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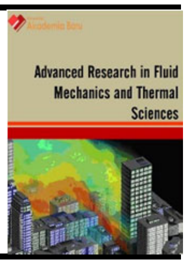

A Two-Phase Pressure Drop Model for Homogenous Separated Flow for Circular Tube Condenser, Examined with Four Modern Refrigerants

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ABSTRACT

This study presents an analytical model for the calculation of two-phase pressure drops for homogeneous separated flow for circular tube condensers. Four different refrigerants were used to examine the pressure-drop model, and some parameters were selected for a trial test, such as vapor quality, inner tube diameter, and the mass flux at a given saturated temperature. The model demonstrates strong validity compared to previous works. The saturated temperature is given at 40°C because most average condensing temperatures are equal to, or are approaching, this value. The pressure-drop model was examined with four refrigerants: R134a, R507A, R600a, and R1234xyh. These refrigerants had different critical pressures, critical temperatures, and molecular weights. Three inner tube diameters were used to examine the model at 6mm, 8mm, and 12mm, and the mass flux was100, 200, 300, 500, and 800 kg/sec. m^2, with vapour quality ranging from 0.1 to 0.9 to 0.1 steps. The results indicate a good agreement of the two-phase pressure-drop behaviour for completely variable parameters compared with previous works. Finally, these findings suggest that this model can contribute to the design of heat exchangers and/or condensation tubes for other refrigerants.

Keywords:
Two-phase pressure drop model, condenser tube, four refrigerants.

1. Introduction

Most condensation processes are encountered in refrigeration, air-conditioning, and condensation equipment. However, condensation can also be present on the inner surface of horizontal tubes. In both cases, the vapor-liquid mixture of refrigerants flows inside the tube with condensation, gradually converting the mixture into saturated liquid. The flow under this condition is a two-phase flow with heat rejected towards the ambient temperature through the wall of the tube. The frictional pressure drop occurs in a two-phase flow inside the tube due to the acceleration and gravitational effects. When the flow is fixed, at that moment, the pressure drop determines the power input of the pumping device. Consequently, the two-phase pressure drop

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inside the tube varies appreciably along the tube axis, depending on the liquid/vapor distribution within the tube. Recently, two main factors are being considered: ozone-depleting potential (ODP) and global warming potential (GWP). These are becoming the deciding criteria for the development of new refrigerants in place of CFC refrigerants due to their contribution to ozone layer depletion and global warming. The pressure drop in two-phase flow is also a major design variation because it governs the pumping power required to transport two-phase fluids and the recirculation rate in circulation systems. Thus, due to all the above reasons, a proper prediction of pressure drop is crucial for the design of a condenser tube. According to available literature, several researchers have studied two-phase flow with pressure drops inside condenser tubes and evaporate for refrigerants. There is a relationship between the pressure drop inside the tube of the condenser/evaporate and the heat transfer coefficient. Naterer presents the relationship between heat transfer coefficients with density gradient ratio [1]. Although there are several researches that deal with two-phase flow with a pressure drop of refrigerants inside tubes of condensers, Hashizume *et al.,* analyzed two-phase flow with a friction pressure drop of simplified models for annular and stratified flow [2]. They compared their results with other calculation methods of frictional pressure drop. Two different types of refrigerates were examined with steam/water and R12. The data were obtained for steam/water (pressure = 1 to 212bar, mass flux =15 to 825kg sec-1m-2, d= 3 to 152mm, and x<0.7)and forR12 (pressure=0.6-33 bar, mass flux = 100–5000kg sec-1m-2,d = 10–23 mm, and x < 0.7). Whalley [3] proposed that the Friedel correlation is the best to date for predication the two-phase pressure drop inside smooth tubes for condensing process if $\mu_l/\mu_v < 1000$. Didi *et al.,* [4] predict a two-phase pressure drop for evaporation in two horizontal tubes with a diameter of 10.92mm and 12.00mm, for five refrigerants (R-134a, R-123, R-402A, R-404A, and R-502) for mass fluxes from 100 to 500 kg sec-1m-2, and vapor quality ranging from 0.04 to 1.0. To predict the pressure drop, they used the methods of Muller-Steinhagen and Heck [5] and Gronnerud [6]. The results indicated that the best method for calculated pressure drop for the annular flow was that of Muller-Steinhagen and Heck. For intermittent flow and stratified-wavy flow, the best method in both cases was that of Gronnerud.

Pamitran *et al.,* [7] present investigations of the two-phase flow pattern transitions of pressure drop for R-22, R-134a, R-410A, R-290, and R-744 in horizontal small steel tubes of 0.5mm, 1.5mm and 3.0mm inner diameters. They obtained experimental data of the heat flux of 5 and 40kWm-2, while the mass flux ranged from 50 to 600 kg sec-1m-2, at saturation temperature, a range of 0–15^{o}C, and quality up to 1.0. They used the flow pattern map by Wang *et al.,* [8] and Wojtan *et al.,* [9] to compare the experimental data. The results indicated that the annular flow rises at lower vapor qualities for evaporation with higher heat fluxes, higher mass fluxes, and smaller inner-tube diameters. Davide *et al.,* present a model that assesses the pressure drop with heat transfer condensation [10]. Their analysis was carried out with several refrigerants, such as R1234ze (E), R32, R134a, and R1234yf. The investigation of the pressure drop effect for a tube with a hydraulic diameter of 0.96 mm, at a saturated temperature of 40^{o}C, and with mass flux, ranged from 200 to 400kg sec-1m-2 with vapor quality ranging from 0.1 to 0.9. They obtained results with a value of R134a and R1234yf; R1234ze (E) displayed the lowest reduced pressure and the highest-pressure gradient within the same working conditions.

From above review of some valuable published literature, it can conclude that the pressure drop of two-phase flow inside the tube of the condenser with different refrigerants is associated with the type of flow model, which related with the methods of pressure drop prediction, and the vapor quality. In this study, three calculation models of pressure drop were presents of Cicchitti *et al.,* [11], McAdams [12] and Dukler *et al.,* [13]. These models expressed the average kinematic viscosities. Otherwise, the homogenous separated two-phase flow model was assumed to calculate the pressure drop for different refrigerants inside the copper tube condenser.

2. The Homogenous Separated Flow Model

Regarding horizontal tubes, the gravity pressure drop component is not relevant as it is only applicable to long vertical tubes. The accelerated pressure-drop gradient develops only if there is a pressure increase at the exit, rather than the inlet. In the case of flow, which is naturally condensed at the outlet, it has less kinetic energy than that condensed at the inlet. Thus, the acceleration component of the pressure gradient is neglected. In addition, for calculations of gravity and acceleration, pressure-drop data is required. Because high vapor density on the condenser causes very high pressure, if the vapor velocity is compared at any given vapor quality (x) and mass fluxes (G), it will be always be less than that of the evaporator. Thus, a stratified flow regime will develop in the condenser, bringing the flow closer to another. Results evaluated from any common void-fraction model, in this stratified region, would be inaccurate. Thus, the only consideration of frictional pressure drop is furthermore justified.

3. Mathematical Formulation

The homogenous separated flow model consists of a special case in whichthe vapour and liquid velocities are not only considered constant but also equal. This model is known as the homogeneous separated flow model. This model converts the two-phase flow into a single-phase pseudo-fluid flow. The properties are taken as averages from both phases. Throughout the condensation film inside tubes, a variety of flow regimes can occur, depending on the length and inner diameter of the tube, its orientation, the heat and mass fluxes along the tube, and relevant fluid properties. For instance, when condensation occurs in the horizontal tube at high condensation rates, the flow passes through different regimes as the fluid takings from the inlet (as vapour quality x near 1.0) at the exit is (x < 0.0), as in Figure 1. The basic equations for horizontal tube condensation from the steady homogeneous flow model in the reduced form are given as
Continuity Equation (1)

$$\dot{m} = \rho u A$$

Momentum Equation (2)

$$\dot{m} du = -A \partial p - dF - \rho A g dz$$

Here, the wall shear force dF, or wall shear stress, τ_w which acts inside the area of the tube can be expressed as

$$dF = \tau_w (Pdz) \tag{3}$$

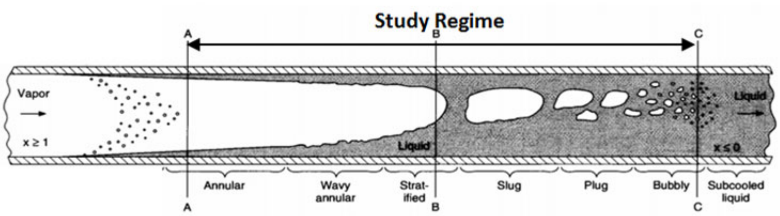


Fig. 1. The flow regime and liquid/vapor distribution inside the tube

The two-phase friction factor for the averaged properties at laminar flow,

$$f_{TP} = 0.079\left[\frac{Gd}{\overline{\mu}_{TP}}\right]^{[-0.25]} \tag{4}$$

Where G is mass flux, d is the inside tube diameter and $\overline{\mu}_{TP}$ is the averaged two-phase viscosity. Collier gives the averaged properties taken for pseudo single-phase fluid from the liquid and vapour properties [14]. The averaged fluid density is given as

$$\frac{1}{\overline{\rho}} = \left[\frac{x}{\rho_g} + \left(\frac{1-x}{\rho_l}\right)\right] \tag{5}$$

Then, for the mean two-phase viscosity of the condensation process, $\overline{\mu}$ is limiting conditions, at, **Error! Reference source not found.** and at $x \le 0$, $\overline{\mu} = \mu_l$ for three models are

McAdams model

$$\frac{1}{\overline{\mu}} = \left[\frac{x}{\mu_g} + \left(\frac{1-x}{\mu_l}\right)\right] \tag{6}$$

Cicchitti model

$$\overline{\mu} = x\mu_g + (1-x)\mu_l \tag{7}$$

and

Dukler model

$$\overline{\mu} = \overline{\rho}\left[\frac{x\mu_g}{\rho_g} + \frac{(1-x)\mu_l}{\rho_l}\right] \tag{8}$$

The average kinematic viscosity of two-phase can be obtained from equations 6, 7, and 8 as

$$\overline{\mu}_{TP} = \frac{\rho_g \rho_l}{4\left[(x\mu_g) + (1-x)\overline{\rho}\mu_l\right]} \tag{9}$$

The frictional pressure drop equation that was obtained from the above-average properties for a horizontal tube of internal diameter d is

$$\Delta p = \frac{2 f_{TP} G^2 L}{3 \bar{\rho} d} \tag{10}$$

Where, Δp pressure drop, f_{TP} two-phase friction factor, G is mass flux, L is the tube length, $\bar{\rho}$ is average density, d is the inner tube diameter.

4. Results and Discussion

Analysis to determine the pressure drop using the homogenous separated flow model was undertaken to select between the Cicchitti, McAdams, and Dukler kinematics viscosity models. The method proposed here predicts only the average kinematics viscosity components of the refrigerants, whereas the test of the three models involved. The liquid and vapor properties of R134a, R507A, R600a, and R1234xyh at a saturation temperature of 40°C were taken into consideration for performing the analysis presented by ASHRAE [15]. A trial was then conducted to verify the analytical models and bring its behavior for data using Microsoft EXCEL. Table 1 summarizes the test condition operations.

Table 1
A summary of the test condition operations

Refrigerants	- Halocarbon Refrigerants
	Ethane Series
	R134a, R1234xyh
	Azeotropic Blends
	R507A
	- Hydrocarbon Refrigerants
	R600a
Saturated Temperature	40°C
Inner-tube diameter	6, 8, and 12mm
Tube Length	0.8 m
Mass Flux (G)	100, 200, 300,550, 800 kg $sec^{-1}m^{-2}$
Vapour Quality (x) %	Ranged from 0.1 to 0.9

4.1 Average Densities and Average Kinematics Viscosities of R134a, R507A, R600a, And R1234xyh

The homogenous separated flow model considers the averaged flow properties. In this study, the condenser tube wall is taken at adiabatic conditions. Therefore, the averaged properties are density and kinematic viscosity. The averaged density is given by a single equation. These values were obtained at various vapor qualities. The obtained values of average densities are shown in Figure 2.

The figure clearly shows the similarity values of average densities between the R134a and R1234xyh at a saturated temperature of 40°C, while the behaviour for R507A and R600a are not similar to average densities. To determine the averaged kinematics viscosities, there are three linear kinematics viscosity models at a saturated temperature of 40°C for the four refrigerates R134a, R600a, R507A, and R1234xyh. Figure 3 shows the behaviour of the three models by Cicchitti, McAdams, and Dukler. The figure seems to demonstrate that the Cicchitti model has a higher value of kinematic viscosity of the four refrigerant types. McAdam’s and Dukler’s models look closely at the trend of values and behaviour.

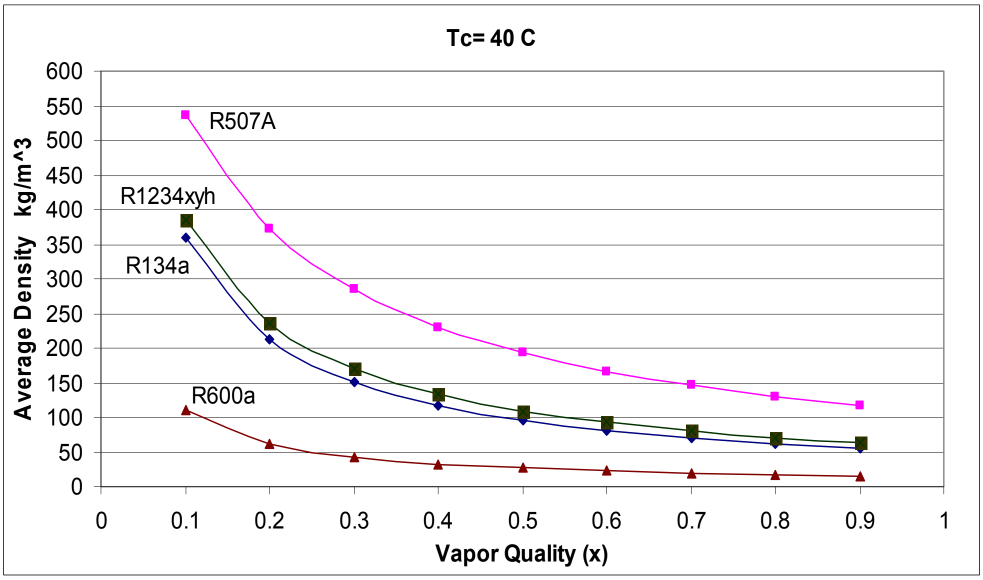
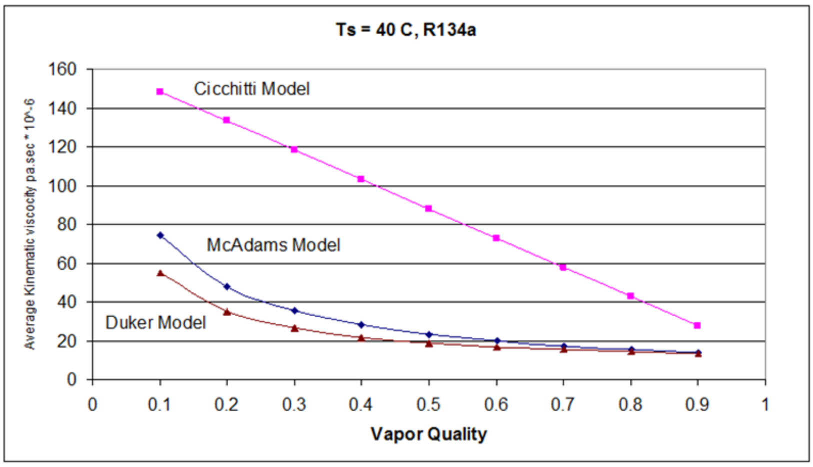


Fig. 2. Average densities of R134a, R507A, R600a, and R1234xyh at a condensing saturation temperature of 40°C



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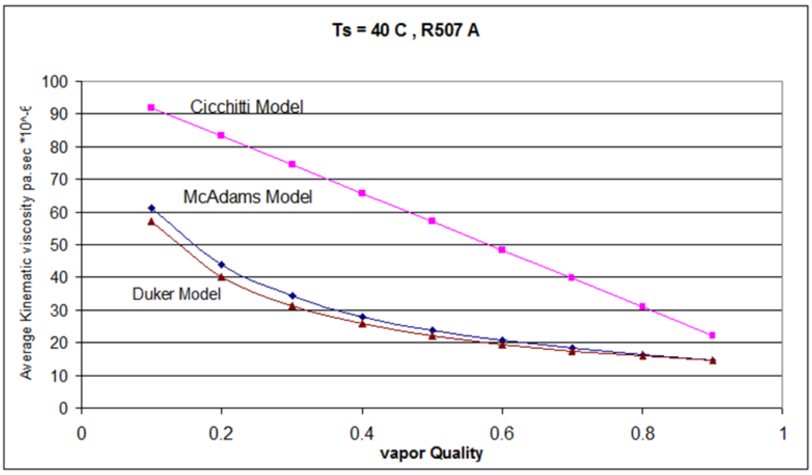

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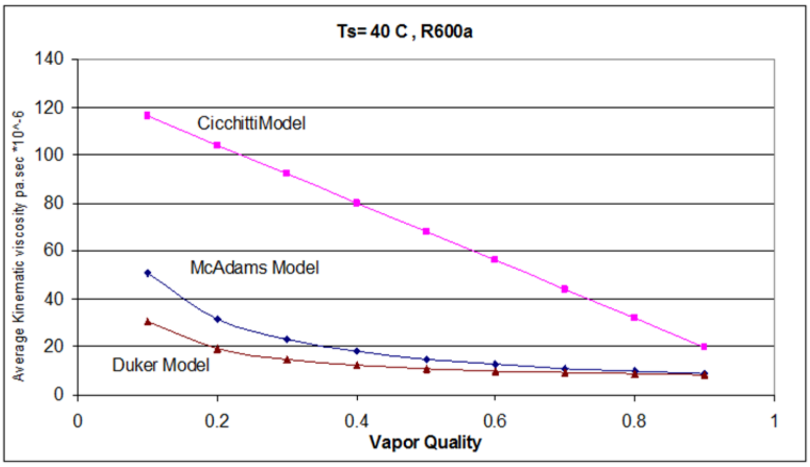

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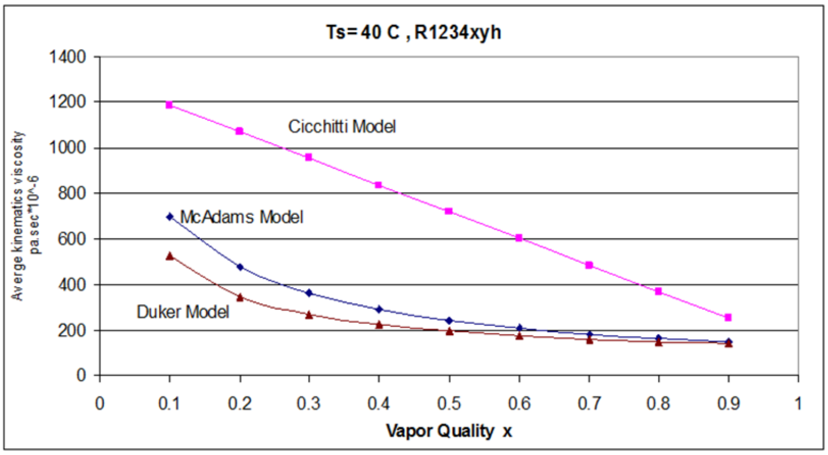

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Fig.3. Average kinematics viscosity of refrigerants (a-R134a, b-R507A, c-R600a, d-R1234xyh) at a saturated temperature of 40°C

4.2 Comparison of Pressure Drop

Pamitran *et al.,* compared the pressure drop model of the present work to the same experimental conditions of previous work [7]. The pressure drop comparisons are illustrated in Figure 4. The pressure drop was strongly affected by the physical properties of the refrigerants, such as average density and average kinematic viscosity, exhibited in Figure2 and Figure 3 respectively. Therefore, it is clear that in the present work model comparison, R134a has the lowest pressure drop.

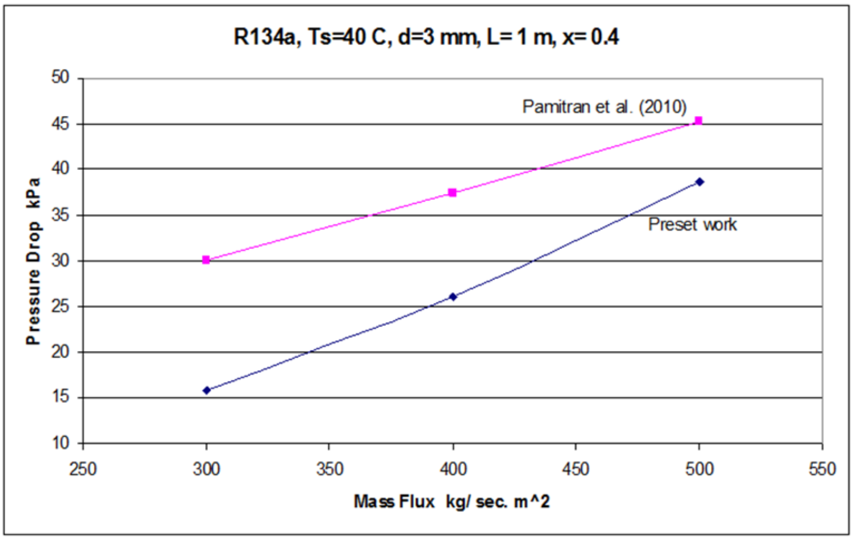
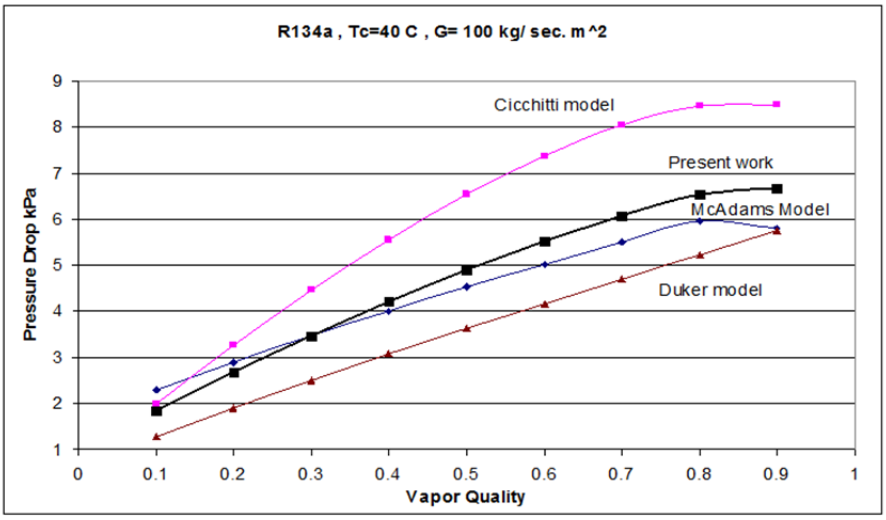


Fig.4. Comparison of pressure drops of the present work of R134a, Ts= 40°C, d= 3mm, L= 1m, and x=0.4

4.3 Present Model of Pressure Drop

Figure 5 presents the results of pressure drop for three models [Cicchitti, McAdams, and Dukler] for four refrigerants (R134a, R507A, R600a, and R1234xyh) at a saturation temperature of 40O°C, an inner tube diameter of 8mm, and L = 80cm. The Figure shows the pressure drop vs. vapour quality of the three models. As predicted, a pressure drop occurred as the two-phase pressure dropped due to the condensation of the refrigerant inside the tube, so when the vapour quality increased the pressure drop increased too. The Figures illustrate the trend of the present work regarding the relationship between the two-phase pressure drops within vapour quality and the percentage of maximum accuracy values of pressure drops predict in the present work for the three models: Cicchitti model 19%, McAdams model ±4%, and Dukler model 17%.



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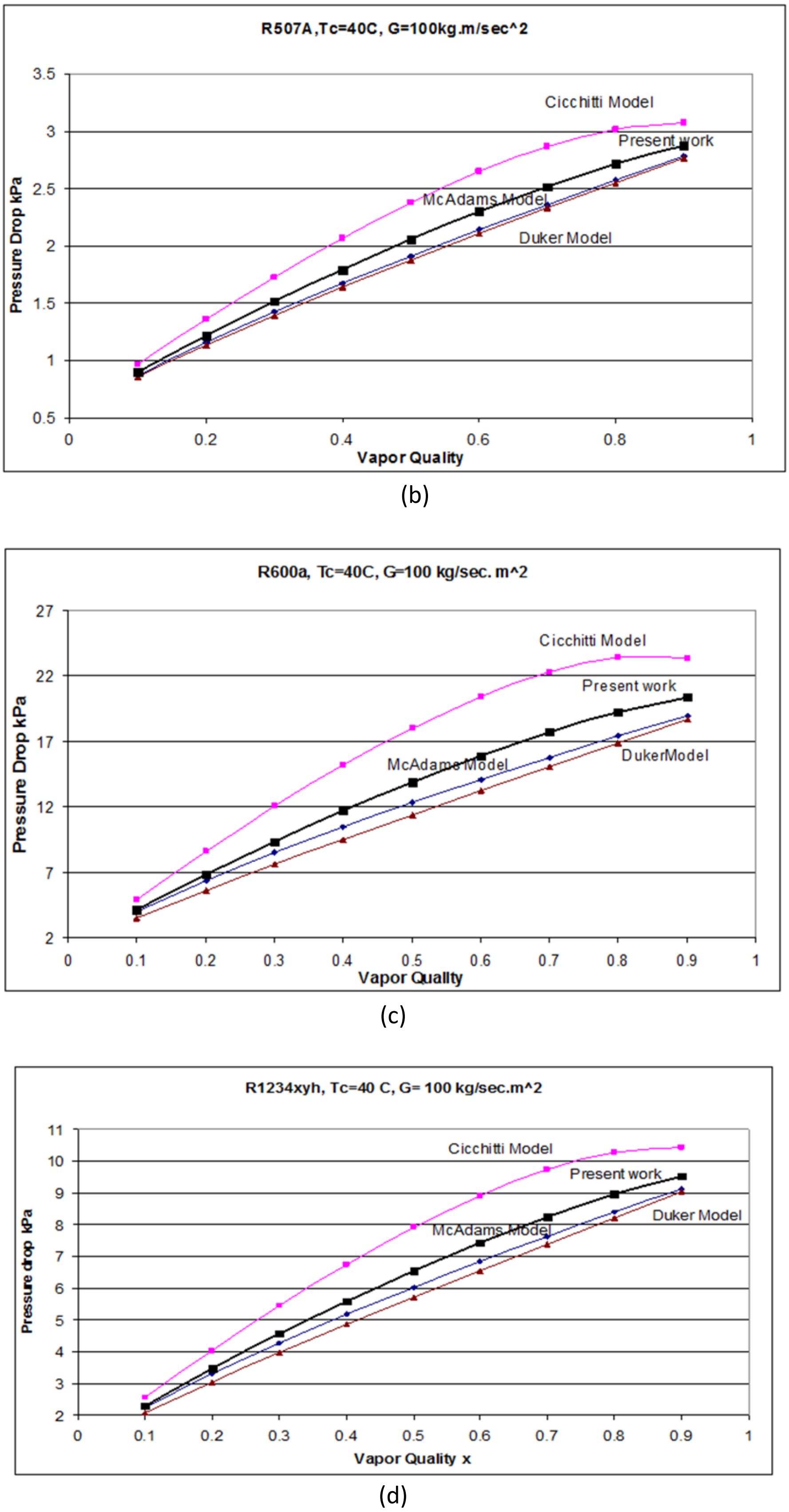
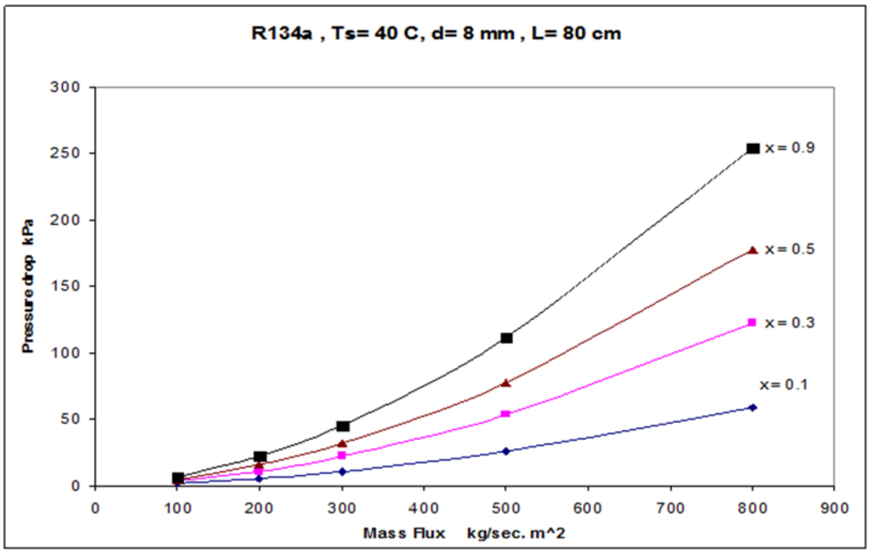


Fig. 5. Comparison of pressure drops of the present work with three models for four refrigerants at a saturated temperature of40°C, mass flux of 100kg sec^{-1}m^{-2}, inner tube diameter at 8mm, and tube length of 80cm for a-R134a, b-R507A, c-R600a, and d- R1234xyh

4.4 Effects of Mass Flux and Vapour Qualities on Pressure Drop

Figure 6 shows the effect of the mass flux of the refrigerants on the pressure drop of the present model at different vapour qualities. The performance of the predicted two-phase pressure drop model influenced by parameters such as vapour quality and mass flux was checked. The Figures illustrate the same results as those of Pamitran *et al.,* which demonstrate that the frictional pressure drop of gas flow is higher than the frictional pressure drop of liquid flow [7].



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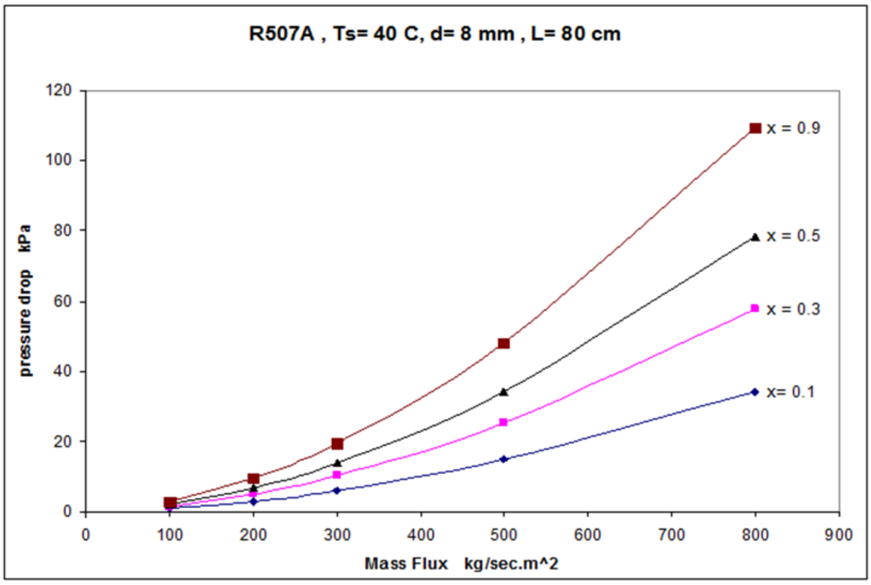

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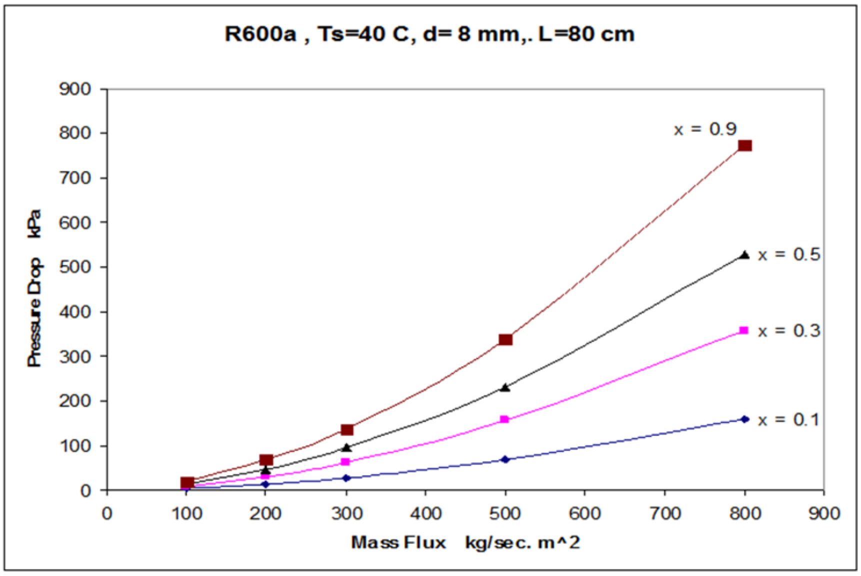

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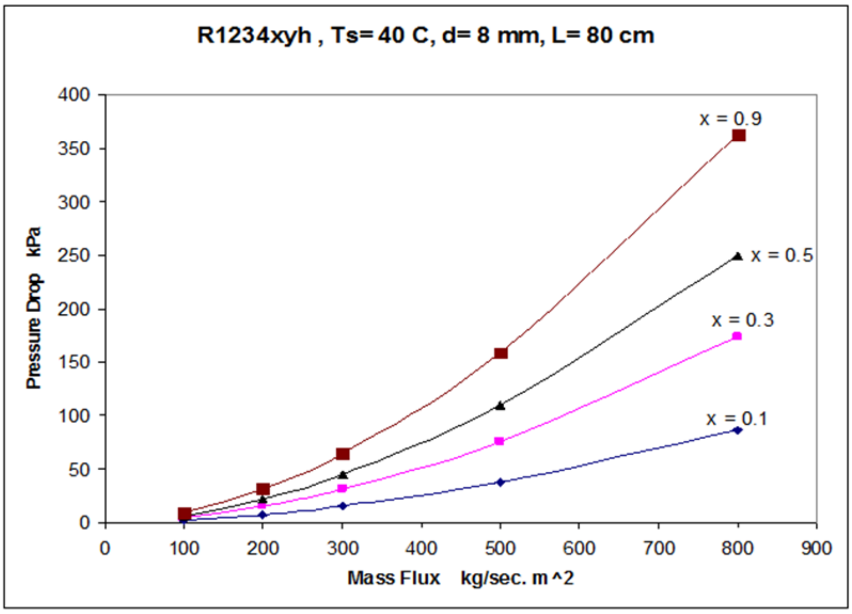

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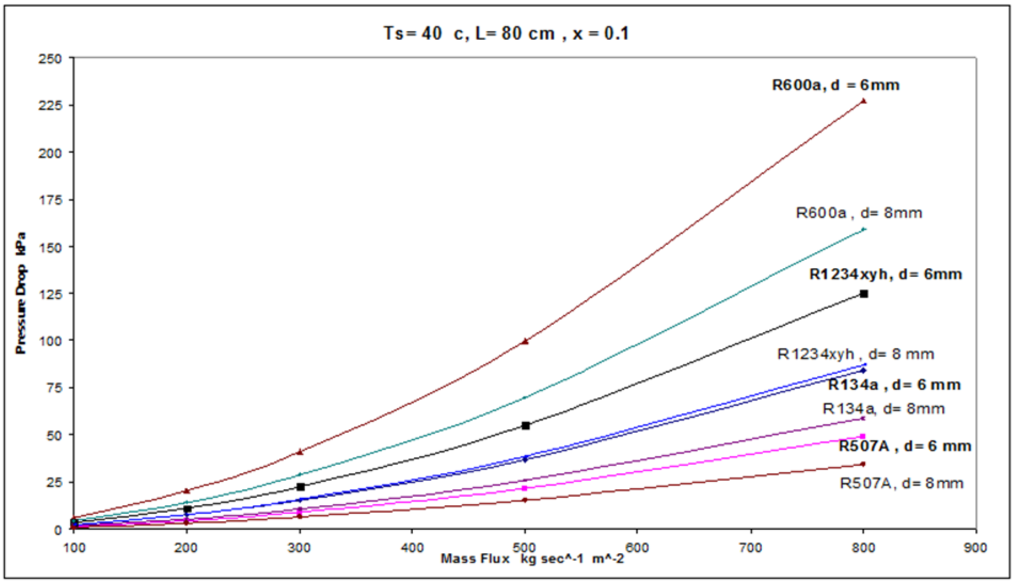
Fig.6. Effect of the mass flux of the refrigerants on the pressure drop of the present model at a different vapour quality at a saturated temperature of 40^{o}C: a-R134a, b-R507A, c- R600a, d- R1234xyh

4.5 Effects of The Inner Tube Diameters and Vapour Quality on The Pressure Drop

Figure 7 shows the effect of inner tube diameters and mass fluxes of different refrigerants on the pressure drop model. The Figures can illustrate that the effect of pressure drop in the smaller diameter tubes is higher than that of the larger diameter tubes. This is because the smaller inner tube diameter gives in a higher wall shear stress, resulting in a higher friction factor, which then causes higher frictional pressure drops. Figure 7-a shows that the presence of higher pressure drop at vapour quality (x= 0.1) for both inner tube diameter of 6mm and 8mm causes more friction pressure of the mist mixture (the mixture content of more liquid than refrigerant), while in Figure 7-b, the vapour quality (x = 0.9) for the inner tube diameter of 6mm, 8mm, and 12mm, the range of pressure drop is less than at (x= 0.1), especially for the tube diameter 12mm,because the overall mixture is vapour.



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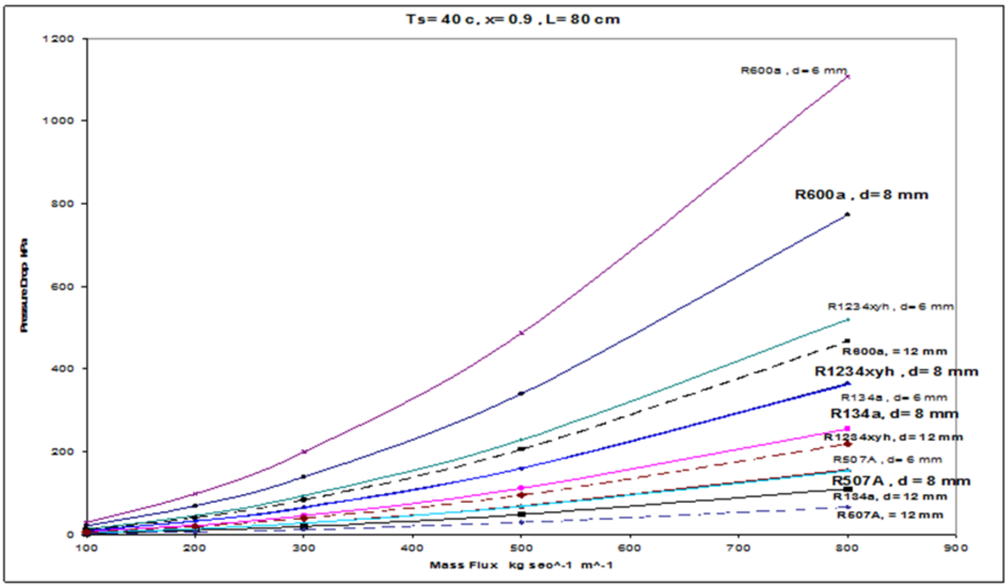

(b)

Fig. 7. Effect of the mass flux of the refrigerants at different inner tube diameters on the pressure drop of the present model at a different vapour quality and a saturated temperature of 40^{o}C: a- for tube diameter 6 and 8mm, x=0.1, b- for tube diameters 6, 8, and 12mm, x= 0.9

5. Conclusions

An analytical investigation was conducted to present a model of two-phase pressure drops for the homogenous separated flow of condensation for different refrigerants in a horizontal tube. As mentioned in the literature, there are many reasons for the calculation and prediction of a two-phase pressure drop. The main reasons are to overcome the acting forces inside the condenser tube using a pumping power device. The model has provided the basis for the average density and average kinematic viscosity of the refrigerant. The model was tested theoretically to provide results to compare with previous work and to agree with the present work of two-phase pressure drop. Four refrigerants were tested in copper tubes of 8mm and 6mm inner diameter and a length of 0.8m at a saturated temperature of 40°C. The average deviations obtained from the results were compared with the three models of Cicchitti, McAdams, and Dukler, which were 19%, ±4%, and 17% respectively. The following conclusions are summarised from the discussed results.

1- Refrigerants: the results show the behaviour of all examined refrigerants is the same, unless there are deviations between them within the thermo-physical properties. The pressure drop for each one of the examined refrigerants from the highest to the lowestareR600a, R1234xyh, R134a, and R507A.
2- Mass flux: the effects of the mass flux indicated the flow condition of the refrigerant inside the tube. This study presents a pressure drop model without any imperative of the flow map inside the tube.
3- Average density and average kinematics viscosity: in this study, it was assumed the flow is a homogenous separated two-phase flow model, and this flow model considers the physical properties of the refrigerants at average values.
4- The pressure drop of the present model: a two-phase pressure drop model was presented and run with four different refrigerants at a given saturated temperature. The obtained results from this analytical model match the reliability and trend curve of literature by Didi *et al.,* [4] and Pamitran *et al.,* [7].
5- Inner tube diameter effects: this proved that the correct behaviour of the present model is the smaller inner tube diameter, which gives a high-pressure drop because of the high shear stress on the inner contact of the refrigerant with the tube walls. When the inner-tube diameter is large, the pressure drop is lower, because there is less shear stress of the refrigerant inside the tube.

A parametric study two-phase pressure drop model was undertaken to estimate the influence of the test variables. The chosen representative values of the test variables of four refrigerants, different inner tube diameters, and mass fluxes had a tough effect on the entire range of vapour quality. Finally, this model can contribute to the designing of condensers and/or heat exchangers with an inner tube diameter, ranging from 6mm to 12mm, with other different refrigerants.

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