

**INVESTIGATION OF FREE-CONVECTION
CONDENSATION ON A SINGLE HORIZONTAL COPPER
TUBE**

BY
AHMAD ADEL OBEIDAT

A Thesis Presented to the
DEANSHIP OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

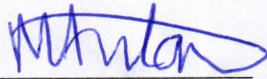
MASTER OF SCIENCE

In
MECHANICAL ENGINEERING

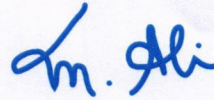
MAY 2024

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS
DHAHRAN- 31261, SAUDI ARABIA
DEANSHIP OF GRADUATE STUDIES

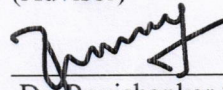
This thesis, written by **Ahmad Adel Obeidat** under the direction of his thesis advisor and approved by his thesis committee, has been presented and accepted by the Dean of Graduate Studies, in partial fulfillment of the requirements for the degree of **MASTER OF SCIENCE IN MECHANICAL ENGINEERING**.



Prof. Mohammad Antar
Department Chairman



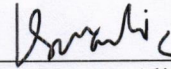
Dr. Hafiz Ali
(Advisor)



Dr. Ravishankar Sathyamurthy
(Member)



Dr. Suliman Al-Homidan
Dean of Graduate Studies



Dr. Usman Ali
(Member)

Date



© Ahmad Obeidat
2024

To my parents Kholod and Adel
To my brothers, Hasan and Mohammad
To my grandfather Ahmad

ACKNOWLEDGMENTS

In the intricate tapestry of my academic journey, my deepest gratitude extends to Allah Almighty, the guiding light throughout my life.

To my family, you are my unwavering support system, with my father's encouragement and my mother's endless love. To my brother Hasan and Mohammad—your unwavering support has been my constant source of strength.

I am deeply grateful to my best friend, Ghazal Safadi, for her continuous support and love throughout my master's thesis journey. Her unwavering encouragement has been a source of strength and inspiration.

Heartfelt appreciation to my advisor, Dr. Hafiz Ali, and committee members, Dr. Usman and Dr. Ravishnkar. Their constant encouragement, insightful feedback, and friendly support have truly shaped the path of this research. Their positive influence has been a key factor in the successful development of this study.

With gratitude to the Department of ME for fostering an environment of academic excellence and research innovation throughout my journey. A Special appreciation to my instructors for imparting valuable knowledge and guidance, shaping my academic and research endeavors.

To friends, colleagues, and classmates, integral contributors to this experience—your shared moments have added vibrant colors to the canvas of academia.

Lastly, to everyone who supported and assisted, even though words, your collective impact has transformed this endeavor into a shared accomplishment. I am profoundly thankful for the indelible mark each of you has left.

TABLE OF CONTENTS

ACKNOWLEDGMENTS	IV
TABLE OF CONTENTS	V
LIST OF TABLES	VIII
LIST OF FIGURES	IX
LIST OF ABBREVIATIONS	X
ABSTRACT	XI
ملخص الرسالة.....	XIII
CHAPTER 1 INTRODUCTION	1
1.1 Introduction	1
1.2 Problem Statement	2
1.3 Aims and Objectives	3
CHAPTER 2 LITERATURE REVIEW	4
2.1 Introduction	4
2.2 Free convection condensation	5
2.2.1 Theoretical studies	5
2.2.2 Theoretical studies	13
2.2 Forced Convection Condensation	16
2.3.1 Theoretical Studies.....	16
2.3.2 Experimental Studies	21

CHAPTER 3 METHODOLOGY AND EXPERIMENTAL APPARATUS.....	27
3.1 General Apparatus Layout	27
3.2 Test Section	27
3.3 Test Tube	28
3.4 Secondary Condenser	28
3.5 Boiler Power	28
3.6 Experimental Procedure.....	29
3.7 Data Processing.....	30
3.7.1 Rate of Heat Transfer	30
 CHAPTER 4 MEASUREMENTS OF HEAT TRANSFER FOR STEAM-ETHANOL MIXTURES CONDENSING ON A HORIZONTAL SMOOTH TUBE	 35
4.1 Introduction	35
4.2 Pure Steam Condensation on Horizontal Smooth Tubes	36
4.3 Results for condensation of steam-ethanol mixture on horizontal smooth tubes.	38
4.3.1 Visual Observation	43
4.3.2 Enhancement Ratio.....	51
 CHAPTER 5 MEASUREMENTS OF HEAT TRANSFER FOR PURE STEAM AND STEAM-ETHANOL MIXTURES CONDENSING ON A WIRE WRAPPING TUBE	 53
5.1 Introduction	53
5.2 Pure Steam Condensation on Horizontal Wire-Wrapping Tubes	54
5.3 Marangoni Condensation on Horizontal Wire-Wrapping Tubes	58
Visual Observation	60
Enhancement Ratio.....	61

CHAPTER 6 HEAT TRANSFER ENHANCEMENT PREDICTING OF NEW CONDENSATION FLUIDS ON PIN-FIN TUBES WITH	78
6.1 Introduction	78
6.2 Results.....	79
CHAPTER 7 CONCLUSION	83
7.1 General Conclusion.....	83
7.2 Steam-ethanol on plain tube	83
7.3 Steam-ethanol on wire-wrapping	84
7.4 Pin-Fin tube	85
REFERENCES	87
APPENDIX A: ESTIMATION OF UNCERTAINTIES IN EXPERIMENTAL MEASUREMENTS.....	94
A.1 Introduction	94
A.2 Application to the Present Investigation	95
A.2.1 Test-Section Vapor Velocity	95
A2.2 Heat Flux.....	96
A.2.3 Overall Heat Transfer Coefficient	97
VITAE	101

LIST OF TABLES

Table 1 Results of Uncertainty Analysis for Test Section Vapor Velocity	99
Table 2 Results of Uncertainty Analysis for Smooth Tube: Heat Flux Measurement	99
Table 3 Results of Uncertainty Analysis for Smooth Tube: Heat Flux Measurement	99
Table 4 Results of Uncertainty Analysis for Smooth Tube: Heat Flux Measurement	99
Table 5 Results of Uncertainty Analysis for Smooth Tube: Heat Flux Measurement ...	100
Table 6 Results of Uncertainty Analysis for Smooth Tube: Heat Flux Measurement ...	100
Table 7 Results of Uncertainty Analysis for Smooth Tube: Heat Flux Measurement ...	100

LIST OF FIGURES

Figure 1 Wire-wrapping tubes used in present investigation.....	34
Figure 2 Comparing Nusselt's 1916 theory with pure steam data on smooth tubes.	37
Figure 3 presents snapshots of film-wise condensation on horizontal smooth tube.....	37
Figure 4 Comparison of the heat flux against vapor side surface temperature difference for three ethanol mass concentrations	40
Figure 5 Heat flux and the temperature difference for six ethanol concentrations.	42
Figure 6 CHTC vs temperature difference for six ethanol concentrations.	42
Figure 7 Condensate appearance as the temperature difference increases(0.0125%).	45
Figure 8 Condensate appearance as the temperature difference increases (0.025%).	46
Figure 9 Condensate appearance as the temperature difference increases (0.05%).	47
Figure 10 Condensate appearance as the temperature difference increases (0.075%).	48
Figure 11 Condensate appearance as the temperature difference increases (0.1%).	49
Figure 12 Condensate appearance as the temperature difference increases (0.3%)	50
Figure 13 The enhancement ratio (ϵ) pure steam.	52
Figure 14 Heat flux for plain tube and wire-wrapping tube with 4 different pitches	56
Figure 15 CHTC for plain tube and wire-wrapping tube with 4 different pitches.	56
Figure 16 Visual observation of the condensate film at 4 different pitches.	57
Figure 17 Heat flux for wire-wrapping tubes.	64
Figure 18 CHTF wire-wrapping tubes at different pitches.....	66
Figure 19 Heat flux for wire-wrapping tubes at different ethanol concentrations.	68
Figure 20 Condensate appearance as the temperature difference increases (1.6 mm).	69
Figure 21 Condensate appearance as the temperature difference increases (2 mm).	71
Figure 22 Condensate appearance as the temperature difference increases (2.4 mm).	73
Figure 23 Condensate appearance as the temperature difference increases (2.9 mm).	75
Figure 24 Enhancement ratio for Marangoni condensation on wire-wrapping tubes.....	77
Figure 25 Enhancement ratio for Marangoni condensation on wire-wrapping tubes.....	81
Figure 26 Enhancement ratio plotted with circumferential pin spacing and pin height. ..	81
Figure 27 Enhancement ratio with pin spacing-longitudinal pin spacing.....	82
Figure 28 Enhancement ratio circumferential pin thickness and pin height.....	82

LIST OF ABBREVIATIONS

ΔT	Vapor side-to-surface temperature difference $T_v - T_w$
U_v	Vapor velocity
q	Heat flux
α	Convective heat transfer coefficient
T_v	Vapor temperature above the test section
T_w	Outside tube wall surface temperature
q_{Nu}	Heat flux obtained from Nusselt theory (1916)
ρ	Density of condensate
ρ_v	Density of vapor
h_{fg}	Specific enthalpy of vaporization
k	Thermal conductivity of condensate
μ	Viscosity of condensate

ABSTRACT

Full Name : Ahmad Adel Ahmad Obeidat
Thesis Title : INVESTIGATION OF FREE-CONVECTION CONDENSATION ON
A SINGLE HORIZONTAL COPPER TUBE
Major Field : Mechanical engineering department
Date of Degree : June, 2024

Improving heat transfer in condensation tubes is crucial for condenser design. Researchers have explored geometric modifications like pin-fin, integral tubes, and wire-wrapping tubes to enhance heat transfer. However, the impact of vapor mixture composition on surface tension and its effect on heat transfer coefficient enhancement have not been fully explored. Marangoni condensation on plain and wire-wrapping tubes has been studied.

The enhancement with wire-wrapping tubes is attributed to reducing condensate film thickness and increasing active surface area. In Marangoni condensation, surface tension gradients induce instability in the condensate film, shifting the condensation mode to a pseudo-dropwise mode similar to pure fluid dropwise condensation on hydrophobic surfaces. This results in a thinner condensate film, leading to increased heat transfer. Although extensive research has been conducted on Marangoni condensation of steam-ethanol mixtures on plain tubes, investigation into Marangoni condensation on wire-wrapping tubes has yet to be investigated. Marangoni condensation of steam-ethanol mixtures has been examined on both plain and wire-wrapping tubes under atmospheric pressure conditions. The investigation on plain tubes involved exploring six different mass concentrations (0.0125%, 0.025%, 0.05%, 0.075%, 0.1%, and 0.3%) of the mixture. Meanwhile, for wire-wrapping tubes, four different pitches (1.6mm, 2mm, 2.4mm, and

2.9mm) were studied alongside four concentrations of steam-ethanol (0.0125%, 0.05%, 0.1%, and 0.3%). Special precautions were taken to prevent errors caused by the presence of air in the vapor during experimentation. Marangoni condensation on both plain and wire-wrapping tubes show significant enhancements in heat transfer compared with pure steam, with higher heat fluxes and heat-transfer coefficients observed. The wire-wrapping tube demonstrated an enhancement ratio (heat flux or heat-transfer coefficient divided by the corresponding value predicted by Nusselt theory) of 9.9 at an ethanol concentration of 0.1% and a pitch of 2.9 mm. For the plain tube, the maximum enhancement ratio was found to be 3.8 at a concentration of 0.1%. Generally, the enhancement ratio was higher at lower vapor side-to-surface temperature differences and decreased with increasing temperature differences.

Finally, a multi-optimization investigation was done for the optimal geometrical parameters of pin-fin tube, A wide range of geometrical parameters of tubes and pin-fins were investigated. Surface tension and the effect of gravity are both taken into consideration. These parameters are used for predicting the condensation for new and safe refrigerants. A significant improvement in the enhancement ratio is gained by raising the circumference pin count, which is accomplished by reducing the thickness and circumferential pin spacing. Furthermore, doubling the pin height almost doubles the enhancement ratio.

ملخص الرسالة

الاسم الكامل: أحمد عادل أحمد عبيدات

عنوان الرسالة: دراسة التكتاف بالتدفق الحر على أنبوب نحاسي أفقي واحد

التخصص: الهندسة الميكانيكية

تحسين انتقال الحرارة في أنابيب التكتاف أمر بالغ الأهمية لتصميم المكثفات. استكشف الباحثون التعديلات الهندسية مثل أنابيب الزعانف المتكاملة والأنابيب المغلفة بالسلك لتعزيز انتقال الحرارة. ومع ذلك، لم يتم استكشاف تأثير تكوين خليط البخار على التوتر السطحي وتأثيره على تحسين معامل انتقال الحرارة بشكل كامل. تم دراسة التكتاف بطريقة مارانغوني على الأنابيب العادية والأنابيب المغلفة بالسلك.

يُعزى التحسين في الأنابيب المغلفة بالسلك إلى تقليل سمك فيلم المكثف وزيادة مساحة السطح النشط. في تكتاف مارانغوني، تُحدث التدرجات في التوتر السطحي عدم استقرار في فيلم المكثف، مما يؤدي إلى تحول نمط التكتاف إلى نمط يشبه التكتاف قطري الشكل على أسطح كارهة للماء، مما ينتج عنه فيلم مكثف أنحف وزيادة في انتقال الحرارة. على الرغم من أن الأبحاث الواسعة قد تناولت تكتاف مارانغوني لخليط البخار-الإيثانول على الأنابيب العادية، إلا أن البحث في تكتاف مارانغوني على الأنابيب المغلفة بالسلك لم يتم بعد.

تم دراسة تكتاف مارانغوني لخليط البخار-الإيثانول على كل من الأنابيب العادية والمغلفة بالسلك تحت ظروف الضغط الجوي. شملت الدراسة على الأنابيب العادية استكشاف ستة تركيزات كتلية مختلفة (0.0125%، 0.025%، 0.05%، 0.075%، 0.1%، و 0.3%) من الخليط. بينما تم دراسة أربعة مسافات للأنابيب المغلفة بالسلك (1.6 م، 2 م، 2.4 م، و 2.9 م) مع أربعة تركيزات من بخار الإيثانول (0.0125%، 0.05%، 0.1%، و 0.3%). تم اتخاذ احتياطات خاصة لمنع الأخطاء الناتجة عن وجود الهواء في البخار أثناء التجارب.

أظهرت النتائج أن تكتاف مارانغوني على كل من الأنابيب العادية والمغلفة بالسلك أدى إلى تحسينات كبيرة في انتقال الحرارة مقارنة بالبخار النقي، مع ملاحظة تدفقات حرارية أعلى ومعاملات انتقال حرارية أكبر. أظهرت الأنابيب المغلفة بالسلك نسبة تحسين (نسبة التدفق الحراري أو معامل انتقال الحرارة مقسومًا على القيمة المقابلة المتوقعة بواسطة نظرية نوسلت) بلغت 9.9 عند تركيز إيثانول 0.1% ومسافة 2.9 م. أما الأنابيب العادية، فقد وصلت نسبة التحسين القصوى إلى 3.8 عند تركيز 0.1%. بشكل عام، كانت نسبة التحسين أكبر عند الفروقات الصغيرة في درجات الحرارة بين الجانب البخاري والسطح، وانخفضت مع زيادة الفروقات في درجات الحرارة. وأخيرًا، تم إجراء تحقيق متعدد الأمثلة لتحديد أفضل المعلمات الهندسية لأنبوب الزعانف، حيث تم استكشاف مجموعة واسعة من المعلمات الهندسية للأنابيب والزعانف. تم أخذ التوتر السطحي وتأثير الجاذبية في الاعتبار. تُستخدم هذه المعلمات للتنبؤ بتكتاف المواد المبردة الجديدة والأمنة. تم تحقيق تحسين كبير في نسبة التحسين من خلال زيادة عدد الزعانف المحيطية، وهو ما تم تحقيقه عن طريق تقليل سمك الزعنفة والمسافات المحيطية بين الزعانف. علاوة على ذلك، تضاعفت نسبة التحسين تقريبًا عند مضاعفة ارتفاع الزعنفة.

CHAPTER 1

INTRODUCTION

1.1 Introduction

Condensers play a crucial role in engineering applications such as power, process, air conditioning, and refrigeration plants. To make them more cost-effective and efficient, ongoing research in condensation heat transfer is essential. Two primary modes of condensation are dropwise and film-wise.

Dropwise condensation occurs when individual liquid droplets form and grow on a solid surface. These droplets are generally non-wetting, meaning they do not spread and form a continuous film. Instead, they remain as separate droplets on the surface. Film-wise condensation, on the other hand, occurs when a continuous film of condensate forms and flows over a solid surface. In this mode, the surface is "wetted" by the condensate film. Film-wise condensation is the typical mode used in practical condensers, especially in industrial applications such as power plants, refrigeration systems, and air conditioning units.

Since the 1930s, it has been well-established that the heat transfer coefficients in dropwise condensation of steam are significantly greater than those observed in film condensation. This discovery has sparked extensive research focusing on various aspects, including heat

transfer coefficients, theoretical understanding, transition phenomena, and the influence of surface materials. Numerous studies have been conducted over the years to delve deeper into these areas and gain insights into optimizing condensation heat transfer processes.[1]

Marangoni condensation occurs when a binary mixture undergoes condensation, and the surface tension of the low boiling-point component is less than that of the high-boiling point component. This discovery was made by Mirkoverich and Missen in 1961 [2], [3]. On hydrophilic surfaces, Marangoni condensation in the dropwise regime can significantly enhance heat transfer, providing improved heat transfer performance.

1.2 Problem Statement

Condensers play a pivotal role in diverse engineering applications, and continual research in condensation heat transfer is focused on improving their efficiency for enhanced overall system performance. Numerous studies aim to optimize condenser performance by augmenting the convection heat transfer coefficient. This goal has been pursued through various avenues over the years, including modifying test tube geometries, exploring different condensing fluids, and adjusting conditions such as pressure, velocity, and temperature differentials. These efforts seek to make condensers more cost-effective and efficient in their heat transfer processes.

The aim of this project is to contribute new experimental data to the existing knowledge on Marangoni condensation. The focus is on studying the condensation process of downward vertical flow over horizontally positioned, water-cooled, plain tubes and wire wrapped tube. The performance of this system will be evaluated by varying the concentrations of the binary mixture, the diameter of the wire, the pitch, and adjusting the

flow rate of the cooling water. A comparison will be made with previously available data to assess the effectiveness and efficiency of the condensation process under different conditions. Then, a multi-optimization study of heat transfer performance of condensation pin-fin tubes will be conducted to determine the optimal geometrical parameters of pin-fin tubes that leverage their performance and achieve the highest possible enhancement ratio.

1.3 Aims and Objectives

The present project aims to achieve four primary objectives in the field of condensation heat transfer research:

- 1- Construction of a precise and thermally insulated apparatus dedicated to studying saturated vapor condensation on a single horizontal tube under low vapor velocity and atmospheric pressure conditions.
- 2- Expansion of the existing database on Marangoni condensation heat transfer on horizontal copper tubes by investigating various concentrations of binary mixtures to determine the most efficient compositions.
- 3- Investigation and addition of new and precise experimental data on Marangoni condensation heat transfer on horizontal wire-wrapped copper tubes with different wire diameters and pitches, as well as various concentrations of binary mixtures.
- 4- Conducting a multi-optimization study to enhance the heat transfer performance of condensation pin-fin tubes. This involves determining the optimal geometrical parameters of pin-fin tubes, considering the effects of gravity and surface tension, and exploring the potential of two new, low emission fluids (R-417a and R-1234yf) as alternatives to high emission fluids like R-22 and R-143a.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

Condensation heat transfer has been the subject of extensive research for a significant period. Initially, the primary focus was on increasing heat transfer by maximizing the available surface area. However, considerable efforts have been dedicated to enhancing condenser performance while reducing its size.

Later on, researchers made a crucial discovery regarding the significant role of surface tension forces in thinning the condensate layer, which ultimately leads to improved heat transfer. This insight has paved the way for significant advancements in condensation heat transfer.

Researchers have successfully identified the most effective geometries for enhancing heat transfer, starting with plain tubes, and moving on to more advanced configurations like integral tubes and pin-fin tubes. These advancements in tube design have greatly contributed to enhancing condensation heat transfer efficiency and have been applied in various industrial and engineering applications.

Indeed, various factors can influence condensation phenomena, such as the notable heat transfer enhancement achievable in the dropwise condensation regime, particularly in the context of Marangoni Condensation on hydrophilic surfaces. This unique phenomenon occurs due to surface tension gradients induced by temperature variations, causing condensate droplets to coalesce and flow along the hydrophilic surface. Unlike traditional

film-wise condensation, where the condensate forms a continuous film, dropwise condensation in Marangoni regime enhances heat transfer by promoting the continuous removal of condensate droplets from the surface. This leads to improved efficiency and better heat transfer rates, making it a promising approach for various practical applications like heat exchangers and thermal management systems.

The objective of the literature review is to present a current overview of understanding of condensation heat transfer on plain and geometrically improved tubes, covering both experimental and theoretical models, it explores several facets of condensation heat transfer using both theoretical and experimental models. The article additionally incorporates current binary-mixture condensation studies and advancements.

2.2 Free convection condensation

2.2.1 Theoretical studies

Nusselt's theory

In 1916, Nusselt had been started the first model of free-convection condensation for both horizontal tube and vertical plates, he assumed laminar flow, constant temperature along the tube, neglected the shear stress, saturated and rest vapor outside, and the heat is transferred through the condensation film only by conduction. Although these assumptions limited his model, it was essential and the base for all later scientific ones.[4]

The following analytical formulas were obtained for both the vertical plate and horizontal tube, starting with the moment equation with boundary conditions of no-slip and no shear stress at the interface.

For a vertical plate

$$\text{Nu} = 0.943 \left[\frac{\rho(\rho - \rho_v)gh_{fg}L^3}{\mu k \Delta T} \right]^{\frac{1}{4}} \quad (1)$$

For a horizontal tube

$$\text{Nu} = 0.728 \left[\frac{\rho(\rho - \rho_v)gh_{fg}d^3}{\mu k \Delta T} \right]^{\frac{1}{4}} \quad (2)$$

These results are not exclusive to a particular fluid, which is one of the advantages of his work.

Surface Temperature Variation

In his theory, Nusselt assumed that the characteristics of the condensate layer and the wall temperature were uniform. It appeared that the condensate film's uneven thickness along the tube, the temperature will be higher in the upper part of the tube where it is thinner and lower in the rear part where it is nearly infinite. As a result, tests reveal a strong angular dependence of surface temperature. Memory & Rose [5] who had investigated this topic. They observed that a very near approach to the surface temperature distribution is provided by.

$$T_w = a \cos \phi + \overline{T_w} \quad (3)$$

$$A = \frac{a}{\overline{\Delta T}} \quad (4)$$

where the ratio of the outside to inside heat transfer coefficients strongly influences the constant a . The wall temperature approaches the uniform (with angle) coolant temperature more closely when the interior heat-transfer coefficient is high.

The surface temperature is more severely influenced by angle when the outside heat-transfer coefficient is high.

A numerical solution was used to obtain the optimum heat transfer coefficient for a range of values of A (0–1). What was surprising about the results was that the Nusselt constant was equal to 0.7280 for all values of A , and the temperature variation had almost no effect on the mean heat transfer coefficient.

Enthalpy Convection Effect

In general, the latent heat of vaporization outweighs any potential sensible heat contribution to the heat flow for the condensation of vapors at moderate pressure, but in some situations, such as when there is a high pressure and a sizable temperature difference between the vapor and the tube surface, the sensible heat effect will have an impact on the heat transfer.

In 1952, Bromley [6] added thermal capacity to Nusselt's study after it had been overlooked in earlier analyses. Bromley tested two similar condenser tubes with the same temperature and physical parameters besides the heat capacity.

The thickness of the condensation film for both tubes was assumed to be the same in his first approximation, and he then solved the same equations as Nusselt, with the exception that he changed the term containing phase change enthalpy with the conductive heat transfer through the film. Ultimately, he demonstrated that through a straightforward

adjustment of the Nusselt equation, it becomes feasible to estimate the impact of the condensate's heat capacity on the heat transfer coefficient. The simplified equation for vertical tube is:

$$h = 0.943 \left[\frac{k^3 \rho(\rho - \rho_v) g h_{fg} (1 + 0.4 \frac{\Delta t C_p}{h_{fg}})^2}{L \mu \Delta T} \right]^{\frac{1}{4}} \quad (5)$$

For laminar film condensation on a horizontal cylinder, Sparrow & Gregg [7] conducted a boundary-layer study in 1959. In their work, they accounted for the inertia forces and the energy convection term, both of which are absent from Nusselt's theory. The fundamental conservation laws of mass, momentum, and energy apply to the condensation problem with neglecting static pressure gradient in the momentum equation as follow:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (6)$$

$$\rho \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = \rho g \sin \frac{x}{r} + \mu \frac{\partial^2 u}{\partial y^2} \quad (7)$$

$$\rho c_p \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = k \frac{\partial^2 T}{\partial y^2} \quad (8)$$

They began their investigation of the horizontal cylinder-appropriate boundary layer equations. These partial differential equations were then modified to ordinary differential equations.

The following boundary conditions have been used to define their model:

- 1- Due to the no-slip attribute of viscous flow and the impermeability of the wall, both velocity components must disappear at the surface of the cylinder ($u=0$, $v=0$ at $y=0$).
- 2- The temperature of the condensate next to the wall surface must be T_w
($T = T_w$ at $y = 0$).
- 3- At the liquid-vapor interface, the vapor provides almost no shearing resistance to the condensate flow ($\mu \frac{\partial u}{\partial y} = 0$ at $y = \delta$)
- 4- At the liquid-vapor interface, the temperature of the condensate T_{sat}
($T = T_{sat}$ at $y = \delta$)

Based on that they provided the heat-transfer values for the horizontal cylinder of the Prandtl number range between 0.003 to 100 using numerical solutions of the resulting ordinary differential equations. For fluids with any Prandtl number and a small value of $\frac{c_p \Delta T}{h_{fg}}$, the Nusselt number is provided by

$$Nu = 0.733 \left(\frac{g \rho^2 h_{fg} d^3}{\mu k (T_{sat} - T_w)} \right) \quad (9)$$

Which is very close to Nusselt prediction in his theory equation (2.2).

In 1959 also, Sparrow & Gregg [8], did the same but for vertical plate, solutions were found for Prandtl numbers between 1 and 100 for values of the $\frac{c_p \Delta T}{h_{fg}}$ parameter between 0 and 2.

These solutions join those of Blasius, Pohlhausen, Schmidt, Beckmann, and others in the boundary-layer family. Results of heat-transfer are provided. It was discovered that for

Prandtl numbers bigger than 1.0, the Prandtl-number effect—which results from the retention of the acceleration terms—is relatively negligible. Results for heat transfer at low Prandtl numbers (0.003-0.03) are published in their Appendix, and the impact of the acceleration terms is more noticeable.

In 1916, Nusselt accounted for the latent heat of vaporization h_{fg} as follows:

$$h_{fg}^* = h_{fg}(1 + CJa) \quad (10)$$

The value of C was estimated to be 3.8.

In 1950, Bromley proposed that C in his circumstance could be 0.5. Later, Bromley in 1952 made a more accurate prediction of C for film condensation and came up with same formula above, which equates to $C = 0.8$. This calculation was expanded by Rohsenow [9] in 1956 to take into consideration liquid crossflow at the liquid-vapor interface. He determined $C = 0.675$, then he adjusted it to 0.68. The implicit premise behind each of these estimates was that the Prandtl number was extraordinarily large.

For the liquid film during film condensation, Sparrow and Gregg [7], [8] gave a similarity solution to the boundary layer equations in 1959. Their findings demonstrated precisely how Pr and Ja affected the Nusselt number. They observed that the findings of their infinite Pr seemed to support the value of $C = 0.675$. A similar calculation was made for film boiling by Koh in 1962. His findings could not be described analytically, just like the Sparrow-Gregg solution. Additionally, he applied the more complex matching shear stress boundary condition at the films outside border.

In 1987, Sadasivan and Lienhard [10], attempted to determine the approximate value of C for Pr infinity in the film condensation problem. The previous authors found that C is

dependent on Pr and Jakob number, furthermore, when the numerical value of Ja is large enough to make the adjustment crucial, the impact of sensible heat becomes more and more significant in the case of film boiling. They had programmed the numerical results of the Sparrow and Gregg equations as well as the Koh equations modified for the limiting case where $u = 0$ at the liquid-vapor interface. For the range 0.6 to 1000 of Pr, they tested each set of equations over the range of Ja. Pr values below 0.6 are impracticable, and a Pr equal to 1000 is sufficiently close to infinity to render larger values superfluous.

Utilizing the ideal curve fitting technique, they come up with the following formulation:

$$h_{fg}^* = h_{fg} \left[1 + \left(0.683 - \frac{0.228}{Pr} \right) Ja \right] \quad (11)$$

Non condensable gases

The first fundamental transport equation for condensation in the presence of a non-condensable gas was created by Colburn and Houge in 1943 [11], and it was the simplest way to describe mass transfer in a mixture. According to their theory, diffusion over a thin film through the non-condensable layer regulated condensation mass transfer. The sum of two parallel resistances—the contribution of sensible heat and the contribution of latent heat—was how they defined the heat transfer process through mixing. The sensible heat contribution comes from the combination of conduction in the gaseous layer, a further term resulting from the energy of the condensation flow, and the latent heat that condensation releases.

The primary equation describes the heat balance at the mixture interface between heat transfer through the boundary layer and heat transfer through the condensate film, which

implies that the total amount of heat transferred from the gas mixture to the condensate at the interface equals the amount of heat transferred from the interface.

Buoyancy Effect

In 2014, Zhao et al. [12] created a new mathematical model to represent the air condensation on a horizontal cryogenic tube. In order to calculate the film thickness, the interfacial shear, and the heat transfer coefficients under various situations, the effects of superheating, sub-cooling, and buoyancy are examined for the condensation of air as well as other vapors with a focus on the effect of buoyancy. The findings demonstrate that superheating contributes more significantly to the increase of the heat transfer coefficient for air than for the other vapors; for example, when the vapor temperature is increased by 100 or 200 K, the average heat transfer coefficient increases by 10.3% and 24.3%, respectively, whereas the other vapors exhibit values that are much lower. Subcooling also has a deleterious effect on the rise in the heat transfer coefficient. Notably, the presence of buoyancy significantly affects the flow and heat transfer characteristics of superheated air condensation. It results in a reduction of the film thickness and an increase in the heat transfer coefficient, particularly noticeable at higher levels of superheating.

Conclusion

It can be concluded that research over the period after 1916 until now has proven the validity of the simple Nusselt theory results for the condensate film, for laminar natural convection condensation on a vertical plate and horizontal tube, Equations (1) and (2), for most practical purposes. There are currently formulae that are more precise, which cover the most extreme situations where inertia, convection, and vapor shear stress are all considered. However, the simplifying assumptions of the Nusselt (1916) theory should be

sufficient when addressing the condensate film in the majority of instances and when considering more complex geometry.

2.2.2 Theoretical studies

Pressure Effect

Cheng conducted two sets of R-152a condensation tests, one with constant vapor pressure and the other with a changing vapor pressure and heat flux. The condensate layer's physical properties were kept constant for the first set. However, examining the influence of changing physical properties in the second case is critical. The experimental data showed a 15 percent agreement with the Nusselt theory. After then, the condensation behavior of R-152a and R-12 were compared. When comparing R-152a to R-12, the heat transfer coefficient of the inner surface increased by roughly 20-25 percent. This rise is justified by its characteristics of higher latent heat and heat conductivity combined with a low viscosity value. [13]

Non-Condensable gas

In 1967, Kroger and Rohsenow [14] conducted research on the condensation of potassium in the presence of helium and argon, two non-condensable gases. The presence of these gases will influence heat transfer by creating a diffusional resistance. These gases also tend to gather at the cold surface and concentrate there, which reduces the heat transfer. For the instance of potassium condensing in the presence of helium, very good agreement is found in comparison to the theoretical estimation. The experimental results revealed a higher heat transfer coefficient for the potassium-argon mixture. They also generated curves that depict the distribution of the partial pressure and saturation temperature of potassium vapor close

to the cold surface; once more, the measured temperature agreed well with the calculated values.

The main issue has always been that even small amounts of non-condensable gases have a significant negative impact on steam condensation rates. This phenomenon has been researched in the past, particularly regarding forced convection, and in 1973, Al-Diwany and Rose [15], published the results of their measurements of the film condensation of steam in the presence of air, argon, neon, and helium, with special attention paid to avoiding forced convection effects. The findings show that for concentrations of non-condensing gas, heat transmission is reduced more significantly. In this work, the flow straightening section was used to ensure that the entering flow was uniform and normal to the condensing plate. They emphasized that under laminar free convection regimes, condensation rates can be reduced by as much as 50% with a non-condensable gas fraction of no more than 1% by mass. These findings also closely match the predictions of the free convection boundary layer theory.

Marangoni Condensation

Murase et al. experimentally measure heat flux and the temperature difference on the vapor side over a range of steam-ethanol combination compositions at atmospheric pressure using a horizontal condenser tube. In comparison to the heat flux values seen with pure vapor, the heat fluxes are consistently noticeably greater, especially at high vapor velocities. [16] The heat flux is lower than the predicted values for pure steam for larger ethanol concentrations and smaller vapor-to-surface temperature differences. Most of the cases showed that the heat flux decreases at the higher values of vapor side temperature

difference, and the maximum heat flux rate achieved at the lowest ethanol concentration, with the maximum vapor velocity.

Murase et al. [16] had conducted their measurements only at atmospheric pressure and low vapor velocities. Ali et al. [17] expanded on their work. Wide pressure variations were tested, and air in the vapor phase was carefully removed. Different coolant flow rates were looked at together with ethanol concentrations that ranged from 0.025% to 1.0%. Results demonstrated Marangoni condensation to significantly improved heat transfer compared to pure steam, with enhancement factors up to about 5. The compositions of ethanol that were most efficient fell between 0.02 and 0.1%. When the vapor velocity rose from 0.8 to 7.5 m/s at atmospheric pressure, heat transfer coefficients increased by 50%, and when the velocity grew from 5 to 15 m/s at 14 kPa, they increased by about 22.22%. The investigation included visual evidence of Marangoni condensation on the tube surface under various circumstances.

In 2021 Jivani et al.'s research extended Murase et al.'s and Ali et al.'s earlier work by focusing on lower ethanol mass concentrations and higher vapor-side temperature disparities. This study included a thorough analysis of ethanol mass concentrations ranging from 0.001% to 1.0%, all carried out at atmospheric pressure. For all ethanol mass concentrations and vapor velocities, it was shown that the heat flux and heat transfer coefficient increased, peaked, and then began to decline as the vapor side temperature differential increased. The range of mass ethanol concentrations from 0.01% to 0.025% showed the highest heat transfer coefficients. Notably, a vapor side temperature differential of 4 K and a mass concentration of ethanol of 0.01% were required to reach the highest enhancement ratio of 5.5.[18]

Ali et al. conducted an experimental investigation on the condensation of steam-ethanol mixtures using a finned copper tube. The boiler contained ethanol concentrations ranging from 0.025% to 1.0%. The range of vapor-to-surface temperature differences at atmospheric, 55 kPa, and 14 kPa pressures was studied. Maximum upstream vapor velocities during condensation were 15.0 m/s and 7.5 m/s at 14 kPa and atmospheric pressure, respectively. The study demonstrated a significant improvement in heat transfer, around three times higher than that of pure steam, for different vapor-to-surface temperature differentials. The results were contrasted with available data for steam-ethanol condensation on smooth tubes and pure steam condensation on finned tubes under similar pressure, vapor velocity, and vapor-to-surface temperature differential circumstances.[19]

2.2 Forced Convection Condensation

2.3.1 Theoretical Studies

In the earlier discussion, the condensate layer was moving due to gravity while the vapor around the tube was at rest. Numerous studies have been conducted in the past to examine the impact of vapor velocity on heat transfer for condensation on horizontal tubes for pure vapors. High vapor velocities will actually cause the condensation layer to experience shear stress at the vapor-liquid contact site. The shear force thins the liquid layer locally, which improves heat transmission.

The film is thinner on the top half and thicker on the bottom half for a vertical downflow. The vapor boundary layer separates after the separation point, which is situated at an angle that is typically between 80° and 180° from the top of the tube. This separation is what results in the thickening. Below the line of separation, the vapor moves in the direction counter to gravity, resulting in thickening and significantly reduced heat transmission.

Additionally, the vapor pressure gradient, which is more significant while the vapor is flowing due to the dynamic component of the pressure, is the source of this separation. This gradient has a positive forward portion that tends to thin the film and a negative rear portion that has the opposite effect.[20]

Shekrladze-Gomelaury [21] assumed that the shear stress at the liquid/vapor contact is equal to the vapor's loss of momentum. They ultimately developed a correlation for the condensation of pure vapor by using the idea that the condensation process is essentially equivalent to suction surrounding the tube on which condensation occurs. As a result, vapor boundary layer separation may be considered to be nonexistent. Then in 1974 Honda and Fuji [22] effectively tested that correlation, after analytically analyzing the scenario of slanted vapor flow within a plane orthogonal to the axis of the tube.

The analysis of [21] was expanded by DENN and Mills in 1969 [23] to produce solutions at normal gravity. For the purpose of analyzing regionally varying fluid characteristics, they established a reference temperature. They also showed that, except for low Prandtl number fluids, there is almost no inaccuracy when using the Nusselt assumptions to evaluate locally variable characteristics using that reference temperature, with the end result being extremely similar to boundary layer theory.

Analytical research on the effects of forced flow, non-condensable gas, and varied thermophysical parameters on laminar film condensation of pure and binary vapors at the forward stagnation point of a horizontal cylinder was conducted by Denny and South in 1972 [24]. Results from this investigation were obtained over a wide range of parameters, including bulk velocity, saturation pressure, temperature difference between the bulk and the wall, mass fraction of non-condensable gas, and cylinder radius. As anticipated, under quiet

conditions, the effect of the non-condensable gases was less severe than free convection. The heat transfer coefficient triples in value between 0.3 and 30 meter per second, exceeding the HTC for pure steam (Nusselt formulation) at higher levels. Heat flux grew until the wall-bulk temperature difference reached 30 K, beyond which it tended to an asymptotic value that might even be three times greater than the one reported at saturation circumstances.

In 1972, Fujii et al. [25] studied a two-phase boundary layer flow model with the condensate film's remaining intact for the entire surface and predicted the heat transfer coefficient of the vapor flowing on horizontal tube. By extending their analysis, they were able to solve the boundary layer equations while ignoring the pressure term analytically and numerically in the momentum equation, energy convection, liquid subcooling, and assuming a velocity profile to avoid the separation issue. Schmal [26] obtained the same equations as [25], but he did away with gravitational forces while continuing to disregard liquid convection and subcooling. Equation (12) displays the numerical output of the average heat transfer coefficients. On the average coefficients of heat transfer rate, there was a considerable amount of agreement between his work and the outcomes of the experiments.

$$Nu_{AVG} = X(1 + \frac{0.276}{X^4 Fr H})^{0.25} Re^{0.5} \quad (12)$$

Where:

$$X = 0.9 (1 + \frac{1}{RH}) \quad (13)$$

Fr: Froude number defined by author as

$$Fr = \frac{U_{\infty}^2}{gD} \quad (14)$$

H: nondimensional number of condensations defined as

$$H = \frac{C_p(T_s - T_w)}{PrL} \quad (15)$$

R: ratio defined as

$$R = \frac{\rho\mu}{\bar{\rho}\bar{\mu}} \quad (16)$$

SMM Ferreira [27] conducted a numerical study of the effects of non-condensable gases, gravitational field, and superheat on the laminar forced convection film condensation in 1973 to develop a more suitable analysis for forced convection condensation of vapor flowing normally to an isothermal circular cylinder. Both pure vapor and vapor coupled with an inert gas required this investigation to be conducted. He also included the gravitational effects, initially by having the gravitational field parallel to the flow field and then by having it anti-parallel to it. The study will also be expanded to take into account how vapor superheat affects a vapor's natural flow toward the axis of an isothermal circular cylinder during the condensation process. Results showed that higher values of the thermal driving force and non-condensable gas concentration can be found to correspond to lower values of heat transfer. The reduction of heat transmission was greater than 20% when the hardest conditions—a thermal driving force of 100°F and a 5 percent free stream gas

concentration—were used. His research also showed that the influence of the superheated vapor was not very noticeable when the concentration of the non-condensable gas was 2 percent with a thermal driving force of 80°F.

In 1979, Gaddis [28] developed models that took into account two-phase flow and used series expansions for the resulting ordinary differential equations to solve the boundary equations for liquid and vapor moving perpendicular to a tube for laminar film condensation. In the condensate film's energy equation, surface tension, viscous dissipation, and pressure are ignored. The distribution of the local values of the Nusselt number is provided by his numerical solutions.

In 1999, Homescu and Panday [29] were interested in the way turbulence in two phases can affect the condensation of a saturated vapor falling in vertical direction around a horizontal tube. This study applied an implicit finite difference analysis for solving boundary layer equations coupled at the condensed film – vapor interface by equating shear stress, longitudinal velocity, and mass flux at the interface. In their attempt to select the most accurate model among different available turbulence models, they found that adopting Kato's and Pletcher's models in the liquid phase and vapor phases, respectively, led to good agreement between the numerical and experimental results. This analysis also proved its potential to predict the condensation characteristics amidst turbulent vapors, and it revealed that turbulence has significant influence on the flow and heat transfer properties. The consequences of turbulence in a condensate film and the phase-change balance equation were examined theoretically using Colburn's analogy by Yang and Lin [30] (2005). The impacts of these on the average rate of heat transfer were also assessed, taking

into account the non-isothermal requirement for interfacial eddy diffusivity in the energy connection and non-uniform wall temperature change.

With an increase in the variable amplitude of the wall temperature, it was found that the eddy diffusivity effect's inclusion reduced the average heat transfer coefficient. The condition of the wall temperature has a negligible impact on the mean heat transfer coefficient as well.

In 2009, a two-dimensional, two-phase laminar film condensation from downward-flowing steam and lighter non-condensable air onto a horizontal tube was numerically constructed by Chen and Lin [31]. The governing equation was iterated concurrently at each axial cross-section until convergence, and then it was solved using the finite volume method. The simultaneous effects of inlet-to-wall temperature difference and inlet air concentration, the Reynolds number and other dimensionless number of the ratio between the gravity to the velocity of the mixture, were clarified by numerical analysis of both the local condensate film thickness and heat transfer characteristics. The results showed that as the tube temperature dropped, the local Nusselt number and film thickness both increased significantly.

2.3.2 Experimental Studies

At high velocity and, for a refrigerant, at intermediate vapor velocity, there are notable differences between theory and experiment heat transfer results. These could be the result of flaws in the theory, errors in the experiments, or even both. Measurements using ethanediol (ethylene glycol), whose properties allow for high vapor velocities to be generated under practical operating settings, as well as with refrigerant-113, have been made to cast more light on these concerns by Lee et al.[32]

This analysis had validated discrepancies between theory and experiment for R-113 that Honda [33] observed previously in 1982. The reported heat transfer coefficients at the higher vapor velocities greatly exceed the predicted values, and the rate at which the coefficient increases with velocity is faster than predicted by theory. For ethanediol, the lower mean heat transfer coefficients may possibly be partly explained by the tube's very high temperature variation as stated by Fujii in 1981. The mean Nusselt numbers produced by the analytical calculations were closer to the experimental values. However, careful comparisons revealed that the predicted temperature drop across the condensate film was, over the majority of the tube, significantly smaller than what was actually seen. Past the vapor boundary layer separation point, the theoretical temperature drop was substantially greater than the actual value.

Film condensation of downward flowing R-113 vapor on single smooth horizontal tubes at near atmospheric pressure was examined over a wide range of vapor velocity and condensation temperature difference by Honda et al. A dye tracer was used to visualize the flow of condensate. Three main flow regimes can be seen: flat surfaces, two-dimensional waves, and three-dimensional waves. The condensate film abruptly thickened in the last flow regime at an angle around 1.75 rad from the tube top. In the thick film region, condensate was seen to be turbulently mixing. The laminar two-phase boundary-layer theory was compared to the preceding heat transfer results for R-113 and R-21. It was discovered that the point at which the experimental results deviated from the theoretical prediction depended on a dimensionless number that provided a threshold for switching between smoothing and wavy condensation surfaces.

The first to investigate experimentally the condensation of steam with the presence of air as a non-condensable gas on an isothermal surface was D.F. Othmer in 1929 [34]. He constructed a methodical experimental bench and investigated the effects of the temperature, pressure, and concentration of the non-condensable gases that were mixed with the steam. He has established the first applicable correlation for condensation deterioration brought on by non-condensable gas. The findings of measured data on a stagnant environment-placed, 1.22 m long, 76.2 mm diameter copper tube of the overall heat transfer rate for a range of surface temperatures revealed a reduction in the condensation heat transfer coefficient of nearly 50% was observed in the presence of 0.5% non-condensable gas volume fraction, and the partial pressure and the saturation temperature are also decreased when compared to the case of condensation with pure steam that predicted by Nusselt.

In order to obtain precise heat-transfer data for film-wise condensation from pure vapors (steam and R113) and vapor-gas mixtures (steam-air, steam-hydrogen, R113-air, and R113-hydrogen) flowing vertically downward over a smooth horizontal tube, Lee and Rose [35] summarized prior experimental processes and systems and constructed a more perfect setup in 1983. Thermocouples implanted in the tube wall were used to measure the surface temperature. Coolant characteristics were measured and compared to condensate collection results to determine the coolant heat flux. The electrical power input to the boiler was used to determine the vapor mass flow rate. The current pure vapor results, for both steam and R113, at moderate velocities, were generally consistent with theoretical equation, which included the assumption of uniform wall temperature. He attributed these inconsistencies to the non-uniformity of wall temperature by noting that the coefficients are somewhat

smaller at the higher velocities produced with steam than those predicted by theory equation. A straightforward approximation equation based on boundary-layer theory produces results for all four vapor-gas mixes that are in very good agreement with the measured mass transfer resistance. The current findings for steam-air mixes are well in line with past findings.

$$Nu \widetilde{Re}^{-1/2} = x \left(1 + \frac{0.276F}{X^4} \right)^{1/4} \quad (17)$$

Whereas:

$$X = 0.9(1 + 1/G)^{1/3} \quad (18)$$

Pin-fin tubes utilize the advantage of fins to enhance the heat transfer performance. The existence of fins contributes to magnifying the surface area, and further enhancement is brought due to the shorter and thinner boundary layers resulting from small pin height to tube circumference ratio, which reduces the thermal resistance of the boundary layers. However, condensation retention or flooding between the tube's fins at the bottom causes a loss in "active" surface area, which in turn causes a reduction in the heat transfer to that portion of the tube. Several studies have shed light on the performance of pin-fin tubes and investigated the effect of various parameters, including the forming material, the geometrical aspects, and the condensate fluid on the heat transfer performance of pin-fin tubes.

Sukhatme et al. [36] revealed the findings of experimental investigations on heat transfer during film condensation of R-11 vapor on single horizontal integral-fin tubes by altering fin density, semi-vertex angle, and fin height. They improved the tubes by adding grooves

with different heights which resulted in an enhancement ratio of 12.3. Briggs [37] introduced an experimental study through which they analyzed the efficiency of condensation of R113 and steam on three-dimensional pin-tubes. They emphasized their superior performance over two-dimensional fin tubes with less accumulation of condensate. Baisar et al. [38] provided an experimental investigation on the effect of circumferential as well as longitudinal pin spacing on the steam condensation on pin-fin tubes. By using pin-fin tubes, heat transfer was much more enhanced compared to integral-fin tubes with even similar dimensions. The heat transfer performance of pin-fin tubes manufactured from different materials was studied by Ali et al. [39] where they used ethylene glycol and R-113. This study addressed the effect of thermal conductivity and pin height on heat transfer of pin-fin tubes. Ali et al. [40] studied the effect of pin geometry on heat transfer performance of ethylene glycol condensed on pin-fin tubes. The tubes enhancement was equivalent to double the active area enhancement, and it was directly proportional to the pin height and inversely proportional to the circumferential pin spacing.

Pin-fin tubes made of different materials (copper and stainless steel) were experimentally tested by Qin et al. [41]. The results indicated that copper tubes performed better thanks to their greater pin heights and higher thermal conductivity. Optimal profiles were obtained for vertical fins with varied cross sections by Kundu et al. [42]. Fins were subjected to saturated vapor condensation under free convection. An extension of this study was conducted by Kundu [43] and Kundu and Ghosh [44] in which they included both free and forced convection condensation respectively. Pins' conjugate conduction effects were considered. However, by selecting fin profiles, it was possible to ignore surface tension

effects and model the condensation process using gravity drainage alone, and gravity with vapor shear.

The work of Ali et al. [45] revealed that pin-fin tubes showed higher condensate retention angles compared with integral-fin tubes having identical geometry. The article proposed a formula for the retention angle on pin-fin tubes which provides results with at least 85% accuracy. A semi-empirical correlation has been established for condensation on horizontal pin-fin tubes by Ali et al. [46] based on the methodology employed in the models by Rose [47] and Briggs and Rose [48] to account for the combined influence of gravity and surface tension. The dependency of heat transfer performance on the fluid and geometry of tubes was proven in this work, which opens the horizon for optimization attempts. Ali [49] established a model that can predict the condensate flooding on horizontal pin-fin tubes as a result of free convection by following the principles of static force equilibrium. Various fluid and tube geometry combinations were tested using this model and compared with experimental results collected from the literature. For most of the available data, error of prediction did not exceed $\pm 15\%$. Ali et al. [50] highly accurate data were published for the condensation heat transfer of square wires wrapped around horizontal tubes utilizing steam as a fluid.

CHAPTER 3

Methodology and Experimental Apparatus

3.1 General Apparatus Layout

Vapor will be produced by an electrical immersion heater fitted in a stainless-steel boiler, yielding 3 kW of total electric power. The boiler's vapor turned 180 degrees, released vertically into the testing area, and then condensed within the test tube. Any extra vapor will condense in the secondary condenser situated beneath the testing area. The obtained liquid was circulated back to the boiler using a downward pipe, facilitated by gravitational force. Cooling water was delivered to the test tube by a 0.5 hp pump connected to a float-style flow meter. All equipment components were constructed from stainless steel and were well-insulated to reduce heat loss.

3.2 Test Section

The test section comprises a circular structure with a cross-sectional shape, featuring an inner diameter of 70 mm. Positioned at the front of this section is a transparent Pyrex window, enabling visual monitoring during the condensation process. Horizontally oriented, the test tube is fixed with PTFE (polytetrafluoroethylene), commonly known as Teflon. Upstream, within the vapor stream, resides a sealed thermocouple pocket. Teflon inserts are fixed at the inner and the outer of tube surface, serve to thermally insulate the inactive part of the test tube, located within the mixing box, ensuring uniform heat transfer lengths on both inner and outer surfaces while preventing axial heat transfer. In regions of the test tube not exposed to vapor, these inserts reduce the internal diameter by 8 mm.

Teflon mixing boxes are affixed to the entrance and exit of the test tube to facilitate measurement of steady-state coolant temperatures.

3.3 Test Tube

This study employs a smooth copper tube with a total length of 190 mm, of which 70 mm is subjected to condensing vapor. The internal tube diameter is 11 mm, while the external diameter is 13 mm. For the wire-wrapping study, a copper wire with a diameter of 0.5 mm is used on the same copper tube.

Within the test section, three J-type thermocouples are incorporated: the first thermocouple is situated above the test tube to measure the vapor temperature before condensation occurs on the tube, while the second and third thermocouples are employed to monitor the temperature of the coolant water before and after passing through the test section. These thermocouples are inserted into the Teflon mixing box to ensure consistent and steady temperature measurements.

3.4 Secondary Condenser

After the test section's partial condensation, the leftover vapor traveled to the second condenser, where it underwent full condensation, turned into liquid, and then went back to the boiler. The centrifugal pump immediately supplied the cooling water required for the auxiliary condenser.

3.5 Boiler Power

A 3KW immersion heater is employed to produce vapor from a 12-liters water tank. The power input to the boiler is measured using an analog voltmeter, which gauges the resistance of the heater.

3.6 Experimental Procedure

Before installation in the mixing box, procedures were undertaken to prepare the test tube, ensuring film-wise condensation for experiments involving pure steam, and eliminating any contamination or residue from previous use. The test tube was first submerged in a solution containing Aseculfame Potassium, Acidifier Phosphoric Acid, and Carbonated Water. Active agitation during immersion ensured the removal of all trapped air bubbles, ensuring thorough contact between the solution and the tube surface. After immersion, the tube underwent sequential rinsing with tap water and distilled water. Finally, a small amount of water was applied over the tube to ensure the formation of a continuous film. Following these steps, the treated tube was then positioned within the test section.

Before turning on the heater, cooling water was circulated through the test tube at different flow rates, covering the range used in the condensation tests. This was done to evaluate the temperature difference caused by frictional dissipation in both the tube and mixing boxes. Later, the temperature difference resulting from frictional dissipation was subtracted from the temperature difference observed after the condensation process. This allowed for the isolation of the temperature rise specifically associated with the condensation phenomenon.

After powering on the device, it ran continuously for about thirty minutes to reach steady-state conditions and remove any residual air from the system through a small hole placed after the test section. Once the temperature and pressure of the vapor stabilized, the cooling water pump was operated at different flow rates covering the required range. Measurements were then taken for the temperature and pressure of the vapor before the test tube, along

with the temperature of the inlet and outlet coolant water, at these different coolant flow rates.

After adjusting the coolant flow rate, a thirty-second stabilization period was allowed before data collection. Visual monitoring through the Pyrex glass in the test section ensured continuous observation of the test tube to prevent dropwise condensation during pure steam experiments.

Once all flow rates were tested, the heater was switched off, while the coolant water and secondary condenser continued to operate for 30 minutes. This ensured complete condensation of all vapors in the system, returning it to the boiler as a liquid.

3.7 Data Processing

3.7.1 Rate of Heat Transfer

The heat transfer rate was determined by measuring the flow rate of the coolant water and the temperature difference between the coolant water at the inlet and outlet of the mixing box.

$$Q = \rho_c V_c C_{pc} \Delta T_c \quad (20)$$

Q = Rate of heat transfer, W

ρ_c = Density of the coolant water, Kg/m³

V_c = Volumetric flow rate of the water, measured from flow meter, m³/s

C_{pc} = Specific heat at constant pressure, J/Kg.K

ΔT_c = Temperature difference between inlet and outlet coolant water, K.

The heat flux is calculated as the rate of heat-transfer by the outside surface area as follow,

$$q = \frac{Q}{\pi DL} \quad (21)$$

D = The outside diameter of the tube, m

L = The length of the tube, m

To calculate the velocity of the generated vapor

$$U_v = \frac{m_v}{\rho_v A_{ts}} \quad (22)$$

Where:

m_v = Vapor mass flow rate, Kg/s

A_{ts} = Cross sectional area of the test-section.

The vapor mass flow rate approaching the tube was calculated by observing a steady flow energy balance between the boiler and the output of the auxiliary condenser. This calculation assumed negligible changes in potential and kinetic energy, as well as losses from the boiler input.

$$m_v = \frac{Q_b}{c_{pc}(T_{sat} - T_r) + h_{fg}} \quad (23)$$

Where:

Q_b = The boiler power input, W

T_{sat} = Saturation temperature, K

h_{fg} = Specific enthalpy calculated at T_{sat} , J/Kg

T_r = Temperature of the liquid returning to the boiler, K

The vapor side convective heat transfer coefficient and tube surface temperature were calculated indirectly due to the complexity of the wire-wrapping tube with different pitches. Direct measurement of the tube surface temperature using thermocouples was impractical for our experiment. Instead, we relied on an indirect method based on the thermal resistance of conduction through the tube's thickness and convection on both the coolant and vapor sides.

The overall heat transfer coefficient is calculated based on the 3 resistances as follows:

$$\frac{1}{U_0 A_0} = \frac{1}{\alpha_c A_i} + \frac{\ln \left(\frac{D}{D_i} \right)}{2\pi l K_w} + \frac{1}{\alpha A_0} \quad (24)$$

Whereas,

U_0 = The outside overall heat transfer coefficient, W/m² K

A_0 = The outside surface tube area, m²

A_i = The inside surface tube area, m²

α_c = Convective heat transfer coefficient of the coolant side, W/m² K

α = Convective heat transfer coefficient of the vapor side, W/m² K

D_i = Inside tube diameter. m

D = outside tube diameter. m

l = The length of the pipe exposed to condensation, m

K_w = Thermal conductivity of the copper at average surface temperature, W/m K

The equation above is used to calculate the Convective heat transfer coefficient of the vapor side, thus, the temperature of both inside and outside wall surface.

The Sieder-Tate correlation, developed in 1936, is a widely used method for estimating the Nusselt number in turbulent flow within pipes. This correlation provides a means to calculate the convective heat transfer coefficient on the coolant side of pipes.

For convective heat transfer coefficient of the coolant side

$$Nu_c = a Re_c^{0.8} Pr_c^{1/3} \left(\frac{\mu_c}{\mu_w} \right)^{0.14} \quad (25)$$

In this correlation, the constant a is determined through least square fit using the direct approach for calculating the surface temperature of the walls. Typically, a has a value of 0.04. The μ_c is evaluated at the average coolant temperature between the inlet and outlet. However, the μ_w is determined at the inside wall temperature, which is initially unknown. An iterative process is employed until the wall temperature converges within a specified tolerance, below 0.001 Kelvin. This iterative approach ensures an accurate estimation of the wall temperatures.

Pitch

1.6 mm



2 mm



2.4 mm



2.9 mm



Figure 1 Wire-wrapping tubes used in present investigation.

CHAPTER 4

Measurements of Heat Transfer for Steam-Ethanol

Mixtures Condensing on a Horizontal Smooth Tube

4.1 Introduction

This chapter presents the results of an experimental investigation into condensation heat transfer on horizontal smooth tubes. Initially, measurements of pure-steam condensation heat transfer were conducted to verify the accuracy and reliability of the experimental setup, instruments, and methodologies. Subsequently, data pertaining to steam-ethanol mixture concentrations of varying compositions were incorporated into the study. The primary significance of this dataset is its ability to clarify the various heat transmission mechanisms, transition zone, drop-wise, and film-wise condensation during the process of Marangoni condensation of steam-ethanol mixtures.

For each experiment, a smooth copper tube with an outside diameter of 13 mm and an inner diameter of 11 mm was employed. The active heat transfer zone of the tube was 70 mm long. The test segment's pressure was measured with a digital manometer, and the result showed that the air pressure was 101 kPa. In the experimental stage, particular care was taken to reduce errors caused by non-condensable gases, such air. During each experiment, a consistent vapor velocity of 0.48 m/s was maintained.

The coolant flow rate was systematically increased in increments of 2 l/min, ranging from 6 to 20 l/min. At each flow rate, three J-type thermocouples were strategically positioned:

one for measuring the input coolant water temperature, another for monitoring the outlet coolant water temperature, and a third embedded just before the test section to record the temperature of the steam preceding the test tube. Additionally, visual observations were facilitated through the capture of videos and images documenting the diverse condensation modes observed at various stages of the experiments.

Consistent with the validation objective, the mass fractions chosen were the same as those obtained by Jivani et al. (2021) in their experiments. To be more specific, ethanol mass fractions of 0.025%, 0.05%, and 0.1% were used in order to align with the findings of Jivani et al. Furthermore, in order to enhance the dataset with new perspectives, three previously untested concentrations were added: 0.0125%, 0.075%, and 0.3%. The primary objective of this chapter is to add more data pertaining to steam-ethanol concentrations while concurrently validating the experimental apparatus through the replication of prior findings.

4.2 Pure Steam Condensation on Horizontal Smooth Tubes

Certainly, before proceeding to discuss the results regarding Marangoni and wire-wrapping effects, it is crucial to validate the experimental results in the context of earlier investigation. This validation process involves comparing the experimental results obtained from the smooth tubes with Nusselt's theory for free convection, as proposed in Nusselt (1916). In Figure 4.1, the experimental data for pure steam condensation on smooth tubes are compared with Nusselt's theory (1916). The plot illustrates the heat flux plotted against the vapor side-to-surface temperature difference.

It is observed that current experimental results marginally exceed the predictions of Nusselt's theory (1916), mainly because of the relatively small downward vapor velocity.

However, overall, figure 2 shows that there is a clear and good agreement between the theoretical predictions and the experimental data.

Videos were recorded during each condensation experiment to ascertain the mode of heat transfer. As expected, the mode of heat transfer observed during the condensation of pure steam on smooth tubes was film-wise condensation. Figure 3 presents snapshots of film-wise condensation captured at various flow rates.

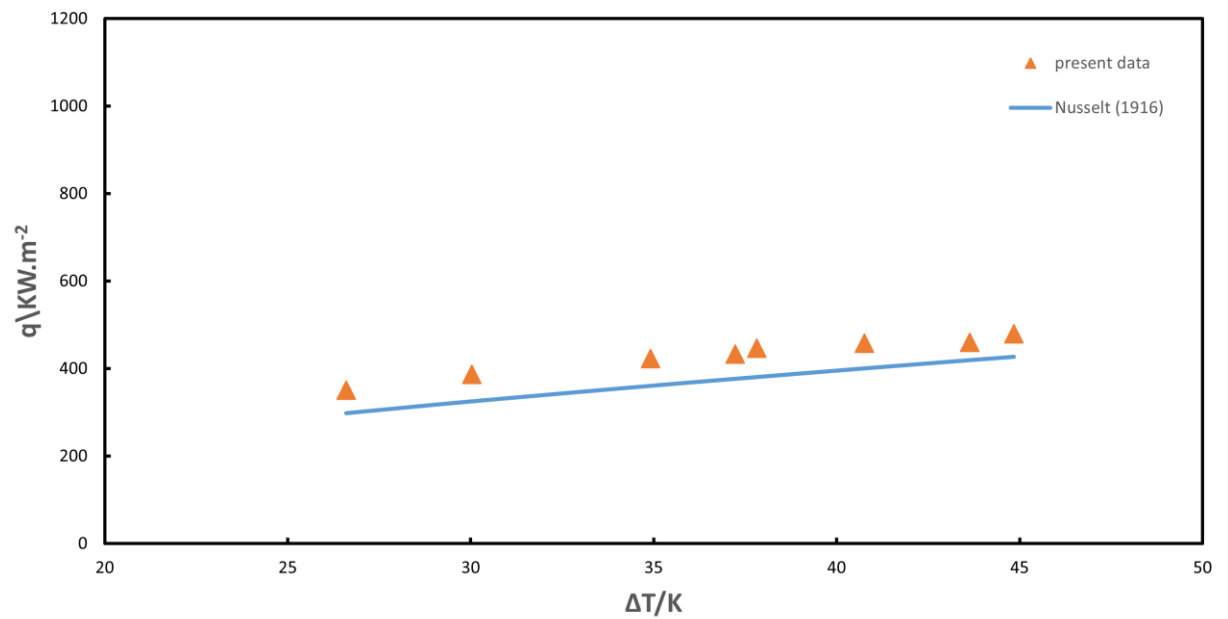


Figure 3 Comparing Nusselt's 1916 theory with pure steam data on smooth tubes.

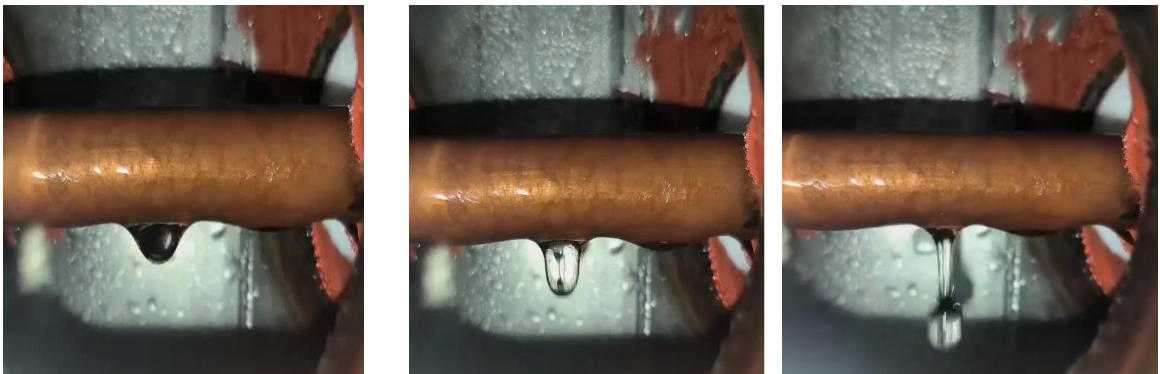


Figure 2 presents snapshots of film-wise condensation captured on horizontal smooth tube.

4.3 Results for condensation of steam-ethanol mixture on horizontal smooth tubes.

Figures 4 illustrate the relationship between heat flux and vapor side-to-surface temperature difference across various flow rates for three ethanol mass concentrations (0.025%, 0.05%, 0.1%). These plots provide a comparison with previous experimental data on steam-ethanol condensation by Jivani et al. (2021), presented for the same concentrations and vapor velocity. The solid green line represents theoretical predictions based on Nusselt's (1916) equation for pure steam (equation 2.2). The current experimental data are denoted by red triangles, while previous data from Jivani et al. are represented in blue. For reasons of validity, these three concentrations were duplicated. Figures show excellent agreement with the earlier findings, confirming the validity and dependability of the present collected data. The agreement between the results of the present experiment and the previous ones gives confidence in the accuracy of the data that was obtained.

The relationships between heat flux and convective heat transfer coefficient and the difference in temperature between the vapor side and the surface are shown in Figures 5 and 6 for six different ethanol mass concentrations (0.0125%, 0.025%, 0.05%, 0.075%, 0.1%, and 0.3%). The main objective of this section was to expand the focus of the investigation to identify the transitional stage of the condensation mode across a wider range of steam-ethanol concentrations.

Through the visual observations and the plotting of corresponding data points on the heat flux and convective heat transfer coefficient graphs against vapor side-to-surface temperature difference, the transition phase became clear. Notably, this transition was first observed at lower ethanol mass fractions (0.0125%). At this concentration, initial visual indicators included the presence of small drops on the tube surface at very low vapor side-

to-surface temperature differences. Subsequently, a pronounced dominance of film-wise condensation was observed.

Visual observations and the convective heat transfer coefficient clearly showed a transition for the next four concentrations: 0.025%, 0.05%, 0.075%, and 0.1% from dropwise condensation, with small drops covering the tube at low vapor side-to-surface temperature, to wavy film-wise condensation, with relatively bigger drops at the higher vapor side-to-surface temperature. The heat transfer coefficient reaches its highest value at low vapor-to-surface temperature differences, according to figures, and dropwise condensation can be seen visually. The convective heat transfer coefficient falls as the vapor side-to-surface temperature differential increases, and a shift from the dropwise to the wavy film-wise mode is noted. During the transition phase, there is a relative decrease in the corresponding heat flux values, because higher vapor side-to-surface temperatures are typically associated with higher heat fluxes. To signify the beginning and end of the transition region.

Furthermore, as the vapor side-to-surface temperature difference increases, a wavy film-wise condensation mode is observed, accompanied by a significant decrease in the heat transfer coefficient. These observations suggest that at higher vapor side-to-surface temperatures, complete film-wise condensation is expected. This observation aligns perfectly with earlier findings.

For the 0.3% steam ethanol concentration, an interesting trend is observed in the convective heat transfer coefficient. Initially, the coefficient remains relatively low at low vapor side-to-surface temperature differences. However, as the vapor side-to-surface temperature difference increases, the convective heat transfer coefficient starts to rise, reaching a maximum before beginning to decrease again.

Visual observations at this stage reveal a transition from wavy film-wise condensation with large drops to dropwise condensation with relatively smaller drops appearing on the tube surface. Interestingly, despite the low heat flux values at low vapor side-to-surface temperature differences, the heat flux values exhibit a significant increase compared to previous concentrations.

These trends are attributed to the combined effects of a change in condensate mode in the liquid phase and the diffusion resistance in the vapor phase. Importantly, these observations align with the findings of Jivani observations.

(a)

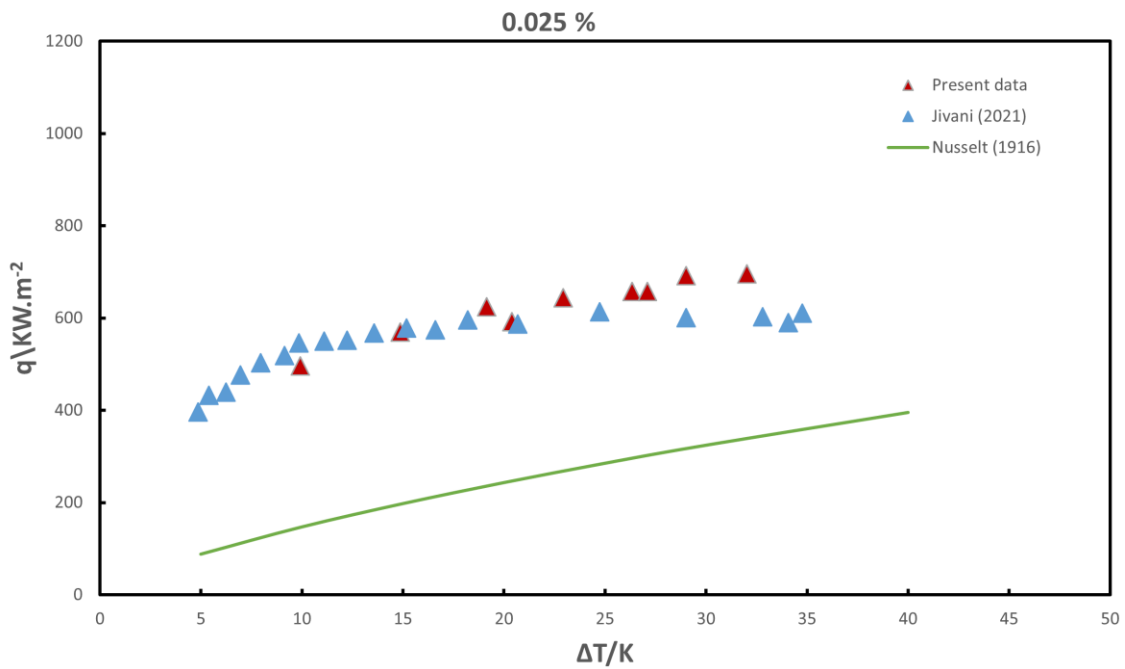
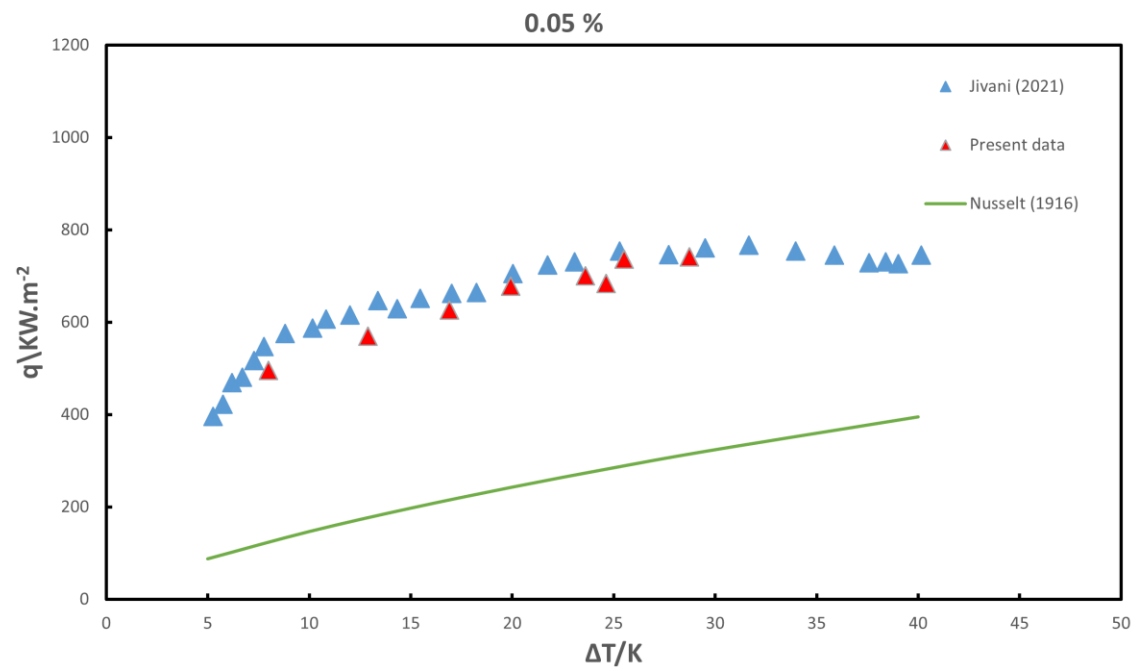
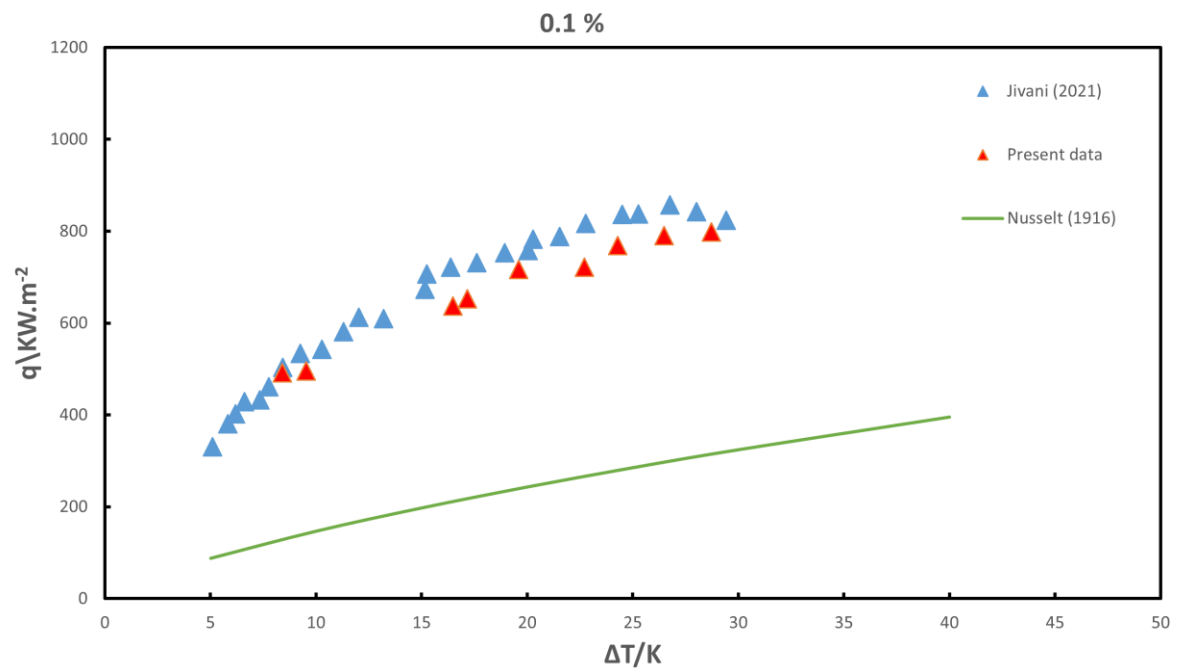


Figure 4 (a)-(c) illustrates a comparison of the variation in heat flux against vapor side-to-surface temperature difference for three ethanol mass concentrations: (a) 0.025%, (b) 0.05%, and (c) 0.1%. The vapor pressure in the test section is approximately 101 k

(b)



(c)



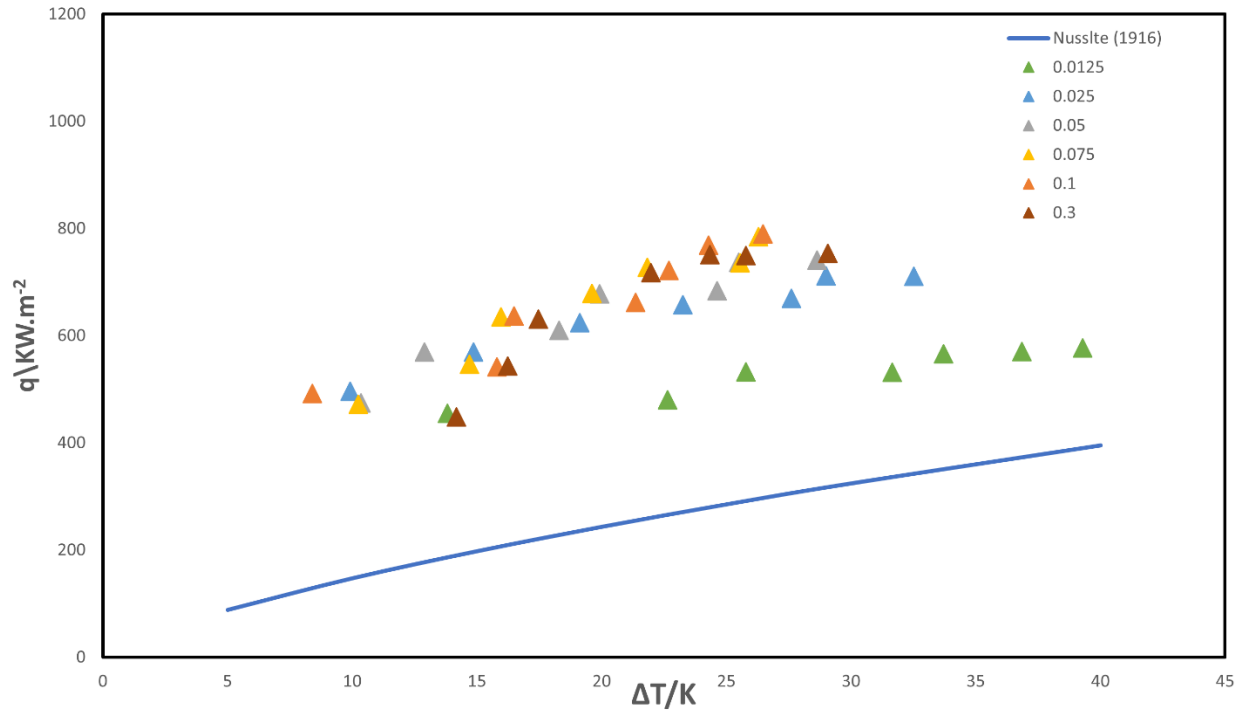


Figure 5 Heat flux and the difference in temperature between the vapor side-to-surface temperature difference for six different ethanol mass concentrations (0.0125%, 0.025%, 0.05%, 0.075%, 0.1%, and 0.3%).

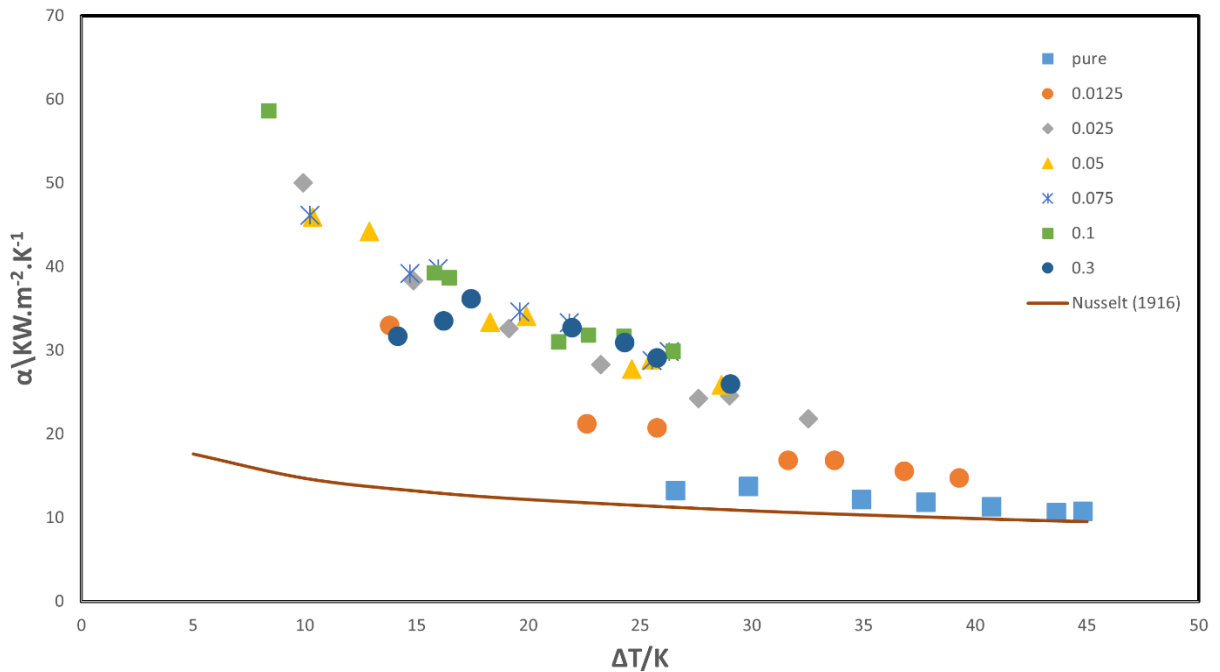


Figure 6 Convective heat transfer coefficient and the difference in temperature between the vapor side-to-surface temperature difference for six different ethanol mass concentrations (0.0125%, 0.025%, 0.05%, 0.075%, 0.1%, and 0.3%).

4.3.1 Visual Observation

Steam-ethanol condensation experiments were recorded using a 108-megapixel camera, showcasing one of the highest-resolution camera sensors available. Figure 7 shows the change of condensate appearance as the vapor side-to-surface temperature difference increases for a constant mass concentration (0.0125%), visual observation is compared with the heat transfer coefficient graph shown with visual images. Initially, at low vapor side-to-surface temperature differences, a mixture of drops and rivulets is observed. As the vapor-to-surface temperature difference increases, the presence of rivulets and drops diminishes, indicating a transition to complete film-wise condensation. These visual observations are in complete alignment with the behavior of the convective heat transfer coefficient. At very low flow rates, the convective heat transfer coefficient exhibits significant elevation before decreasing to values very close to those observed in the case of pure steam.

Figures 8-11 illustrate the behavior of ethanol mass concentrations (0.025%, 0.05%, 0.075%, and 0.1%) at different flow rates (6, 12, and 18 liters/min). Initially, the condensate exhibits characteristics of dropwise condensation, accompanied by higher heat transfer coefficients. As the condensation process progresses, the condensate gradually transitions into a wavy film, coinciding with a decrease in the heat transfer coefficient. This trend closely follows the behavior of the convective heat transfer coefficient. It is anticipated that at higher vapor side temperature differences, the expected mode of condensation would be film-wise condensation. However, this mode of condensation was not observed due to the limited availability of coolant flow rate.

The behavior of a 0.3% ethanol concentration at different flow rates is shown in Figure 12. As the difference in temperature between the vapor side and the surface increases the

convective heat transfer coefficient increases initially before peaking. Visual studies show that the condensation process first occurs in a dropwise manner, with big droplets first appearing and then gradually changing into denser, smaller drops. As the heat transfer coefficient lowers, the condensate appearance goes back to a relatively bigger drops with additional rises in the vapor side-to-surface temperature difference. There is also a hint of a wavy film as increasing in the vapor side-to surface temperature difference.

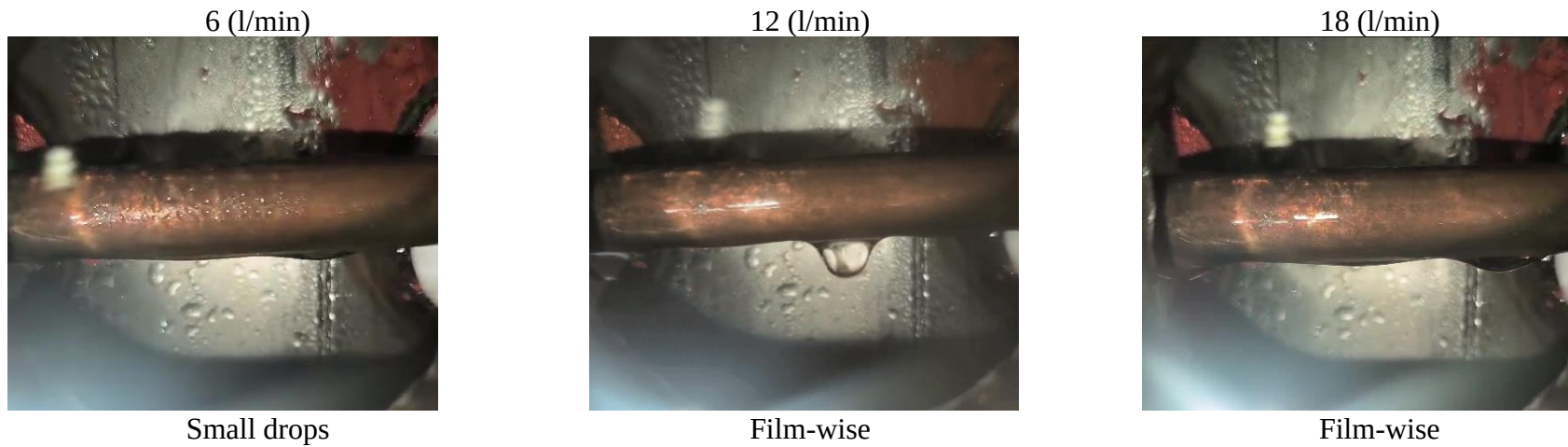
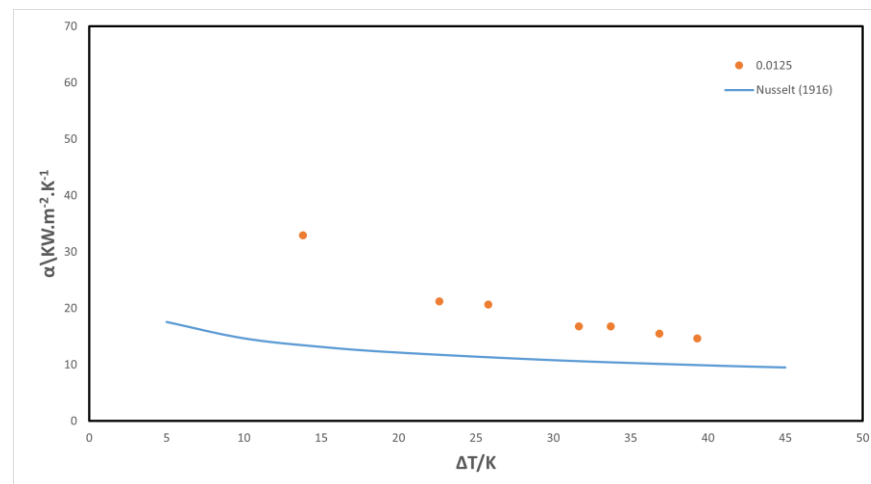
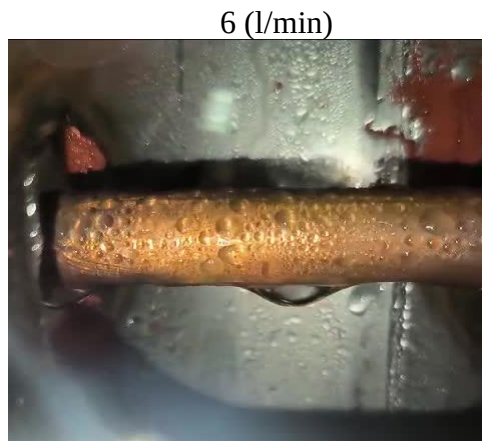
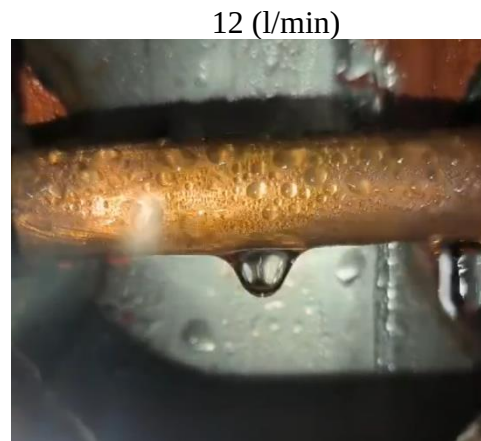


Figure 7 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a constant mass concentration (0.0125%)

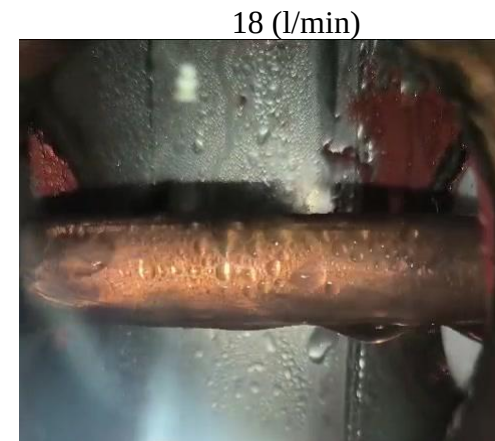




Small drops

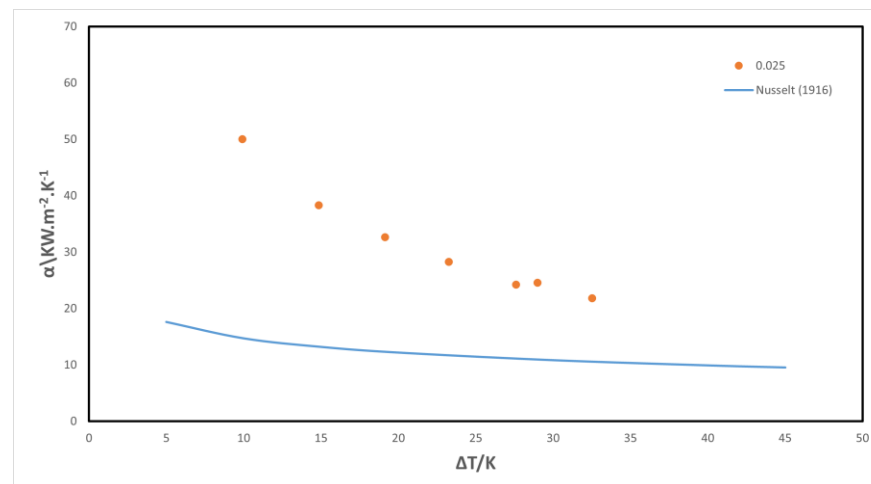


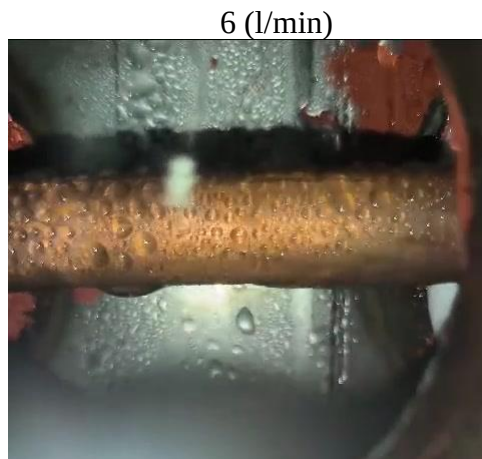
Large drops



Wavy film-wise

Figure 8 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a 0.025% concentration.

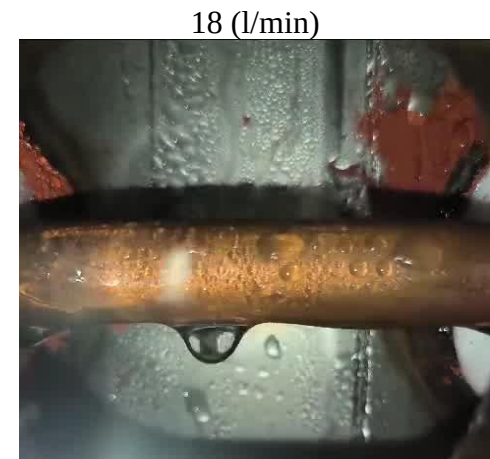




Small drops

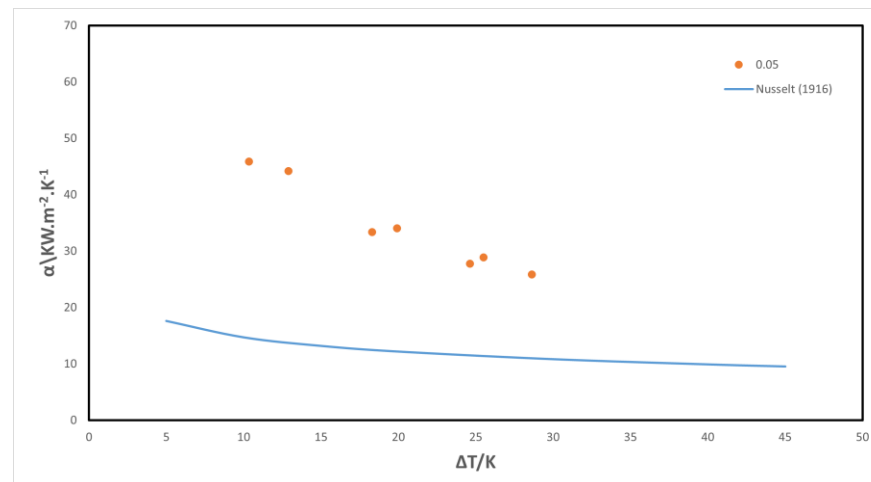


Large drops



Wavy film-wise

Figure 9 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a 0.05% concentration.



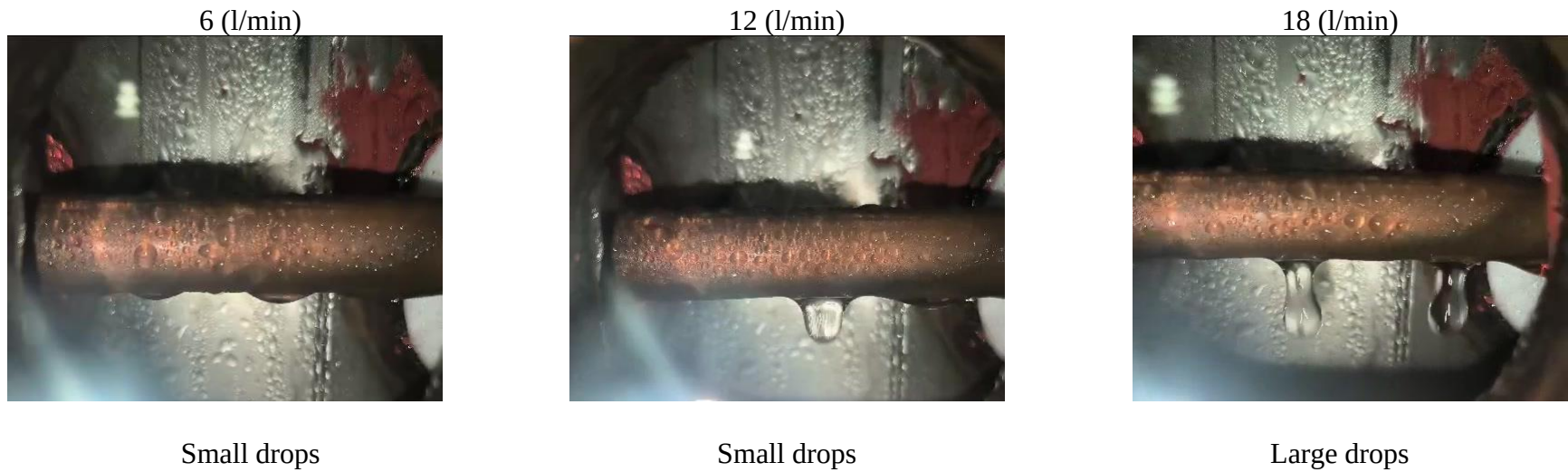
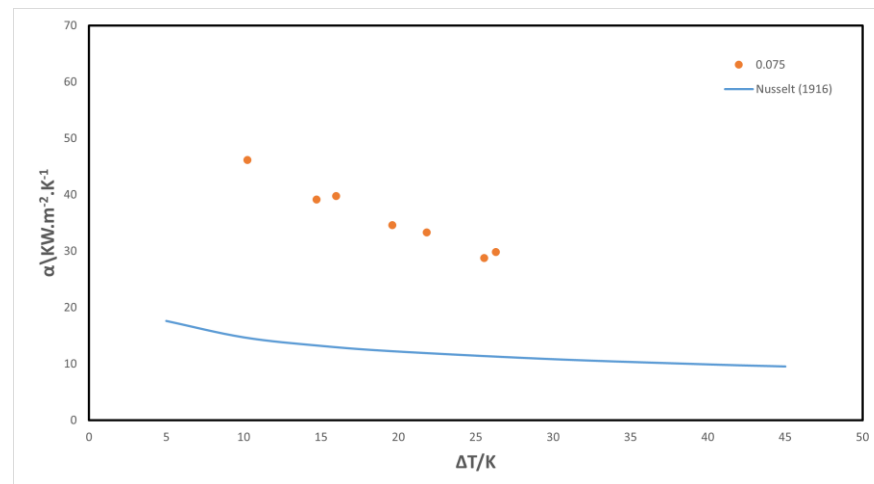
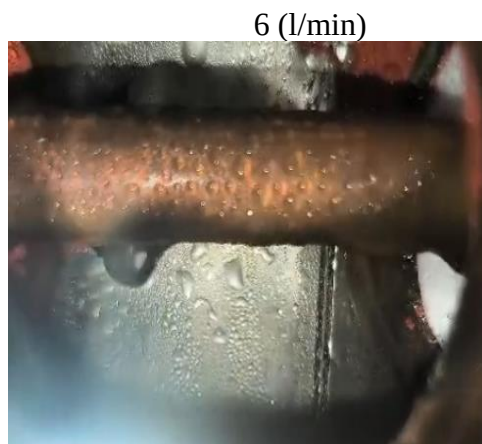
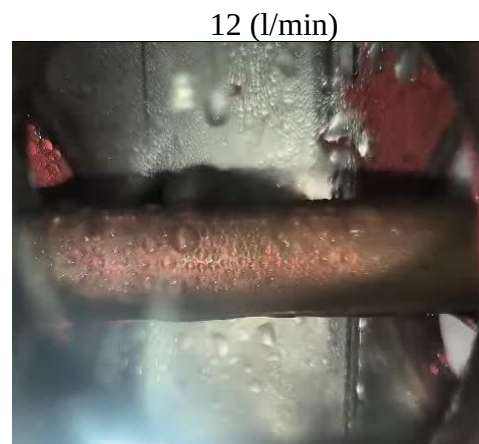


Figure 10 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a 0.075% concentration.





Small drops

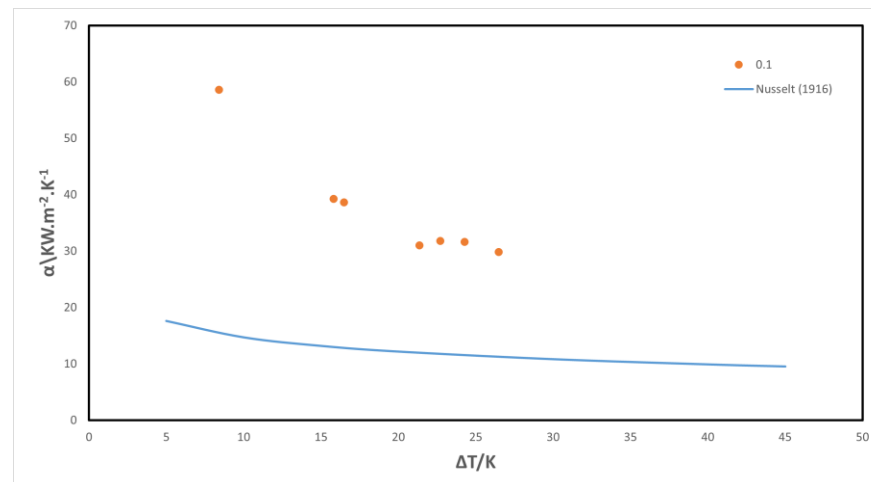


Small drops



Large drops

Figure 11 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a 0.1% concentration.



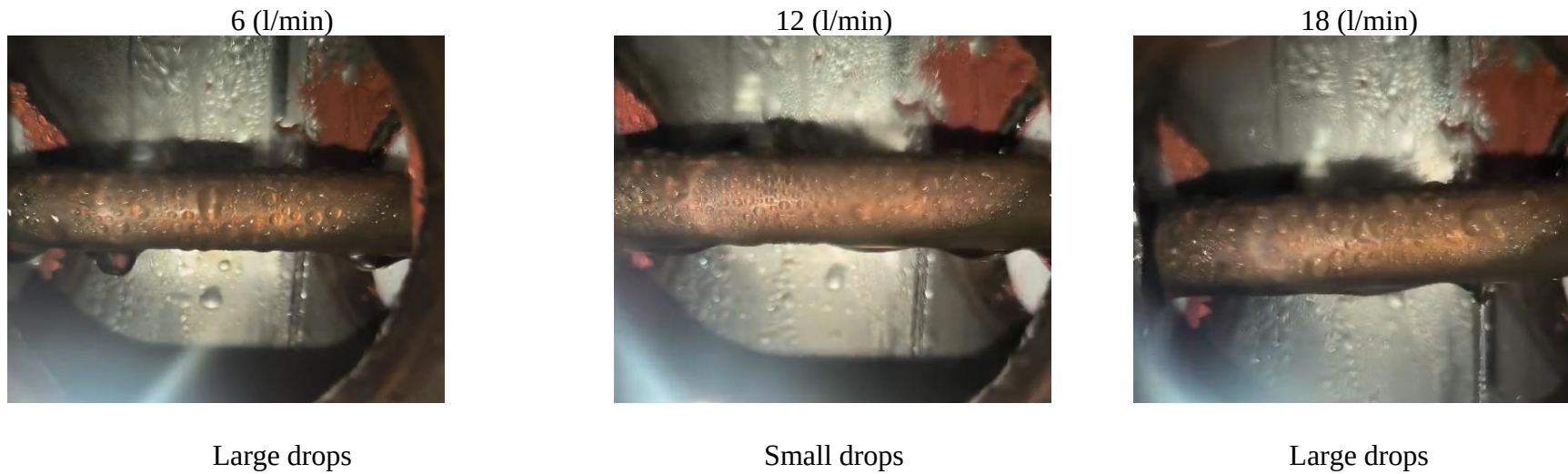
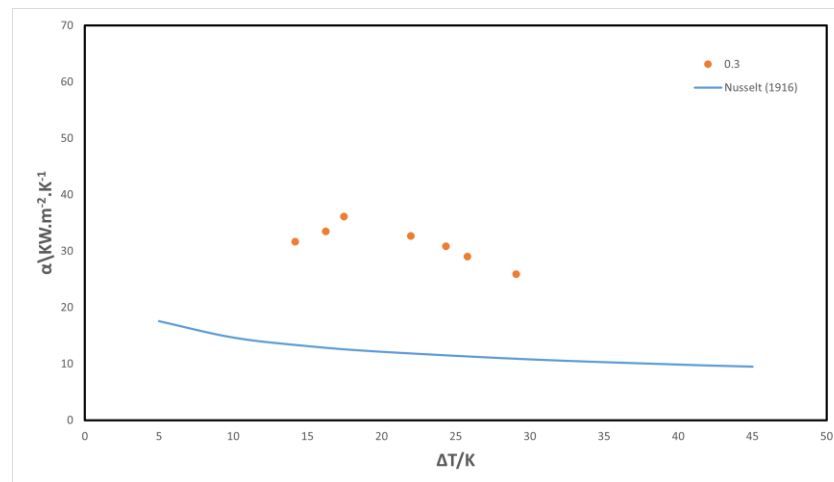


Figure 12 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a 0.3% concentration



4.3.2 Enhancement Ratio

The enhancement ratio (ϵ) is calculated by comparing the heat flux of the steam-ethanol mixture at a particular vapor side-to-surface temperature difference with the theoretical heat flux values for pure steam obtained from Nusselt's (1916) theory at the same vapor side-to-surface temperature difference.

In Figure 13, the enhancement ratio for steam-ethanol mixtures is depicted across various steam-ethanol mass concentrations at a constant vapor velocity of 0.46 m/s. Across all mass concentrations, the enhancement ratio surpasses unity throughout the entire range of vapor side-to-surface temperature differences. The enhancement ratio exhibits a strong dependence on both ethanol mass concentration and vapor side-to-surface temperature difference.

Notably, the enhancement ratio for all concentrations except for 0.3% decreases as the vapor side-to-surface temperature increases. This trend mirrors the observed behavior of the convective heat-transfer coefficient with respect to vapor side-to-surface temperature difference.

For an ethanol concentration of 0.0125%, the enhancement ratio was notably high at 2.4 during low vapor side-to-surface temperature differences. Subsequently, as the vapor side-to-surface temperature difference increased, the enhancement ratio exhibited a nearly constant value of approximately 1.5. At a concentration of 0.3%, the enhancement ratio initially rises with an increase in vapor side-to-surface temperature difference, peaking at a maximum value. However, as the vapor side-to-surface temperature difference continues to increase beyond this point, the enhancement ratio gradually decreases.

At the lowest vapor side-to-surface temperature differential ($\Delta T = 8.4$ K) for the 0.1% ethanol concentration, the maximum enhancement ratio of 3.8 was recorded.

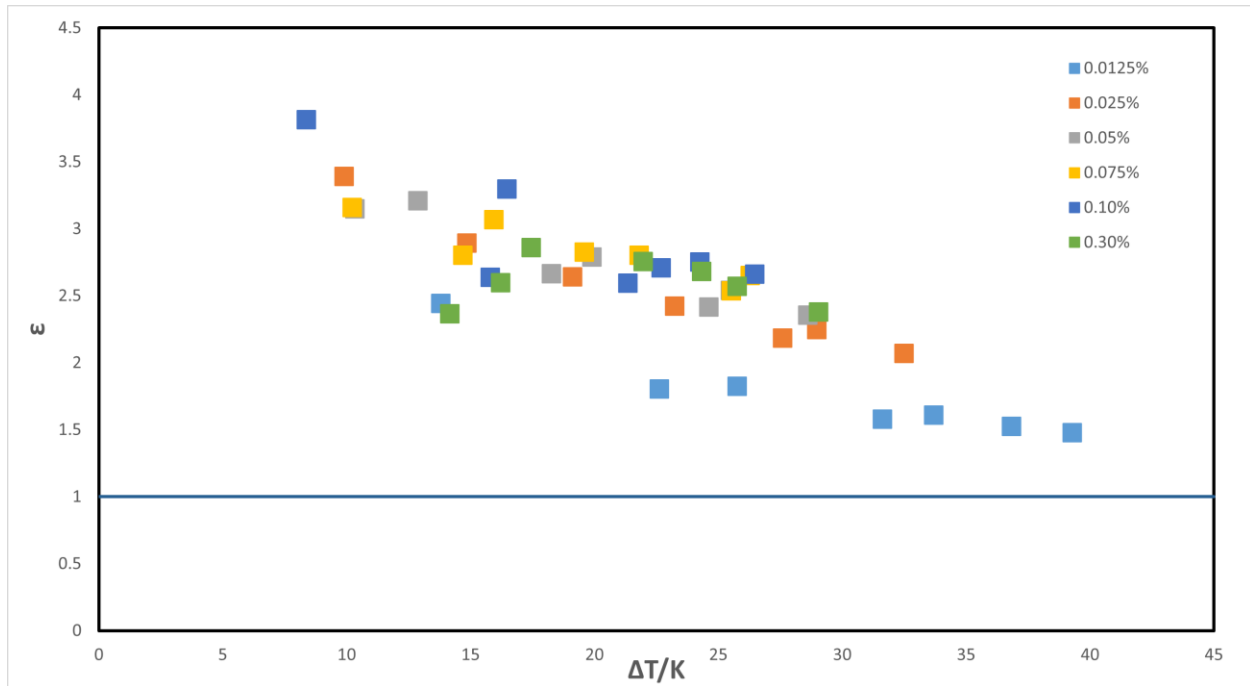


Figure 13 The enhancement ratio (ϵ) comparing the heat flux of the steam-ethanol mixture at a particular vapor side-to-surface temperature difference with the theoretical heat flux values for pure steam obtained from Nusselt's (1916) theory at the same vapor side

CHAPTER 5

Measurements of Heat Transfer for Pure Steam and Steam-Ethanol Mixtures Condensing on a Wire Wrapping Tube

5.1 Introduction

This chapter outlines an experiment examining condensation heat transfer on wire-wrapping tubes and measures the enhancement ratio for Marangoni condensation of binary mixtures on these tubes. The enhancement from wire-wrapping is attributed to reducing condensate thickness by creating a line surface that changes the tube's surface shape, affecting surface tension. While the enhancement ratio of wire-wrapping is lower than that of integral-fin and pin-fin tubes, it remains the most cost-effective method as it requires no machining.

The same experimental setup utilized in the previous chapter was used, employing a smooth copper tube with an outer diameter of 13 mm and an inner diameter of 11 mm, measuring 65 mm in length. A copper wire with a diameter of 0.5 mm was utilized in all experiments.

Initially, measurements focused on pure-steam condensation heat transfer, varying wire pitches from 1.6 mm to 2.9 mm, while maintaining a constant vapor velocity of 0.48 m/s and operating at atmospheric pressure. Subsequently, the study integrated data concerning steam-ethanol mixtures of varying compositions at different pitches. The primary objective

was to investigate Marangoni condensation on the wire-wrapping tube across different steam-ethanol concentrations and wire pitches, aspects that had not been previously investigated.

In addition to quantitative measurements, visual observations were recorded through the capture of videos and images, documenting the condensation modes observed throughout the experiments.

5.2 Pure Steam Condensation on Horizontal Wire-Wrapping Tubes

Heat flux and convective heat transfer coefficient are shown against the vapor side-to-surface temperature differential in Figures 14 and 15. The results of steam condensation are compared in these figures between a simple tube and a wire-wrapping tube with a wire diameter of 0.5 mm and pitches of 1.6 mm, 2 mm, 2.4 mm, and 2.9 mm, respectively.

The data reveals a significant enhancement in heat transfer for wire-wrapped tubes across all pitches compared to plain tubes. This enhancement can be explained by the reduction of the condensate film between the wire rounds due to wire wrapping, impacting the surface tension of the condensate, and consequently increasing the heat flux, as indicated by Murase. Furthermore, as highlighted by Ali et al., the wire-wrapping tube's active area—which is the unflooded space between the wire rounds—is noticeably larger than the plain tube's.

Observing the figures, it's evident that the highest heat transfer is achieved with a pitch of 2.4 mm. This can be explained to the maximal active surface area available compared to other pitches. In the case of 2.9 mm pitch, there's diminished fluid retention at the lower part of the tube, resulting in a reduced active surface area due to fewer wire turns on the tube, ultimately yielding lower heat flux compared to the 2.4 mm pitch.

In contrast, the 1.6 mm pitch demonstrates the least heat transfer enhancement. This is primarily because a significant portion of the tube is covered by the wire rounds, leading to parts of the tube being flooded with condensate, thus impeding the effective heat transfer.

Across all pitches, the convective heat transfer coefficient values surpassed those of the plain tube. Notably, at low vapor side-to-surface temperature differences, the coefficients were elevated, gradually decreasing as the vapor side-to-surface temperature difference increased. The peak values were attained with a 2.4 mm pitch, where at a 10 K vapor side-to-surface temperature difference, the convective heat transfer coefficient reached $60 \text{ KW.m}^{-2} \text{ K}^{-1}$, marking a substantial enhancement.

Figure 16 visually illustrates the separation of the condensate film, identified as the primary factor contributing to the increase in heat transfer. While the separation may not be distinctly clear in the picture, it was clearly observed in the videos. In the case of the 1.6 mm pitch, the tube was nearly flooded by the condensate, resulting in minimal enhancement. Conversely, for other pitches, the flooded area was less pronounced, leading to a more effective active area for heat transfer.

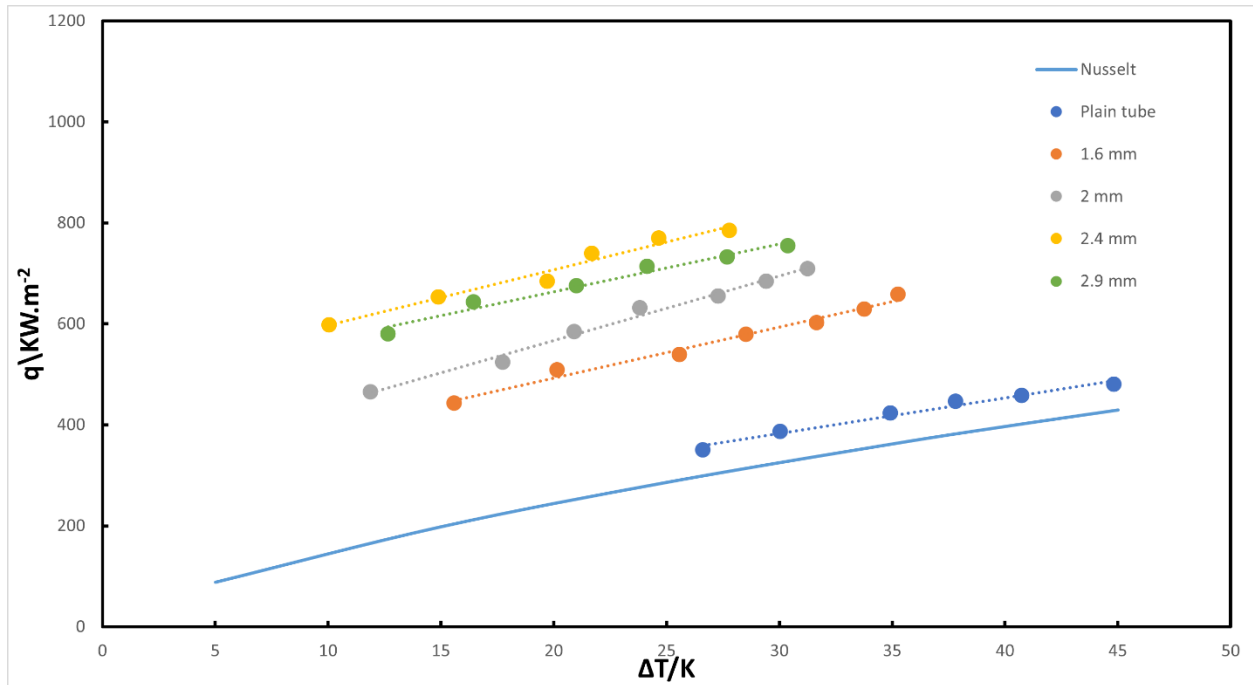


Figure 14 Heat flux against the vapor side-to-surface temperature difference for plain tube and wire-wrapping tube with 4 different pitches (1.6 mm, 2 mm, 2.4 mm, and 2.9 mm).

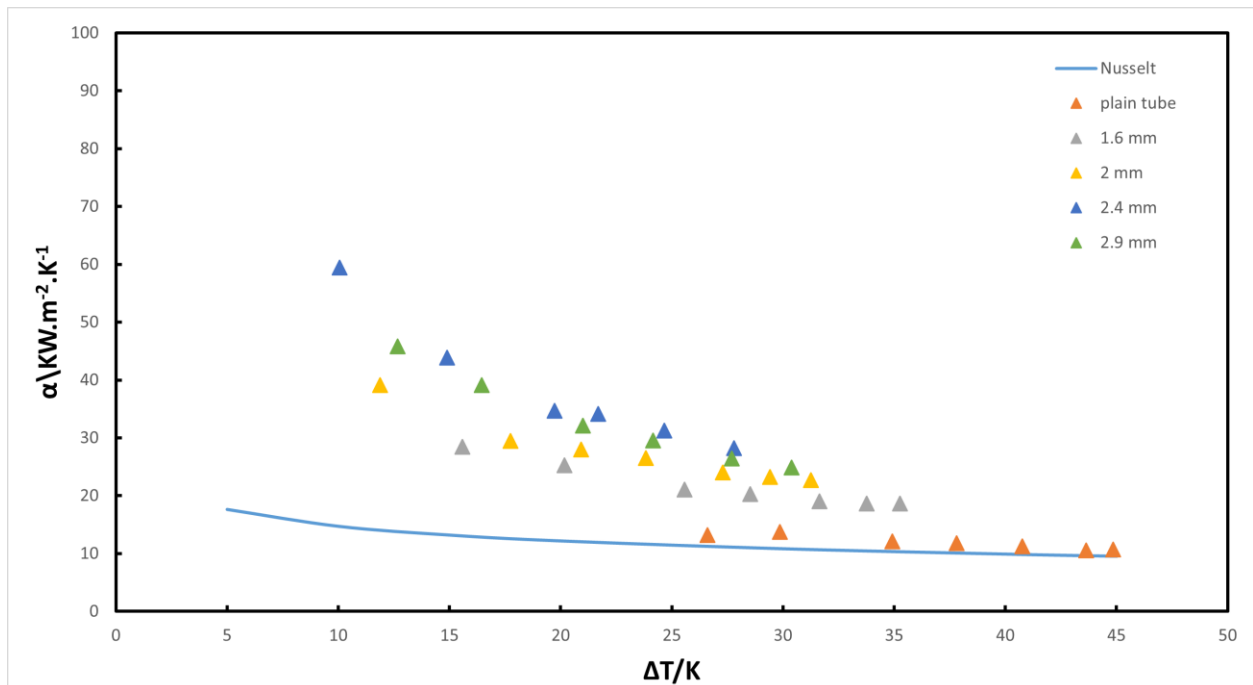
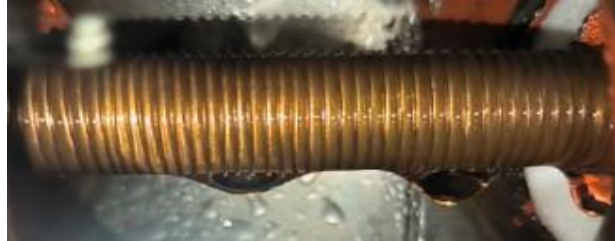


Figure 15 convective heat transfer coefficient against the vapor side-to-surface temperature difference for plain tube and wire-wrapping tube with 4 different pitches (1.6 mm, 2 mm, 2.4 mm, and 2.9 mm).

Pitch (mm)

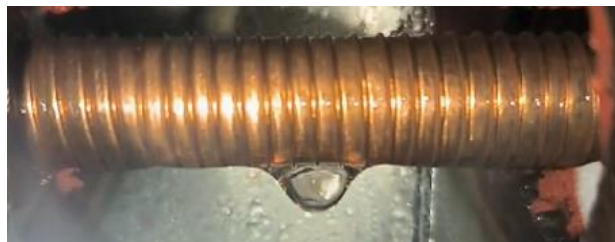
1.6



2



2.4



2.9



Figure 16 Visual observation of the condensate film at 4 different pitches.

5.3 Marangoni Condensation on Horizontal Wire-Wrapping Tubes

In this section, Marangoni condensation on wire-wrapped tubes is investigated, involving four different steam-ethanol concentrations (0.0125%, 0.05%, 0.1%, and 0.3%), combined with four distinct pitches (1.6 mm, 2 mm, 2.4 mm, and 2.9 mm), utilizing a copper wire with a 0.5 mm diameter. Figure 17 depicts the correlation between heat flux and vapor side-to-surface temperature difference across various flow rates, to observe the impact of steam-ethanol concentrations on wire-wrapped tubes.

The figure demonstrates that across all steam-ethanol concentrations, the heat flux values surpass those observed with pure steam. This phenomenon can be explained by the reduction in surface tension of the condensate film after its separation, influenced by the wire turns as discussed in the previous section. Now, the introduction of a small amount of ethanol further alters the separated condensate film, transforming it into small droplets dispersed between the wire rounds. This restructuring enhances heat transfer by reducing the condensate retention at the lower part of the tube, more efficient condensate removal and increasing the active surface area available for heat exchange.

The results revealed that the minimum enhancement was consistently observed at a steam-ethanol concentration of 0.0125% for all pitches, except for the 2.9 mm pitch where the minimum enhancement was observed at 0.3%. Conversely, the maximum enhancement was consistently observed at a concentration of 0.1%, which closely mirrors the scenario observed with the plain tube.

At low flow rates, associated with low vapor side-to-temperature differences, there was a significant enhancement in heat transfer, especially noticeable at low temperature differences. At such low temperature differences, the condensate film is broken up into

very small drops. However, as the temperature difference increases, the enhancement in heat transfer reduced gradually. This decrease in enhancement is accompanied by the growth of the drops, leading to a more continuous condensate layer.

Figure 17 depicts the convective heat transfer coefficient plotted against the vapor side-to-surface temperature difference. The values peak at low temperature differences before declining with increases in temperature difference and flow rates. The impact of reducing retention by adding ethanol to the wire-wrapping tubes significantly affects the convective heat transfer coefficient, consistently yielding higher values compared to pure steam across all pitches.

Furthermore, a decreasing trend is observed in the vapor side-to-surface temperature difference associated with the increasing convective heat transfer coefficient. This can be attributed to heightened heat transfer facilitated by the reduction in surface tension and condensate retention on the lower part of the tube. Consequently, the surface temperature of the tube increases while the vapor temperature remains constant, leading to a reduction in temperature difference between them.

Figure 5.6 illustrates the relationship between heat flux and vapor side-to-surface temperature difference across various flow rates, aiming to assess the influence of wire-wrapped tube pitches on Marangoni condensation.

The figure demonstrates that across all pitches, the heat flux values exceed those observed with pure steam. Interestingly, the maximum enhancement is consistently observed at a pitch of 2.9 mm for all steam-ethanol concentrations, attributed to minimal retention. However, at a concentration of 0.3%, the maximum enhancement shifts to a pitch of 2.4 mm. Conversely, the minimum enhancement is consistently observed at a pitch of 1.6 mm,

primarily due to a high number of wire rounds causing the tube to be fully flooded with condensate, thus hindering effective heat transfer.

Visual Observation

The experiments on Marangoni condensation on wire-wrapping tubes were meticulously recorded. Figures 20-23 showcase the evolution of condensate appearance as the vapor side-to-surface temperature difference increases for all pitches at varying steam-ethanol concentrations. Visual observations are juxtaposed with the heat transfer coefficient graph, providing a comprehensive understanding.

Initially, at a pitch of 1.6 mm and 0.0125% ethanol concentration, enhancement is only noticeable at low temperature differences. Subsequently, the convective heat transfer coefficient values become nearly identical to those observed with pure steam. However, for higher concentrations (0.05%, 0.1%, and 0.3%), a clear enhancement is evident at low temperature differences. As the temperature difference increases, the flooding area of the tube gradually expands, leading to a reduction in the convective heat transfer coefficient values.

For the 2 mm pitch, particularly at low flow rates, the condensate film undergoes a transformation into numerous drops that start to emerge between the wire rounds. This phenomenon enhances heat transfer by reducing the thickness of the condensate film. As the temperature difference increases, the number of drops decreases. This reduction in the number of drops impacts the values of the convective heat transfer coefficient, as depicted in the figures for all concentrations. Interestingly, the lowest reduction is observed at a

concentration of 0.1%. Visual observations corroborate this, as they indicate that the number of droplets continues to appear even at high temperature differences.

In the case of pitches of 2.4 mm and 2.9 mm, there is an increase in the number of droplets while the retention decreases. Consequently, the active area of the tube surface increases. Even at high flow rates associated with higher temperature differences, a number of droplets appear on the surface of the tube. Notably, the maximum convective heat transfer is observed at these pitches, especially at concentrations higher than 0.0125%.

Enhancement Ratio

In Figure 24, the enhancement ratio for Marangoni condensation of steam-ethanol mixtures on wire-wrapping tubes is illustrated across various steam-ethanol mass concentrations and wire pitches, maintaining a constant vapor velocity of 0.46 m/s. Remarkably, the enhancement ratio consistently exceeds unity across all mass concentrations and wire pitches, spanning the entire range of vapor side-to-surface temperature differences.

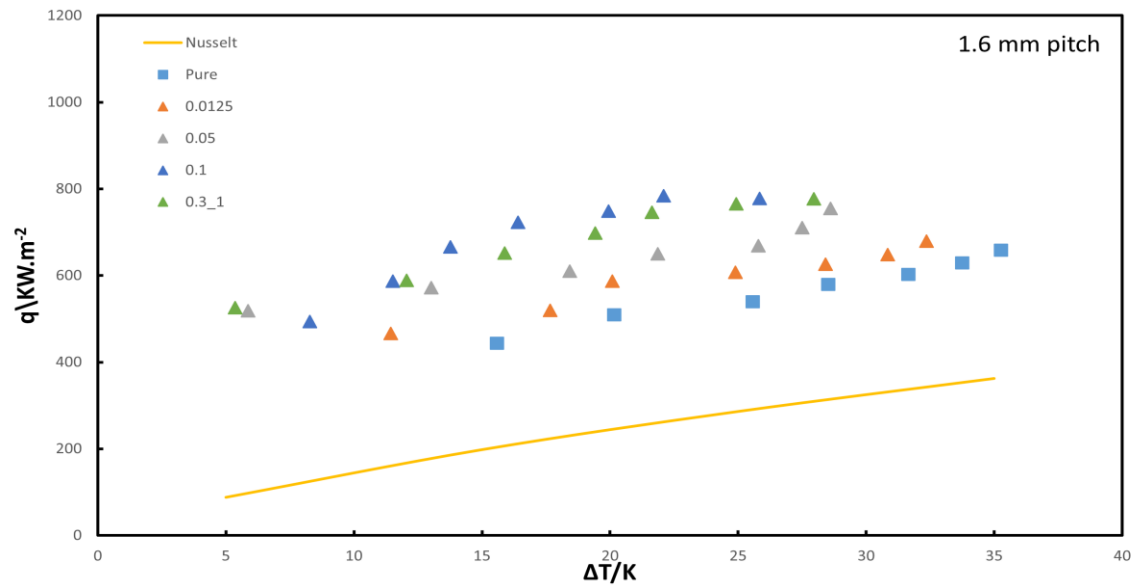
Notably, the enhancement ratio is particularly significant, especially in the case of a pitch of 2.9 mm. The maximum enhancement ratio was 9.9, occurring at a temperature difference of 3.5 K, which is considered a substantial enhancement compared to the maximum enhancement recorded for Marangoni condensation on a plain tube in the previous chapter, which was 3.8 at a temperature difference of 8.4 K. Additionally, the minimum enhancement achieved at this pitch is recorded to be 3.1 at a temperature difference of 22 K, still indicating a noteworthy enhancement.

Similarly, for a pitch of 2.4 mm, a significant enhancement is observed, particularly at a concentration of 0.05 ethanol, with an enhancement ratio of 8 at a temperature difference of 4.6 K. For both pitches mentioned above, the trend of the enhancement ratio is high at

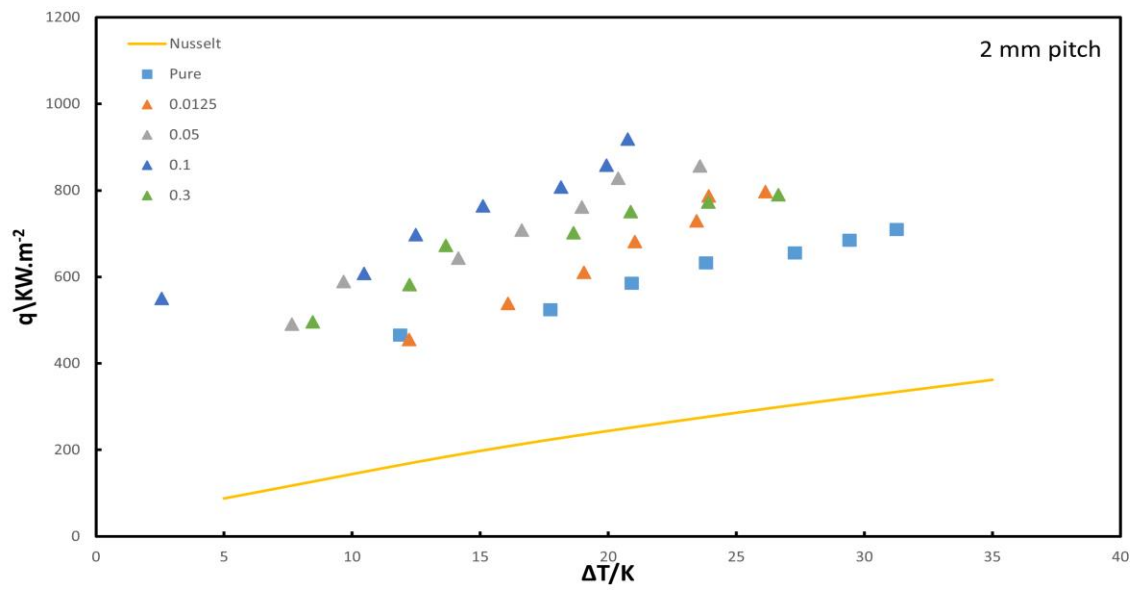
low temperature differences before gradually reducing. This trend mirrors the observed behavior of the convective heat transfer coefficient concerning vapor side-to-surface temperature difference.

Conversely, for pitches of 1.6 mm and 2 mm, the trend of the enhancement ratio appears almost linear, especially at temperature differences larger than 10 K.

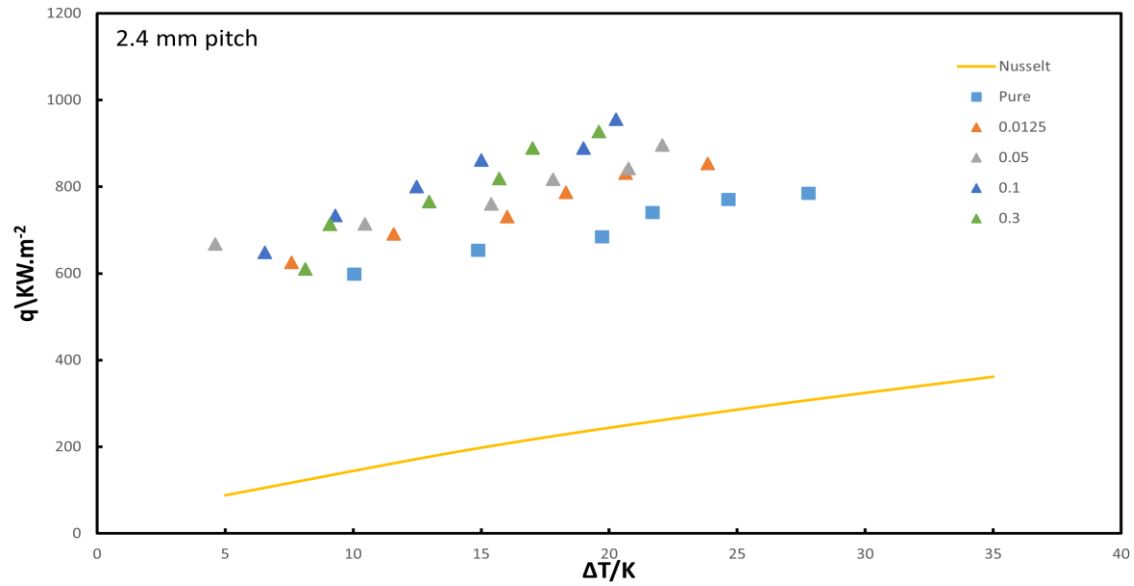
(A)



(B)



(C)



(D)

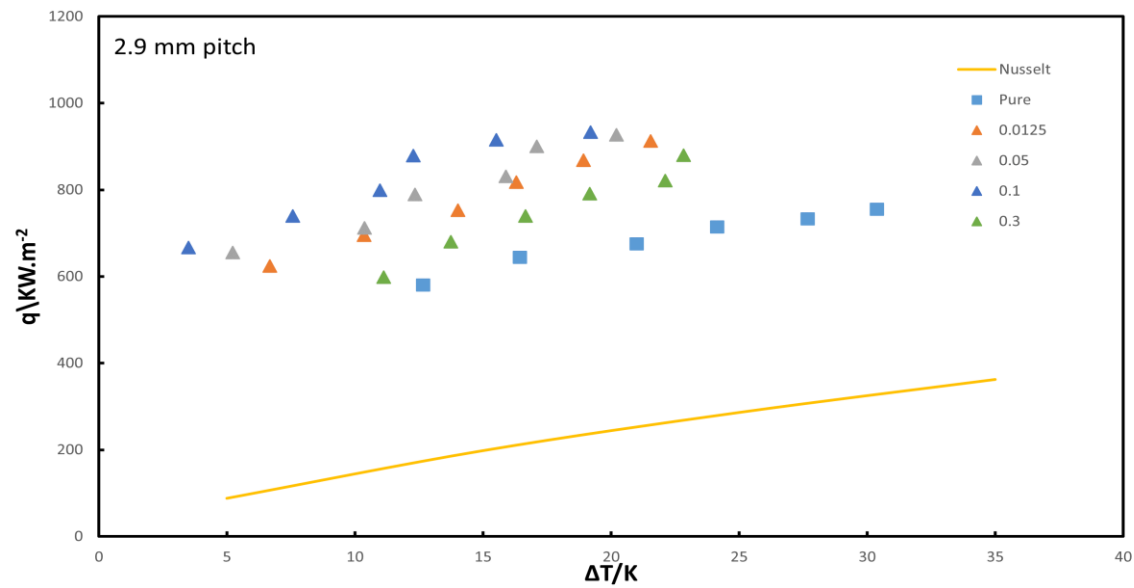
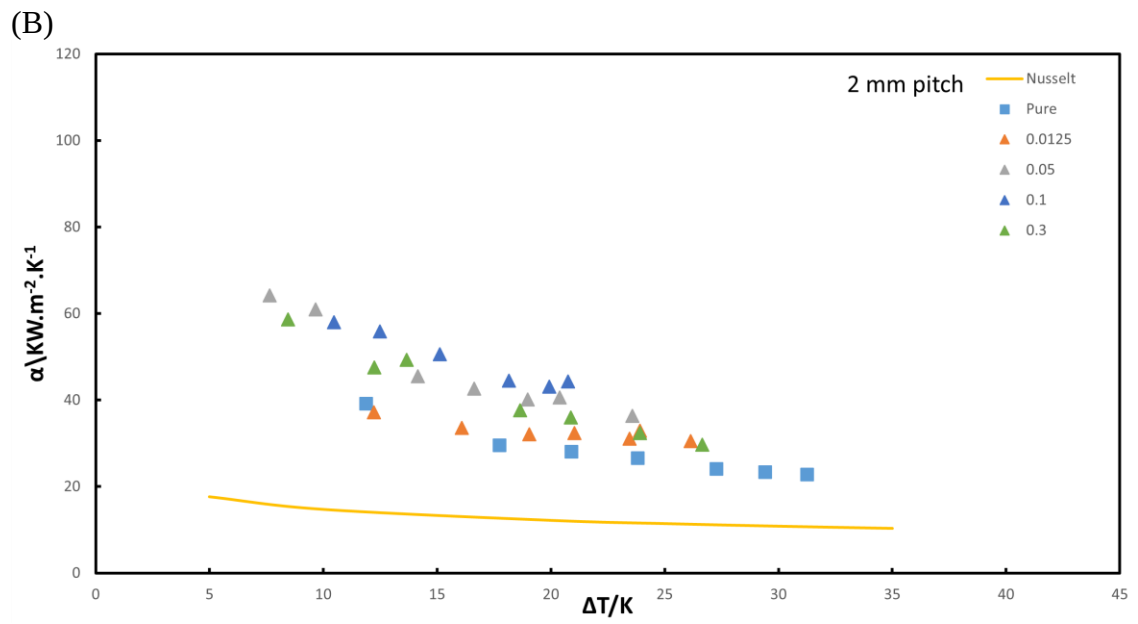
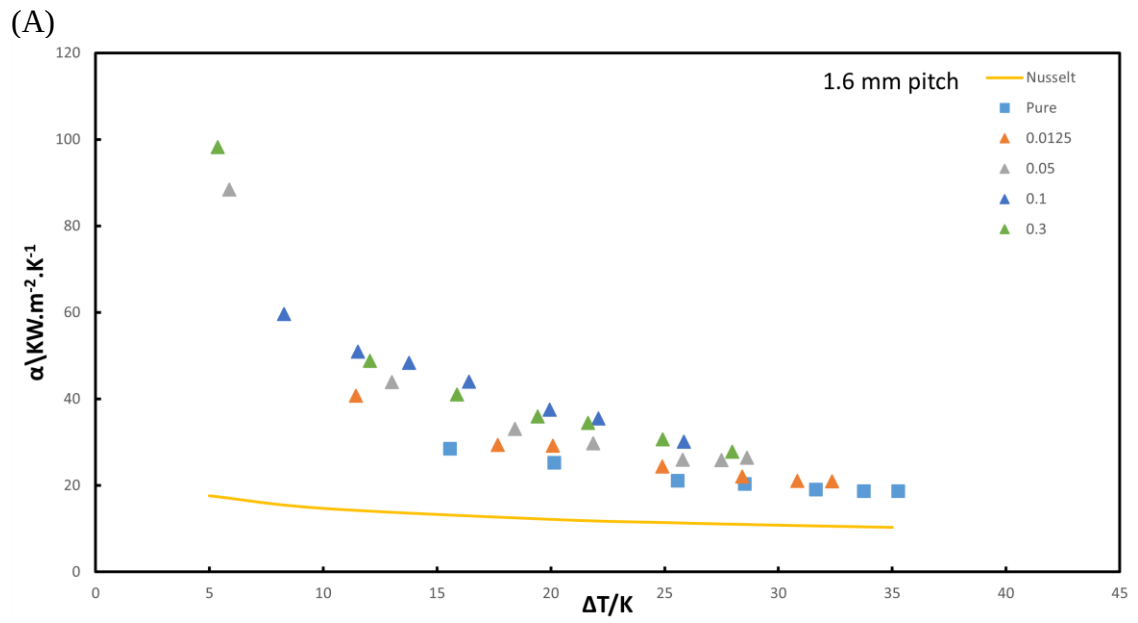


Figure 17 Heat flux against the vapor side-to-surface temperature difference for a wire-wrapping tube at different pitches and 4 steam-ethanol concentrations (0.0125%, 0.05%, 0.1%, and 0.3%).



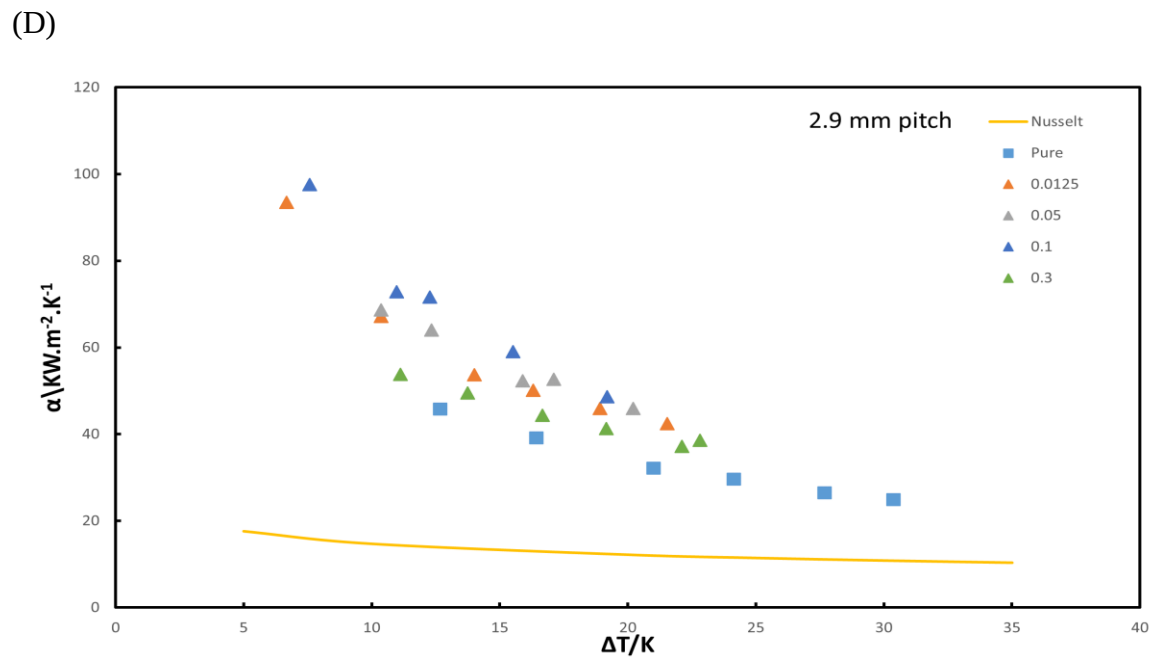
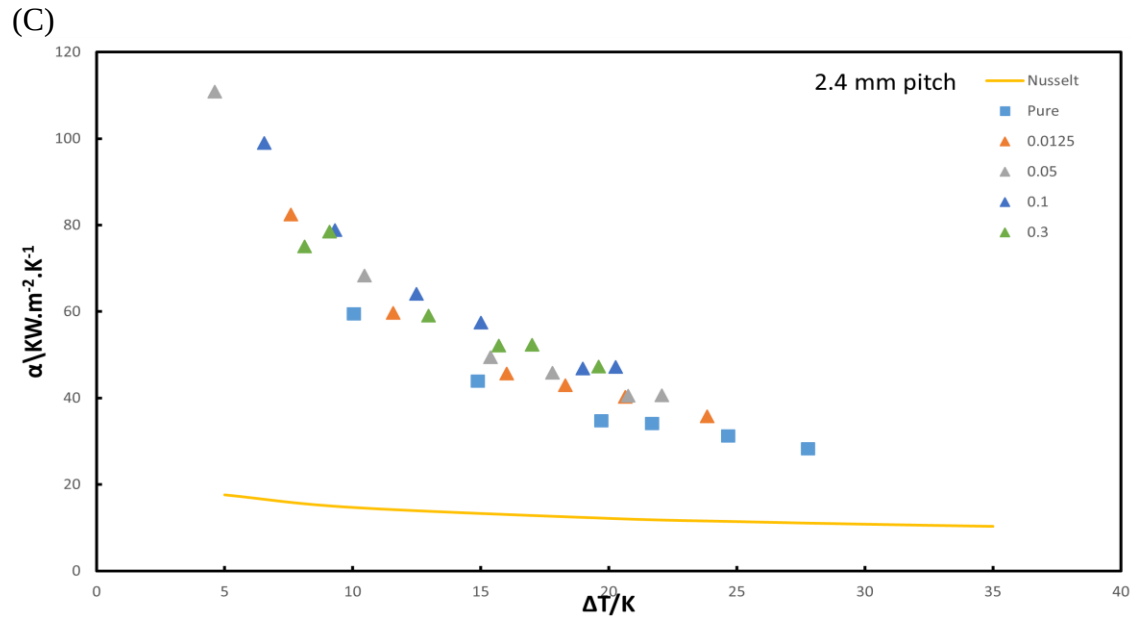
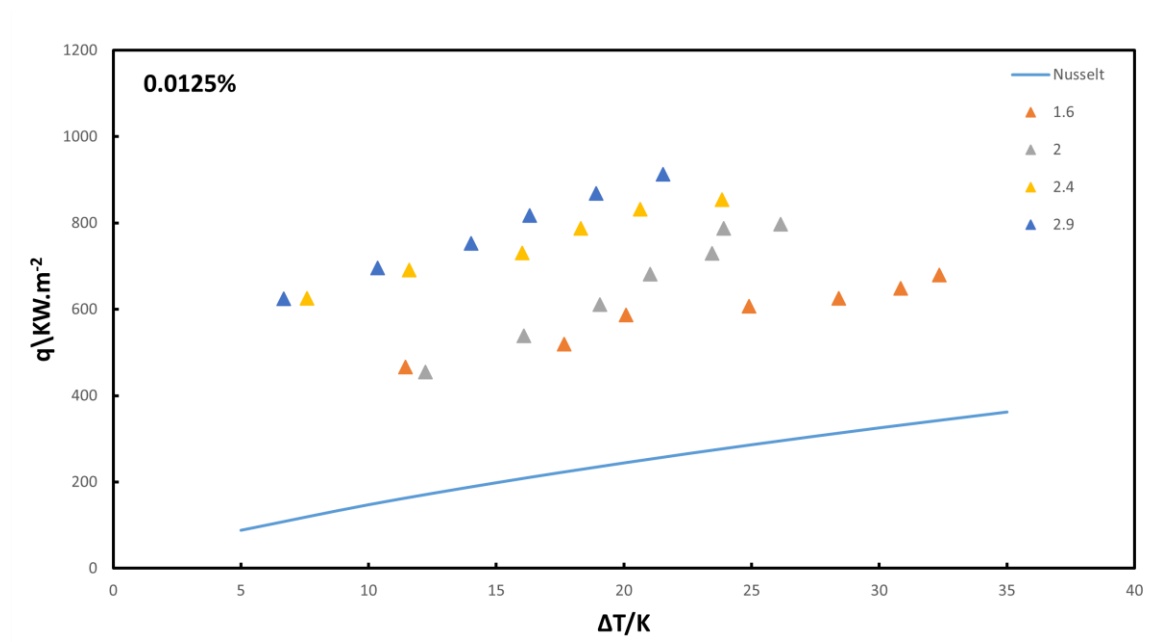
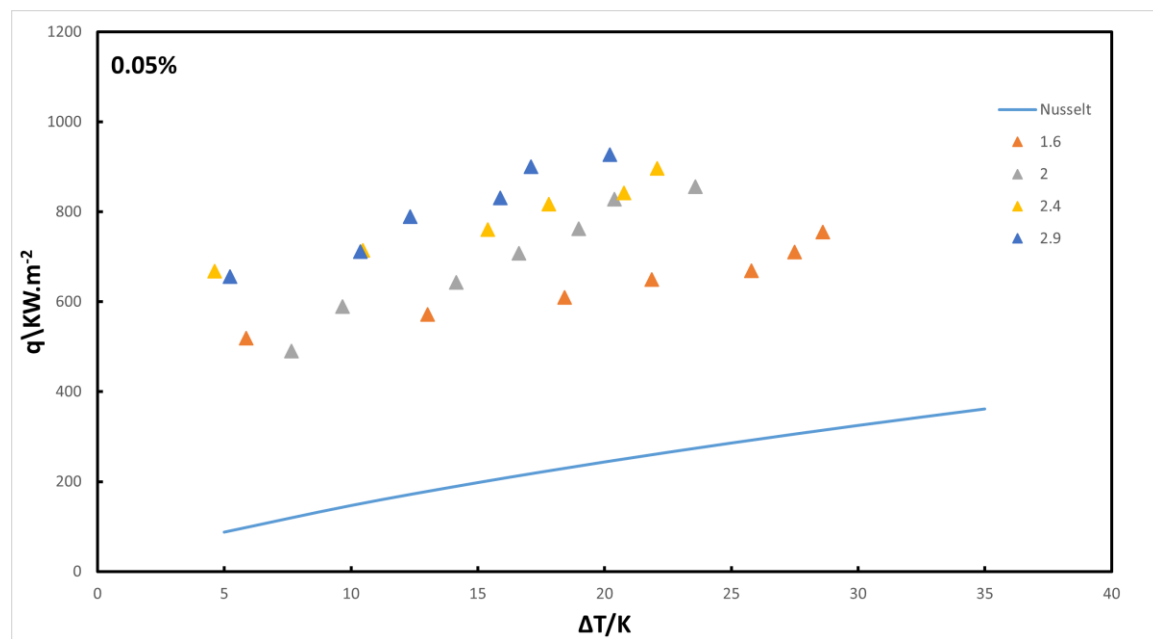


Figure 18 Convective heat transfer coefficient against the vapor side-to-surface temperature difference for a wire-wrapping tube at different pitches and 4 steam-ethanol concentrations (0.0125%, 0.05%, 0.1%, and 0.3%).

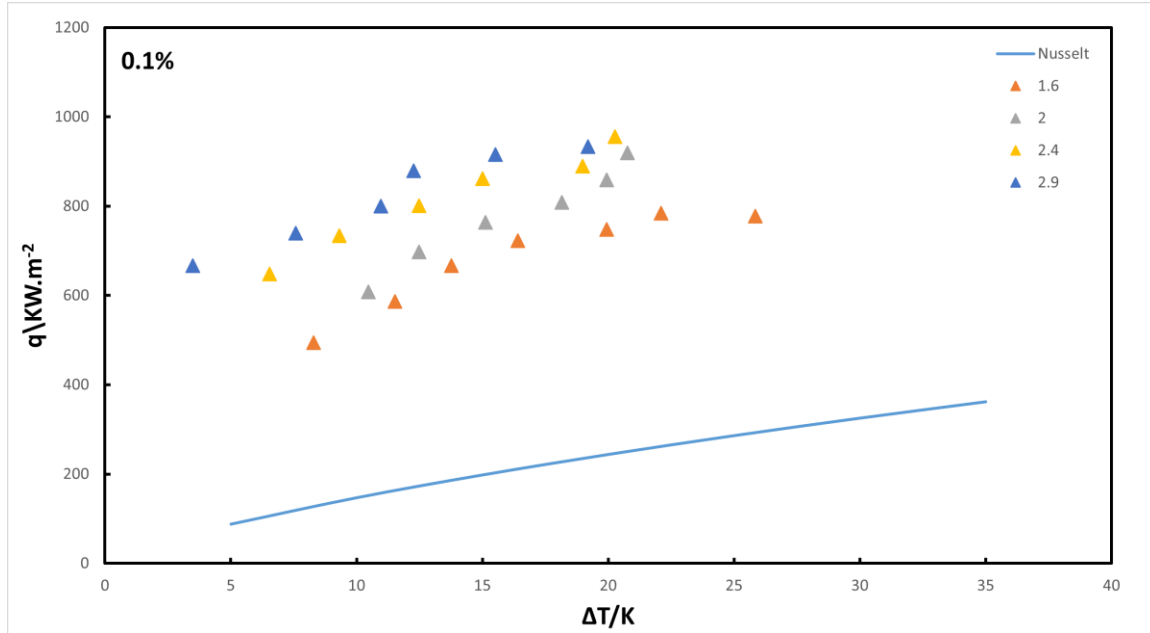
(A)



(B)



(C)



(D)

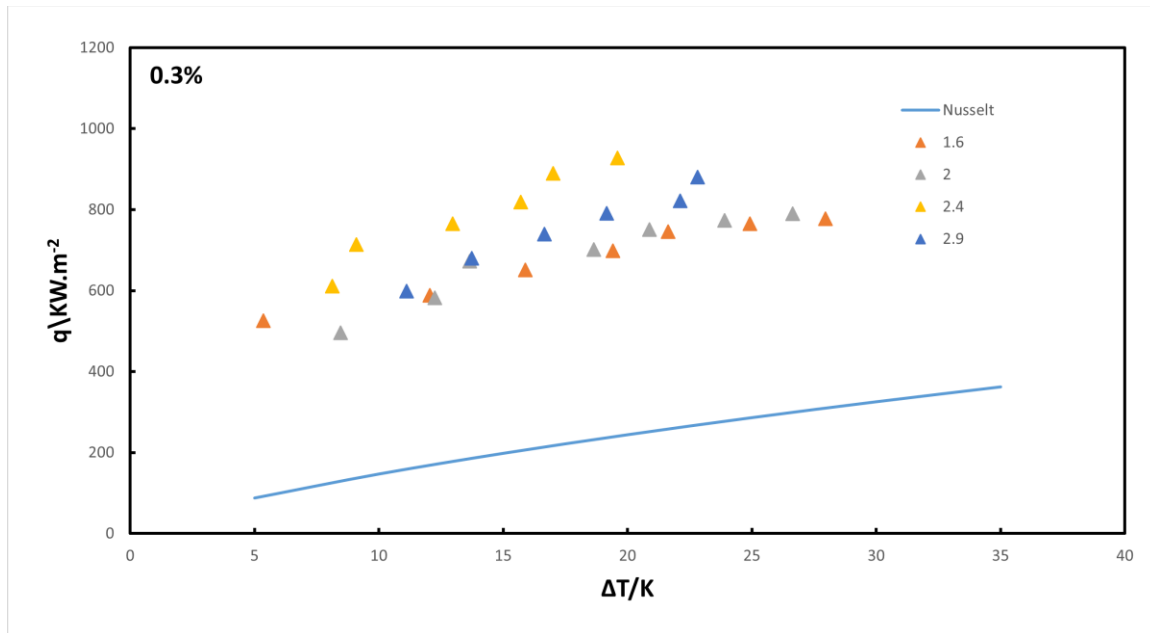


Figure 19 Heat flux plotted against the vapor side-to-surface temperature difference for a wire-wrapping tube at different ethanol concentrations with 4 pitches (1.6 mm, 2mm, 2.4 mm, and 2.9 mm).

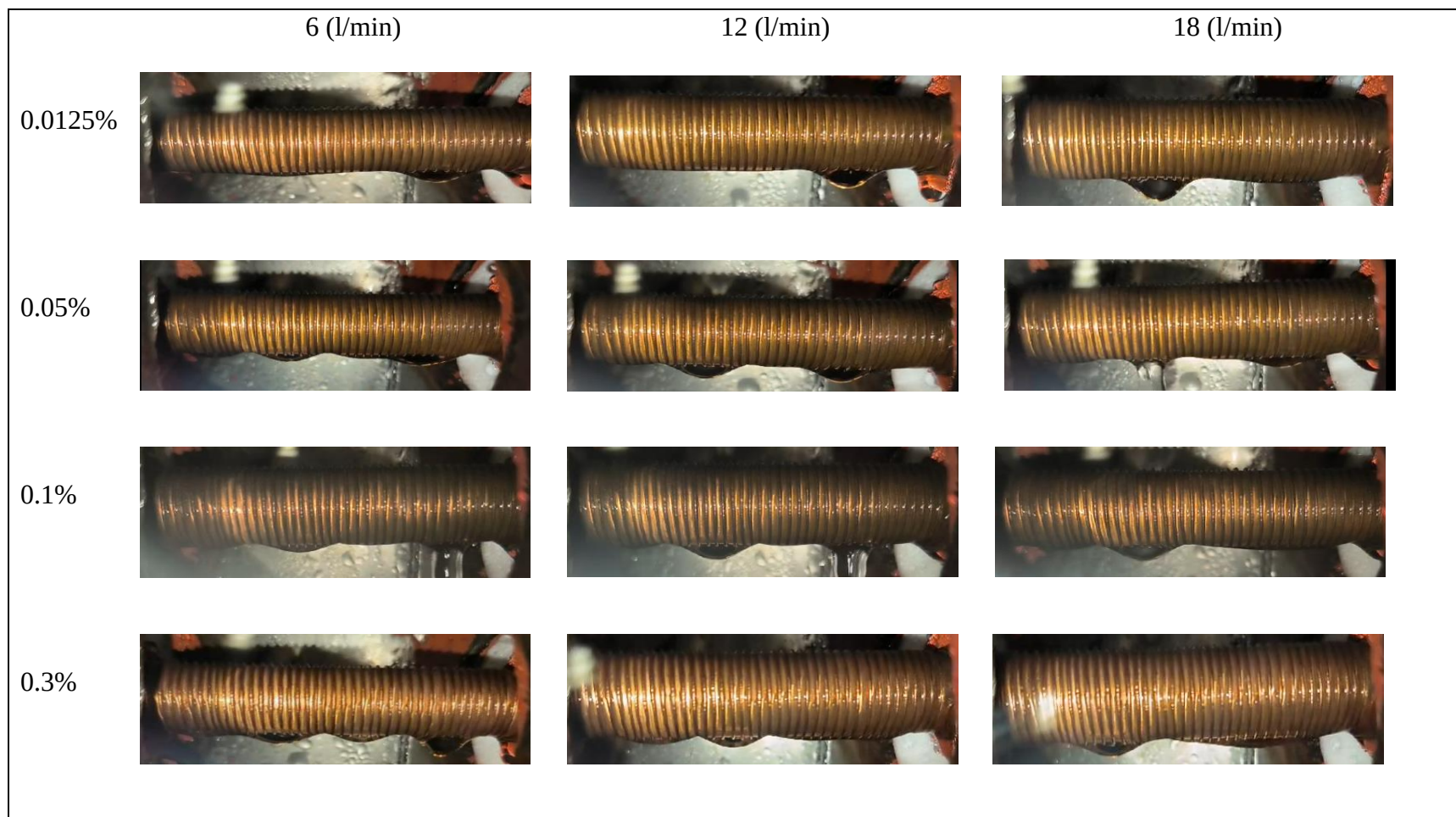
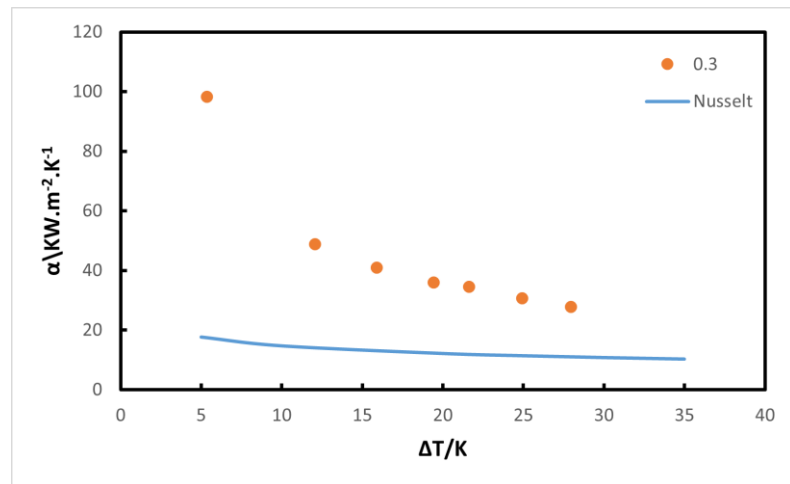
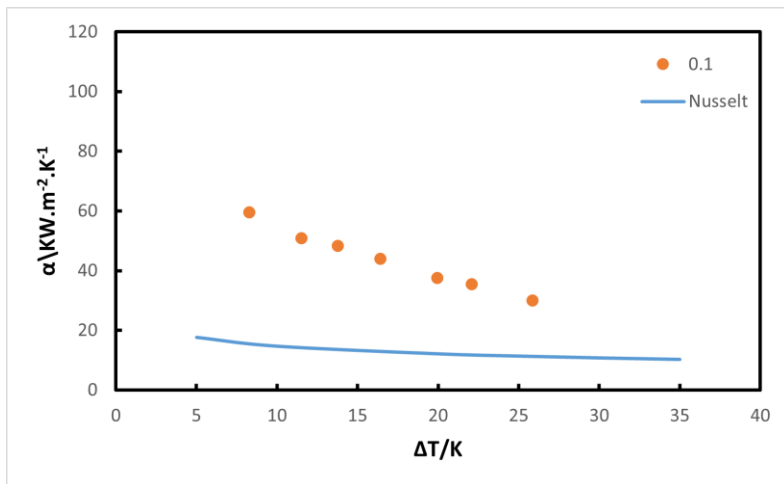
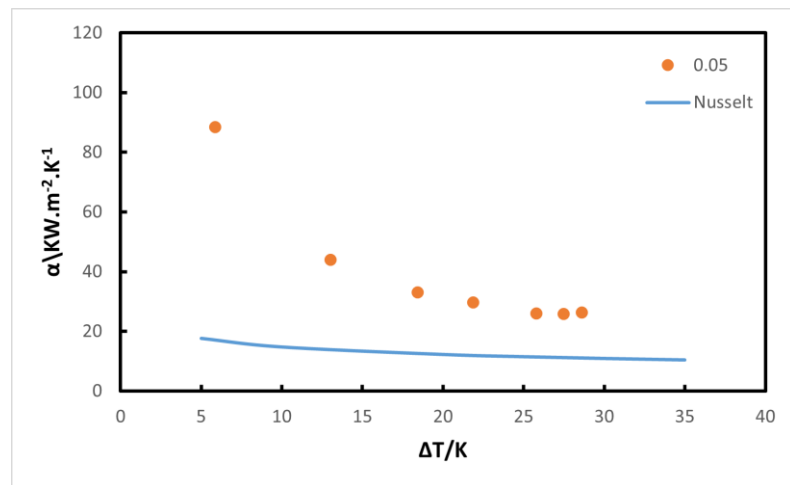
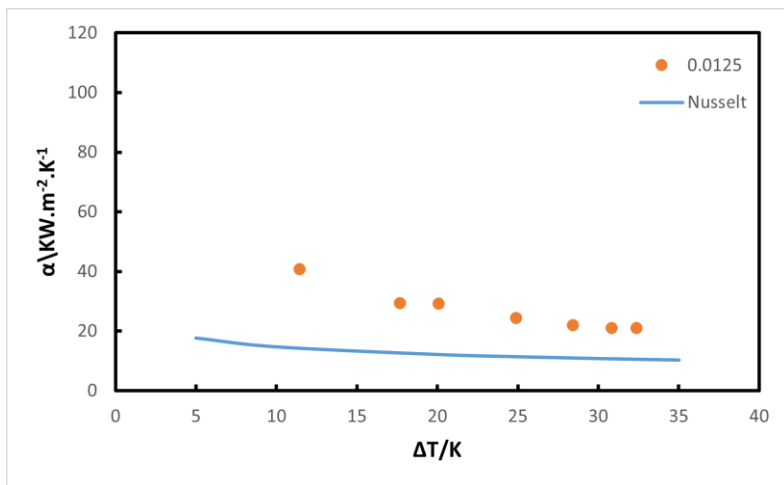


Figure 20 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a 1.6 mm pitch.

1.6 mm



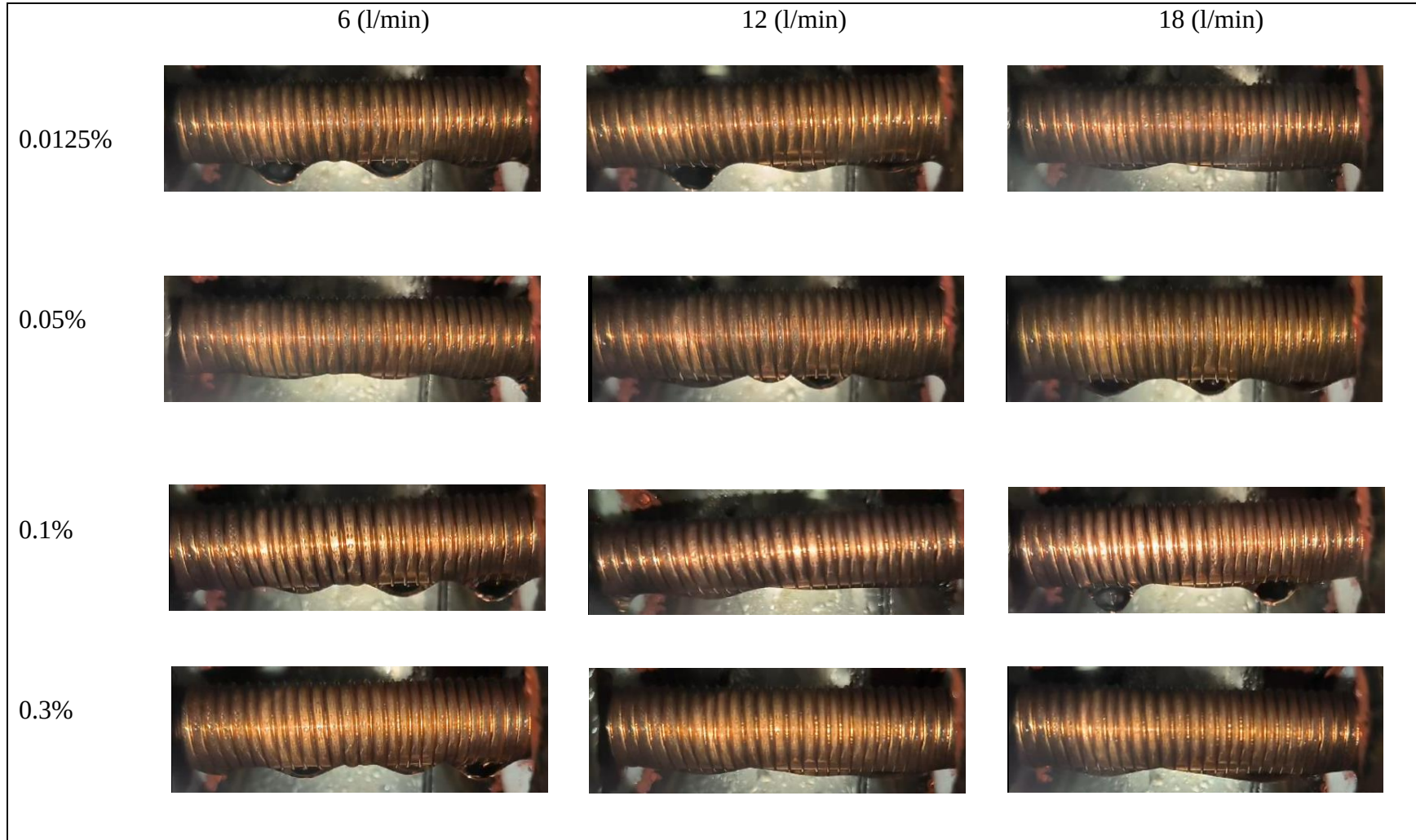
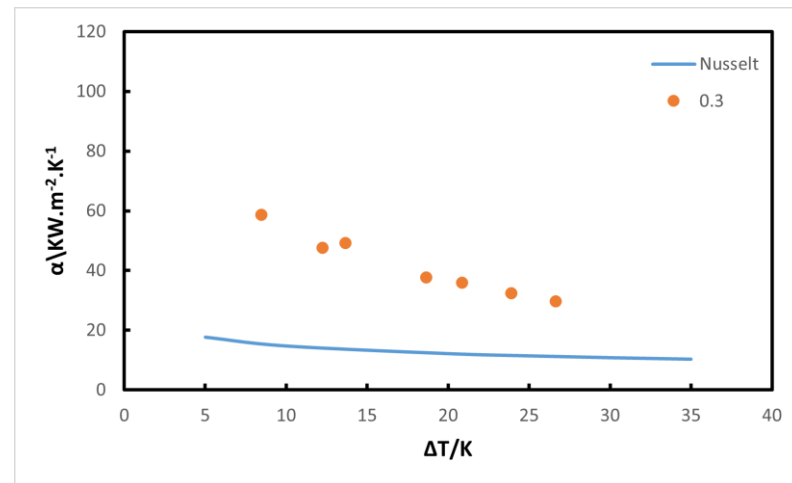
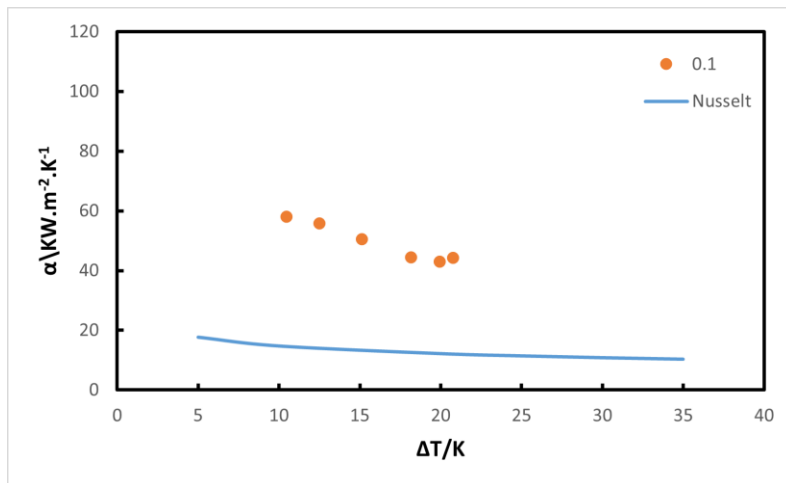
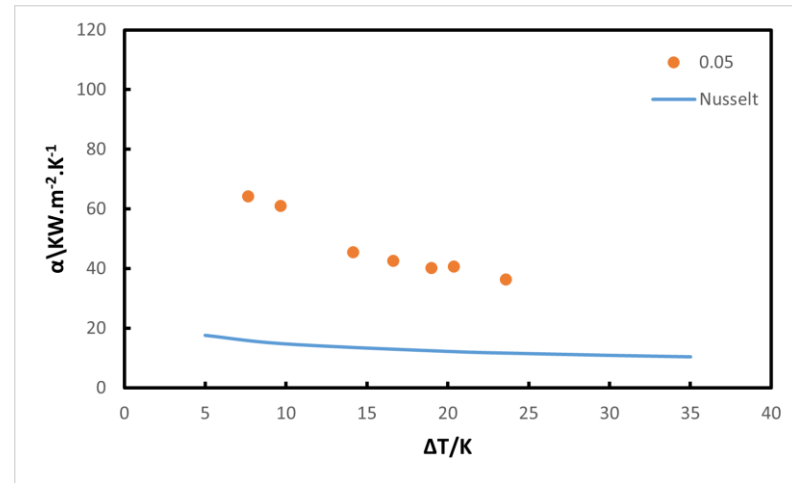
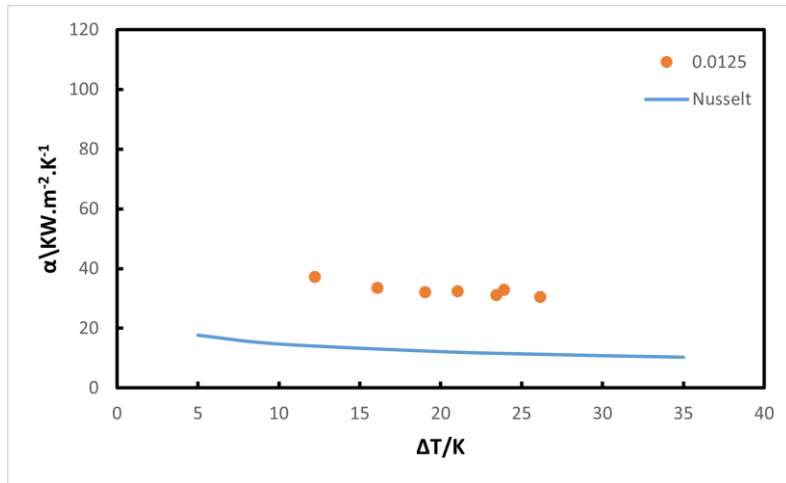


Figure 21 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a 2 mm pitch.

2 mm



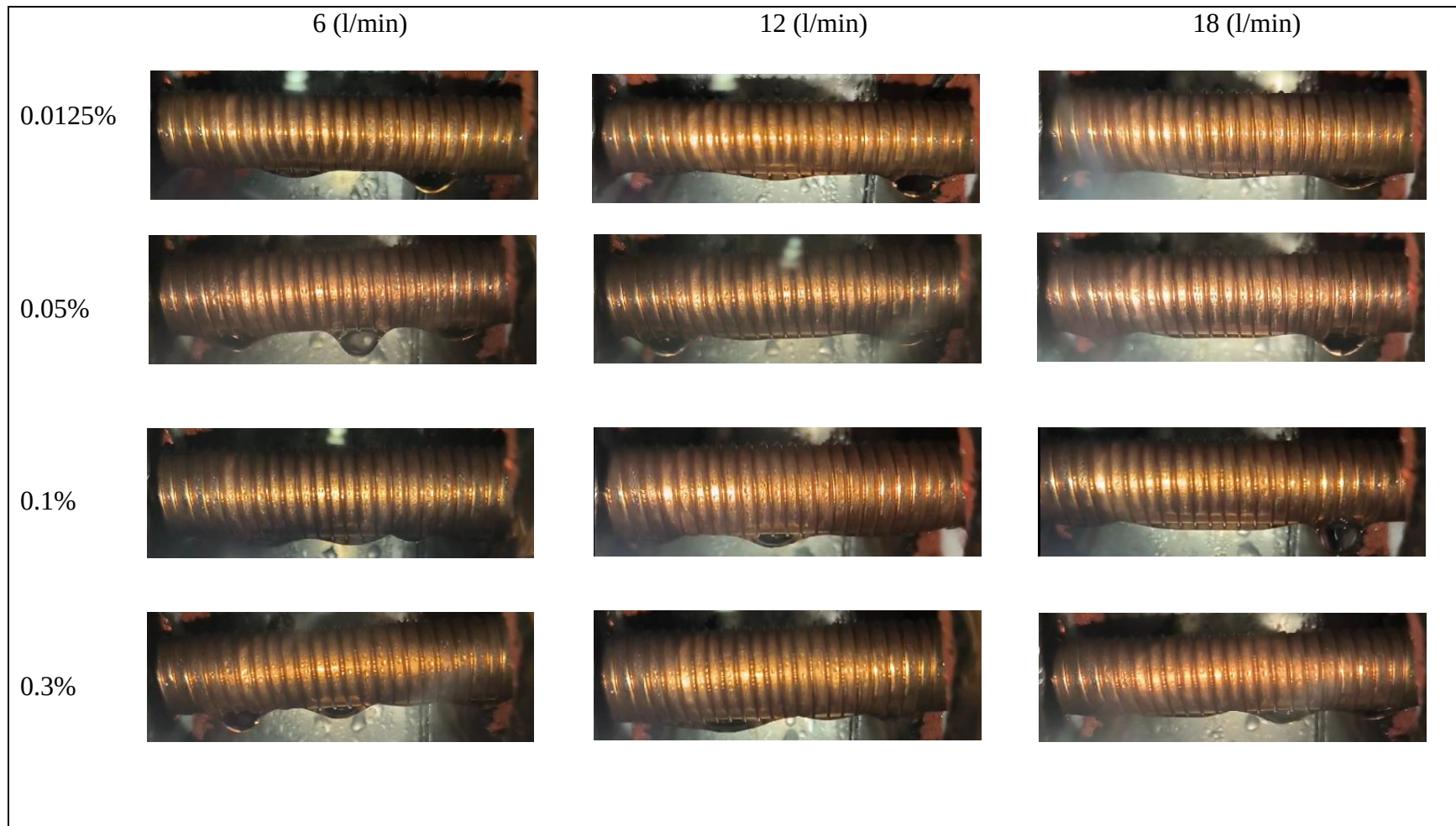
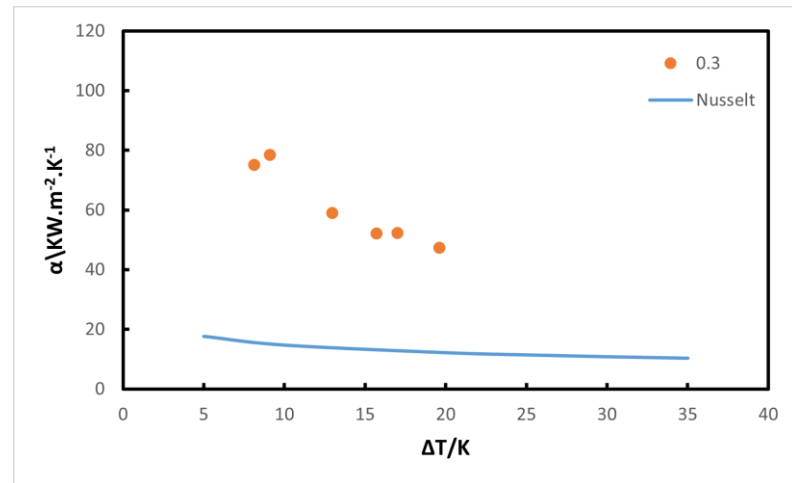
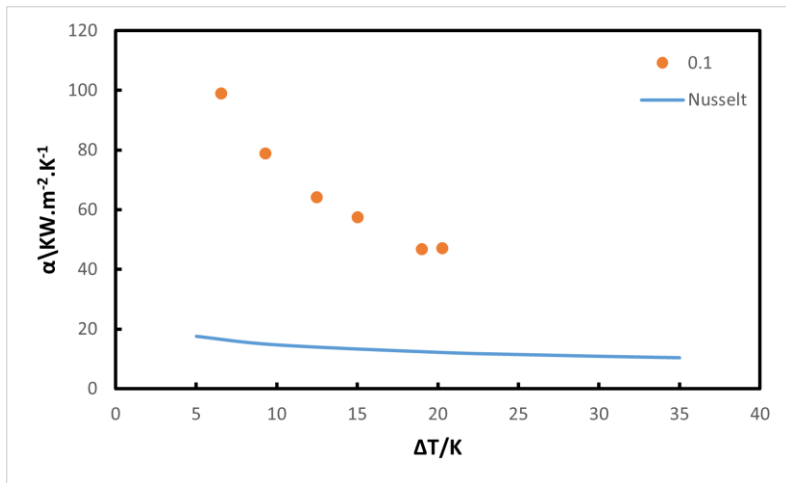
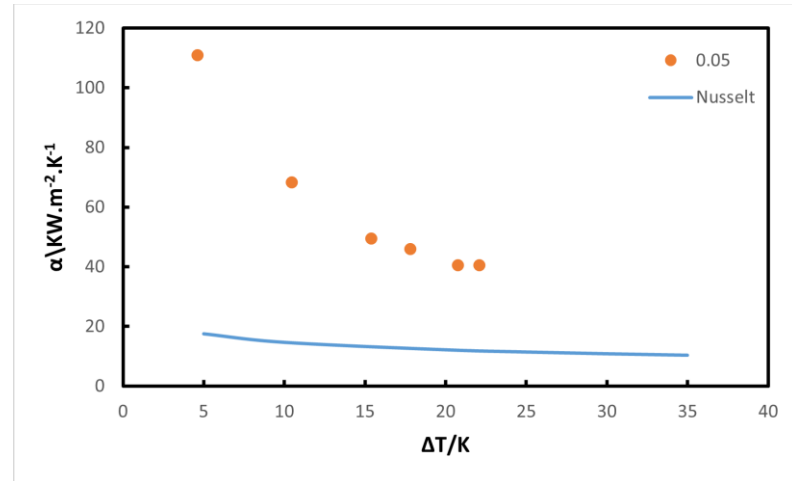
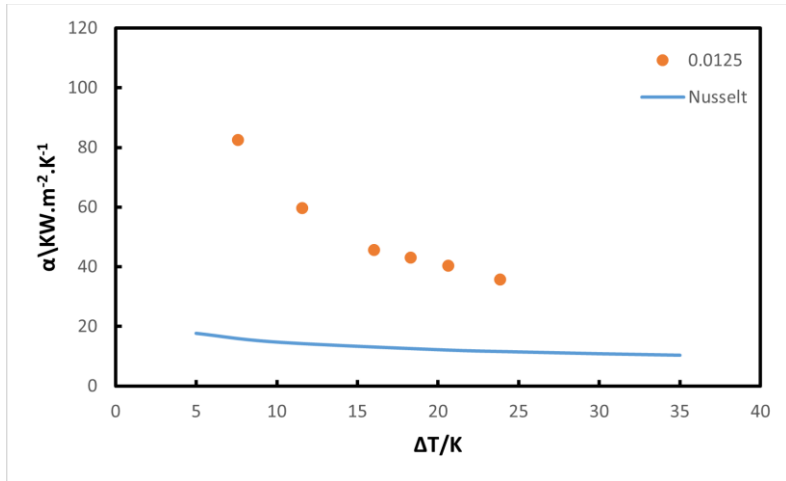


Figure 22 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a 2.4 mm pitch.

2.4 mm



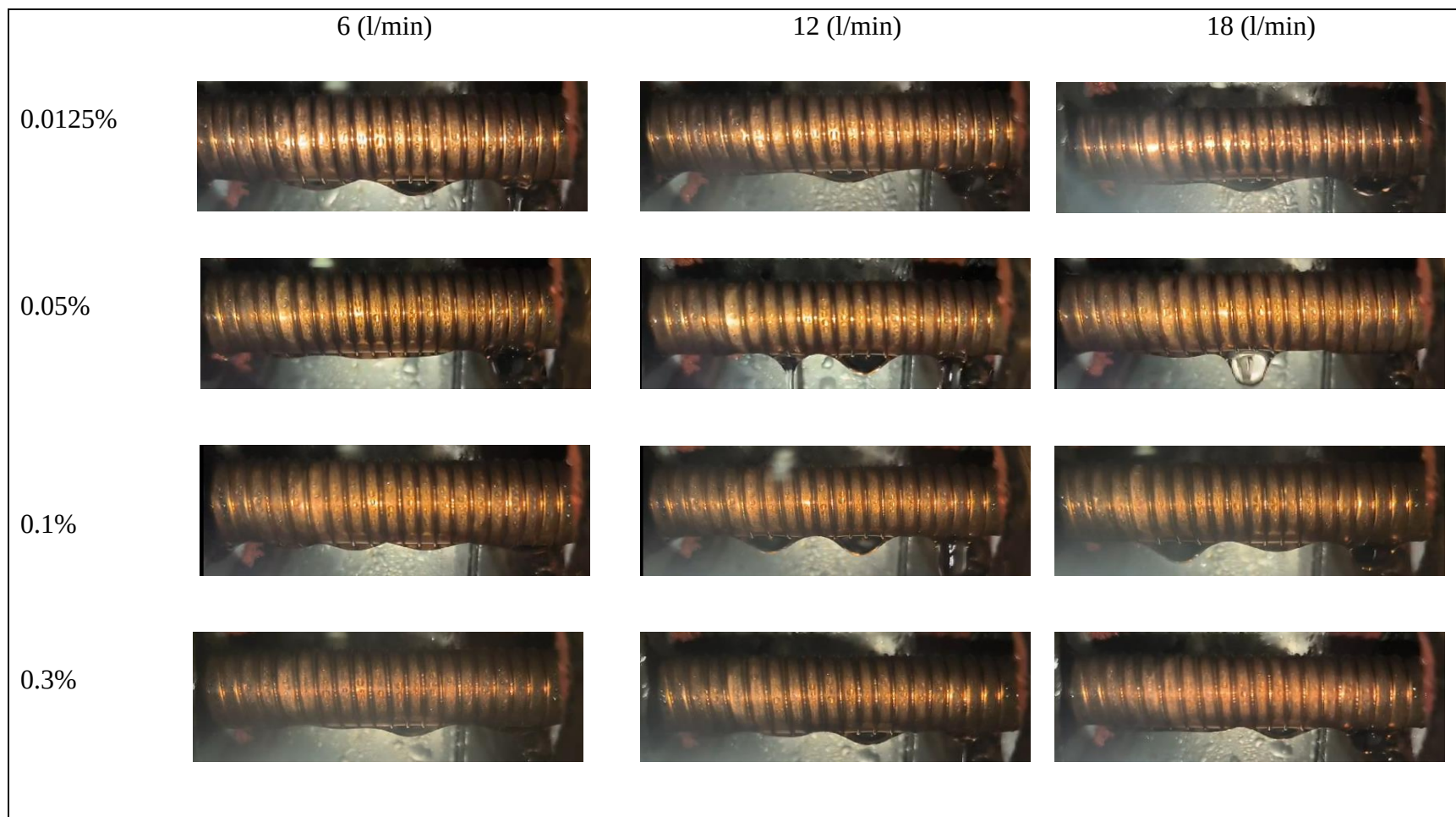
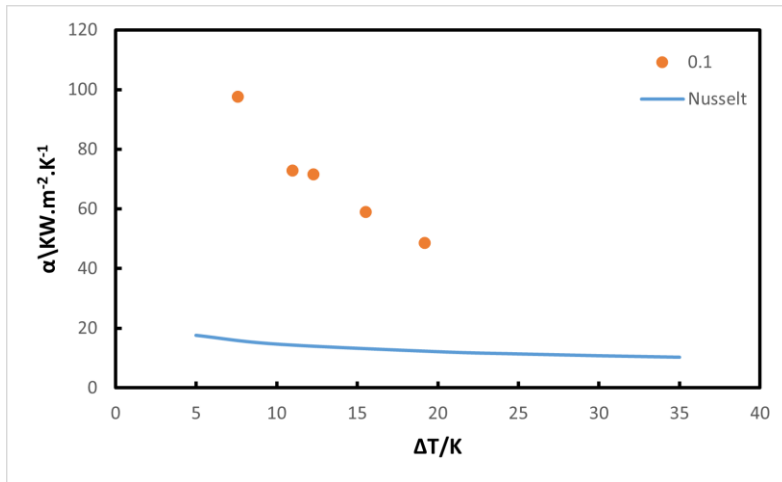
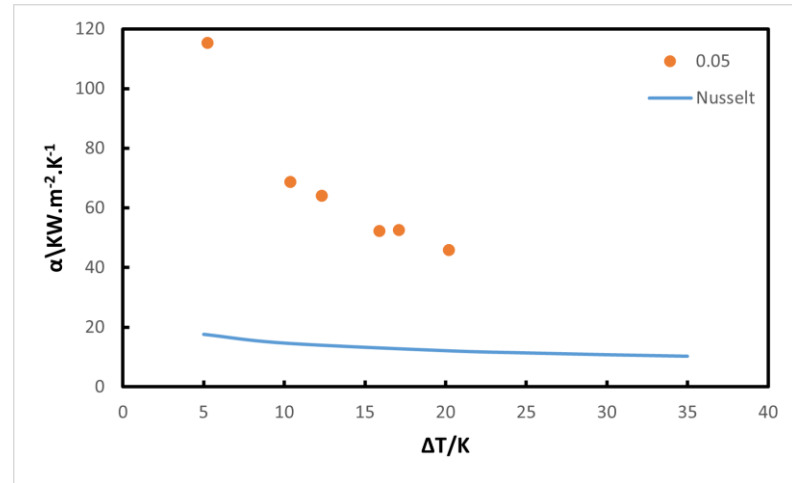
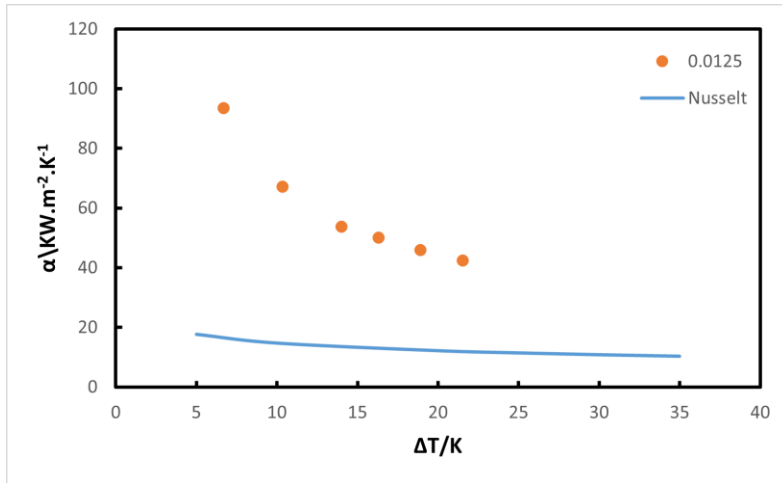
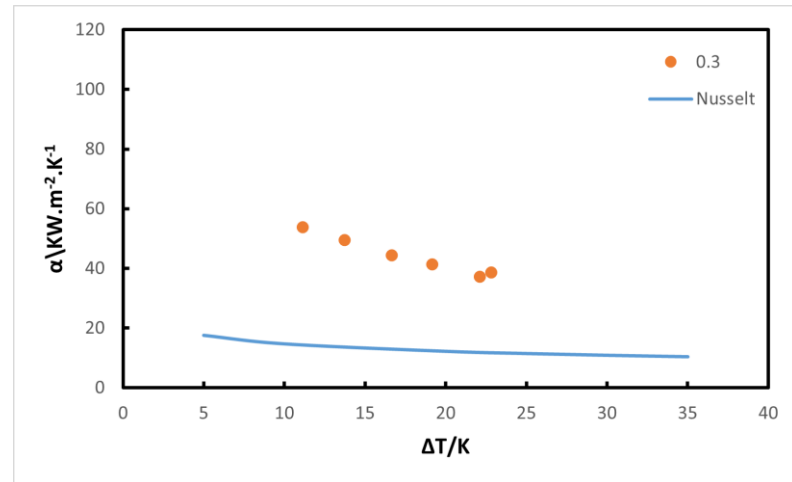


Figure 23 Change of condensate appearance as the vapor side-to-surface temperature difference increases for a 2.9 mm pitch.

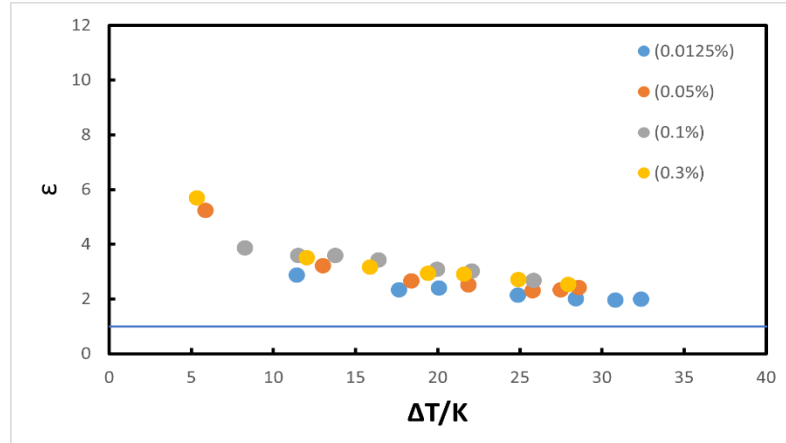
2.9mm



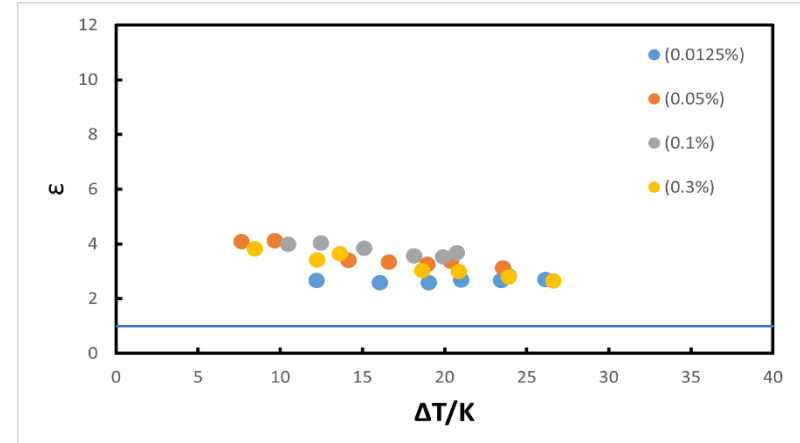
1.6 mm



2 mm



2.4 mm



2.9 mm

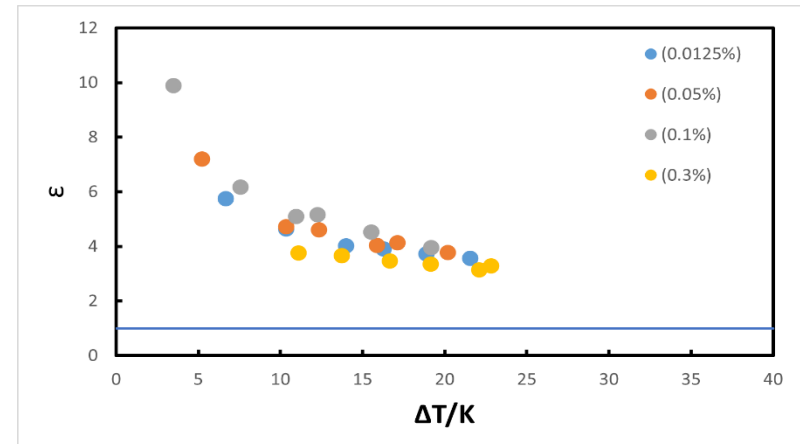
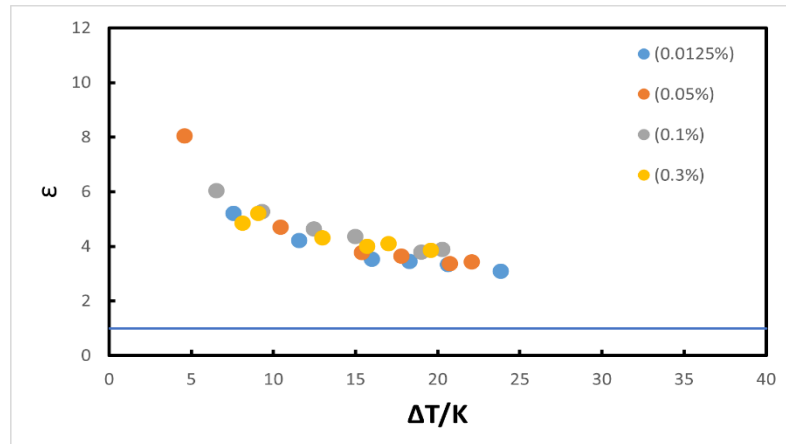


Figure 24 The enhancement ratio (ε) comparing the heat flux of the steam-ethanol mixture at a particular vapor side-to-surface temperature difference with the theoretical heat flux values for pure steam obtained from Nusselt's (1916) theory at the same vapor side

CHAPTER 6

Heat Transfer Enhancement predicting of New

Condensation Fluids on Pin-Fin Tubes with

6.1 Introduction

Pin-fin tubes leverage fins to boost heat transfer performance by increasing surface area. The presence of fins amplifies surface area, while shorter and thinner boundary layers, resulting from a small pin height to tube circumference ratio, reduce thermal resistance. However, condensation retention or flooding between fins at the tube's bottom decreases the "active" surface area, leading to reduced heat transfer in that section. Various studies, as discussed in Chapter 2, have examined pin-fin tube performance, and explored the impact of parameters such as material, geometry, and condensate fluid.

Maximizing heat transfer in condensation tubes is crucial for their design. Previous researchers have noted significant improvement upon adding pin-fins, measured by enhancement ratios for different tube and pin-fin geometries. This prompted current research to conduct a multi-optimization study aiming for the highest possible enhancement ratio. Investigating a wide range of geometrical parameters for tubes and pin-fins, such as pin height, longitudinal and circumferential pin spacing, and pin thickness while keeping tube diameter constant, was carried out to determine optimal ranges for each geometry. Both gravity and surface tension effects were considered. The optimal

geometries determined from the multi-optimization study are applied to predict the performance of pin-fin tubes using two new, low-emission fluids: R-417a and R-1234yf. This assessment aims to evaluate the feasibility of substituting these low-emission fluids for high-emission and commercial refrigerant fluids like R-22 and R-143a.

6.2 Results

A multi-optimization investigation was conducted to determine the highest enhancement ratio for R-417A and R-1234yf fluids, focusing on the geometrical dimensions of the tube. Parameters like circumferential pin thickness, pin height, longitudinal pin spacing at the pin root, and circumferential pin spacing were varied while keeping tube diameter and longitudinal pin tip thickness constant at 12.7 mm and 0.5 mm.

Figure 25 illustrates the enhancement ratio for these fluids with variations in pin height and circumferential pin thickness. Minimizing circumferential pin thickness was found to yield the highest enhancement ratio, as it increases the number of pins around the circumference, enhancing heat transfer through the pin tips. Additionally, maximizing pin height was crucial for achieving the highest enhancement ratio.

Figure 26 depicts how heat transfer varies with changes in pin height for both fluids, while keeping circumferential pin thickness and pin spacing constant and adjusting circumferential pin spacing. The highest heat transfer gain occurs at smaller values of circumferential pin spacing, which aligns with earlier findings. This relationship is influenced by both circumferential pin thickness and spacing, as fewer pins are present at larger values of these parameters. Consistent with previous results, maximum heat transfer is observed at the highest pin height, expanding the effective area for heat transfer along

the flank. Increasing pin height from 0.2 mm to 2.2 mm led to enhancement ratios of 3.61 for R-417A and 3.85 for R-1234yf.

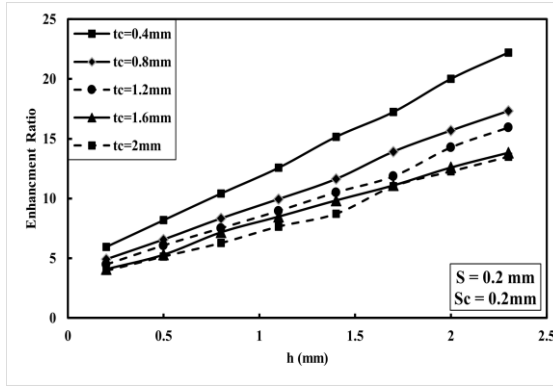
Figure 27 illustrates how the enhancement ratio correlates with circumferential pin spacing and longitudinal pin spacing, ranging from 0.2 mm to 2.2 mm. It's noted that maximum heat transfer occurs with the smallest longitudinal pin spacing. There's potential for even greater heat transfer enhancements with narrower fin spacing, below the minimum investigated value of 0.2 mm.

With a single longitudinal pin pitch, the number of pins remains constant, but their density on the tube increases, leading to further heat transfer improvements. Heat transfer primarily occurs through the pin roots in unflooded areas of the tube, while reducing longitudinal pin spacing decreases heat transfer from the pin root, an area partially affected by condensation retention, but increases heat transfer from the pin tip, unaffected by condensation.

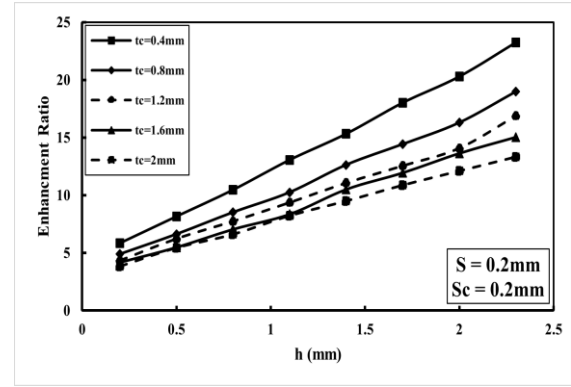
The enhancement ratio for R-417A and R-1234yf increased to 2.83 and 2.95, respectively, when longitudinal pin spacing decreased from 2 mm to 0.2 mm.

Figure 28 (a) compares the enhancement ratios of R-417A and R-22 across variations in circumferential pin thickness and pin height while maintaining optimal circumferential and longitudinal pin spacing. The results show a slight difference between the two fluids, particularly at pin heights below 1.5 mm, suggesting that R-417A could effectively replace R-22.

Similarly, Figure 28 (b) compares enhancement ratios between R-1234yf and R-134A. The results reveal a notable similarity in enhancement ratio values, indicating that R-1234yf could be a promising substitute for R-134A.

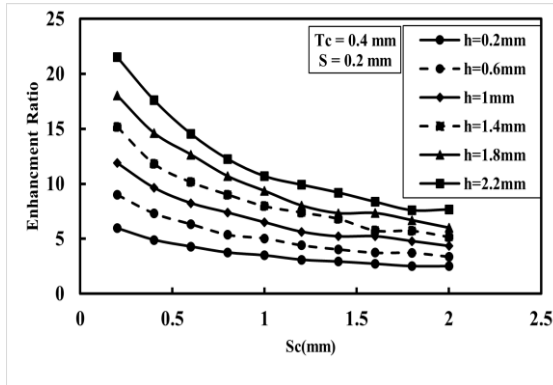


(a) R-417A

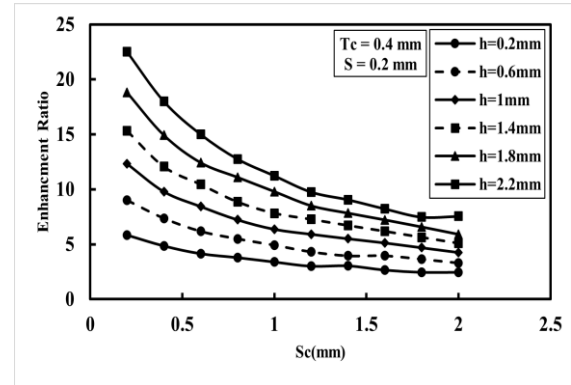


(b) R-1234yf

Figure 25 The enhancement ratio (ϵ) comparing the heat flux of the steam-ethanol mixture at a particular vapor side-to-surface temperature difference with the theoretical heat flux values for pure steam obtained from Nusselt's (1916) theory at the same

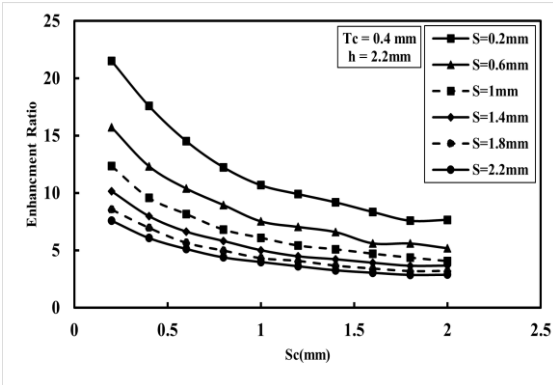


(a) R-417A

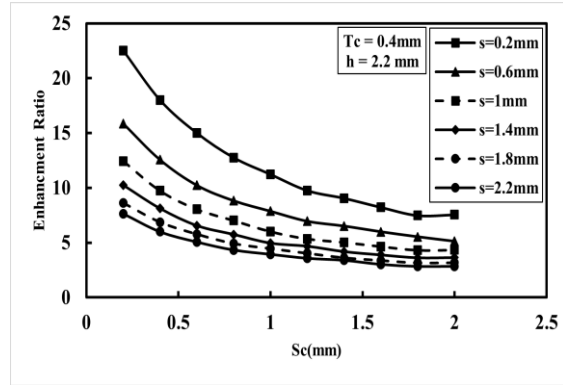


(b) R-1234yf

Figure 26 Heat-transfer enhancement ratio plotted with circumferential pin spacing and pin height.

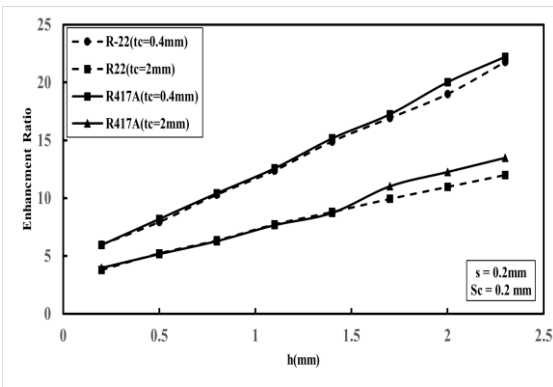


(a) R-417A

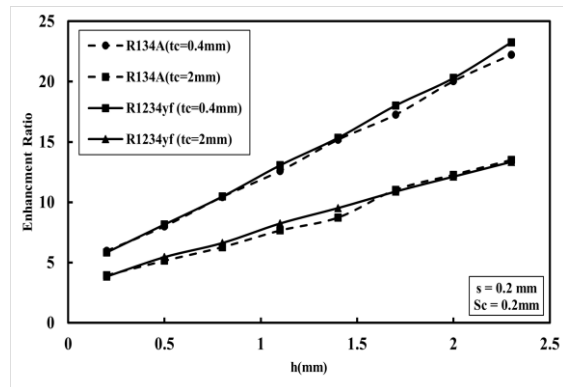


(b) R-1234yf

Figure 27 Heat-transfer enhancement ratio plotted with circumferential pin spacing and longitudinal pin spacing.



(a)



(b)

Figure 28 Heat-transfer enhancement ratio plotted with circumferential pin thickness and pin height.

CHAPTER 7

CONCLUSION

7.1 General Conclusion

Previous experimental studies extensively explored condensation of steam-ethanol mixtures on plain tubes, consistently finding that the addition of a small amount of ethanol improves performance compared to pure steam condensation. However, investigations into Marangoni condensation on wire-wrapping tubes have never been investigated before.

Experimental investigations of Marangoni condensation were conducted on horizontal smooth tubes and wire-wrapping. These experiments involved varying ethanol concentrations, wire pitches and coolant water flow rate, while maintaining constant vapor velocity. The obtained data showed good agreement with previous theoretical and experimental findings, indicating significant enhancement in condensation performance for wire-wrapping tubes, which is open up the horizon for further investigation.

Regarding pin-fin tubes, numerous theoretical and experimental investigations have been conducted for various fluids and refrigerants. In the current study, a theoretical investigation was undertaken to identify optimal geometrical parameters aimed at increasing tube surface area to enhance heat transfer.

7.2 Steam-ethanol on plain tube

The comparison of experimental data with previous studies by Jivani et al. demonstrates excellent agreement, validating the reliability of the collected data. By expanding the

investigation to include a wider range of ethanol concentrations, the transitional stages of condensation modes were identified.

Notably, visual observations and analysis of heat flux and convective heat transfer coefficient reveal distinct transitions at various ethanol concentrations. At lower concentrations, below 0.0125% the transition from wavy film-wise to complete film-wise condensation was observed at low vapor side-to-surface temperature differences.

Interestingly, at higher ethanol concentrations, of 0.3%, a unique trend in convective heat transfer coefficient and enhancement ratio was observed, with a temporary increase to reach the peak before decreasing again. This phenomenon corresponds to a transition from wavy film-wise to pseudo dropwise condensation to wavy film-wise again at higher flow rate, these modes were captured with a high-resolution camera.

The enhancement ratio calculated for steam-ethanol mixtures reveals significant improvements in heat transfer efficiency compared to theoretical predictions for pure steam, the maximum enhancement ratio of 3.8 was recorded.

7.3 Steam-ethanol on wire-wrapping

The investigation into Marangoni condensation on wire-wrapped tubes reveals how steam-ethanol concentrations and tube pitches impact heat transfer. By adding ethanol, heat transfer improves as the condensate film transforms into small droplets, converting from film-wise into a pseudo drop-wise condensation, reducing retention and expanding active surface area. Additionally, wire wrapping decreases film thickness and increases tube surface area, contributing to enhanced heat transfer. Both factors were thoroughly investigated to understand their combined effect. Minimum enhancement is consistently observed at lower ethanol concentrations, while maximum enhancement is at 0.1%, similar

to plain tubes. At low flow rates and temperature differences, significant enhancements occur, but decrease as temperature difference increases. The convective heat transfer coefficient peaks at a low flow rate before declining.

Heat flux exceeds pure steam values across all pitches, with the maximum enhancement consistently observed at 2.9 mm pitch, except for 0.3% of ethanol concentration, where it shifts to 2.4 mm. The visual observation of pitches of 2.4 mm and 2.9 mm showed increased droplet numbers and reduced retention, leading to expanded active surface area and maximum convective heat transfer, particularly noticeable at higher concentrations.

The enhancement ratio for Marangoni condensation on wire-wrapping tubes exceeds unity across various steam-ethanol concentrations and wire pitches. Notably, at a pitch of 2.9 mm, the maximum ratio reaches 9.9, showing substantial improvement compared to plain tube condensation. Similarly, at a pitch of 2.4 mm, significant enhancement is observed, especially at 0.05% ethanol concentration, with an enhancement ratio of 8. The trend of the enhancement ratio reflects the behavior of the convective heat transfer coefficient concerning temperature difference. Overall, these findings highlight the potential of wire-wrapping tubes for enhancing Marangoni condensation.

7.4 Pin-Fin tube

A comprehensive multi-optimization study was undertaken on horizontal pin-fin tubes, utilizing Hafiz's model to incorporate the joint influences of surface tension and gravity. The investigation yielded significant insights. Optimal geometrical dimensions resulted in remarkable enhancements in the enhancement ratio for R-417A and R-1234yf, reaching impressive values of 21.5 and 22.5, respectively. The ideal geometry featured minimized circumferential pin thickness, circumferential pin spacing, and longitudinal pin spacing,

coupled with maximized pin height. Notably, the number of pins emerged as the most influential factor in enhancing heat transfer, with its efficacy directly linked to geometric dimensions. Furthermore, comparison of enhancement ratios demonstrated excellent agreement and minimal deviation between R-134a and R-1234yf, as well as R-22 and R-417A, indicating the potential of R-417A and R-1234yf as optimal substitute.

References

- [1] J. W. Rose, "Dropwise condensation theory and experiment: A review," *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, vol. 216, no. 2, 2002, doi: 10.1243/09576500260049034.
- [2] V. V. Mirkovich and R. W. en, "Non-filmwise condensation of binary vapors of miscible liquids," *The Canadian Journal of Chemical Engineering*, vol. 39, no. 2. 1961. doi: 10.1002/cjce.5450390207.
- [3] V. V. Mirkovich and R. W. en, "A study of the condensation of binary vapors of miscible liquids: Part 2: Heat transfer co-efficients for filmwise and non-filmwise condensation," *Can J Chem Eng*, vol. 41, no. 2, 1963, doi: 10.1002/cjce.5450410208.
- [4] W. Nusselt, *Die Oberflächenkondensation des Wasserdampfes*, vol. 60, no. 27. 1916.
- [5] S. B. Memory and J. W. Rose, "Free convection laminar film condensation on a horizontal tube with variable wall temperature," *Int J Heat Mass Transf*, vol. 34, no. 11, 1991, doi: 10.1016/0017-9310(91)90236-8.
- [6] L. A. Bromley, "Effect of Heat Capacity of Condensate," *Ind Eng Chem*, vol. 44, no. 12, 1952, doi: 10.1021/ie50516a052.
- [7] E. M. Sparrow and J. L. Gregg, "Laminar Condensation Heat Transfer on a Horizontal Cylinder," *J Heat Transfer*, vol. 81, no. 4, 1959, doi: 10.1115/1.4008210.
- [8] E. M. Sparrow and J. L. Gregg, "A Boundary-Layer Treatment of Laminar-Film Condensation," *J Heat Transfer*, vol. 81, no. 1, 1959, doi: 10.1115/1.4008118.

- [9] W. M. Rohsenow, "Heat Transfer and Temperature Distribution in Laminar-Film Condensation," *J Fluids Eng*, vol. 78, no. 8, 1956, doi: 10.1115/1.4014125.
- [10] P. Sadasivan and J. H. Lienhard, "Sensible heat correction in laminar film boiling and condensation," *Journal of Heat Transfer*, vol. 109, no. 2. 1987. doi: 10.1115/1.3248122.
- [11] A. P. Colburn and O. A. Hougen, "Design of Cooler Condensers for Mixtures of Vapors with Noncondensing Gases," *Ind Eng Chem*, vol. 26, no. 11, 1934, doi: 10.1021/ie50299a011.
- [12] Z. Zhao, Y. Li, L. Wang, Z. Liu, and J. Zheng, "Flow and heat transfer characteristics of ambient air condensation on a horizontal cryogenic tube," *Cryogenics (Guildf)*, vol. 62, 2014, doi: 10.1016/j.cryogenics.2014.03.021.
- [13] B. Cheng and W. Q. Tao, "Experimental study of R-152a film condensation on single horizontal smooth tube and enhanced tubes," *Journal of Heat Transfer*, vol. 116, no. 1. 1994. doi: 10.1115/1.2910874.
- [14] D. G. Kroger and W. M. Rohsenow, "Condensation heat transfer in the presence of a non-condensable gas," *Int J Heat Mass Transf*, vol. 11, no. 1, 1968, doi: 10.1016/0017-9310(68)90060-4.
- [15] H. K. Al-Diwany and J. W. Rose, "Free convection film condensation of steam in the presence of non-condensing gases," *Int J Heat Mass Transf*, vol. 16, no. 7, 1973, doi: 10.1016/0017-9310(73)90144-0.
- [16] T. Murase, H. S. Wang, and J. W. Rose, "Marangoni condensation of steam–ethanol mixtures on a horizontal tube," *Int J Heat Mass Transf*, vol. 50, no. 19–20, pp. 3774–3779, Sep. 2007, doi: 10.1016/j.ijheatmasstransfer.2007.02.011.

- [17] H. Ali, H. S. Wang, A. Briggs, and J. W. Rose, "Effects of vapor velocity and pressure on marangoni condensation of steam-ethanol mixtures on a horizontal tube," *J Heat Transfer*, vol. 135, no. 3, 2013, doi: 10.1115/1.4007893.
- [18] S. Jivani, J. H. Liu, J. H. Pu, and H. S. Wang, "Marangoni condensation of steam-ethanol mixtures on a horizontal smooth tube," *Exp Therm Fluid Sci*, vol. 128, Oct. 2021, doi: 10.1016/j.expthermflusci.2021.110434.
- [19] H. Ali, M. S. Kamran, H. M. Ali, and S. Imran, "Condensation heat transfer enhancement using steam-ethanol mixtures on horizontal finned tube," *International Journal of Thermal Sciences*, vol. 140, pp. 87–95, Jun. 2019, doi: 10.1016/j.ijthermalsci.2019.02.033.
- [20] C. Bonneau, C. Josset, V. Melot, and B. Auvity, "Comprehensive review of pure vapour condensation outside of horizontal smooth tubes," *Nuclear Engineering and Design*, vol. 349, 2019. doi: 10.1016/j.nucengdes.2019.04.005.
- [21] I. G. Shekriladze and V. I. Gomelauro, "Theoretical study of laminar film condensation of flowing vapour," *Int J Heat Mass Transf*, vol. 9, no. 6, 1966, doi: 10.1016/0017-9310(66)90092-5.
- [22] H. Honda and T. Fujii, "EFFECT OF THE DIRECTION OF ON-COMING VAPOR ON LAMINAR FILMWISE CONDENSATION ON A HORIZONTAL CYLINDER.," 1974. doi: 10.1615/ihct5.1200.
- [23] V. E. Denny and A. F. Mills, "Laminar film condensation of a flowing vapor on a horizontal cylinder at normal gravity," *J Heat Transfer*, vol. 91, no. 4, 1969, doi: 10.1115/1.3580233.

- [24] V. E. Denny and V. South, "Effects of forced flow, noncondensables, and variable properties on film condensation of pure and binary vapors at the forward stagnation point of a horizontal cylinder," *Int J Heat Mass Transf*, vol. 15, no. 11, 1972, doi: 10.1016/0017-9310(72)90037-3.
- [25] F. Tetsu, U. Haruo, and K. Chikatoshi, "Laminar filmwise condensation of flowing vapour on a horizontal cylinder," *Int J Heat Mass Transf*, vol. 15, no. 2, 1972, doi: 10.1016/0017-9310(72)90071-3.
- [26] J. S. d'Avila and M. Schmal, "CONDENSATION BY COMBINED BODY FORCE AND FORCED CONVECTION ON VERTICAL PLATES.," *Rev Bras Tecnol*, vol. 9, no. 1–2, 1978.
- [27] S. M. Martins Ferreira, "Forced convection condensation of vapor flowing around a circular cylinder: Effect of the presence of inert gas, gravitational field and superheating," *The Chemical Engineering Journal*, vol. 6, no. 2, 1973, doi: 10.1016/0300-9467(73)80022-X.
- [28] E. S. Gaddis, "Solution of the two phase boundary-layer equations for laminar film condensation of vapour flowing perpendicular to a horizontal cylinder," *Int J Heat Mass Transf*, vol. 22, no. 3, 1979, doi: 10.1016/0017-9310(79)90003-6.
- [29] D. Homescu and P. K. Panday, "Forced convection condensation on a horizontal tube: Influence of turbulence in the vapor and liquid phases," *Journal of Heat Transfer*, vol. 121, no. 4. 1999. doi: 10.1115/1.2826078.
- [30] S. A. Yang and Y. T. Lin, "Turbulent film condensation on a non-isothermal horizontal tube - Effect of eddy diffusivity," *Appl Math Model*, vol. 29, no. 12, 2005, doi: 10.1016/j.apm.2005.02.012.

- [31] C. K. Chen and Y. T. Lin, "Laminar film condensation from a downward-flowing steam-air mixture onto a horizontal circular tube," *Appl Math Model*, vol. 33, no. 4, 2009, doi: 10.1016/j.apm.2008.05.003.
- [32] W. C. Lee, S. Rahbar, and J. W. Rose, "Film condensation of refrigerant-113 and ethanediol on a horizontal tube—effect of vapor velocity," *J Heat Transfer*, vol. 106, no. 3, 1984, doi: 10.1115/1.3246710.
- [33] H. Honda, S. Nozu, and T. Fujii, "VAPOUR TO COOLANT HEAT TRANSFER DURING CONDENSATION OF FLOWING VAPOUR ON A HORIZONTAL TUBE.," in *Heat Transfer, Proceedings of the International Heat Transfer Conference*, 1982. doi: 10.1615/ihtc7.410.
- [34] J. Huang, J. Zhang, and L. Wang, "Review of vapor condensation heat and mass transfer in the presence of non-condensable gas," *Applied Thermal Engineering*, vol. 89, 2015. doi: 10.1016/j.applthermaleng.2015.06.040.
- [35] W. C. Lee and J. W. Rose, "Forced convection film condensation on a horizontal tube with and without non-condensing gases," *Int J Heat Mass Transf*, vol. 27, no. 4, 1984, doi: 10.1016/0017-9310(84)90025-5.
- [36] S. P. Sukhatme, B. S. Jagadish, and P. Prabhakaran, "Film condensation of R-11 vapor on single horizontal enhanced condenser tubes," *J Heat Transfer*, vol. 112, no. 1, 1990, doi: 10.1115/1.2910350.
- [37] A. Briggs, "Enhanced condensation of r-113 and steam using three-dimensional pin-fin tubes," *Experimental Heat Transfer*, vol. 16, no. 1, 2003, doi: 10.1080/08916150390126469.

- [38] M. Baisar and A. Briggs, "Condensation of steam on pin-fin tubes: Effect of circumferential pin thickness and spacing," in *Heat Transfer Engineering*, 2009. doi: 10.1080/01457630902921014.
- [39] H. M. Ali and A. Briggs, "Condensation heat transfer on pin-fin tubes: Effect of thermal conductivity and pin height," *Appl Therm Eng*, vol. 60, no. 1–2, 2013, doi: 10.1016/j.applthermaleng.2012.08.020.
- [40] H. M. Ali and A. Briggs, "Enhanced condensation of ethylene glycol on single pin-fin tubes: Effect of pin geometry," *J Heat Transfer*, vol. 134, no. 1, 2012, doi: 10.1115/1.4004714.
- [41] P. Qin, Z. Zhang, T. Xu, X. Gao, and S. Wang, "Experimental investigation on condensation heat transfer of R134a on single horizontal copper and stainless steel three-dimensional finned tubes," in *AIP Conference Proceedings*, 2013. doi: 10.1063/1.4816903.
- [42] B. Kundu and K. S. Lee, "Optimum configurations of vertical fins under condensation of saturated vapor," *International Journal of Refrigeration*, vol. 34, no. 4, 2011, doi: 10.1016/j.ijrefrig.2011.01.002.
- [43] B. Kundu, "Approximate analytical method for prediction of performance and optimum dimensions of pin fins subject to condensation of quiescent vapor," *International Journal of Refrigeration*, vol. 32, no. 7, 2009, doi: 10.1016/j.ijrefrig.2009.04.006.
- [44] B. Kundu and G. K. Ghosh, "An approximate analytical prediction about thermal performance and optimum design of pin fins subject to condensation of saturated

- steam flowing under forced convection,” *International Journal of Refrigeration*, vol. 32, no. 5, 2009, doi: 10.1016/j.ijrefrig.2008.12.006.
- [45] H. M. Ali and A. Briggs, “An investigation of condensate retention on pin-fin tubes,” *Appl Therm Eng*, vol. 63, no. 2, 2014, doi: 10.1016/j.applthermaleng.2013.11.038.
- [46] H. M. Ali and A. Briggs, “A semi-empirical model for free-convection condensation on horizontal pin-fin tubes,” *Int J Heat Mass Transf*, vol. 81, 2015, doi: 10.1016/j.ijheatmasstransfer.2014.10.008.
- [47] J. W. Rose, “An approximate equation for the vapour-side heat-transfer coefficient for condensation on low-finned tubes,” *Int J Heat Mass Transf*, vol. 37, no. 5, 1994, doi: 10.1016/0017-9310(94)90122-8.
- [48] A. Briggs and J. W. Rose, “Effect of fin efficiency on a model for condensation heat transfer on a horizontal, integral-fin tube,” *Int J Heat Mass Transf*, vol. 37, no. SUPPL. 1, 1994, doi: 10.1016/0017-9310(94)90045-0.
- [49] H. M. Ali, “An analytical model for prediction of condensate flooding on horizontal pin-fin tubes,” *Int J Heat Mass Transf*, vol. 106, 2017, doi: 10.1016/j.ijheatmasstransfer.2016.10.088.
- [50] H. M. Ali, M. Z. Qasim, and M. Ali, “Free convection condensation heat transfer of steam on horizontal square wire wrapped tubes,” *Int J Heat Mass Transf*, vol. 98, 2016, doi: 10.1016/j.ijheatmasstransfer.2016.03.053.

Appendix A: Estimation of Uncertainties in Experimental Measurements

A.1 Introduction

Since it is impossible to exactly reproduce the coolant and vapor conditions on two different occasions the present results are essentially single sample measurements. Due to this fact the method of Kline and McClintock (1953) (see also Moffat (1988)) was used to estimate the uncertainties in the measurements (i.e. coolant flow rates, temperatures etc) and calculate the propagation of these uncertainties in the derived quantities (i.e. heat flux, heat transfer coefficients etc).

The final expression to describe a variable can be written as follow,

$$x = x_{measured} + \delta x \quad (A.1)$$

Where:

x	The final best estimate of the variable
$x_{measured}$	The measured value of the variable
δx	The uncertainty in the measured value estimated from for instance fluctuations in instrument readings, scale graduations, calibration equations, etc

The final measurements are expected to lie within $\pm \delta x$ of the true value. In general, the final result of an experiment, x_R , will be a function of several measured quantities with each having a different uncertainty value, i.e.

$$x_R = f(x_1, x_2, \dots x_n) \quad (A.2)$$

Kline and McClintock (1953) suggested the following expression for calculating the

resulting uncertainty, δx_R , in the dependent variable, x_R ,

$$\delta x_R = \left[\left(\frac{\partial f}{\partial x_1} \delta x_1 \right)^2 + \left(\frac{\partial f}{\partial x_2} \delta x_2 \right)^2 + \cdots + \left(\frac{\partial f}{\partial x_n} \delta x_n \right)^2 \right]^{0.5} \quad (\text{A.3})$$

Eqn. (A.3) can be non-dimensionalised to give,

$$\frac{\delta x_R}{x_R} = [(x_1 \delta x_1)^2 + (x_2 \delta x_2)^2 + \cdots + (x_n \delta x_n)^2]^{0.5} \quad (\text{A.4})$$

A.2 Application to the Present Investigation

In the present investigation, the uncertainty levels were found in the test-section vapor velocity, U_v , heat flux on the outside of the test tube, q , overall heat-transfer coefficient, U .

A.2.1 Test-Section Vapor Velocity

The test section vapor velocity was calculated using Eqn. (22),

$$U_v = \frac{\dot{m}_v}{\rho_v A_{ts}} \quad (\text{A.5})$$

Thus, the fractional uncertainty in the test section vapor velocity depends on the test section vapor density, test section diameter and vapor mass flow rate as follow,

$$\frac{\delta U_v}{U_v} = \left[(X_{\rho_{vts}} \delta x_{\rho_{vts}})^2 + (X_{d_{ts}} \delta x_{d_{ts}})^2 + (X_{\dot{m}_v} \delta x_{\dot{m}_v})^2 \right]^{0.5} \quad (\text{A.6a})$$

Where,

$$X_{\rho_{vts}} = \frac{-\frac{\partial \rho_{vts}(T_{sat})}{\partial T_{sat}}}{\rho_{vts}(T_{sat})} \quad (\text{A.6b})$$

$$X_{d_{ts}} = -\frac{2}{d_{ts}} \quad (\text{A.6c})$$

$$X_{m_v} = \frac{1}{m_v} \quad (\text{A.6d})$$

The uncertainty level in the test-section vapor density, $\delta x_{\rho_{vts}}$, was measured from the vapor temperature of $\pm 0.05 \text{ kg/m}^3$, and the uncertainty level of test section diameter, $\delta x_{d_{ts}}$, was estimated from manufacturing tolerance to be $\pm 0.0005 \text{ m}$. The uncertainty in vapor mass flow rate, δx_{m_v} , was measured by Lee (1982) using the same method as here but for a different experimental apparatus, by collecting and weighing the condensate at the exit of the auxiliary condenser and comparing the results with the calculated value of vapor mass flow rate from Eqn. (22). He found that the two methods agreed to within $\pm 1.5 \%$ over the whole range of vapor mass flow rate. The uncertainty level δx_{m_v} can therefore be estimated as $0.015 m_v$. Substituting this value along with Eqn. (13) into Eqn. (10) reduces the final term of Eqn. (10) to $(0.015)^2$, i.e. the fractional uncertainty in the vapor velocity is independent of the vapor mass flow rate.

A2.2 Heat Flux

The heat flux on the outside of the test tube was calculated from Eqns. (20) and (21). By combining these we have,

$$q = \frac{m_c C_{pc} \Delta T_c}{\pi D L} \quad (\text{A.7})$$

Memory (1989) assumed negligible error in the property equations and showed that the uncertainty in C_{pc} due to the uncertainty in the measured coolant temperature was negligible. The fractional uncertainty in heat flux can then be written as,

$$\frac{\delta q}{q} = \left[(X_{m_c} \delta x_{m_c})^2 + (X_{\Delta T_c} \delta x_{\Delta T_c})^2 + (X_D \delta x_D)^2 + (X_L \delta x_L)^2 \right]^{0.5} \quad (\text{A.8a})$$

Where,

$$X_{m_c} = \frac{1}{m_c} \quad (\text{A.8b})$$

$$X_{\Delta T_c} = -\frac{2}{\Delta T_c} \quad (\text{A.8c})$$

$$X_D = -\frac{1}{D} \quad (\text{A.8d})$$

$$X_L = -\frac{1}{L} \quad (\text{A.8e})$$

The uncertainty level in coolant mass flow rate, m_c , was estimated to be ± 0.5 l/min from calibration experiments performed by Briggs (1991). The coolant temperature difference was found using a J-type thermocouple with an uncertainty level of ± 0.005 K. The uncertainties in tube diameter and length were estimated from manufacturing tolerances to be ± 0.0001 and ± 0.0005 m respectively.

A.2.3 Overall Heat Transfer Coefficient

The following expression was used to calculate the overall heat-transfer coefficient,

$$U = \frac{q}{T_{sat} - \left(\frac{T_{in} + T_{out}}{2} \right)} \quad (\text{A.9})$$

The fractional uncertainty in $\mathbf{U}$ can be found as,

$$\frac{\delta q}{q} = \left[(X_q \delta x_q)^2 + (X_{T_{sat}} \delta x_{T_{sat}})^2 + (X_{T_{in}} \delta x_{T_{in}})^2 + (X_{T_{out}} \delta x_{T_{out}})^2 \right]^{0.5} \quad (\text{A.10a})$$

Where,

$$X_q = \frac{1}{q} \quad (\text{A.10b})$$

$$X_{T_{sat}} = -\frac{U}{q} \quad (\text{A.10c})$$

$$X_{T_{in}} = X_{T_{out}} = \frac{U}{2q} \quad (\text{A.10d})$$

The uncertainty in heat flux, δx_q , was found from Eqn. (A.8a), T_{sat} , T_{in} , and T_{out} were measured using thermocouples before the test tube, inlet and outlet of the test tube with an estimated uncertainty of ± 0.005 K.

Test-Section Vapor Velocity

U_c (L/min)	$(X_{\rho_{vts}} \delta x_{\rho_{vts}})^2 / 10^{-6}$	$(X_{d_{ts}} \delta x_{d_{ts}})^2 / 10^{-6}$	$(X_{m_v} \delta x_{m_v})^2 / 10^{-6}$	$(\delta U_v / U_v) \times 100\%$
6	5.77	187	225	2.045
18	5.77	187	225	2.045

Table 1 Results of Uncertainty Analysis for Test Section Vapor Velocity

Heat Flux

U_c (L/min)	$(X_{m_c} \delta x_{m_c})^2 / 10^{-5}$	$(X_{\Delta T_c} \delta x_{\Delta T_c})^2 / 10^{-5}$	$(X_D \delta x_D)^2 / 10^{-5}$	$(X_L \delta x_L)^2 / 10^{-5}$	$(\delta q / q) \times 100\%$
6	694.44	1.83	5.92	5.10	8.41
18	77.16	8.80	5.92	5.10	3.11

Table 2 Results of Uncertainty Analysis for Smooth Tube (pure steam): Heat Flux Measurement

U_c (L/min)	$(X_{m_c} \delta x_{m_c})^2 / 10^{-5}$	$(X_{\Delta T_c} \delta x_{\Delta T_c})^2 / 10^{-5}$	$(X_D \delta x_D)^2 / 10^{-5}$	$(X_L \delta x_L)^2 / 10^{-5}$	$(\delta q / q) \times 100\%$
6	694.44	1.08	5.92	5.10	8.41
18	77.16	6.08	5.92	5.10	3.07

Table 3 Results of Uncertainty Analysis for Smooth Tube (steam-ethanol mixture 0.0125%): Heat Flux Measurement

U_c (L/min)	$(X_{m_c} \delta x_{m_c})^2 / 10^{-5}$	$(X_{\Delta T_c} \delta x_{\Delta T_c})^2 / 10^{-5}$	$(X_D \delta x_D)^2 / 10^{-5}$	$(X_L \delta x_L)^2 / 10^{-5}$	$(\delta q / q) \times 100\%$
6	694.44	9.28	5.92	5.10	8.40
18	77.16	3.24	5.92	5.10	3.02

Table 4 Results of Uncertainty Analysis for Smooth Tube (steam-ethanol mixture 0.1%): Heat Flux Measurement

Overall Heat Transfer Coefficient

U_c (L/min)	$(X_q \delta x_q)^2 / 10^{-5}$	$(X_{T_{sat}} \delta x_{T_{sat}})^2 / 10^{-5}$	$(X_{T_{in}} \delta x_{T_{in}})^2 / 10^{-5}$	$(X_{T_{out}} \delta x_{T_{out}})^2 / 10^{-5}$	$(\delta U/U) \times 100\%$
6	694.44	1.83	5.92	5.10	8.41
18	77.16	8.80	5.92	5.10	3.11

Table 5 Results of Uncertainty Analysis for Smooth Tube (pure steam): Heat Flux Measurement

U_c (L/min)	$(X_q \delta x_q)^2 / 10^{-5}$	$(X_{T_{sat}} \delta x_{T_{sat}})^2 / 10^{-5}$	$(X_{T_{in}} \delta x_{T_{in}})^2 / 10^{-5}$	$(X_{T_{out}} \delta x_{T_{out}})^2 / 10^{-5}$	$(\delta U/U) \times 100\%$
6	694.44	1.83	5.92	5.10	8.41
18	77.16	8.80	5.92	5.10	3.11

Table 6 Results of Uncertainty Analysis for Smooth Tube (steam-ethanol mixture 0.0125%): Heat Flux Measurement

U_c (L/min)	$(X_q \delta x_q)^2 / 10^{-5}$	$(X_{T_{sat}} \delta x_{T_{sat}})^2 / 10^{-5}$	$(X_{T_{in}} \delta x_{T_{in}})^2 / 10^{-5}$	$(X_{T_{out}} \delta x_{T_{out}})^2 / 10^{-5}$	$(\delta U/U) \times 100\%$
6	694.44	1.83	5.92	5.10	8.41
18	77.16	8.80	5.92	5.10	3.11

Table 7 Results of Uncertainty Analysis for Smooth Tube (steam-ethanol mixture 0.1%): Heat Flux Measurement

Vitae

Name :Ahmad Adel Obeidat

Nationality :Jordanian

Date of Birth :8/3/1998

Email :aobeidat143@gmail.com

Address :Irbid-Masqat Street- Building 503-21166

Academic Background :Bachelor's degree in mechanical engineering , Jordan University of Science and Technology, Jordan

Work Experience :Graduate Teaching Assistant, Mechanical Engineering Department, College of Engineering & Physics, King Fahd University of Petroleum & Minerals, Saudi Arabia: Spring 2022 - Present.

Research Output : A PREDICTION OF HEAT TRANSFER ENHANCEMENT OF CONDENSATION FOR NEW FLUIDS ON PIN-FIN TUBE.

Obeidat, A. and Ali, H.M., 2023. A PREDICTION OF HEAT TRANSFER ENHANCEMENT OF CONDENSATION FOR NEW FLUIDS ON PIN-FIN TUBE. In *International Heat Transfer Conference Digital Library*. Begel House Inc..