amendment to the 1987 Montreal Protocol is again driving a transition to new types of refrigerants in response to their negative environmental impact [1].

GWP is a measure of a compound's potential to trap heat in the atmosphere over some time horizon when compared to carbon dioxide, which has a GWP of 1. High GWP refrigerants, like R134a with a GWP of 1430, continue to be used for their high cooling efficiency needed in commercial and industrial applications. However, the shift towards more environmentally friendly refrigerant mixtures requires low GWP refrigerants to be investigated to ensure they maintain efficient and reliable operations. Hydrofluoroolefins (HFOs) are rapidly gaining acceptance in the heating, ventilation, air conditioning, and refrigeration industry (HVAC&R). These fluids have comparable thermophysical properties to HFCs with a lower global warming potential. However, they are often considered mildly flammable (A2L rating according to ASHRAE), which increases safety concerns.

To decrease the GWP and flammability risk, one option is to use mixtures of refrigerants. Thus, the focus of this study is the zeotropic refrigerant mixture, R471A. R471A is a mixture of HFOs R1234ze(E) and R1336mzz(E) (78.7/17.0 weight%) and HFC R227ea (4.3 weight%) with an overall GWP of 148 [2]. The HFO components contribute to the low GWP, while the small fraction of HFC enhances its heat transfer performance and decreases the flammability risk.

The phase change heat transfer behavior of this fluid must be understood to design heat exchangers in next generation equipment. Currently, there is limited data on the in-tube evaporative behavior of low-GWP zeotropic refrigerant mixtures. This study addresses this gap by investigating the convective evaporative heat transfer characteristics of R471A in horizontal tubes.

Navy Chillers

Centrifugal chillers play a critical role in maintaining thermal control on United States Navy vessels, supporting essential machinery and systems, as well as the comfort of the crew. These systems provide cooling for a variety of applications on the ship, such as propulsion machinery, living quarters for the sailors, and refrigerated storage. The need for reliable, energy-efficient cooling solutions is heightened by the environment aboard Naval ships, where space is limited, and equipment is exposed to extreme conditions.

The refrigeration systems aboard Navy ships utilize a water-cooled vapor-compression cycle, which is a common method for transferring heat. In the US, the most common refrigerant used is R134a, an HFC that is subject of global phase down. US Navy chillers are designed to efficiently transfer heat in a compact package with a long service life. Chilled water is supplied to various ship processes, and heat is rejected to the ocean [3]. However, these systems can be costly and require a lot of space which is limited on a Navy ship. Furthermore, there are concerns about using mildly flammable HFO refrigerants that have gained acceptance in the commercial sector.

The chillers used in ships are based on the vapor-compression cycle, which consists of four main components: the compressor, condenser, expansion valve, and evaporator [4]. The refrigerant circulates through the system, absorbing and releasing heat as it undergoes phase change between liquid and vapor. A separate chilled water loop transfers the heat to the refrigerant through a heat exchanger. A schematic of this system can be seen in Figure 1.

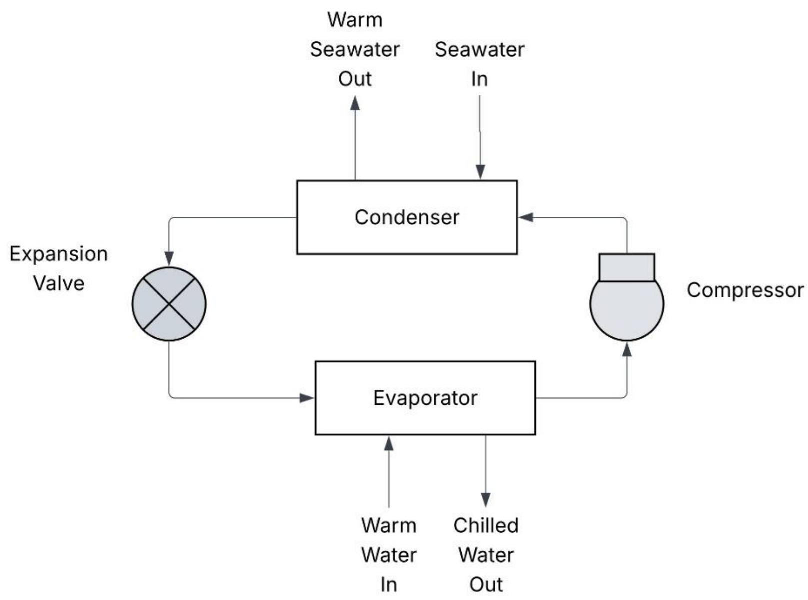

Figure 1: Simple schematic of water-cooled vapor-compression cycle aboard Navy ships.

The cycle begins when the refrigerant enters the compressor. The compressor's role is to increase the pressure of the refrigerant, which raises its temperature and produces a high pressure vapor. High temperature is needed so heat transfer can occur with the cooling water in the next component of the cycle, the condenser.

The high temperature refrigerant flows into the condenser, where it is cooled by the surrounding environment (seawater) and releases the heat it absorbed from the evaporator. As the refrigerant loses heat, it condenses from a high-pressure vapor into a high-pressure liquid. The condenser facilitates the phase change by transferring the heat from the refrigerant to the environment.

Next, the high-pressure liquid refrigerant is passed through the expansion valve, a crucial component that controls the flow of refrigerant between the condenser and the evaporator. The expansion valve causes a drop in pressure, and as the refrigerant expands, its temperature also decreases, and it flashes to a two-phase liquid/vapor mixture.

Finally, the low pressure, low temperature refrigerant enters the evaporator, where it absorbs heat from what the chilled water loop collects from the environment. The heat absorption causes the refrigerant to evaporate, leading to a cooling effect. This is important for the overall performance of the system, as the efficiency of heat absorption during the evaporation process influences the cooling capacity of the chiller system.

Zeotropic Mixtures

Temperature glide, a range of temperatures at which a refrigerant evaporates and condenses within a cycle at a constant pressure, determines how a refrigerant is categorized. Pure substances exhibit no temperature glide, since they evaporate and condense at a single saturation temperature. Azeotropic refrigerant mixtures behave similarly, in which the mixture evaporates and condenses at the same temperature, the saturation temperature [5]. However, a refrigerant can also be categorized as a zeotropic mixture. A zeotropic mixture has components that have different volatilities, causing them to evaporate and condense over a range of temperatures, yielding a temperature glide [5]. This is because each component of the mixture has its own boiling point and dew point.

A temperature glide is measured from the dew point of the mixture, where its vapor phase first condenses into a liquid, to the bubble point of the mixture, where its liquid phase first evaporates into a vapor [5]. Depending on what is present in the mixture and its composition, it determines the saturation temperatures seen in zeotropic mixtures. The presence of a temperature glide can have a negative effect on the heat transfer due to the introduction of sensible heat transfer and mass transfer resistances [6].

When a liquid zeotropic mixture is heated to its bubble point, the more-volatile component evaporates preferentially. Over time, an accumulation of the less-volatile component results in the liquid side, while the vapor phase contains a high concentration of the more-volatile component. This separation creates a concentration gradient at the liquid-vapor interface. For the less-volatile component to evaporate, it must travel through the interface, and as a result, mass transfer resistances must be considered [7]. This is ultimately responsible for the degradation in heat transfer associated with zeotropic mixtures, compared to pure substances, since the evaporation rate is adversely affected [7].

Thesis Organization

This thesis is composed of the following chapters:

- Chapter 2 reviews previous work on the evaporation of zeotropic refrigerant mixtures in horizontal tubes. Common experimental approaches are introduced, emphasizing differences in methodology. Models and correlations of the heat transfer of zeotropic mixtures are also reviewed. Existing research gaps that motivate this study are mentioned as well.
- The experimental setup used and approach to conduct this study is detailed in Chapter 3. Data analysis methodology is defined for calculating heat transfer coefficients.

- Chapter 4 presents the evaporation heat transfer results from experiments. Trends in the data are explored. The measured data is then compared to existing models from literature to determine predictive performance.
- Chapter 5 presents the conclusions from this investigation and ways to further research in this field.

Chapter 2 Literature Review

Many experimental studies have been conducted to understand the heat transfer performance of zeotropic refrigerant mixtures. Early investigations focused on improving experimental setups to get the best performance from these mixtures. The experimental setup of a heat exchanger generally consists of a compressor, condenser, expansion valve, evaporator and measurement instruments such as flow meters and thermocouples. Variables such as tube diameter and their internal surface structure can be adjusted to evaluate their influence on heat transfer performance.

This review focuses on the evaporation of zeotropic refrigerant mixtures in horizontal tubes. It begins with an overview of past experiments involving zeotropic mixtures in horizontal tubes, followed by a discussion of the limited studies on R471A. This review also covers numerical modeling efforts that focus on evaporation in horizontal tubes and concludes by identifying current gaps in the literature.

Experimental Work

Several experimental testing parameters influence the evaporation performance of zeotropic mixtures. Key parameters include heat flux, mass flux, pressure, tube diameter, and mixture composition. Uchida *et al*. studied the condensation and evaporation characteristics of multiple zeotropic mixtures in smooth, grooved, and cross-grooved horizontal tubes [8]. Their focus was on how the tube's surface affected heat transfer efficiency. Their results showed that cross-grooved tubes delivered the best heat transfer performance in both condensation and evaporation, with coefficients up to three times higher than smooth tubes and 20-40% higher than grooved tubes [8]. This improvement is a result of the enhanced mixing of the flow from the fins, which reduces mass transfer effects. Although cross-grooved tubes offer enhanced heat transfer performance, they also increase pressure drop during evaporation, which must be considered when designing systems [8]. However, the experiments conducted in this study were run using smooth tubes, which offer a lower pressure drop and are widely used throughout literature.

Barraza *et al.* conducted experiments to determine how heat flux, mass flux, pressure, tube diameter, and mixture composition affect the heat transfer coefficient of mixtures [9]. Specifically, their goal was to collect data at lower temperatures, from cryogenic to room temperature, which is an area with limited previous research. They tested three different mixtures (composed of methane, ethane, and propane) in tubes with diameters of 0.5 mm, 1.5 mm, and 3 mm [9]. The composition by volume of these mixtures was varied as well. Their results indicated that heat flux and pressure had little impact on the heat transfer coefficient across the mixtures and tube diameters tested [9]. Increasing the mass flux, tube roughness and reducing the tube size all resulted in higher heat transfer coefficients [9]. Conversely, diluting the mixture composition generally decreased the heat transfer coefficient [9]. However, it was found that convective boiling was the dominant factor influencing the effectiveness of heat transfer [2].

In more recent studies, researchers have focused on evaporation at higher saturation temperatures instead of cryogenic conditions, which are more relevant for air conditioning applications. The heat transfer coefficients of common refrigerant mixtures such as R134a, R245fa, and their combination were studied at high saturation temperatures by Guo *et al.* [10].

Their goal was to evaluate the impact of R134a on R245fa, which has a low system efficiency when it is used on its own. R245fa is often used for high-temperature applications due to its high critical temperature. Guo *et al.* used 10 mm inner diameter tubes and varied the saturation temperature of the refrigerants between 55-95°C [10]. From their studies, they concluded that R134a has minimal effect on the heat transfer and pressure drop of the mixture, as its heat transfer coefficient closely resembled that of R245fa [10]. These results highlight the importance of carefully selecting refrigerant mixtures to ensure optimal heat transfer performance. Although R134a is well known for its efficient heat transfer, it is unable to portray this when it is mixed with different refrigerants.

Currently, there is a push for new refrigerant blends with low GWP to phase out high-GWP refrigerants like R134a. R134a belongs to the HFC refrigerant group, which has a low ODP, but a high GWP. This means these refrigerants contribute to global warming when they are released into the atmosphere. For environmental considerations, transitioning towards refrigerants with a low ODP and GWP is essential. This review examines R471A, a new refrigerant blend of R1234ze(E), R227ea, and R1336mzz(E), which exhibits both a low ODP and GWP [11]. Compared to R404A, R471A's environmental impact index (EII) values are lower under all operating conditions [12]. EII, developed by Yildirim *et al*, is a factor that evaluates the environmental impact of vapor compression refrigeration systems [6]. EII was developed based on energy and exergy analyses and further proves that R471A is an environmentally friendly alternative [13]. Also, this refrigerant is non-flammable, making it great for applications requiring a high amount of refrigerant, without violating refrigerant laws [14]. However, for practical applications, R471A must demonstrate similar performance to existing refrigerants to maintain system efficiency.

Because of its recent development, limited experimental data is available on the performance of R471A. Recently, Gao *et al.* conducted condensation experiments to compare R471A with common refrigerants like R134a and R404A [2]. Specifically, they focused on R471A's performance in a commercial medium-temperature refrigerated open display cabinet. First, looking at R471A's thermodynamic properties, its lower vapor pressure (around 50% lower than both R134a and R404A) suggests lower leak rates for the system [2]. To account for this, modifications were made to the system, including a larger compressor displacement and a l2.5 mm diameter tube in the evaporator. After running the tests, they found that while maintaining the product temperature in the cooler, R471A showed 8% to 18% lower 24-hour energy consumption compared to R404A [2]. Additionally, R471A showed up to 60% lower emissions than R404A [2]. These results suggest a promising potential of R471A for future refrigeration technologies.

A summary of experimental operating conditions and tube diameters in literature are summarized in Table 2. This list does not include all evaporation experiments in tubes; however, it shows that experimental data is limited for zeotropic mixtures in small diameter tubes, in which this work addresses.

Table 1. Summary of Experimental Findings

Reference	Refrigerants	D [mm]	Operating Conditions	Key Findings
Barraza *et al.* (2016) [9]	$CH_4/C_2H_6/C_3H_8$, R14/R23/R32/R134a, CH_4/C_2H_6	0.5, 1.5, 3	Temperature glide: 103-246 K, 129-275 K, 132-208 K	Dilution of the mixtures increases the temperature glide and deteriorates the heat transfer coefficient. Granryd's correlation provides the best prediction for the data since its prediction is less dependent on flow conditions.
Guo *et al.* (2020) [10]	R134a, R245fa, R134a/R245fa	10	Mass Flux: 100, 200, 300 $\frac{kg}{m^2s}$ Saturation Temperature: 35-85°C (R134a), 55-95°C (R245fa) Pressure: 1109-1694 kPa (R134a/245fa) Quality: 0.2-1 Heat Flux: 6, 12, 24 $\frac{kW}{m^2}$	Heat transfer coefficient of the mixture is very close to that of 245fa. Heat transfer coefficients increase slightly when saturation temperatures change by 10 degrees Celsius.
Gao *et al.* (2022) [2]	R404A, R471A	12.7	Mass Flow Rate: 0-1000$\frac{kg}{hr}$ Temperature: -20 – 125°C	These experimental conditions were conducted in a refrigerated open display case application. R471A showed 8% to 18% lower 24-hour energy consumption than R404A. R471A showed up to 60% lower emissions than R404A.
Uchida *et al.* (1996) [8]	R32/R125/R134a	Smooth: 6.4 Grooved: 6.4 Cross-grooved: 6.1	Mass Flux: 100-500$\frac{kg}{m^2s}$ Quality: 0.1-0.9 Heat Flux: 10$\frac{kW}{m^2}$	Cross-grooved tube showed highest heat transfer coefficients for condensation and evaporation.

Hossain *et al.* (2013) [15]	R1234ze(E), R32, R410A, R1234ze(E)/ R32	4.35	Mass flux: 150-445$\frac{kg}{m^2s}$ Saturation Temperature: 5°C, 10°C Saturation Pressure: 843 kPa Vapor Quality: 0.08-1 Temperature Glide: 8.3 K	Heat transfer coefficients of R1234ze(E) are lower than R1234ze(E) and R32 mixture, R410A and R32.
Zhang *et al.* (2019) [16]	R134a, R32, R1234ze, R1234yf, R407C, R417A	1.5-14	Mass Flux: 80-1100$\frac{kg}{m^2s}$ Saturation Pressure: 200-4156 kPa Vapor Quality: 0.01-1 Heat Flux: 1.3-89.6 $\frac{kW}{m^2}$ Temperature Glide: 0.004-89 K	Correlations presented by Mishra *et al*, Thome *et al,* and Shah *et al* presented the best predictive performance.

Modeling

In addition to experimental work, heat transfer correlations are developed to predict the evaporative performance of refrigerants like R471a and to design heat exchangers. These correlations consist of equations and models derived from experimental data to predict heat transfer coefficients. Zhang *et al.* evaluated various correlations using several zeotropic mixtures to determine the most accurate ones [16]. They show that many existing correlations were developed with limited experimental data and are not accurate over a wide range of operation conditions [16]. Therefore, Zhang *et al.* focused on identifying or developing correlations that can work across a wide array of mixtures and conditions. After testing multiple correlations, they determined that those presented by Mishra *et al.* [17], Shah-GW87 [18], and Thome-GW87 [19] provided the best predictive performance, with mean absolute percentage deviations of less than 50% [16]. Zhang *et al.* also developed two new correlations: one based on physics and another based on regression. Their mean absolute percentage deviations were 29% and 24.6%, respectively [16]. Notably, their review did not compare models with any data for evaporation of R471A. Precise modeling of flow boiling can lead to better system performance, especially when considering new refrigerant alternatives like R471A.

One of the most widely used and reliable models for predicting evaporative heat transfer is Chen's correlation. This correlation accounts for both nucleate and convective boiling regions, recognizing that both occur in some proportion throughout two-phase flow [20]. Originally tested against experimental data for water and organic fluids, Chen's correlation demonstrated agreement with the data from two-phase flow. Average deviations were typically under 15%, confirming its general applicability and accuracy [20].

Building on this correlation, Shin *et al* [21] extended Chen's ideas and incorporated mass transfer effects that develop from composition differences between the liquid and vapor phases during evaporation. These composition changes cause a resistance to heat transfer which needs to be considered when calculating the heat transfer coefficient. To account for the resistance, they modified the convective boiling enhancement factor of Chen's correlation and added a correction factor. This new correlation was tested on R32/R134a and R290/R600a and achieved errors of less than 15% and the heat transfer coefficients aligned closely with the experimental data [21]. Due to its reliability and adaptability, Chen's correlation continues to be used and developed into new correlations, particularly for evaluating new refrigerants like R471A.

Gungor and Winterton [22] also built upon Chen's correlation by introducing modifications to better suit horizontal channels. To achieve this, they revised the forced convection enhancement factor (F) and the pool boiling suppression factor (S) [22]. In a study done by Thome *et al* [23], ten heat transfer correlations were compared to the experimental results of their study with R134a. The correlations were grouped into four categories: strictly convective, superposition, strictly empirical, and flow pattern based. Gungor and Winterton's correlation, falling under the superposition category, demonstrated a notable predictive performance, ranking second overall with a mean error of 22.5% [23]. This proves its effectiveness and supports its use in evaporative analysis.

The heat transfer correlations of Zhang, Thome-GW87, Mishra, Gungor-Winterton, and Guo were selected for this study to evaluate the evaporative heat transfer performance of R471A, as presented in Table 3. These correlations were chosen based on their widespread use in literature, as described above, as well as their accuracy amongst different zeotropic mixtures. Comparing these correlations is important in examining the accuracy of our experimental data and the reliability of the model.

Table 2. Existing Correlations Used in this Study

Name	Correlation
Mishra [17]	$$\alpha_{Mishra} = 21.75\left(\frac{1}{X_{tt}}\right)^{0.29} Bo^{0.23}\alpha_{lo}$$ $$Bo = \frac{q}{Gh_{fg}}$$ $$X_{tt} = \left(\frac{\rho_v}{\rho l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}\left(\frac{1-x}{x}\right)^{0.9}$$ $$\alpha_{lo} = 0.023\left(\frac{k_l}{D}\right)Re_l^{0.8}Pr_l^{0.4}$$
Zhang [16]	If T*≤ 0.06 $$\alpha_{Zhang} = \left[\left(F_c\alpha_{Cooper,nb}\right)^2 + (\alpha_{Mishra})^2\right]^{0.5}$$ Else $$\alpha_{Zhang} = \left[1 + 3000(F_cBo)^{0.86} + 1.12\left(\frac{x}{1-x}\right)^{0.75}\left(\frac{\rho_l}{\rho_v}\right)^{0.41}\right]\alpha_{lo}$$ $$\alpha_{Cooper,nb} = 35M^{-0.5}q^{-0.67}P_r^{0.12}(-\log_{10}P_r)^{-0.55}$$ $$T^* = \frac{(T_b - T_d)}{T_{sat}}$$ $$P_r = \frac{P}{P_{crit}}$$ $$F_c = \left[1 + \frac{\alpha_{id}}{x}(T_d - T_b)\left[1 - e^{\left(-\frac{Bx}{\rho_l h_{fg}\beta_l}\right)}\right]\right]^{-1}$$ $$B = 1$$ $$\beta_l = 0.0003\ [\frac{m}{s}]$$

	$$\alpha_{id} = \left(\alpha_{cb}{}^{2.5} + \alpha_{Cooper}{}^{2.5}\right)^{\frac{1}{2.5}}$$ $$Fr_l = \frac{G^2}{qD\rho_l^2}$$ If $Fr_l \geq 0.25$ $$\alpha_{cb} = 2.15\alpha_{lo}\left(0.29 + \frac{1}{X_{tt}}\right)^{0.85}$$ Else $$\alpha_{cb} = 2.838\alpha_{lo}\left(0.29 + \frac{1}{X_{tt}}\right)^{0.85} Fr_l^{0.2}$$
Thome-GW87 [19]	$$\alpha_{Thome} = F_{mix}\alpha_{lo}$$ $$F_{mix} = 1 + 3000(BoF_c)^{0.86} + 1.12\left(\frac{x}{1-x}\right)^{0.75}\left(\frac{\rho_l}{\rho_v}\right)^{0.41}$$
Gungor-Winterton (GW) [24]	$$\alpha_{GW} = F_{GW}\alpha_{lo} + S_{GW}\alpha_{Cooper}$$ $$\alpha_{Cooper} = 55M^{-0.5}q^{0.67}P_r^{0.12}(-\log_{10} P_r)^{-0.55}$$ If $Fr_l \geq 0.05$ $$F_{GW} = 1 + 2.4 \cdot 10^4 Bo^{1.16} + 1.37X_{tt}^{-0.86}$$ $$S_{GW} = (1 + 1.15 \cdot 10^{-6} F_{GW}{}^2 Re_l^{1.17})^{-1}$$ Else $$F_{GW} = (1 + 2.4 \cdot 10^4 Bo^{1.16} + 1.37X_{tt}^{-0.86})Fr_l^{0.1-2Fr_l}$$ $$S_{GW} = (1 + 1.15 \cdot 10^{-6} F_{GW}{}^2 Re_l^{1.17})^{-1} Fr_l^{0.5}$$
Guo [25]	$$\alpha_{Guo} = \left[(F_{Guo}\alpha_{lo})^2 + \left(S_{Guo}\alpha_{Cooper}\right)^2\right]^{0.5}$$ $$F_{Guo} = \left(1 + 0.063x\,\mathrm{Pr}_l{}^{-0.69}\left(\frac{\rho_l}{\rho_v}\right)\right)^{4.4}$$

	$S_{Guo} = (1 - 0.084 F_{Guo}^{0.17} Re_l^{0.39})^{-1}$

A summary of other modeling findings in existing literature is summarized in Table 4. This list does not include all evaporation modeling efforts; however, it shows that modeling data is limited for zeotropic mixtures, in which this work addresses.

Table 3. Summary of Modeling Findings

Reference	Refrigerant	Key Findings
Shin *et al*. (1996) [21]	R32/R134a, R290/R600a	Adapted Chen's correlation to include correction factors to account for mass transfer effects. Experiments achieved errors of less than 15% and the heat transfer coefficients aligned closely with the experimental data.
Chiou *et al*. (1996) [26]	R22/R114, R22/R124, R32/R134a	Incompleteness of dry out phenomenon of mixtures may be due to difference in flow pattern between pure and mixed refrigerant. Pressure gradient for mixtures increases with flow quality.

Zhang *et al*. (2019) [16]	R134a, R32, R1234ze, R1234yf, R407C, R417A	Two correlations based on physics and regression were developed and have a better predictive performance.

Research Gap

The search for refrigerants that meet the needs of modern society is increasingly important due to global warming concerns and the phase out of high GWP refrigerant mixtures. R471A, with its low ODP and GWP, has gained attention as a potential alternative. However, there is limited data available on its evaporative performance, a crucial factor needed to maintain the efficiency of Naval systems. This gap in study presents a challenge for those looking to transition their systems to use more environmentally friendly refrigerants while maintaining system efficiency. This study addresses this gap by providing experimental data on the evaporative efficiency of R471A and comparing it with the predictions of existing correlations. The findings will lead to a more informed transition to alternative refrigerants and determine whether R471a is a viable solution for future applications.

Chapter 3

Methods and Materials

This chapter provides details on the experimental facility used in this study, including its design and operation, measurements, and methods used to calculate the heat transfer coefficient from experimental data.

Experimental Setup

The experimental setup, shown in Figure 2, was designed to measure the heat transfer coefficients for the evaporation and condensation of a refrigerant inside a horizontal tube. The experimental facility consists of two loops: a refrigerant loop and a water mixture loop. A schematic of these loops can be seen in Figure 3.

Figure 2. Photo of the experimental setup.

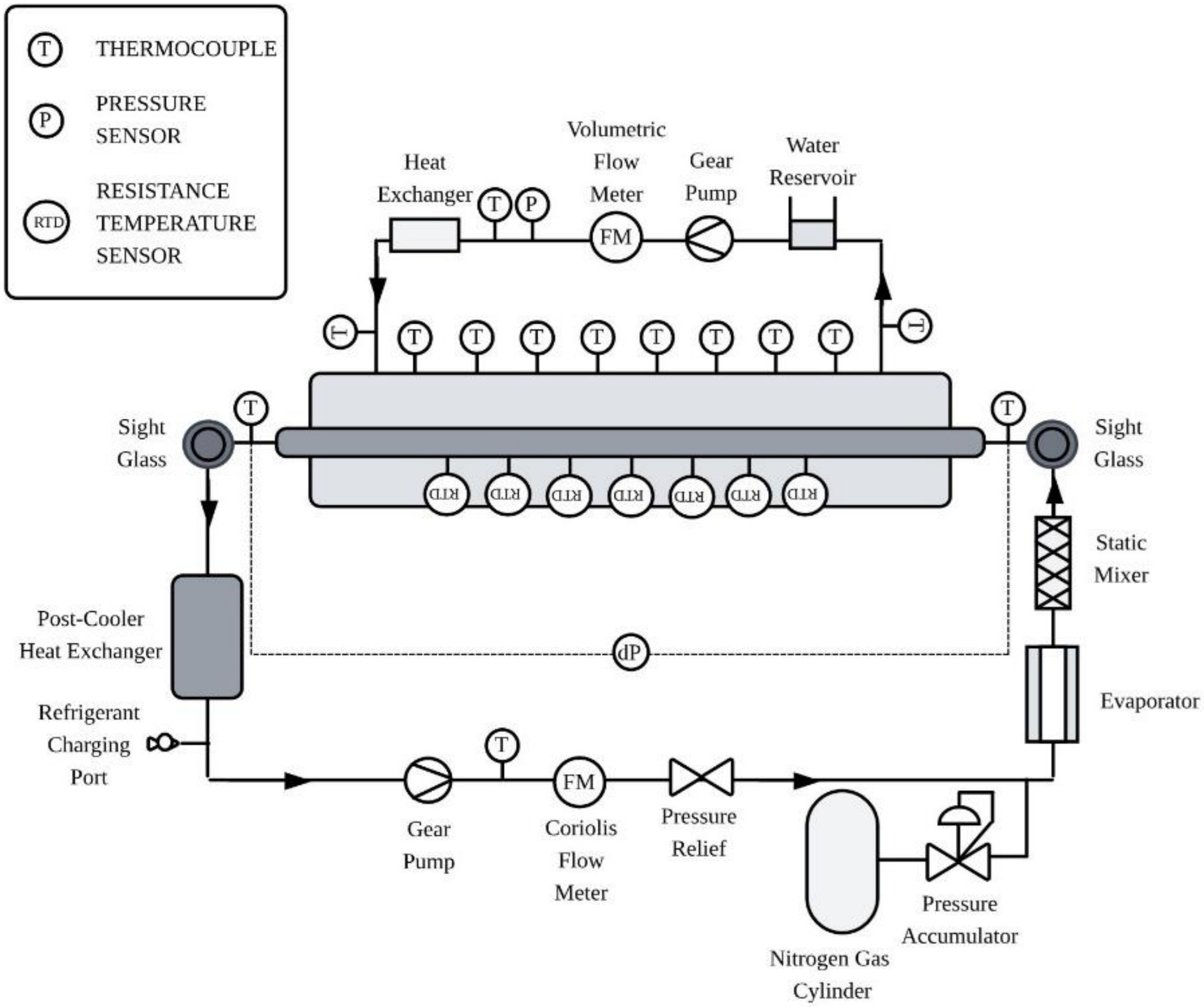

Figure 3. Schematic of the refrigerant and water loops.

After the charge port for the refrigerant, the refrigerant loop begins with a Micropump GC Series gear pump driven by a Baldor Explosion Proof DC Motor. The motor speed is controlled by an Eaton Variable Frequency Drive, which adjusts the refrigerant mass flux. The refrigerant mass flow rate is measured using a Micro Motion ELITE CMF010M Coriolis flow meter connected to a Micro Motion 2700 transmitter.

After leaving the pump, the refrigerant flows through the pressure relief device. This device keeps the pressure at its max value but will release some refrigerant to lower the pressure if it exceeds this value. The refrigerant then encounters a nitrogen-charged piston accumulator. This device stores excess liquid refrigerant and uses nitrogen pressure to put it back into the system when needed. This keeps a steady, pressurized supply of refrigerant to the evaporator. When moving through the evaporator, the refrigerant absorbs heat, but stays in a liquid phase. Finally, before the test section, the refrigerant moves through a static mixer which mixes the refrigerant to ensure its temperature is the same in the whole tube.

Next comes the test section, which is a counter flow, tube-in-tube heat exchanger. The refrigerant flows through the inner tube and enters the test section as a subcooled liquid. An ethylene-glycol/water mixture is used as the heating fluid and flows in the annulus of the tube in the opposite direction. Heat transfer from the hot water mixture to the refrigerant causes the refrigerant to evaporate along the length of the test section and to exit as a superheated vapor. Sight glasses at the inlet and outlet of the test section are used to verify that the refrigerant is subcooled at the inlet and superheated at the outlet.

The refrigerant then flows through a post-cooler heat exchanger before returning to the pump inlet. The post-cooler is seen after the test section to ensure that the refrigerant vapor does not flow through the pump. After the refrigerant passes through post-cooler, it returns as a subcooled liquid and flows towards the pump, thus completing the loop.

The water mixture loop consists of a reservoir, a pump, and a heat exchanger. The Watlow heater supplies the necessary energy to warm the water to the desired temperature. The heater power is controlled by a Watlow temperature control board to regulate the water temperature. The water flow rate is measured using an Emerson Rosemount Magnetic Flowtube and can be used to make flow rate adjustments. The water passes through the test section, before returning to the reservoir to complete the loop. The water reservoir is filled with an ethylene-glycol/water solution. Ethylene-glycol lowers the freezing point and raises the boiling point of plain water, which ensures that the heat transfer fluid remains in a liquid phase over a wider temperature range.

Test Section

The test section is a counter flow, tube-in-tube heat exchanger, with refrigerant flowing inside the inner tube and water flowing through the annulus, as seen in Figure 4. The inner tube of the tube-in-tube test section is a copper tube with an inner diameter of 4.62 mm and an outer diameter equal to 6.35 mm. The outer tube has an inner diameter of 14.73 mm and an outer diameter equal to 21.59 mm.

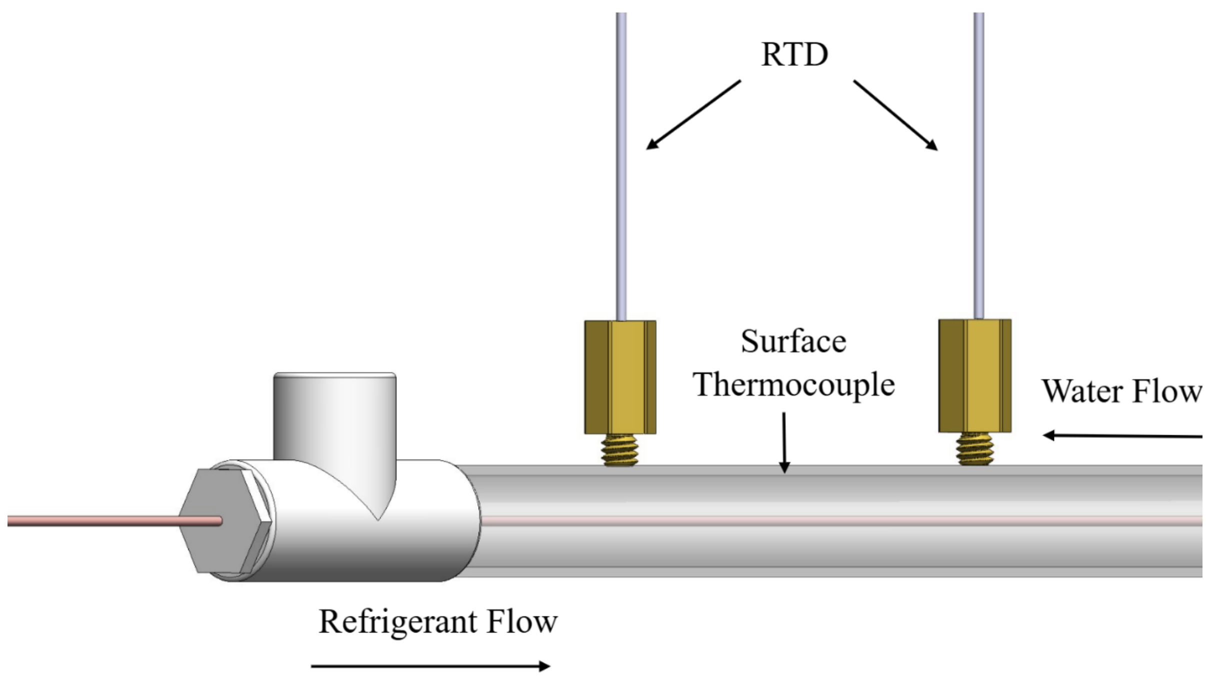

Figure 4: Rendering of test section with RTD and thermocouple locations.

Resistance temperature detectors (RTDs) are evenly distributed along the test section. The RTDs are inserted halfway into the annulus to measure the water temperature along the test section. The test section is divided into seven sections, measuring around 15 cm each, which is defined by the spacing of the RTDs, which is shown in Figure 5. Two additional RTDs are placed at the inlet and outlet of the test section.

Thermocouples are also used along the length of the test section. Two thermocouples are used to measure the inlet and outlet temperature of the refrigerant. Also, in the middle of each segment, thermocouples were soldered to the surface of the tube to measure the outer surface temperature. Heat transfer coefficients and the average quality can then be calculated in each segment as the refrigerant evaporates along the tube. This is through the measurements provided by the RTDs at the inlet and outlet at each segment and the thermocouples at the surface of the tube.

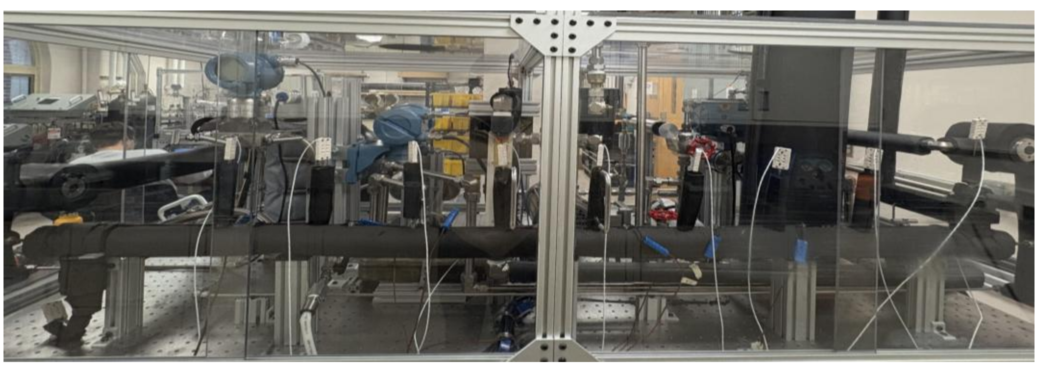

Figure 5. Photo of the test section.

Refrigerant Charging Procedure

To avoid refrigerant cross-contamination, a refrigerant charging and recovery procedure was used. First, the refrigerant loop, refrigerant hoses, and other pipe fittings were evacuated using a vacuum pump to remove any impurities from the system. Once a vacuum of 0.027 kPa (0.2 torr) or less was achieved, the refrigerant could be charged into the loop. The refrigerant was always charged as a liquid using a standard refrigerant hose.

In order to recover the refrigerant, a recovery unit was used to remove all refrigerant from the system and store it in a recovery cylinder. The system would then be evacuated, as described above, to eliminate the possibility of cross-contamination. This procedure allowed refrigerant charging and recovering to be consistent and ensured no refrigerant escaped to the atmosphere. It also guaranteed that the composition of the refrigerant in the loop is close to the intended composition provided by the manufacturers.

Data Reduction

Data reduction refers to the process of converting experimental measurements, including temperature, pressure, and flow rate, into useful quantities such as heat transfer coefficients, vapor quality, or mass flux. The data reduction method used to calculate refrigerant evaporation heat transfer coefficients is described in the following section.

Prior to recording each data set, steady-state conditions needed to be achieved. This was confirmed when temperature, absolute pressure, differential pressure, and mass flow rate readings varied less than 0.1°C, 10kPa, 0.1kPa and 0.1g/s, respectfully, over a four-minute interval. Once steady-state was achieved, data was collected for another four-minute interval at a frequency of 1 Hz. Data processing was performed using Engineering Equation Solver [27] software using refrigerant property data obtained from REFPROP 10 [28].

Heat Transfer Coefficients

To evaluate the heat transfer coefficient of each segment, the heat input that drives evaporation of the refrigerant needs to be determined. To do this, the heat transferred to the water is considered as it is assumed to be equal to the heat absorbed by the refrigerant. The heat supplied to each segment, Q_i, is calculated using Equation (3.1), where $\dot{m}_w$ is the water mass flow rate, $c_{p,w}$ is the specific heat of water calculated at the average water temperature for the segment, and $\Delta T_{w,i}$ is the water temperature change across the segment. The subscript i indicates the segment number.

$$\dot{Q}_i = \dot{m}_w c_{p,w} \Delta T_{w,i} \tag{3.1}$$

The refrigerant enters the test section in a subcooled state. Its enthalpy can be determined for each subsequent section in the test section. Considering the water side again, the refrigerant enthalpy can be determined using the measured pressure and temperature. Equation (3.2) describes this, where h_r is the refrigerant enthalpy and $\dot{m}_r$ is the refrigerant mass flow rate. As the enthalpy changes, the vapor quality at the inlet and outlet of each section can be calculated using REFPROP 10 [28]. The heat flux, q, in each segment is given by Equation (3.3) where D_i is the refrigerant inner tube diameter and L is the length of the segment. A constant heat flux is assumed for each segment.

$$h_{r,out,i} = h_{r,in,i} - \frac{\dot{Q}_i}{\dot{m}_r} \tag{3.2}$$

$$q = \frac{\dot{Q}_i}{\pi D_i L} \tag{3.3}$$

The inner wall temperature of the tube, $T_{wall,in}$, is calculated in Equation (3.4) using the radial heat conduction equation for a cylindrical wall. $T_{wall,out}$, is the measured outer wall temperature, L is the length of the segment, k_c is the thermal conductivity of copper, and D_o is the refrigerant tube diameter. Conduction in the axial direction is negligible due to its small contribution to the total heat transferred compared to radial heat transfer.

$$T_{wall,in,i} = T_{wall,out,i} - \frac{\dot{Q}_i \ln\frac{D_o}{D_i}}{2\pi L_i k_c} \tag{3.4}$$

Finally, the heat transfer coefficient, α_i, is evaluated in equation (3.5) where $T_{ref,avg}$ is the average temperature of the refrigerant at that segment.

$$\alpha_i = \frac{q}{T_{wall,in,i} - T_{ref,avg}} \tag{3.5}$$

Chapter 4 Results and Discussion

Heat transfer results are presented in this chapter from evaporation experiments conducted for the zeotropic mixture R471A.

Experimental Heat Transfer Coefficients

Table 4 summarizes the experimental data collection matrix for this study. Evaporation data were collected for R471A at two different dew point temperatures 22°C and 33°C, and for two mass fluxes, 300$\frac{kg}{m^2 s}$ and 500 $\frac{kg}{m^2 s}$. For each condition, the corresponding heat transfer coefficient for each segment, α_{seg}, was calculated. Refrigerant entered the test section as a subcooled liquid and left as a superheated vapor, for complete evaporation.

Table 4. Summary of experimental data collected.

T_{dew} [°C]	Mass Flux (G) $[\frac{kg}{m^2 s}]$
22	300
22	500
33	300
33	500

The vapor quality at each segment, x_{avg} was also calculated, and uncertainties in vapor quality UC_X and heat transfer coefficient $UC\alpha$ were calculated based on instrument uncertainty. The measured heat transfer coefficients for R471A are plotted against vapor quality for each mass flux and dew point temperature in Figure 6. Error bars are used to represent the uncertainties in α_{avg} and X_{avg}.

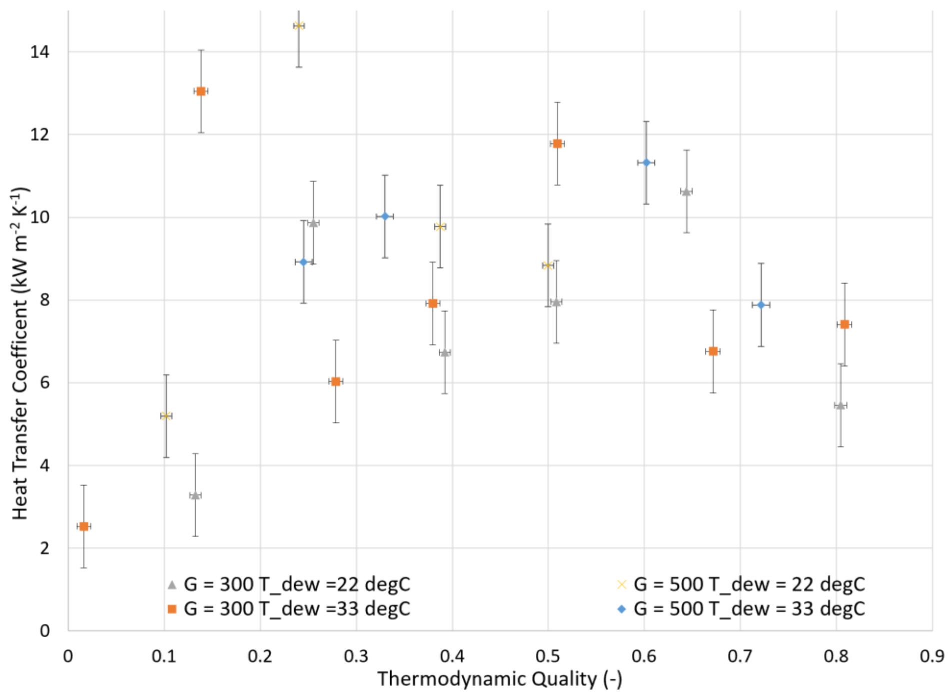

Figure 6. Heat transfer coefficients plotted against vapor quality.

In general, α_{avg} tends to increase with increasing mass flux and decrease with increasing dew temperatures, consistent with prior work. As vapor quality increases, the value of the heat transfer coefficient becomes complicated in zeotropic mixtures. Initially, the heat transfer coefficient increases, but can decrease due to mass transfer effects and composition differences, since the more volatile component evaporates first. These trends are seen throughout Figure 6. Although a few outliers are observed, the higher mass flux, G = 500$\frac{kg}{m^2s}$, corresponds to larger heat transfer coefficients, compared to the lower mass flux, G = 300$\frac{kg}{m^2s}$, for both dew temperatures. Higher mass flux corresponds to higher fluid velocity, reducing the thickness of the boundary layer, allowing heat to transfer more efficiently to the tube wall into the refrigerant.

As vapor quality increases, the heat transfer coefficient initially increases because of enhanced nucleate boiling. However, as vapor quality increases, α is seen to plateau and decrease, which is expected due to mass transfer and composition effects. Finally, as temperature increases, the heat transfer coefficient decreases for a mass flux of 300$\frac{kg}{m^2s}$. However, for a mass flux of 500$\frac{kg}{m^2s}$, a trend is difficult to see. The heat transfer coefficient at 22°C plateaus much before the plateau at 33°C. This makes the heat transfer coefficients difficult to compare. However, considering that the max heat transfer coefficient at 22°C is around 14$\frac{kW}{m^2K}$, while that at 33°C is around 11$\frac{kW}{m^2K}$, it can be inferred that the trend is similar to what is expected.

Predicted Heat Transfer Coefficients

The measured heat transfer coefficients for R471A are compared to the heat transfer correlations developed by Gungor-Winterton, Guo, Mishra, Thome-GW87, and Zhang in Figure 7, 8, 9, 10, and 11, respectfully. These correlations were developed for predicting heat transfer coefficients of zeotropic mixtures and have been widely used throughout literature. Mass fluxes of 300$\frac{kg}{m^2s}$ and 500$\frac{kg}{m^2s}$ at dew point temperatures of 22°C and 33°C were used for each correlation.

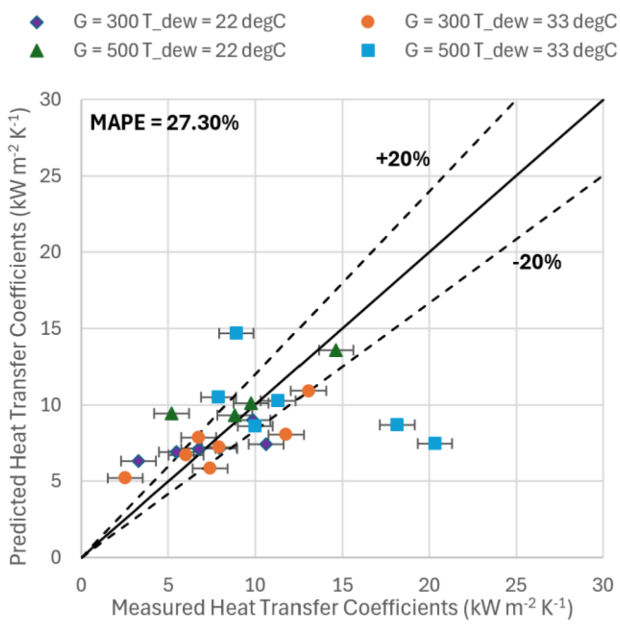

Figure 7. A comparison of measured heat transfer coefficients for R471A against predicted heat transfer coefficients from Gungor-Winterton's correlation [24].

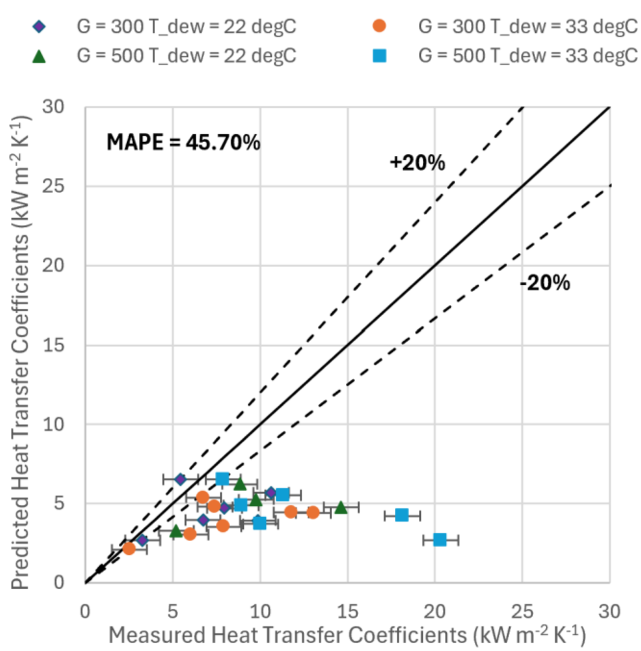

Figure 8. A comparison of measured heat transfer coefficients for R471A against predicted heat transfer coefficients from Guo's correlation [25].

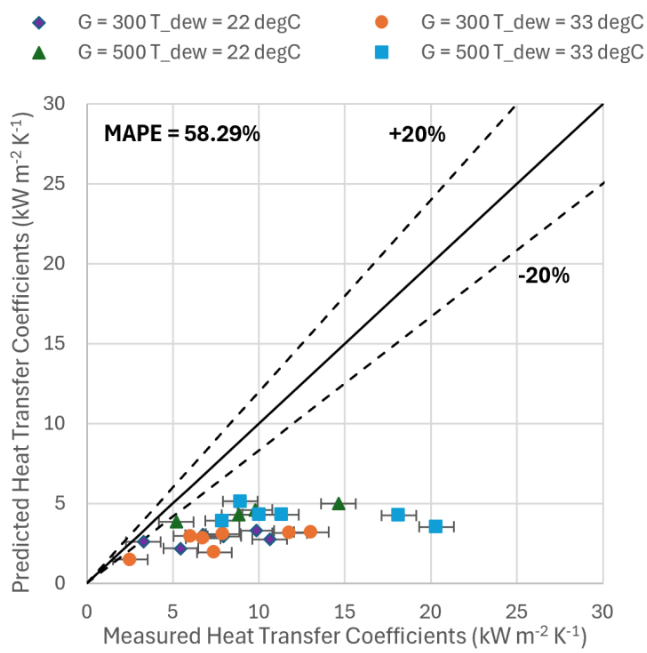

Figure 9. A comparison of measured heat transfer coefficients for R471A against predicted heat transfer coefficients from Mishra's correlation [17].

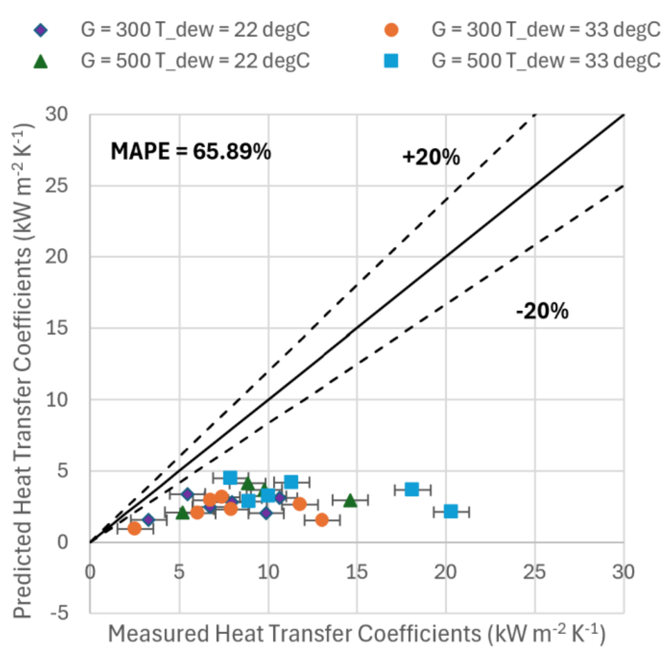

Figure 10. A comparison of measured heat transfer coefficients for R471A against predicted heat transfer coefficients from Thome-GW87's correlation [19].

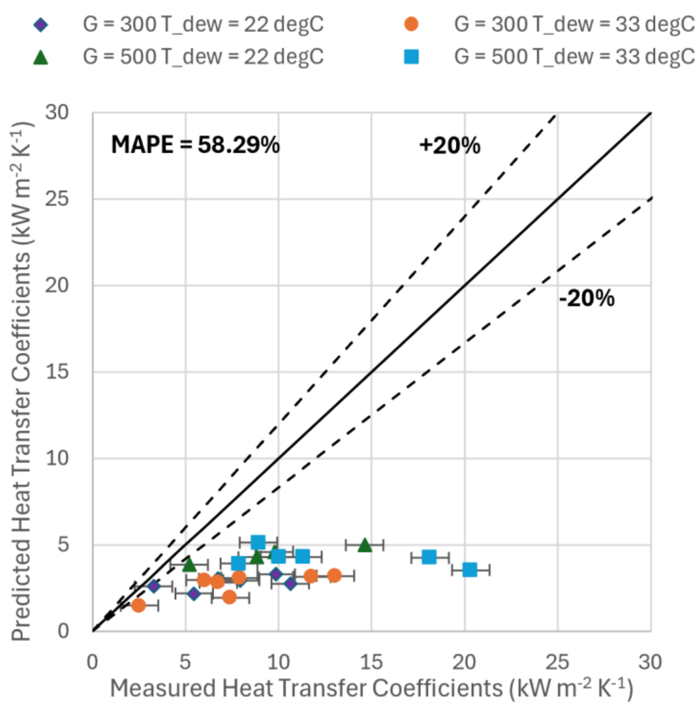

Figure 11. A comparison of measured heat transfer coefficients for R471A against predicted heat transfer coefficients from Zhang's correlation [16].

A total of 23 data points were collected and the *Gungor-Winterton* correlation predicted the most accurate heat transfer coefficients with a Mean Absolute Percentage Error (MAPE) of 27.30%. MAPE is defined in equation 4.1 and it is a statistical equation used to measure how accurate a prediction is compared to actual values.

$$MAPE = \frac{100\%}{n} \sum_{n=1}^{n} \frac{|P_{predicted} - P_{measured}|}{P_{measured}} \quad (4.1)$$

Guo's correlation predicted similarly to the experimental values with a MAPE of 45.70%. Mishra, Thome-GW87, and Zhang predicted the least accurately, with MAPE's of 58.29%, 65.89% and 58.29% respectively.

As shown throughout the figures, the predicted heat transfer coefficients varied across the five correlations evaluated. Considering Gungor-Winterton's model, a MAPE of 27.30% indicates that although some deviation existed, the correlation still reasonably captured the overall heat transfer characteristics of R471A. The performance of this correlation is notable considering its accuracy amongst a limited number of data points. This suggests its potential applicability to low-GWP refrigerants like R471A.

Guo's correlation also performed moderately well. Although less accurate than Gungor-Winterton's, Guo's correlation demonstrated a similar trend to the experimental results. In contrast, the remaining correlations showed larger deviations to the measured heat transfer coefficient values. Although the data set is small, these correlations struggle to align with the measured values. The correlations tend to predict heat transfer coefficients that are less than or equal to 5. These models may not account for unique properties and mass transfer characteristics exhibited by R471A. Overall, these results show that existing correlations exhibit varying predictability when applied to low-GWP zeotropic refrigerant mixtures. The deviations suggest that further development of heat transfer correlations is necessary to accurately predict the evaporation performance of refrigerants with complex thermodynamic behaviors as well as the continuation of experimental studies on these refrigerants.

Chapter 5 Conclusion

Research Scope

Global regulations and environmental policies are phasing out high-GWP refrigerants, typically used in Naval and industrial systems. Refrigerant blends of HFCs and HFOs have emerged as potential replacements for conventional HFC refrigerants. These mixtures combine environmentally favorable HFO components with HFC components that provide desirable thermophysical properties. However, evaluation of their thermal performance is limited, which is critical for reliable chiller operation. The zeotropic mixture, R471A, has gained interest as it reflects both properties of HFCs and HFOs. However, experimental data on its evaporative behavior is limited. This work aims to address that gap by providing experimental data crucial for the design and optimization of systems using R471A.

Conclusions

This study investigated the evaporative heat transfer performance of the low-GWP zeotropic refrigerant mixture R471A in a 4.62 mm horizontal tube. The goals were to evaluate R471A's heat transfer behavior across a range of dew point temperatures and mass fluxes to assess its suitability as a replacement refrigerant for Navy chiller applications. Experiments were conducted at dew temperatures of 22°C and 33°C and mass fluxes of $300\frac{kg}{m^2s}$ and $500\frac{kg}{m^2s}$. Heat transfer coefficients were determined from measured temperatures, pressures, and flow rates across a range of vapor qualities.

Heat transfer measurements were obtained under steady-state conditions in a 1.067m long horizontal tube with counterflow ethylene-glycol/water in the annulus. The refrigerant entered the inner tube as a subcooled liquid and exited as a superheated vapor. Results showed that the heat transfer coefficient increased with mass flux and decreased with increasing dew temperatures. As vapor quality increased, the heat transfer coefficient initially increased, plateaued and then decreased. While some outliers were observed, the trends remained clear.

Five established heat transfer correlations were assessed using the measured data. Correlations developed by Gungor-Winterton, Guo, Mishra, Thome-GW87, and Zhang were chosen based on their widespread use in literature, specifically for zeotropic mixtures. Gungor-Winterton's correlation predicted the most accurately with a MAPE of 27.30%. Guo's correlation came behind this, with a MAPE of 45.70%.

Overall, R471A demonstrated a favorable heat transfer coefficient ranging from 2-14$\frac{kW}{m^2K}$, illustrating its potential as a low-GWP replacement. The data presented in this work will be valuable to researchers and engineers looking to understand the evaporative behavior of R471A.

Recommendations for Future Work

This study provides an initial step towards the integration of environmentally friendly refrigerants. Further research should:

- Expand testing across a wider range of mass fluxes and dew point temperatures to capture the full evaporative behavior of R471A under varied operating conditions.

- Assess additional low-GWP zeotropic mixtures to compare their evaporation performance in small diameter tubes.
- Investigate the effects of lubricating oils, which are present in compressors of real chiller systems and may change heat transfer behavior.
- Examine the two-phase frictional pressure drop to better understand overall system performance and energy consumptio

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