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DETERMINATION OF THE HEAT TRANSFER COEFFICIENT DURING ANNULAR FLOW CONDENSATION IN SMOOTH HORIZONTAL TUBES

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Abstract: In this study, a model describing the relationship between pressure drop and heat transfer coefficients was developed to calculate the heat transfer coefficient under two-phase annular flow conditions. According to these conditions, the single phase liquid flow with temperature distribution which is equivalent to velocity and temperature distributions in the liquid film was determined and the heat transfer coefficient during condensation was calculated from equations provided for the single-phase liquid. The results of the new analytical model were compared with the heat transfer coefficient obtained from experimental study. In this study, the refrigerant (R600a) was circulated inside a smooth horizontal tube with an inner diameter 4 mm and the experimental results were found to be consistent with the mentioned correlations within a range of ±20%. Consequently, to validate the model, the heat transfer coefficient data in the literature were compared with the heat transfer coefficient values of the analytical model study, under the same conditions. A deviation of ±25% was found as a result of the comparison.
Keywords: Hydrocarbon, Pure refrigerant, Condensation, Heat transfer coefficient.

HALKA AKIŞ ŞARTLARINDA PÜRÜZSÜZ YATAY BORUDA YOĞUŞMADA TAŞINIM KATSAYISININ BELİRLENMESİ

Özet: Bu çalışmada, iki fazlı akışlarda halka akış şartlarında taşınım katsayısını hesaplamak için basınç düşümü ve taşınım katsayısı arasında bir model geliştirilmiştir. Halka akış şartlarında, sıvı filmindeki hız ve sıcaklık dağılımlarına eşdeğer tek faz sıvı akışı belirlenerek yoğuşmadaki ısı taşınım katsayısı tek faz sıvı için verilen bağıntılardan yararlanarak hesaplandı. Geliştirilen modelden elde edilen sonuçlar deneysel çalışmadan elde edilen taşınım katsayısı değerleri ile karşılaştırıldı. Deneysel çalışmada R600a (İsobütan) soğutkanının iç çapı 4 mm olan yatay düz boru içerisinde yoğuşması incelendi ve elde edilen taşınım katsayısını değerlerinin mevcut korelasyonlarla ±20% uyumlu olduğu saptandı. Modelin doğruluğunu belirlemek için aynı şartlarda literatürde mevcut taşınım katsayıları ile analitik modelden hesaplanan taşınım katsayıları birbirleri ile mukayese edildi sonuçta ±% 25 lik bir sapma saptandı.
Anahtar Kelimeler: Hidrokarbon, Saf soğutkan, Yoğuşma, Isı taşınım katsayısı.

NOMENCLATURE

d	diameter, [m]
G	mass flux rate, [kgm^{-2}s^{-1}]
h	convective heat transfer coefficient, [Wm^{-2}K^{-1}]
J^*_G	dimensionless gas velocity, $J^*_G = \frac{Gx}{\sqrt{Dg\rho_v(\rho_l-\rho_v)}}$
k	thermal conductivity of the refrigerant,[Wm^{-1} K]
$\dot{m}$	refrigerant mass flow rate, [kgs^{-1}]
dP/dz	pressure gradient, [Pa m^{-1}]
Pr_t	Turbulent Prandtl number, [-]
Re	Reynolds number, [-]
T	temperature, [ºC or K]
x	refrigerant vapor quality, [-]
X	Martinelli parameter, $X = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}$
y	distance from the tube wall, [m]
Greek symbols	
δ	film thickness, [m]
μ	dynamic viscosity, [kgm^{-1}s^{-1}]
ν	kinematic viscosity, [m^{2}s^{-1}]
ρ	density, [kgm^{-3}]
τ	shear stress, [Nm^{-2}]
λ	friction factor, [-]
Subscripts	
i	inside of refrigerant tube
l	liquid
s	saturation
ts	test section
w	tube wall
Superscripts	
+	non-dimensional symbol

INTRODUCTION

Forced convection condensation inside horizontal tubes is a frequently seen and important phenomenon in many refrigeration and air-conditioning systems, power plants and process industries. Heat transfer analysis in such applications is extremely important in the design of condensers. In the design process, for a given set of flow conditions, the designer must predict the two phase heat transfer coefficient. It is desirable to predict the local values inside the tube because the film coefficient is known to vary along the length of the condenser due to changes in the flow patterns. Pure theoretical treatments of the two phase flow during condensation are very difficult to enforce.

The first theoretical calculation procedure for predicting heat transfer coefficients in condensation was suggested by Nusselt (1916). He assumed a laminar film flowing down over a vertical plate, without waves. This scenario also assumed that a linear temperature profile was present throughout the film.

The classical models for turbulent film flow were developed by Rohsenow et.al. (1956) and Dukler (1960). Rohsenow et.al. (1956) investigated turbulent film condensation assuming the universal velocity profile in the condensate film and constant turbulent prandtl number: Pr_t=1. A linear distribution of shear stress between the wall and film surface was used. The local heat transfer coefficients were obtained from the energy equations. Dukler (1960) assumed that Pr_t=1 but, unlike the method of Rohsenow et.al. (1956), turbulent diffusivity for the momentum of the Deissler model in the near wall region and Von Karman's in the region far from the wall were used, respectively. Condensation heat transfer was obtained by solving conservation equations for the condensate film.

Carpenter and Colbourn (1951) started the use of shear-based correlations for annular flow condensation. They argued that the resistance to heat transfer in the turbulent liquid flow was inside the laminar sublayer and they estimated its thickness. Traviss et. al. (1972) used the Von-Karman's equations of turbulent flow in pipes to represent the velocity distribution in the liquid film and presented a correlation treating the local Nusselt number as a function of various system parameters for different ranges of Reynolds numbers. Chisholm (1980) developed equations for the local film heat transfer coefficients of turbulent film from an approximate boundary layer treatment and compared the results with experimental data using other well-known predictive methods.

Chitti and Anand (1994) presented a model based on annular flow and proposed a method to predict the local heat transfer coefficient for forced convective condensation inside smooth horizontal tubes. They used the Prandtl mixing length theory, Van-Driest hypothesis and Reynolds analogy for the analysis. Hulburt and Newell (1996) developed condensation heat transfer coefficients and pressure drop models during annular flow for 3<d<10 mm using an empirical interfacial shear relationship developed by Asali et.al. and data from Sacks and Dobson to validate their model.

Akers et.al. (1960) presented a new model in a horizontal tube for annular flow which was known as the "equivalent Reynolds number". This number was substituted in a single phase heat transfer equation, a function of the liquid Prandtl and Reynolds numbers, to predict the two phase condensation Nusselt numbers. Moser et.al. (1998) presented an equivalent Reynolds number model that was based on the model of Akers et al. for the prediction of the condensation heat transfer coefficient. They indicated that a single phase heat transfer equation requires a correction factor, because the driving temperature difference for a single phase and an annular two phase flow were different. Cavalini et.al. (2001) recommended the usage of the Kosky and Staub model, which is related with the heat transfer coefficient and the frictional pressure gradient through the interfacial shear stress.

The above discussion shows that there are various models concerned with the solution of the two-phase flow phenomena from single phase-flow models for the prediction of condensation heat transfer coefficients in tubes. The main objective of the current study was to develop a semi empirical model for predicting heat transfer coefficients during condensation of pure refrigerants flowing inside a smooth tube. The theoretical model, which calculates the convection heat transfer coefficient has been developed by means of the single phase equations that are equivalent with the velocity and temperature distribution of the annular liquid film in two phase flow conditions. The validity of the new semi empirical model is evaluated with available experimental data.

ANALYTICAL MODEL

The following assumptions are made for the calculation of in-tube condensation heat transfer coefficients for a turbulent annular film flow and the physical model of the problem is shown in Figure 1.

The assumptions:

- Annular flow configuration
- The thickness of the liquid film is uniform around the entire circumference of the tube
- Equivalent single-phase flow is assumed to be turbulent and fully developed.
- Surface temperature is constant around the entire length of the tube.
- Refrigerant properties are constant.
- Liquid entrainment in the vapor core region is neglected.
- The vapor is at the saturation temperature.

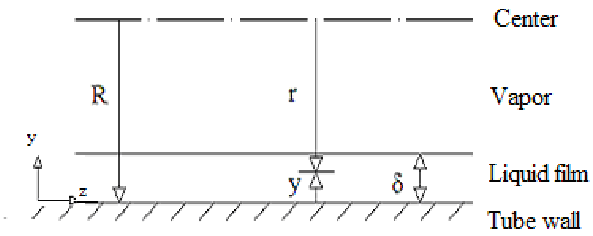


Figure 1. Section situation for the annular flow condensation.

Liquid annular film of thickness δ passes through the considered tube with R diameter. When pressure on the section is assumed to be constant, the pressure drop occurring in steam during the flow inside the tube will appear to be exactly the same for the liquid. First the velocity of the single-phase liquid is equivalent to the pressure drop in the two-phase flow allowing for the Reynolds number to be determined. This method determines whether the single-phase liquid is laminar or turbulent.

The pressure gradient for the fully-developed flow is

$$\frac{dP}{dz} = \lambda \frac{1}{d} \rho \frac{u^2}{2} \tag{1}$$

When the flow is turbulent, the friction coefficient (2003) during single-phase flow inside smooth tubes is,

$$\lambda = \frac{0.316}{Re^{0.25}} \qquad 3.10^3 < Re < 2.10^4 \tag{2}$$

$$\lambda = \frac{0.184}{Re^{0.2}} \qquad 2.10^4 < Re < 2.10^6 \tag{3}$$

When the friction coefficient is expressed in velocity as shown in Eq. (4)

$$\lambda = \frac{0.316}{\left(\frac{u_m d}{\nu}\right)^{0.25}} \tag{4}$$

and substituted in Eq. (1), the velocity value is found as a function of the pressure drop.

$$u_m = \left(\frac{dP}{dz} \frac{2d^{1.25}}{0.316 \nu^{0.25} \rho}\right)^{\left(1/1.75\right)} \tag{5}$$

When the pressure drop is known, the velocity is calculated using Eq. (5). With the velocity value (u_m), the Reynolds number is calculated to determine the flow regime.

$$Re = \frac{u_m d}{\nu} \tag{6}$$

The calculated Reynolds number determines whether the single phase flow is laminar or turbulent. After establishing the flow regime, the pressure drop during fully-developed flow inside the tube with a length of L and diameter of d is calculated using the following equation for shear stress:

$$\tau_w = \frac{d}{4} \frac{dP}{dz} \tag{7}$$

The velocity distribution should be known to calculate the thickness of the liquid film in turbulent flow.

Von Karman (Çengel, 2007) assumed the velocity profile of the liquid layer in single phase flow and used the law of the wall to describe the liquid layer. This velocity profile was comprised of three sections; laminar sub-layer, buffer and turbulent layer. Flow characteristics are quite different in various regions, and therefore, it is difficult to obtain an analytical correlation for the velocity profile during the entire flow. Thus, experimental results from various studies were considered to develop the following correlations for the universal velocity distribution:

$$0 < y^+ < 5 \quad \text{Laminar bottom layer} \quad u^+ = y^+ \tag{8}$$

$$5 \le y^+ < 30 \quad \text{Buffer zone} \quad u^+ = 5.0 \ln y^+ - 3.05 \tag{9}$$

$$30 \le y^+ \quad \text{Turbulent zone} \quad u^+ = 2.5 \ln y^+ + 5.05 \tag{10}$$

Dimensionless velocity distribution during turbulent flow is expressed as

$$u^+ = \frac{u}{u^*}, \qquad u^* = \sqrt{\frac{\tau_w}{\rho}} \tag{11}$$

The dimensionless wall distance is expressed as

$$y^+ = y \frac{u^*}{\nu_l} \tag{12}$$

The rate of liquid flowing in the tube surface is calculated by the following equations for annular flow.

$$\dot{m}_l = \dot{m}_{total} (1 - x) \tag{13}$$

$$\dot{m}_l = \int_0^\delta 2\pi\rho u r dr \tag{14}$$

Because the mass flux and vapor quality values during the flow are known, the condensed liquid amount is determined using Eq.(13). The liquid film thickness y flowing on the tube surface is found by cross equalization of Eq.(13) and (14). The variation by layers is expressed in Eq.(16).

$$\dot{m}_l = \dot{m}_{laminar} + \dot{m}_{buffer} + \dot{m}_{turbulent} \tag{15}$$

$$\dot{m}_l = \left(-2\pi\rho \frac{\nu_s^2}{u^*}\right) \left\{ \int_0^5 y^{+2} dy^+ + \int_5^{30} (5.0 \ln y^+ - 3.05) y^+ dy^+ + \int_{30}^{\delta^+} (5.05 + 2.5 \ln y^+) y^+ dy^+ \right\} \tag{16}$$

Although the thickness of the liquid film is found by Eq.(16), the value of y^+ that would meet the value of the condensed amount is found with the iteration method. The combined value of y^+ value the $\dot{m}_l$ value in Eq. (16) is searched for in the expression for the laminar bottom layer. If this value is unattainable, the buffer zone is considered and the solution is iterated. Where these two layers do not suffice, the value meeting the expression in Eq. (16) is found to determine the thickness of the liquid film.

Next, temperature distribution inside the film should be determined. Depending on the actual zone where the liquid film is involved, the dimensionless temperature distribution expressions developed by Martinelli (Kakac, 1980) are employed. The dimensionless temperature distribution expressions developed by Martinelli have been defined in y^+ dimensionless parameters separately for three layers to be used in identifying the velocity profile. Dimensionless temperature distributions by zones are expressed as follows, assuming $\zeta = \frac{\varepsilon_h}{\varepsilon_m} = 1$:

For the laminar zone $0 < y^+ < 5$

$$\frac{T_w - T}{T_w - T_c} = \frac{\zeta \Pr \frac{y}{y_1}}{\zeta \Pr + \ln(1 + 5\zeta \Pr) + 0.5 \ln \frac{Re}{60}\sqrt{\frac{\lambda}{2}}} \tag{17}$$

For the buffer zone $5 \le y^+ < 30$

$$\frac{T_w - T}{T_w - T_c} = \frac{\zeta \Pr + \ln[1 + \zeta \Pr(\frac{y}{y_1} - 1)]}{\zeta \Pr + \ln(1 + 5\zeta \Pr) + 0.5 \ln \frac{Re}{60}\sqrt{\frac{\lambda}{2}}} \tag{18}$$

where T_c is the tube centerline temperature and the y_1 value is the value of the y at $y^+=5$.

For the turbulent zone $y^+>30$

$$\frac{T_w - T}{T_w - T_c} = \frac{\zeta \Pr + \ln(1 + 5\zeta \Pr) + 0.5 \ln \frac{Re}{60}\sqrt{\frac{\lambda}{2}}\frac{y}{R}}{\zeta \Pr + \ln(1 + 5\zeta \Pr) + 0.5 \ln \frac{Re}{60}\sqrt{\frac{\lambda}{2}}} \tag{19}$$

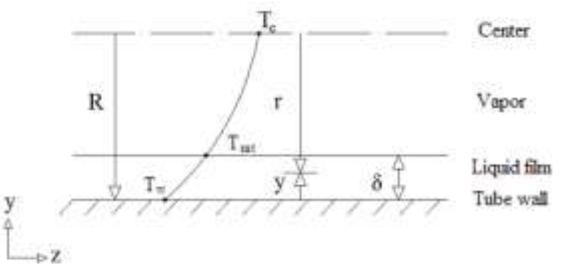


Figure 2. Temperature profile during annular flow and single-phase inside the horizontal tube.

Figure 2 shows the sectional view of the flow inside the horizontal tube under annular flow conditions. By replacing T_s with $T_{y=\delta}$, the temperature value in the expression representing the temperature distribution in a single-phase liquid, and the temperature distribution and axis temperature (T_c) in the equivalent single-phase liquid flow are determined.

After the axis temperature based on the thickness of the liquid film is calculated with these equations, the derivative of the expression yielding the temperature distribution in the laminar bottom layer by y is taken to identify the heat flux passing on the surface. Once the temperature change of $\frac{dT}{dy}$ is found, the heat flux passing on the surface is obtained using Fourier's law :

$$\dot{q} = -k \left.\frac{dT}{dy}\right|_{y=0} \tag{20}$$

The amount of heat passing by the surface is calculated with Newton's law of cooling. In the experimental study, the (T_s-T_w) differential was measured directly with a gauge developed on the data collection unit as illustrated in Figure 9. Detailed information about the measurement method is given in the part where the experimental study is explained.

$$\dot{q} = h(T_{sat} - T_w) \tag{21}$$

Combining Eq.(20) and (21), the local heat transfer coefficient is determined using the following equation:

$$h = \frac{-k \left.\frac{dT}{dy}\right|_{y=0}}{(T_{sat} - T_w)} \tag{22}$$

The calculation procedure of the proposed model is shown in Figure 3.

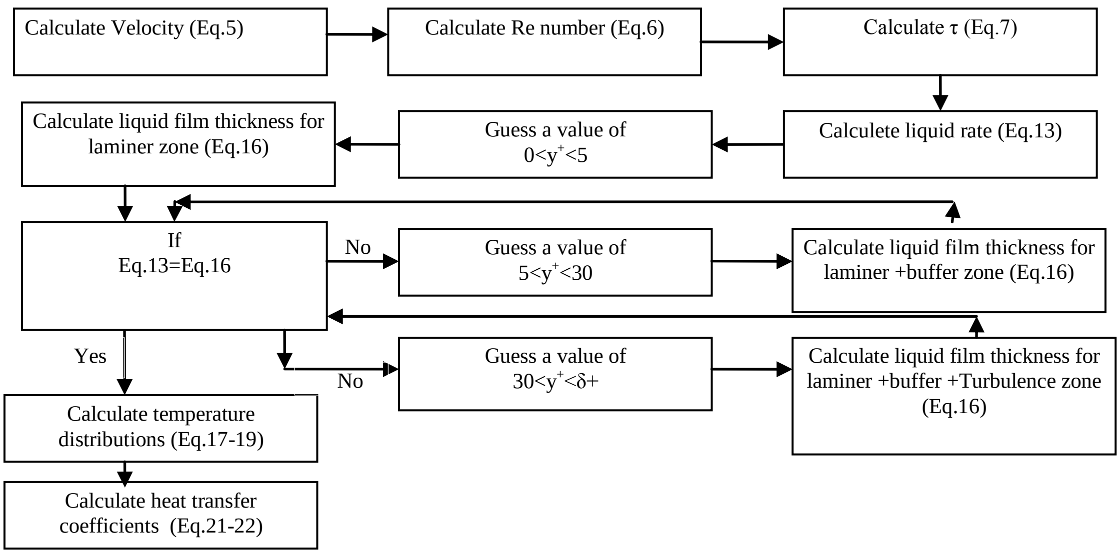


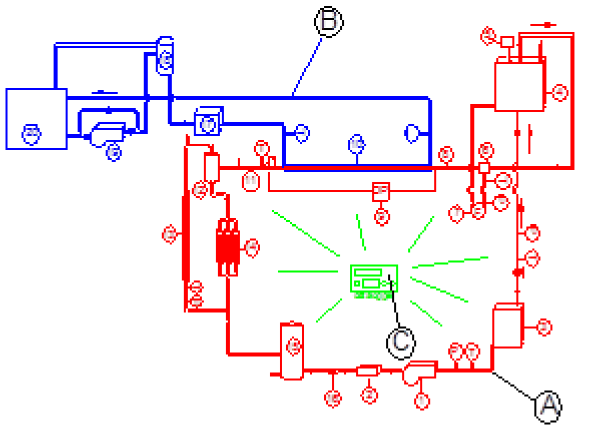
Figure 3. Flow chart of iteration proces.

EXPERIMENTAL APPARATUS AND METHOD

To validate the calculation modeling, an experiment was performed for the condensation heat transfer of R600a in horizontal smooth tubes. The experimental apparatus and the data reduction method are described the following section.

Experimental apparatus

The experimental apparatus prepared for this study, which is illustrated Figures 4 (a) and (b) is comprised of three major units and 20 components, including the refrigerant cycle, cooling water cycle and data collection unit (2008).



(a)

1- Refrigerant pump 2- Filter 3- Heater 4-Evaporator 5- Level control 6- Mixing chamber
7- Coriolis effect flow meter 8- Sight glass 9- Differential pressure sensor 10- Test section
11- Sight glass 12- Liquid-vapor separator 13- Condenser 14- Liquid collectors 15-Receiver 16- Sight glass 17- Water flow meter 18- Water tank 19- Water Pump 20-Water reservoir
A-Refrigerant Loop, **B**-Water Loop, **C**-Data acquisition system

(b)

Figure 4. (a) Schematic diagram of experimental apparatus, **(b)** Photographs of the experimental apparatus.

The refrigerant was transferred to the test area by an ex-proof Lewa Ecodos diaphragm pump that had a preciously adjustable stroke and flow rate. To identify the flow rate of the refrigerant, a Coriolis-type flow meter with a measuring range of 0-250 kg/h was employed. The flow meter measures with a precision of ±10%. To ensure that the refrigerant pumped to the test tube had the desired vapor quality at the inlet of the test tube, an evaporator was installed in the system.

It was necessary to obtain different vapor qualities in the experimental study. When the steam leaving the evaporator passes through the specially-designed mixing chamber, it amalgamates with the liquid surface at a determined rate to form an annular flow without any droplets. The liquid-steam mixture in annular form passes through the specially-designed 10-cm sight glass that has the same diameter as the test tube.

The test area was comprised of two concentric circular tubes. The inner tube is made of copper and the outer one has a Plexiglas construction. To determine the pressure drop occurring at the inlet and outlet of the test area, a Smar differential pressure transmitter calibrated at a measuring range of 0-500 mmwg with a precision of ±0.075% was installed in the apparatus. A T-type thermocouple with a diameter of 0,3 mm was installed on the test tube to measure the surface temperature.

In the experimental study, the (T_s-T_w) differential was measured directly with a gauge developed on the data collection unit as illustrated in Figure 5. Thus, to minimize the temperature related measuring errors, the T_s and T_w temperatures were not measured based on another T_{ref}, but rather the surface temperature was directly measured based on the reference temperature T_s. Thus, the (T_s-T_w) temperature differential was obtained in a single step.

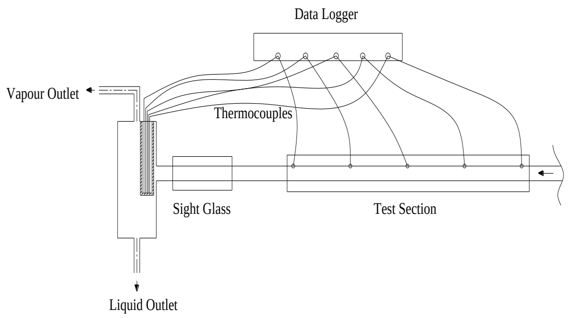


Figure 5. Schematic illustration of the (T_s-T_w) differential.

The refrigerant leaving the test area reaches a liquid/vapor separator in which the liquid and vapor content of the refrigerant is separated. The refrigerant flows to the pressure-resistant graduated cups to determine the quantity of the condensate. However, the vapor is driven to the air-cooled condenser installed in the apparatus where it condenses and is then transferred to the receiver in liquid form. To bring the cooling water received from a container to the desired temperature, it is first heated and then accumulated under constant temperature in a reservoir installed in the apparatus.

To prevent the water that will be transferred to the test area at constant temperature from getting impacted by vibrations from the pump, it is accumulated under another reservoir fitted close to the test area. From this reservoir, the cooling water flows to the test area under its own static pressure. After cooling water leaves the reservoir and passes through a turbine-type Honsberg flow meter with a capacity of 2-10 l/h to reach the test area. The temperature of the cooling water is measured by means of RTDs fitted on the inlet and outlet of the test area. The flow of the cooling water and the refrigerant in the test area was designed in a counter-flow pattern to ensure an almost constant (T_s-T_w) temperature differential along the tube.

In the cooling cycle during which the refrigerant was circulating, pressures were measured by means of Bourdon Haenni-type pressure transmitters with a measuring range of 0-10 bars and precision of ±0.2%. Temperatures were measured using RTDs. Physical and thermodynamic characteristics of the R600a refrigerant were retrieved from the REFPROP (version 7.0) software.

RESULT AND DISCUSSION

The objective of the study was to suggest an alternative calculation procedure to determine the heat transfer coefficients for annular flow condensation inside horizontal tubes. The experimental study was performed in a smooth tube with an inner diameter of 4 mm for a saturation temperature range of 30-43ºC and approximate mass flux range of 47- 116 kg/m^{2}s. It was determined both visually and by Breber et.al. (1980) flow maps that the flow regime where the experimental study was carried out was annular.

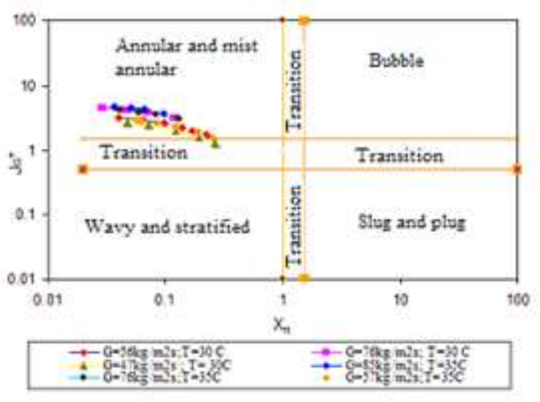


Figure 6. Variation of the flow regimes for of R600a at different mass flux rates (Breber et.al.,1980).

Figure 6 shows the flow regime transitions for six test plotted using a flow pattern map suggested by Breber et.al. (1980). All the curves are drawn for the parameter JG* as a function of the Lockhart-Martinelli parameter which depends on local quality and saturation properties of the condensing fluid. The six tests plotted indicate that most of the points at which the local h values are calculated are in the annular flow regime. Figure 7 shows that the flow was in the annular form. Therefore, the assumption of annular flow for the model development is a valid.

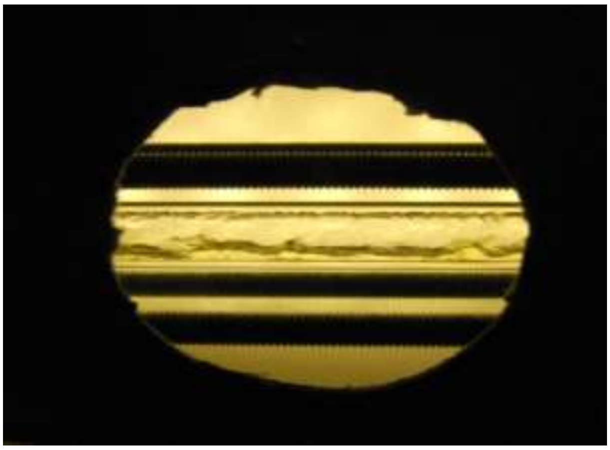


Figure 7. Photograph of the annular form through the sight glass during the experiments.

The experimental results were compared with Moser et.al. (1998) and Cavallini et.al (2001) correlations. The graphs of Figure 8 (a) and (b) show a comparison of the predicted local heat transfer coefficients and the experimental results. The heat transfer coefficient values obtained from the experimental study are consistent and have a deviation range of ±20%. On the other hand, the deviations of experimental and predicted values published in the literature are given in Table 1.

Table 1. The deviations of experimental and predicted values published in the literature.

Author	**Fluids**	**Diameter (mm)**	Deviations(%)
Hulburt and Newell (1996)	R22, R134a, R12,	3mm-10 mm	± %25
Cavallini et. al.(2001)	R22,R134a, R125, R236ea	8 mm	± %20
Moser et. al.(1998)	R22,R12,R134a, R11, R125	3,14mm< D < 20mm	± %20
Agra and Teke(2008)	R600a	4 mm	± %20

Figure 9 shows the comparison between the analytical model developed in the current study and the experimental results. The figure indicates that most of the experimental values are within a range of ±%10. This range indicates that the model is accurate. The small deviation value(±%10) can be explained by two main reasons. Firstly by using a high speed camera the flow regime determinated as annular flow certainly. Secondly, difference of Ts-Tw value was measuremed directly in one step. Therefore, the temperature related measuring errors minimized. Finally, the model that is based on a semi-theoretical approach is suitable for the prediction of heat transfer coefficients accurately for annular flow. The developed analytical model was tested with the experimental data obtained by Dobson. In his study the condensation of R134a refrigerant in a horizontal smooth tube was examined (Hulbert and Newell, 1996). The heat transfer coefficients were obtained as a result of the experiments that were carried out in a test tube with an inner diameter of 3.14 mm and length of 0.94 m for mass flux values of 300 kg/m^{2}s and 650 kg/m^{2}s. Our results were compared with those obtained under the same conditions and the values were found to be in accordance with a deviation of ±25%. Figure 10 shows the graphs for two test runs in which the local heat transfer coefficients that were obtained both experimentally and analytically, were plotted versus the quality. The local heat transfer coefficients were also plotted in each case. The predicted values, seen in the graphs, are in accordance with the experimental results. (Figure 11)

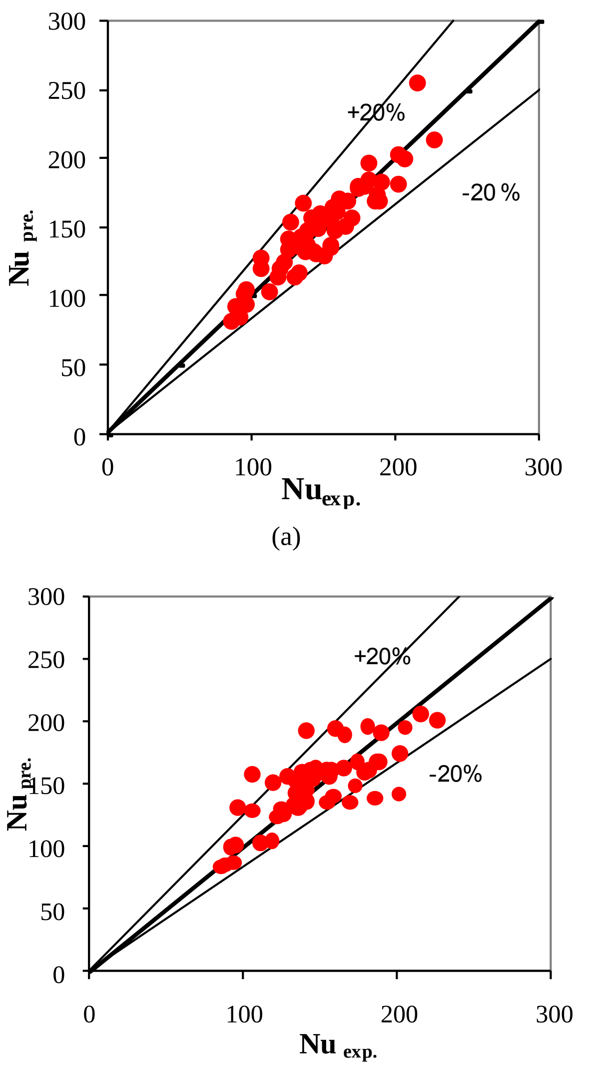


Figure 8. Comprasion experimental data with other correlations **a)** Cavallini's correlation (2001) **b)**Moser correlation (1998).

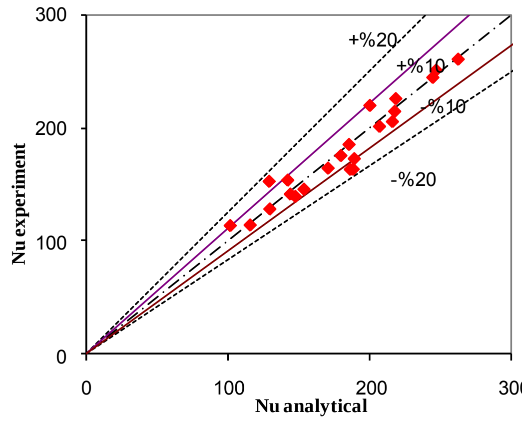


Figure 9. Comprasion of analytical model with experimental data for R600a.

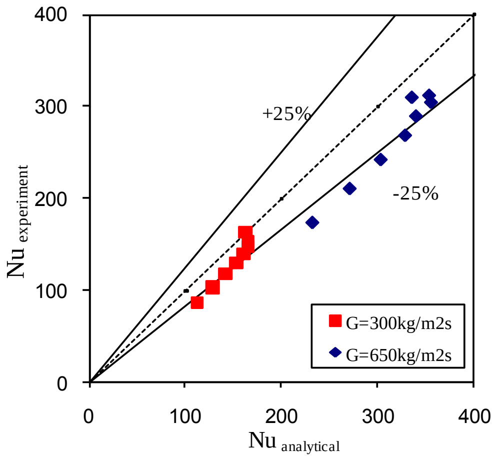


Figure 10. Comprasion of analytical model withexperimental data for R134a (Hulbert and Newell, 1996).

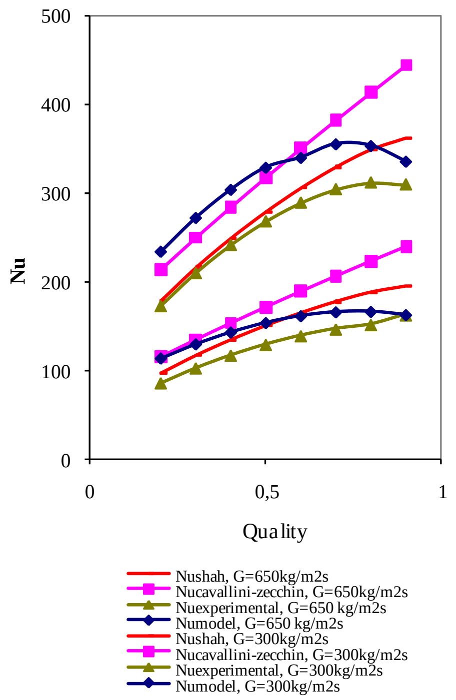


Figure 11. Variation of local heat transfer coefficient with quality for R134a.

In summary, the comparing of the proposed analytical model with the experimental results and other correlations indicate that the model can predict accurate values. Finally the model is based on a semitheoretical approach, the heat transfer coefficient values can be predicted fairly well.

CONCLUSION

An analytically and experimental study of condensation heat transfer in the smooth tube was performed and the following conclusions were obtained:

(1) A model was developed to calculate the heat transfer coefficient during condensation in an annular flow regime.

(2) The model was based upon using the pressure drop inside the tube to determine the heat transfer coefficient .

(3) The condensation of the R600a in a smooth tube with an inner diameter of 4 mm for a saturation temperature range of 30-43ºC and approximate mass flux range of 47- 116 kg/m^{2}s was experimentally studied. It was determined that the flow regime during which the experimental study was carried out was annular. The heat transfer coefficient values obtained through experiments carried out with R600a were found to be in accordance with the correlations of Moser et.al. (1198) and Cavallini et.al (2001), at a deviation of ±20%.

(4) The heat transfer coefficients calculated with the model developed for annular flow conditions were compared with those obtained experimentally. The results were at a deviation of ±20% (Figure 9).

(5) The model was compared with the experimental data obtained by Dobson (Hulbert and Newell, 1996) with the R134a refrigerant and the results were found to be in accordance with a deviation of ±25%.

(6) The model studied can be used to calculate heat transfer coefficients during condensation when pressure drops are known.

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