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HEAT TRANSFER ENHANCEMENT DURING DOWNWARD LAMINAR FLOW CONDENSATION OF R134A IN VERTICAL SMOOTH AND MICROFIN TUBES

Ahmet Selim DALKILIÇ*, İsmail TEKE* and Somchai WONGWISES**
*Yildiz Technical University Mechanical Engineering Faculty Department of Mechanical Engineering
34349 Beşiktaş, İstanbul, Turkey, dalkilic@yildiz.edu.tr
**King Mongkut's University of Technology Thonburi Engineering Faculty Department of Mechanical Engineering
10140 Bangmod, Bangkok, Thailand, somchai.won@kmutt.ac.th

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Abstract: This paper presents an experimental comparison of the laminar film condensation heat transfer coefficients of R134a in vertical smooth and micro-fin tubes having inner diameters of 7 mm and lengths of 500 mm. Condensation experiments were performed at a mass flux of 29 kg m^{-2} s^{-1}. The pressures were between 0.8 and 0.9 MPa. The original smooth tube heat transfer model was modified by a well-known friction factor to account for the heat transfer enhancement effects due to the presence of micro-fins on the internal wall surface during annular flow regime conditions. Alterations of the local heat transfer coefficient, and condensation rate along the tube length during downward condensing film were determined, considering the effects of the temperature difference between the saturation temperature and the inner wall temperature of the test tubes, and the condensing temperature on these items. The results show that the interfacial shear stress is found to have significance for the laminar condensation heat transfer of R134a under the given conditions due to its better predictive performance than the classical solution neglecting the interfacial shear stress effect. A comparison of the condensation heat transfer coefficients was also done according to the condensing pressures. New empirical correlations of the condensation heat transfer coefficient belonging to the test tubes are proposed for practical applications.
Keywords: Condensation, Heat transfer coefficient, Downward flow, Laminar flow, Micro-fin tube, Nusselt theory, Annular flow.

İÇ YÜZEYİ PÜRÜZSÜZ VE MİKRO KANATLI DÜŞEY BORULARDA R134A'NIN LAMİNER AKIŞINDAKİ YOĞUŞMASI SIRASINDA MEYDANA GELEN ISI TRANSFERİ İYİLEŞMESİNİN ARAŞTIRILMASI

Özet: Bu çalışmada, iç çapı 7 mm ve uzunluğu 500 mm olan iç yüzeyi pürüzsüz ve mikro kanatlı borular içinde akan R134a'nın laminar film yoğuşması ısı transferi katsayısının deneysel karşılaştırılması sunulmuştur. Yoğuşma deneyleri 29 kg $m^{-2}s^{-1}$ kütlesel akılarında yapılmıştır. Basınçlar 0.8-0.9 MPa değerleri arasındadır. Halka akışı şartlarında geçerli olan iç yüzeyi pürüzsüz boruya ait olan ısı transferi modeli, literatürde yaygın olarak kullanılan bir sürtünme katsayısı ile boru iç yüzeyindeki mikro kanatların varlığından ötürü meydana gelen ısı transferi iyileşmesini hesaba katacak şekilde modifiye edilmiştir. Düşey olarak yoğuşan film esansındaki yerel ısı transferi katsayılarındaki değişim ve boru boyunca oluşan yoğuşma miktarı, doyma sıcaklığı ile boru iç yüzeyindeki sıcaklık farkı ve yoğuşma sıcaklığı da dikkate alınarak bulunmuştur.Sonuçlar arayüzey kayma gerilmesinin çalışmada belirtilen şartlarda R134a'nın laminer yoğuşma ısı transferi üzerinde öneme sahip olduğunu ve önerilen modelin arayüzey kayma gerilmesinin etkisinin ihmal edildiği geleneksel çözümden daha iyi sonuçlar vermesiyle göstermiştir. Ayrıca, yoğuşma basınçlarına göre ısı transferi katsayıları da karşılaştırılmıştır. Pratik uygulamalar için test edilen borulara ait yeni amprik yoğuşma ısı transferi eşitlikleri önerilmiştir.
Anahtar Kelimeler: Yoğuşma, Isı transferi katsayısı, Düşey akış, Laminer akış, Micro kanatlı boru, Nusselt teorisi, Halka akış.

NOMENCLATURE

A_i	tube inside surface area, m^2
A_{mf}/A_s	augmentation ratio
c_p	specific heat capacity at constant pressure, J $kg^{-1}K^{-1}$
d	inside diameter of tube, m
Exp	experiment
G	mass flux, kg $m^{-2}s^{-1}$
g	gravitational acceleration, m s^{-2}
h	heat transfer coefficient, W $m^{-2}K^{-1}$
i	specific enthalpy, j kg^{-1}

i_{fg}	latent heat of condensation, J kg^{-1}
i_{fg}'	correction factor included latent heat of condensation, J kg^{-1}
k	thermal conductivity, W m^{-1}K^{-1}
l	tube length, m
L	characteristic length
l_f	fin length, m
m	mass flow rate, kg s^{-1}
n	number of fins per unit length
Nu	Nusselt number
P	pressure, MPa
Pr	Prandtl number
Re	Reynolds number
Q	heat transfer rate, W
T	temperature, °C
u	axial velocity, m s^{-1}
w	tube thickness, m
x	vapour quality
y	wall coordinate
z	axial coordinate
$\Delta T_{r,sat}$	$T_{r,sat}$-T_{wi} , K

Greek symbols

δ	film thickness, m
δ^*	dimensionless film thickness
μ	dynamic viscosity, kg m^{-1}s^{-1}
ρ	density, kg m^{-3}
ρ_g^*	fictitious vapour density defined by Carey (1992), kg m^{-3}
β	spiral angle, rad
τ	shear stress, N m^{-2}

Subscripts

avg	average
cond	condensate
corr	correlation
eq	equivalent
exp	experimental
F	frictional term
G	gravitational term
g	gas/vapour
i	inlet
l	liquid
M	momentum term
o	outlet
ph	preheater
r	refrigerant
sat	saturation
T	total
TS	test section
w	water
wi	inner wall

INTRODUCTION

The taking cognisance of the need to increase the thermal performance of heat exchangers, thereby effecting savings of energy, material and cost, as well as a consequential mitigation of environmental degradation, has led to the development and use of many heat transfer enhancement techniques. In general, enhancement techniques can be divided into two groups: active and passive. Active techniques require external forces, e.g., electrical field, acoustic or surface vibration. Passive techniques require special surface geometries, such as rough surface, extended surface for liquids etc., or fluid additives. Both techniques have been used by researchers for 140 years to increase heat transfer rates in heat exchangers. If two or more of these techniques are utilised together to achieve enhancement, this is referred to as compound enhancement.

The usage of micro-fin tubes has increased the heat transfer performance of tubes with relatively low pressure drop increases in commercial and air conditioning applications since the 1980s. Micro-fins improve heat transfer in both single-phase and two-phase applications, and are one of the most efficient and common heat transfer enhancement mechanisms for heat exchangers due to their superior heat transfer performance.

Many experimental investigations have been performed to determine the effects of fin geometry, tube diameter, refrigerant, etc., on the condensation heat transfer and pressure drop performance of micro-fin tubes. The presence of the micro-fins inside the tube enhances the heat transfer by providing an increased surface area. They cause not only uniform liquid film distribution around the circumference of the tube, but also turbulence induced in the liquid film.

Many studies on condensation have been done with horizontal micro-fin tubes. Wang and Honda (2003), Cavallini et al. (1999), Chamra et al. (1996), and Schlager (1990) have proposed correlations and theoretical models to predict the heat transfer, and they have made intensive comparisons of previously proposed correlations with a large body of experimental data at the same time. Helically grooved, 18° helix angled, horizontal micro-fin tubes have been used in air conditioners recently, because of their better heat transfer performance compared to smooth tubes.

However, there are few studies on the condensation of R134a with down flow in vertical micro-fin tubes. Briggs et al. (1998) have used large diameter tubes around 20.8 mm with CFC113. The Shah (1979) correlation has commonly been used by researchers for turbulent condensation conditions.

The flow pattern occurring in many real convective condensation applications is that of an annular flow along a tube length, which is characterised by a phase interface separating a thin liquid film from the gas flow in the core region. Researchers, in relation to both analytical and experimental work, have paid attention to this flow regime due to its practical importance.

The first theoretical solution for predicting heat transfer coefficients was proposed by Nusselt (1916). A linear temperature profile through a laminar film flowing downwards without entrainment on a vertical plate was assumed, waves and an interfacial shear effect between the phases were neglected in his analysis. Nusselt-type analysis can be used for convective condensation in round tubes under these conditions. Detailed information on the studies in the literature regarding modifications of the Nusselt theory can be seen in the authors' previous publications (Dalkilic et al., 2009a, 2009b).

Carey (1992) developed a theoretical model to investigate convective condensation in round tubes during an annular flow regime. He modified Nusselt's theory (1916) by taking into account the interfacial shear stress and new simplified equations. An iterative technique for the calculation of the interfacial shear and the determination of the local heat transfer coefficients were proposed in this study. Any alteration of the thermo-physical properties of the refrigerant for condensation was also neglected due to the small pressure drop along the test tube. But it is reported that the analysis in his study cannot be applicable to fully or partially turbulent film flow.

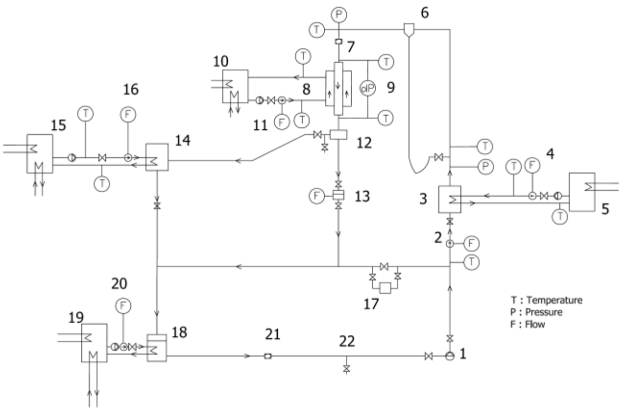
Dalkilic et al. (2009a) used Carey's (1992) theoretical model for the downward condensation of R134a to investigate the local and average heat transfer coefficients in a vertical smooth tube at low mass flux conditions. The calculated results obtained from the modified Nusselt model incorporating interfacial shear stress, the modified Nusselt model with the McAdams correction factor (1954), and the classical Nusselt model (1916), were compared with the experimental data. Comparisons with the data for laminar flow at low mass flux show that the modified Nusselt model without a correction factor predicts the data well. Experimental results show that the interfacial shear stress that was incorporated into the modified Nusselt model (1916) affects the condensation process of R134a in a vertical smooth tube.

The most common passive heat transfer enhancement technique nowadays for condensers is the use of helical micro-fin tubes. In the present study, tests are performed in smooth and micro-fin tubes with the same dimensions and conditions for the purpose of comparison. The helical micro-fin tube, which was produced for commercial usage, used as test tube has a 0.5 m length, 7.94 mm outside diameter, 18° helix angle and 50 fins. To the best of the authors' knowledge, there has been insufficient work dealing with condensation heat transfer of HFC-134a in small diameter micro-fin tubes during downward flow. Although some information is currently available, there still remains room for further research. Moreover, it should also be noted that the reported mass fluxes, heat fluxes, condensation pressures and dimensions of the test tube do not include the parameters presently studied (except for the authors' previous publications (Balcilar et al. (2010a, 2010b), Dalkilic et al. (2007, 2008, 2009, 2010)). The aim of the present study is to determine the heat transfer enhancement, comparing smooth and micro-fin tubes, and investigate the alteration of the local heat transfer coefficients, film thicknesses, and condensation rates along the test tubes. In addition to this, the effect of different experimental parameters such as condensing temperature difference between saturation and wall inlet temperature of the test tube, condensation pressure on the convective heat transfer coefficient of R134a and condensation rate are also shown and then discussed. Moreover, new correlations for the condensation heat transfer coefficient are proposed for practical applications for smooth and micro-fin tubes separately.

EXPERIMENTAL SETUP

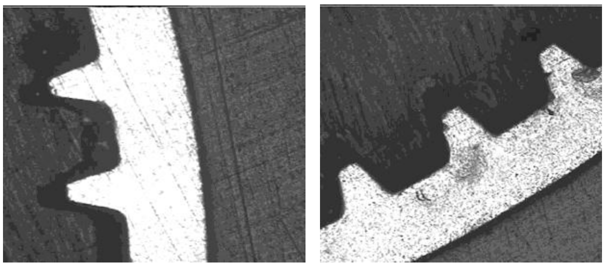
A schematic diagram of the test apparatus is shown in Figure 1. The refrigerant loop consists of an evaporator, test section and condenser loop. The refrigerant is circulated by a gear pump controlled by an inverter. The refrigerant flows in series through the bypass line, a coriolis type refrigerant flow meter which has a sensitivity of 0.1%, an evaporator, a separator, and a sight glass tube, and then enters the test section. The evaporator controls the inlet quality before entering the test section, and consists of a plate heat exchanger designed to supply heat to adjust the inlet quality for the vaporisation of the refrigerant. The circulated water flow rate of the evaporator is measured by a turbine-type flow meter which has a sensitivity of 2%. After exiting the test section, the vapour phase of R134a, which comes from the liquid-gas separator, continues to the condenser. The flow rate of liquid R134a from the liquid-gas separator is measured in a vessel to check the condition of the apparatus. A plate heat exchanger is used as a condenser. The liquid phase of R134a, from the condenser and separator, is collected in a reserve tank which has a water coil to balance the pressure of liquid R134a. There is another sight glass to check the saturated liquid R134a before the refrigerant pump. The pressures are measured by pressure transducers which have sensitivities of 0.5%.

The test section is a vertical counter-flow tube-in-tube heat exchanger with refrigerant flowing in the inner tube and cooling water flowing in the annulus. The inner and outer tubes are made from copper having inner diameters of 7 mm and 16 mm, respectively. The length of the heat exchanger is 0.5 m. A thermostat is used to control the inlet temperature of the water. The flow rate of cooling water is measured using a turbine-type flow meter which has a sensitivity of 1%. The pressure drop is measured by a differential pressure transducer, which has a sensitivity of 0.05%, installed between the inlet and outlet of the test section. The temperatures of the inlet and outlet of the test section are measured by pt100 sensors and T-type thermocouples. A band-type heater is wrapped around the copper tube line from the exit of the evaporator to the inlet of the test tube to control the system pressure of the refrigerant flow.

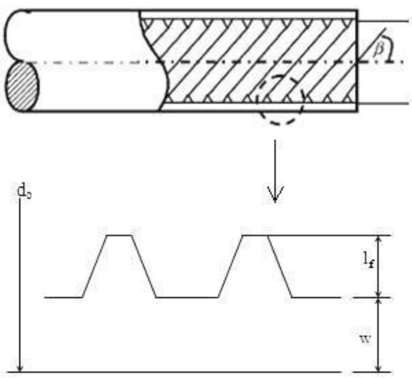


1- Refrigerant Pump 2- Coriolis Flow Meter 3- Evaporator 4- Turbine Flow Meter 5- Thermostat System 6- Liquid/Gas Separator 7- Sight Glass 8- Test Section 9- Differential Pressure Transmitter 10- Thermostat System 11- Turbine Flow Meter 12- Liquid/Gas Separator 13- Scaled Vessel 14- Condenser 15- Thermostat System 16- Rotameter 17- Filter/Dryer 18- R134a Reserve Tank 19- Thermostat System 20- Rotameter 21- Sight Glass 22- R134a Charging Point

Figure 1. Schematic diagram of experimental apparatus.



(a) Cross-section of the micro-fin tube by electron microscope (100x) (Dalkilic, 2007)



(b) Schematic cross-section of the micro-fin tube (Dalkilic and Wongwises, 2009t)

Figure 2. Detailed cross-section of the test tube.

A Panasonic-Nais PLC device was used to record and collect data from all flow meters, pressure transducers and differential pressure transmitters. The computer programme collected 10 types of data and recorded the average values each second using an MS Excel programme spreadsheet.

Figure 2 shows detailed cross section of the tested micro-fin tube which was produced for commercial usage and its geometric parameters can be seen in Table 1.

Table 1. Geometry of the tested micro-fin tube.

Test tube	Micro-fin tube
l (mm)	500
d_o (mm)	7.94
w (mm)	0.28
l_f (mm)	0.15
ß (°)	18
n	100
A_{mf}/A_s	1.38

DATA REDUCTION

The data reduction of the measured results can be analysed as follows:

The Inlet Vapour Quality of The Test Section ($x_{TS,i}$)

$$x_{TS,i} = \frac{i_{TS,i} - i_{l@T_{TS,i}}}{i_{fg@T_{TS,i}}} \quad (1)$$

where $i_{l@T_{TS,i}}$ is the specific enthalpy of the saturated liquid based on the inlet temperature of the test section, $i_{fg@T_{TS,i}}$ is the specific enthalpy of vaporization based on the inlet temperature of the test section, $i_{TS,i}$ is the refrigerant specific enthalpy at the test section inlet and is given by:

$$i_{TS,i} = i_{ph,i} + \frac{Q_{ph}}{m_r} \quad (2)$$

where $i_{ph,i}$ is the inlet specific enthalpy of the liquid refrigerant before entering the pre-heater, m_r is the mass flow rate of the refrigerant, and Q_{ph} is the heat transfer rate in the pre-heater:

$$Q_{ph} = m_{w,ph} c_{p,w} (T_{w,i} - T_{w,o})_{ph} \quad (3)$$

where $m_{w,ph}$ is the mass flow rate of the water entering the preheater, cp,w is the specific heat of water, $(T_{w,i} - T_{w,o})_{ph}$ is the temperature difference between inlet and outlet positions of the preheater.

The Outlet Vapour Quality of The Test Section ($x_{TS,o}$)

$$x_{TS,o} = \frac{i_{TS,o} - i_{l@T_{TS,o}}}{i_{fg@T_{TS,o}}} \quad (4)$$

where $i_{TS,o}$ is the refrigerant specific enthalpy at the test section outlet, $i_{l@T_{TS,o}}$ is the specific enthalpy of the saturated liquid based on the outlet temperature of the test section, and $i_{fg@T_{TS,o}}$ is the specific enthalpy of vaporization. The outlet specific enthalpy of the refrigerant flow is calculated as follows:

$$i_{TS,o} = i_{TS,i} - \frac{Q_{TS}}{m_{ref}} \quad (5)$$

where the heat transfer rate, Q_{TS}, in the test section is obtained from:

$$Q_{TS} = m_{w,TS} c_{p,w} (T_{w,o} - T_{w,i})_{TS} \quad (6)$$

where $m_{w,TS}$ is the mass flow rate of the water entering the test section, and $(T_{w,o} - T_{w,i})_{TS}$ is temperature difference between outlet and inlet position of the test section.

The Average Heat Transfer Coefficient

$$h_{exp} = \frac{Q_{TS}}{A_{wi}(T_{r,sat} - T_{wi})} \quad (7)$$

where h_{exp} is the experimental average heat transfer coefficient, Q_{TS} is the heat transfer rate in the test section, T_{wi} is the average temperature of the inner wall, $T_{r,sat}$ is the saturation temperature of the refrigerant, and A_{wi} is the inside surface area of the test section:

$$A_{wi} = \pi d l \quad (8)$$

where d is the inside diameter of the test tube. l is the length of the test tube. It should be noted that the increase in tube area (A_{mf}/A_s) in Table 1 is considered for the surface area of the micro-fin tube during calculation process.

THE LAMINAR ANNULAR FILM CONDENSATION MODEL

Figure 3 shows the steady-state physical model of downward film condensation of R134a in a vertical tube. The assumptions were made as follows: laminar film flow; saturated state for the vapour of R134a; condensed film of R134a along the tube surface; constant physical properties corresponding to inlet pressure and temperature conditions; no entrainment. A Nusselt-type analysis is valid under these assumptions for the internal convective condensation in a round tube. The interfacial shear effect at the liquid-vapour interface is taken account due to the much greater vapour velocity than the film velocity. It should be noticed that the inertia and downstream diffusion contributions were also neglected.

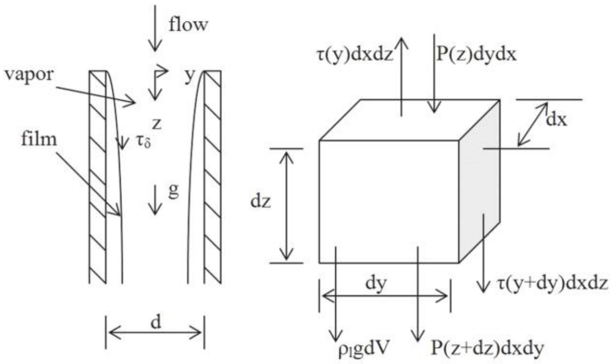


Figure 3. System model for analysis of downward condensation (Dalkilic et al., 2009a).

The differential element's force balance in the control volume is given as follows:

$$\rho_l g dx dy dz + \tau_\delta(y + dy) dx dz + P(z) dx dy = \tau_\delta(y) dx dz + P(z + dz) dx dy \quad (9)$$

The hydrostatic pressure gradient, the frictional pressure gradient and the momentum pressure gradient composes the total two-phase pressure gradient and it can be seen as:

$$\left(\frac{dP}{dz}\right)_T = \left(\frac{dP}{dz}\right)_G + \left(\frac{dP}{dz}\right)_F + \left(\frac{dP}{dz}\right)_M \quad (10)$$

where the hydrostatic pressure gradient is:

$$\left(\frac{dP}{dz}\right)_G = \rho_g g \quad (11)$$

The interfacial shear stress causes the occurrence of the frictional pressure gradient in the vapour (Carey, 1992) and it can be expressed as follows:

$$\left(\frac{dP}{dz}\right)_F = -\frac{4\tau_\delta}{(d - 2\delta)} \quad (12)$$

As a result of the one-dimensional two-phase separated-flow analysis, the momentum pressure gradient can be calculated as follows:

$$\left(\frac{dP}{dz}\right)_M = -G^2 \frac{d}{dz}\left[\frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_l (1-\alpha)}\right] \quad (13)$$

Eq. (14) is obtained from Eq. (13) under the following assumptions: liquid density of R134a is much greater than its vapour density; the variation in vapour quality is bigger than the variation in the void fraction along the test tube (Carey, 1992):

$$\left(\frac{dP}{dz}\right)_M = -\frac{2xdG^2}{\rho_g(d-2\delta)}\frac{dx}{dz} \quad (14)$$

Carey (1992) modified this by means of the usual idealisation on pressure gradient According to his theory; pressure gradient has an equal value in the vapour phase and in the liquid film as expected. A fictitious vapour density is defined to facilitate analysis of the momentum transport in the liquid film, as reported below (Carey, 1992):

$$\rho_g^* g = \rho_g g - \frac{4\tau_\delta}{d-2\delta} - \frac{2xdG^2}{\rho_g(d-2\delta)}\left(\frac{dx}{dz}\right) \quad (15)$$

The velocity gradient can be obtained from Eq. (16) using interfacial shear stress as follows:

$$\frac{du}{dy} = \frac{(\delta-y)(\rho_l-\rho_g^*)g}{\mu_l} + \frac{\tau_\delta}{\mu_l} \quad (16)$$

Integration of Eq. (16) gives Eq. (17) using u=0 at y=0:

$$u = \frac{(\rho_l-\rho_g^*)g}{\mu_l}\left(y\delta - \frac{y^2}{2}\right) + \frac{\tau_\delta y}{\mu_l} \quad (17)$$

The liquid flow rate can be derived from the velocity profile as follows:

$$\dot{m} = \pi d\delta u_{avg}\rho_l = \left[\frac{(\rho_l-\rho_g^*)g\delta^2}{3\mu_l} + \frac{\tau_\delta\delta}{2\mu_l}\right]\pi d\delta\rho_l \quad (18)$$

Eq. (19) can be expressed from the overall mass and energy balance in the case of a falling film without sub-cooling (Carey, 1992):

$$k_l\left(\frac{T_{r,sat}-T_{wi}}{\delta}\right)\pi d dz = i_{fg} d\dot{m} \quad (19)$$

The energy balance in Eq. (19) yields quality gradient as (Carey, 1992):

$$\dot{q} = h_l \pi d\Delta z\Delta T_{sat} = \dot{m}_{cond} i_{fg} \quad (20)$$

$$\frac{dx}{dz} = \frac{4q''}{DGi_{fg}} = \frac{4h_l(T_{r,sat}-T_{w,i})}{DGi_{fg}} \quad (21)$$

Eq. (19) can be rearranged using Eq. (21): (δ=0, x=0)

$$\delta^4 + \frac{4}{3}\frac{\tau_\delta\delta^3}{(\rho_l-\rho_g^*)g} = \frac{4k_l\mu_l(T_{r,sat}-T_{wi})z}{\rho_l(\rho_l-\rho_g^*)gi_{fg}} \quad (22)$$

The film thickness equation, belong to Nusselt's analysis (Nusselt, 1916), can be obtained when the interfacial shear stress effects are omitted as shown in Eq. (23):

$$\delta(z) = \left[\frac{4\mu_l k_l z(T_{r,sat}-T_{wi})}{gi_{fg}\rho_l(\rho_l-\rho_g)}\right]^{1/4} \quad (23)$$

Application of the correction factor to the latent heat of vaporisation per unit mass can be seen from Eq. (24) as follows:

$$i_{fg}' = i_{fg}\left[1+\left(\frac{3}{8}\right)\frac{c_{pl}(T_{r,sat}-T_{wi})}{i_{fg}}\right] \quad (24)$$

The film heat transfer coefficient is shown in Eq. (25) assuming a linear temperature distribution in the film region as:

$$h_l(z) = \frac{k_l}{\delta(z)} \quad (25)$$

The vapour flow can be behaved as a single phase flow in the tube as an approach due to the thin film and the much greater mean velocity of vapour than the liquid velocity at the interface caused by the high viscosity of the liquid phase compared to the vapour phase. Furthermore, vapour velocity is assumed to be zero between the phases in calculations. The interfacial shear can be obtained by means of the conventional single phase correlation with these assumptions (Carey, 1992):

$$\tau_\delta = f_g\left(\frac{\rho_g u_g^2}{2}\right) = f_g\left(\frac{G^2x^2}{2\rho_g(1-4\delta/d)}\right) \quad (26)$$

The friction factor can be expressed for round smooth tubes as (Carey, 1992):

$$f_g = 0.079\left[\frac{Gx(d-\delta)}{\mu_g(1-4\delta/d)}\right]^{-0.25} \quad (27)$$

Cavallini et al. (1997b) proposed a friction factor for round micro-fin tubes including their fin parameters as follows:

$$f_g = \frac{[1.74-2.\log(2Rx_f)]^{-2}}{4} \quad (28)$$

where geometry enhancement factor of micro-fin tube in comparison to the smooth tube is expressed in Eq. (29) as (Cavallini, 1997b):

$$Rx_f = \frac{0.18\left(\frac{l_f}{d}\right)}{(0.1+\cos\beta)} \quad (29)$$

It is possible for the McAdams correction factor (1954) to be used to consider the effects of the waviness and rippling in the film regarding the increase in heat transfer. It can be used for downward laminar film condensation, and it corrects the Nusselt's equation (1916) in terms of the above explanation as follows:

$$h_{wave} = 1.2\frac{k_l}{\delta(z)} \tag{30}$$

Carey (1992) offered an iterative technique for the determination of film heat transfer coefficient and interfacial shear using specified mass flux, tube wall inlet temperature, and condensation pressure and thermo physical properties. Firstly, a value for a film thickness is guessed, the quality gradient is obtained from Eqs. (21) and (25). The interfacial shear stress value is calculated by means of Eqs. (26) and (28), and Eq. (15) gives the fictitious vapour density.

The substitution of the estimated film thickness values, calculated fictitious vapour density and interfacial shear values are performed into Eq. (22). These values are updated and the film thickness value repeatedly estimated until convergence. The level of accuracy judges the acceptable values of estimated film thickness and calculated film heat transfer coefficient. Figure 4 shows this procedure step by step in detail.

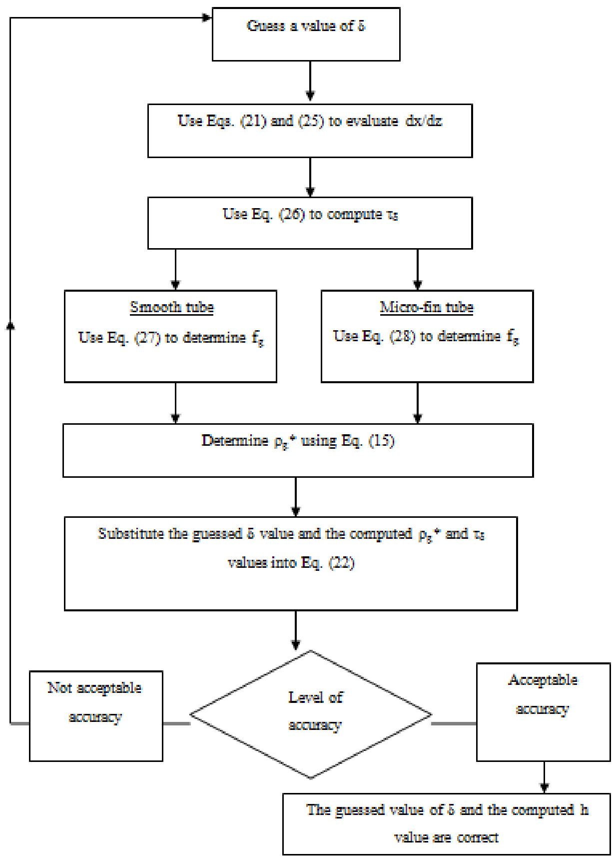


Figure 4. Flow chart of iteration process.

It should be noted that it is possible Nusselt's theory (1916) to be used for both condensation on vertical flat surfaces and condensation outside tubes and also inside the tubes if the tubes are large in diameter, compared with the film thickness. Nusselt (1916) proposed the average convective heat transfer coefficient as follows ($0<Re_l<30$):

$$h_{Nusselt} = 0.943\left[\frac{\rho_l(\rho_l-\rho_g)g i_{fg} k_l^3}{\mu_l l(T_{r,sat}-T_{wi})}\right]^{1/4} \tag{31}$$

RESULTS AND DISCUSSION

The experiments of smooth and micro-fin tubes were done using R134a in tubes at the mass flux of 29 kg $m^{-2}s^{-1}$ and pressures between 0.8-0.9 MPa. The modified Nusselt theory including the interfacial shear effect was used in the calculations of all local heat transfer coefficients. Figures 5-8 show the effect of the temperature difference between the saturated temperature of R134a and inner wall temperature of the micro-fin tube ($\Delta T_{r,sat}$) and the condensation pressure on the local heat transfer coefficients, film thickness and condensation rates along the micro-fin tube's length.

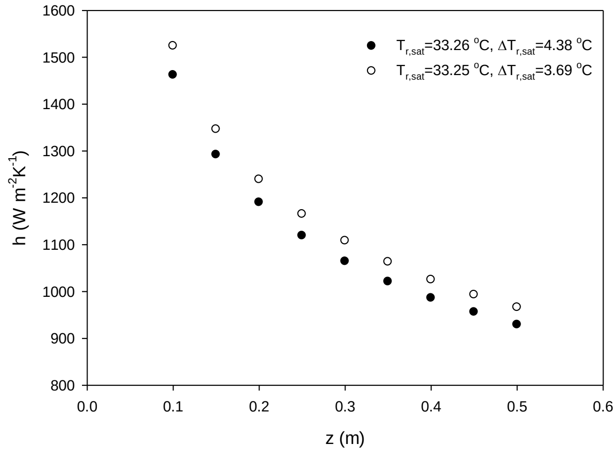


Figure 5. Comparison of local heat transfer coefficients of micro-fin tube at different $\Delta T_{r,sat}$ for the mass flux of 29 kg $m^{-2}s^{-1}$ and a constant condensation temperature.

The decrease in the local condensation heat transfer coefficient and condensation rate of micro-fin tube along the tube length, calculated by the proposed model, can be seen from Figures 5 and 6. It is because of this that the film thickness and hence total condensation rate increase from the top to the bottom of the test tube through the direction of gravity. There is high local condensation rate along the tube length at the tube entrance caused by high vapour velocity and high interfacial shear. It decreases along the tube length with decreasing vapour velocity due to an increase in condensation rate. Oh and Revankar (2005) have the similar characteristics of trend lines in their study.

The effect of pressure, in other saying, the effect of the saturation temperature on the local heat transfer coefficients and condensation rate along the tube length for the micro-fin tube, calculated by the proposed model, can be seen from Figures 7 and 8. Low condensation pressures, in other saying, high temperature differences and change of physical properties at low pressure induce higher local heat transfer coefficients than those of high condensation pressure. These results and the general trend are found to be compatible with the Nusselt (1916) theory in which the average and local heat transfer coefficients are proportional to $\Delta T_{r,sat}^{-0.25}$ and $z^{-0.25}$.

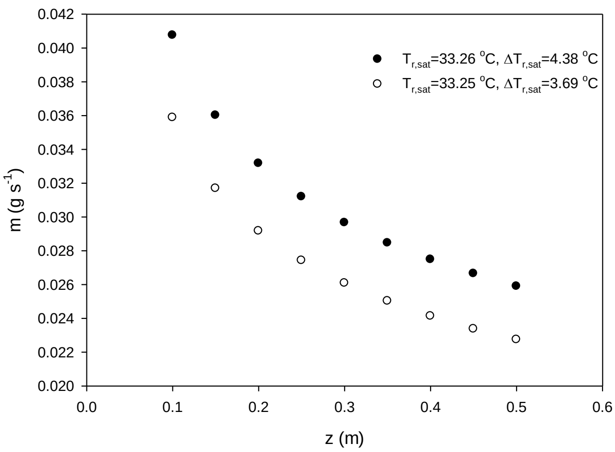


Figure 6. Comparison of local condensation rates of micro-fin tube at different $\Delta T_{r,sat}$ for the mass flux of 29 kg m^{-2}s^{-1} and a constant condensation temperature.

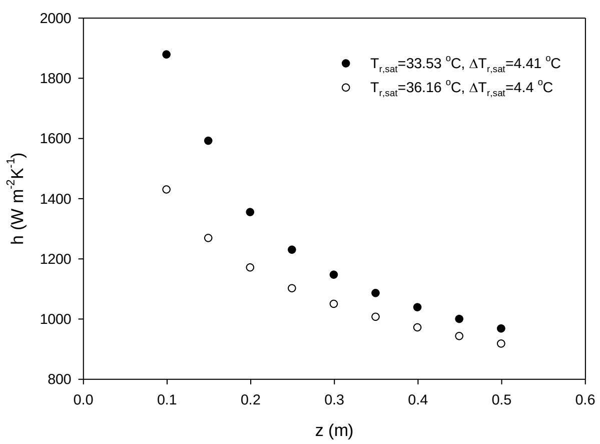


Figure 7. Comparison of local heat transfer coefficients of micro-fin tube at different condensation temperatures (33.53 and 36.16 °C) for the mass flux of 29 kg m^{-2}s^{-1} and $\Delta T_{r,sat}$=4.4 °C.

The data shown in all figures and tables were collected in an annular flow regime in Hewitt and Robertson's (1969) flow pattern map, and also checked by sight glasses at the inlet and outlet of the test section.

The determination of the film heat transfer coefficient for the laminar flow at low mass flux for the smooth and micro-fin tubes was performed by means of the modified Nusselt theory (Carey, 1992), the McAdams correlation (1954) and the classical Nusselt theory (1916).

The modified Nusselt theory in Eq. (22) including the interfacial shear stress effect is found to be the most suitable model for the experimental data from the micro-fin tube according to the analysis of Figure 9. Besides this, it can be also understood from Figure 9 that the investigated experimental conditions do not require the usage of McAdams correction factor (1954) in Eq. (30) to take account the wave effect at the vapour-liquid interface occurred by low mass flux. In addition to this, the classical Nusselt theory in Eq. (31) overestimates the heat transfer coefficients considering the deviation of 30%.

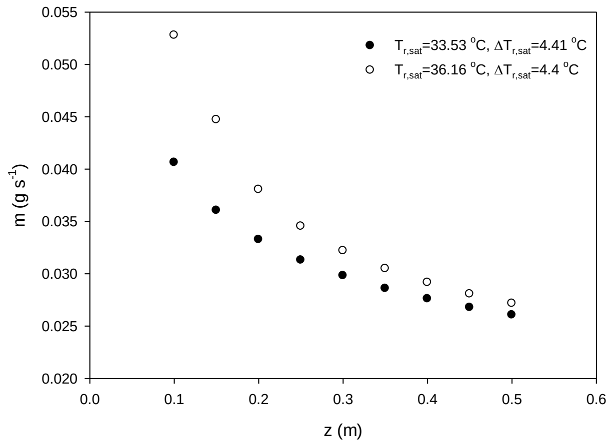


Figure 8. Comparison of local condensation rates of micro-fin tube at different condensation temperatures (33.53 and 36.16 °C) for the mass flux of 29 kg m^{-2}s^{-1}and $\Delta T_{r,sat}$=4.4 °C.

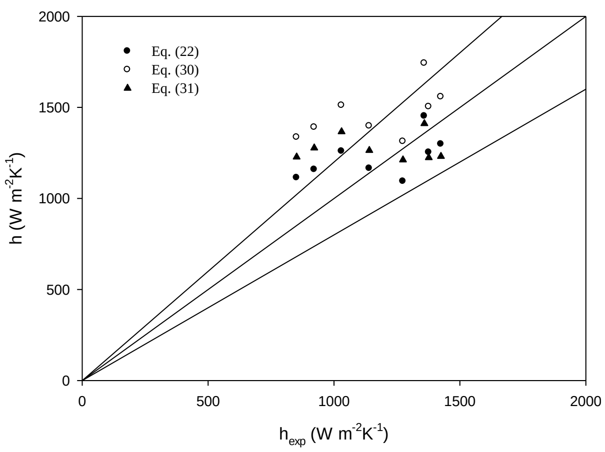


Figure 9. Comparisons between the experimental and calculated condensation heat transfer coefficients for the micro-fin tube under the experimental conditions: G=29 kg m^{-2}s^{-1}, P= 0.8-0.9 MPa, $T_{r,sat}$= 31.73-35.28 ºC, $\Delta T_{r,sat}$= 1.67-4.4 °C, x_{avg} = 0.82-0.99.

The enhancement on the condensation heat transfer coefficients using smooth and micro-fin tubes at the approximately similar experimental conditions, shown in Table 2, such as mass flux, condensing pressure, vapour quality, $\Delta T_{r,sat}$ can be seen from Figure 10. According to the result of analysis, the heat transfer enhancement of 60-82% is gained by means of the usage of micro-fin in comparison to the smooth tube.

Figure 11 shows the effect of the condensation pressure on the heat transfer coefficients. According to Figure 11, the average experimental condensation heat transfer coefficients obtained from Eq. (7) decrease with increasing system pressure as expected.

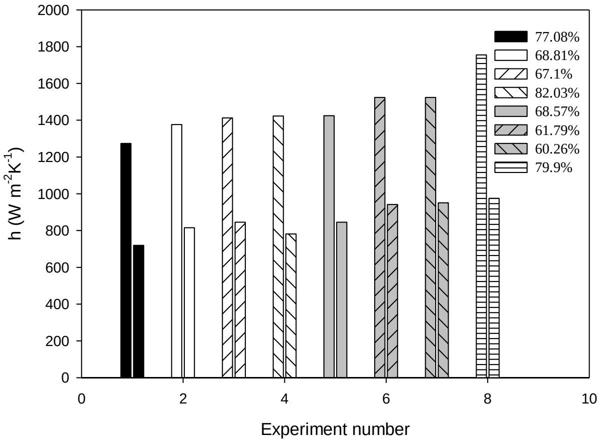


Figure 10. Comparisons between the experimental condensation heat transfer coefficients for the smooth and micro-fin tubes under the experimental conditions:
G=29 kg $m^{-2}s^{-1}$, P= 0.8-0.9 MPa, $T_{r,sat}$= 31.08-36.16 ºC, $\Delta T_{r,sat}$= 2.71-4.86 °C, x_{avg} = 0.8-0.99.

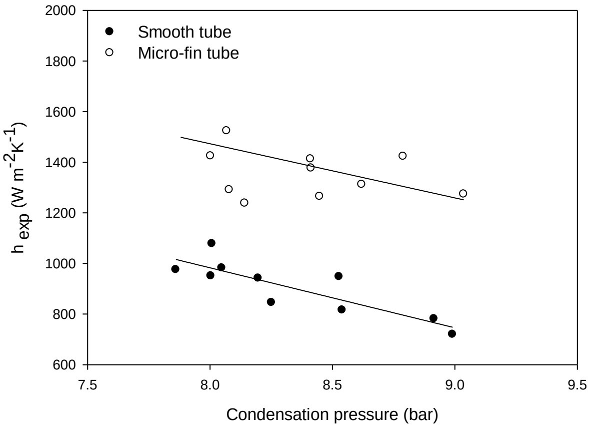


Figure 11. Comparison of average experimental convective heat transfer coefficients of the smooth and micro-fin tubes for the mass flux of 29 kg $m^{-2}s^{-1}$ at pressures of 0.77-0.90 MPa.

Bellinghausen and Renz's (1992) method is used to present correlations for the smooth (Dalkilic, 2009a) and micro-fin tubes separately. Although their correlation's accuracy seems poor with their data (Bellinghausen and Renz, 1992) it can be used in Nusselt-type analysis for the conditions of laminar film with stagnant vapour and a wave-free interface in a smooth tube. Therefore, the low mass flux data (29 kg $m^{-2}s^{-1}$) were used. It should be noted that these correlations simplify the calculations for practical applications. The comparison of the results from the present correlation with the experimental data is shown in Figure 12. The majority of the data fall within ±25% of the proposed correlation.

Table 2. Numerical values used in Figure 10.

Exp. number	Tube type	$T_{r,sat}$ (°C)	G ($kg\ m^{-2}s^{-1}$)	x	$\Delta T_{r,sat}$ (°C)	h ($W\ m^{-2}K^{-1}$)	Augmentation rate (h, %)
1	Smooth	35.982	28.434	0.936	3.989	719.413	77.08
1	Micro-fin	36.169	28.664	0.938	4.405	1273.982	
2	Smooth	34.084	28.65	0.99	3.609	815.385	68.81
2	Micro-fin	33.537	28.432	0.8	4.419	1376.521	
3	Smooth	32.836	28.73	0.88	3.69	845.127	67.1
3	Micro-fin	33.527	28.478	0.88	4.56	1412.718	
4	Smooth	35.669	29.74	0.85	4.861	781.32	82.03
4	Micro-fin	35.141	28.72	0.84	4.676	1422.693	
5	Smooth	32.83	28.73	0.88	3.699	845.127	68.57
5	Micro-fin	31.723	29.75	0.82	4.457	1424.652	
6	Smooth	32.59	28.903	0.938	3.09	941.773	61.79
6	Micro-fin	32.019	28.57	0.925	2.904	1523.75	
7	Smooth	31.73	28.79	0.97	2.76	950.782	60.26
7	Micro-fin	32.019	28.57	0.925	2.904	1523.75	
8	Smooth	31.082	28.972	0.864	3.04	975.225	79.9
8	Micro-fin	31.174	28.81	0.98	2.719	1755.282	

The correlations are presented as:

For the smooth tube (Dalkilic et al. (2009a):
$$\delta^*=7.4.Rel^{\,0.018} \quad (32)$$

For the micro-fin tube:
$$\delta^*=6.19.Rel^{\,0.018} \quad (33)$$

$$h_l=(k_l/L).(1/\delta^*) \quad (34)$$

Liquid Reynolds number:
$$Re_l=\dot{m}/(\pi.d.\mu_l) \quad (35)$$

Liquid Nusselt number:
$$Nu_l=1/\delta^* \quad (36)$$

Dimensionless film thickness:
$$\delta^*=\delta/L \quad (37)$$

Characteristic length:
$$L=(\delta_l^2/g)^{1/3} \quad (38)$$

Nusselt number:
$$Nu_l=h_l.L/k_l \quad (39)$$

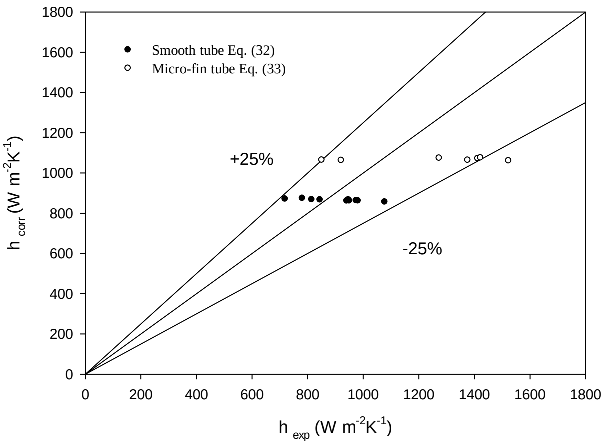


Figure 12. Comparison of the experimental convective heat transfer coefficient with the present heat transfer correlations for the smooth tube and micro-fin tubes.

CONCLUSION

Generally, horizontal micro-fin tubes are used to enhance convective heat transfer during in-tube condensation. According to the review of literature, only a few experimental works (Dalkilic and Wongwises, 2009) have been done for the heat transfer and flow characteristics inside enhanced vertical tubes. There isn't any previous work with the content and parameters in the literature apart from the present paper. Therefore, the present study has a large importance for the development of new types of compact heat exchangers in all industrial fields.

The investigation of the convective heat transfer coefficient of R134a during condensation in vertical downward flow at a low mass flux in smooth and micro-fin tubes was presented in the paper. It should be noted that research on the various parameters used in the present study is still limited. The results from this study are expected to fill the gap in the literature due to the insufficient works regarding condensation inside vertical micro-fin tubes in the literature. The accurate and repeatable heat transfer data for the condensation of R134a in a downward flow at a low mass flux inside smooth and micro-fin tubes were obtained. Detailed investigation and discussion were performed on the effects of various relevant parameters such as condensing temperature, condensation temperature difference, vapour quality and mass flux on the heat transfer.

The investigation of the local and average heat transfer coefficients in vertical smooth and micro-fin tubes at low mass flux conditions were done by means of the theoretical model from Carey (1992). Cavallini et al. (1997b)'s friction factor was added to the model for the micro-fin tube's calculations. The experimental data was compared with the calculated results obtained from the modified Nusselt model incorporating the interfacial shear stress, along with the modified Nusselt model with McAdams correction factor and also with the classical Nusselt model.

As an expected result, the condensation rate reaches the highest value at the pipe entrance where the highest local heat transfer coefficients exist. The film thickness is highest at the end of the tube where the local heat transfer coefficients are lowest. This result also shows the experiments' accuracy and validity.

According to the comparisons with laminar flow at low mass flux data, the modified Nusselt model without a correction factor predicts the data well. As a result of the analysis, the interfacial shear stress that was incorporated into the modified Nusselt model affects the condensation process of R134a in a vertical smooth and micro-fin tube. The classical Nusselt theory and the McAdams heat transfer coefficient cannot predict the data well.

Convective condensation heat transfer coefficients of micro-fin tube are found to be 1.6-1.82 times higher than the smooth tube at the similar experimental conditions.

New correlations for the determination of condensation heat transfer coefficients are proposed belong to the tested tubes separately for practical applications, and as a result of this study, it is possible to simply calculate the condensation heat transfer coefficient with an interfacial shear effect during downward laminar flow in vertical smooth and micro-fin tubes at low mass flux conditions.

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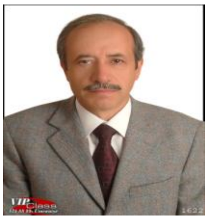
Ahmet Selim DALKILIÇ is currently a Research Assistant of Mechanical Engineering at Yildiz Technical University, İstanbul, Turkey. He received his Ph.D. Degree in Mechanical Engineering from the same university. His current research interest is on enhanced heat transfer, convection heat transfer, condensation, evaporation, boiling heat transfer of new refrigerants and mixture refrigerants and applications in heat exchangers. Dr. Ahmet Selim Dalkılıç is the member of Innovation Explorer for Scientific Researchers Community sponsored by Elsevier and American Society of Mechanical Engineers (ASME). He has been serving as a volunteer associate editor for Scientific Journals International (SJI) - Journal of Mechanical, Aerospace and Industrial Engineering.



İsmail TEKE is currently a professor of mechanical engineering at Yıldız Technical Üniversity,İstanbul, Turkey. He received his PhD degree in mechanical engineering from the same üniversity in 1981. His current research interest is on two phase flow and heat exchangers.

Somchai Wongwises is currently a professor of mechanical engineering at King Mongkut's University of Technology Thonburi, Bangmod, Thailand. He received his Doktor-Ingenieur (Dr.-Ing.) in mechanical engineering from the University of Hannover, Germany, in 1994. His research interests include two phase flow, heat transfer enhancement, and thermal system design. He is the head of the Fluid Mechanics, Thermal Engineering and Multiphase Flow Research Labarotory (FUTURE). He has been serving as an editor for Experimental Thermal and Fluid Science.