



The heat transfer coefficient similarity between binary and single component flow condensation inside plain pipes

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ABSTRACT

To identify the heat transfer mechanisms during flow condensation of binary mixtures inside pipes has motivated vast research in the past decades. While the prediction capabilities of models have substantially improved due to larger experimental data bases and computational tools, the complexity of the models has grown to a level that makes it difficult to identify the dominant flow and fluid properties contributions. Opposite to this trend, in this work we show that the heat transfer coefficient of single and binary component fluids follows a similar scaling law in terms of a two-phase flow Reynolds number, reducing the complexity of the model substantially. This similarity is attributed to an equivalent heat transfer mechanisms between them and the single-phase flow case. Therefore, it is assumed that the dominant heat transfer resistance is located in the conductive sublayer and thus unaffected by either the flow pattern, liquid film thickness or a mass transfer resistance close to the liquid-vapour interface.

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1. Introduction

Condensers based on small pipe diameters are of great relevance and interest in connection with the growing demand for heat exchangers in several applications ranging from condensers of cooling equipment and air conditioning systems, horizontal tubular evaporators of water-desalinating thermal units, heaters of power systems, heat pipes, etc. Furthermore, heat exchangers are the main component in cryogenic processes such as air separation and natural gas liquefaction plants. In these systems, the design and performance of the heat exchangers can affect other major components like compressors and drivers [1]. These facts have motivated the development of more accurate simulation tools [2–5] that depend on heat transfer coefficient models.

During the past decades, models of the heat transfer coefficient during flow condensation of single component fluids inside pipes have been proposed and tested against experimental data bases as discussed in several publications, e.g. [6–11]. In spite of the extensive work, the mechanisms controlling the heat transfer phenomena have not been conclusively identified, and thus most models have been based on heuristic methods. This fact has led to models considering different dimensionless groups and assumptions. In some cases, models with more than 7 dimensionless groups and 10

adjusted parameters have been proposed for predicting the experimental data. The recent use of machine learning techniques have further increased the number of dimensionless numbers up to 20 and the number of fitting parameters up to 12 [12,13]. On the one hand, the increase complexity of the models can reduce the error in the prediction of larger data sets even when the data sets contain larger error and show high dispersion. On the other hand, the increase in the complexity of the models has resulted detrimental to the goal of unveiling the physic of the heat transfer process which in turn should lead to simple models.

For the case of flow condensation of multicomponent fluids, the situation is even more difficult. A common approach has been to include correction factors to the heat transfer coefficient of a single component fluid. However the validity of the assumptions for such corrections remain under debate. In this work, we will show that the heat transfer coefficient during condensation of single and binary component fluids follows a similar scaling law in terms of a two-phase flow Reynolds number defined in terms of the superficial velocities of the liquid and vapour phase. This result reduces the existing complexity related to the modelling of the heat transfer coefficient of binary mixtures during flow condensation inside pipes.

1.1. Literature review

Extensive reviews of heat transfer models for flow condensation of binary and multicomponent mixtures inside pipes can be found

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Nomenclature

$Re_{L0} = GD/\mu_L$	Liquid Reynolds number [-]
$Re_{V0} = GD/\mu_V$	Vapour Reynolds number [-]
$Nu = hD/k$	Nusselt number [-]
$Re_L = G(1-x)D/\mu_L$	Superficial liquid Reynolds number [-]
$Re_V = GxD/\mu_V$	Superficial vapour Reynolds number [-]
$Re_{2\phi} = Re_L + Re_V$	Superficial two-phase Reynolds number [-]
$Pr = c_p\mu/k$	Prandtl number [-]
P	pressure [kPa]
G	mass flux [$kg\ m^{-2}\ s^{-1}$]
c_p	heat capacity [$J\ kg^{-1}\ K^2$]
μ	viscosity [Pa s]
ρ	density [$kg\ m^{-3}$]
x	quality [-]
L	liquid
V	vapor

in [14,15]. The literature review presented in this section focuses on highlighting the difference of the models in terms of the selected dimensionless groups considered in some selected models.

The first mathematical expression known for condensation of a pure fluid is attributed to Nusselt in 1916 for the case of liquid flowing over a surface under the influence of gravity. This model considers a laminar flow and that the heat is transferred from the wall to the fluid mainly by conduction across the film. The temperature at the liquid-vapour interface is set to be the saturation temperature of the vapour. The dominant conduction mechanism across the liquid film implies a linear temperature profile as the convective contribution is disregarded. At high liquid film velocities, the convective contribution is important and the assumption of a linear temperature profile across the film becomes questionable. However, this assumption will be used in the following decades in models related to convective flow boiling and condensation even in very turbulent flowing films.

In 1937, Colburn and Drew [16] presented a model for describing the condensation of a mixture of vapours over a cold surface. The condensation process is sketched in terms of a condensate that separates the vapour mixture from the cold wall. The local heat flux absorbed by the cooling media across the wall is assumed to be the sum of the sensible heat of cooling of the condensate, the latent heat involved in condensing the vapour at the vapour-liquid interface and the sensible heat received from the vapour. The model is based on the film theory which assumes that the resistance to turbulent heat and mass transport is concentrated in thin layers adjoining on both sides of the interface. The vapour-condensate interfacial temperature is determined by assuming that the liquid composition, the vapour composition, and the temperatures of the phases have values very close to those of equilibrium. Based on this assumption, the temperature at the interface will correspond to the boiling point of the equilibrium composition at the liquid side. At the interface on the vapour side, the vapour will be enriched on the most volatile component compared to the bulk composition. This implies that the temperature at the interface is lower than the one from the bulk. Hence, if it is assumed that the heat is transferred mainly by conduction across the film, this results in a lower heat flux. In summary, the assumption introduced by Colburn and Drew leads to the concept of a vapour-condensate interface thermal resistance that has been adopted for different flow conditions in the following decades.

For example, Sparrow and Marschall [17] considered the problem of binary, gravity-driven film flow condensation and proposed a model in terms of conservation laws. The model depends on the

temperature at the interface which is obtained by an iterative procedure. The model considers that the condensate has a composition equal to the one at the liquid phase at the interface. Denny and Jusonius [18] proposed a model for laminar film condensation of binary vapor mixtures undergoing forced flow down a vertical plate based on conservation laws. The analysis of the model is presented in terms of the heat transfer ratio q/q_{Nu} where q_{Nu} is the Nusselt heat transfer. The main observations are (i) an initial rapid decline of q/q_{Nu} is observed as the concentration of the second component increases, and (ii) a minima in q/q_{Nu} is observed for the non-azeotropic binary pairs. These effects are attributed to first order variations of the physical properties in the liquid phase and the vapour-liquid equilibrium curve. Further, it is observed that the effect of the forced flow is pronounced when the velocity of the vapour is increased as a consequence of the vapor-drag. The importance of the vapour phase on the heat transfer coefficient was observed experimentally in previous research related to flow condensation of single components. For instance, Crosser in 1955 [19] pointed out previous research work identifying the direct effect of the vapour velocity on the heat transfer coefficient, namely Jakob, Erk and Eck (1932), Schmidt (1937) and Carpenter and Colburn (1951) (as cited in [19]). However, after the late 70s the importance of the vapour phase on the heat transfer coefficient was disregarded or not explicitly mentioned in most models.

To avoid the difficulties related to the estimation of the resistance to mass transfer in the vapour phase, related to the formulation of Colburn and Drew, Bell and Ghaly [20] considered the assumption of replacing the mass transfer resistance in the vapour phase by an estimate of the heat transfer resistance in the vapour. The model from Bell and Ghaly considers that the total latent heat of condensation and sensible heat of the cooling condensate are transferred through the thickness of the liquid film. It is assumed that the liquid and vapour compositions are in equilibrium at the vapour bulk temperature. This particular assumption is acknowledged to be satisfied at low condensing rates. However, this assumption will be overlooked in the future extensions of the model. The work of Bell and Ghaly [20] includes the application of the model to the design of single-, two- and multi-pass coolant condenser. In the next decades, the model of Bell and Ghaly was adapted to different cases, in particular in-tube condensation, due to its simplicity as compared to the one from Colburn and Drew. Taitel and Tamir [21] presented a generalised model for any number of components considering a diffusion law for the multicomponent system in terms of binary diffusivities. The model considers a stagnant multicomponent mixture where the laminar film of the condensate mixture flows by gravity. Tamir and Merchuk [22] studied experimentally the case considered by Taitel and Tamir [21]. It is concluded that at relatively high condensation rates (corresponding to the difference between the bulk and wall temperature), the composition of the condensate becomes identical with the composition of the bulk of the condensing vapour. This observation implies a simplification in the calculation of the temperature at the interface which is given as the boiling point of the multicomponent mixture to the condensate composition. This observation agrees with the underlying assumption of the model of Bell and Ghaly of considering that the liquid and vapour compositions are in equilibrium at the vapour bulk temperature for low condensing rates.

Research on multicomponent mixtures condensation inside pipes has been extensively studied during the past decades. Some works have followed and/or extended the ideas proposed by Colburn and Drew [16] or the one by Bell and Ghaly [20], while other works have adopted alternative models.

The first mentioned approach have been the most adopted one in general. For example, Cavallini et al. [23] studied condensation of the zeotropic mixture R125/R236ea in a 8 mm smooth tube and

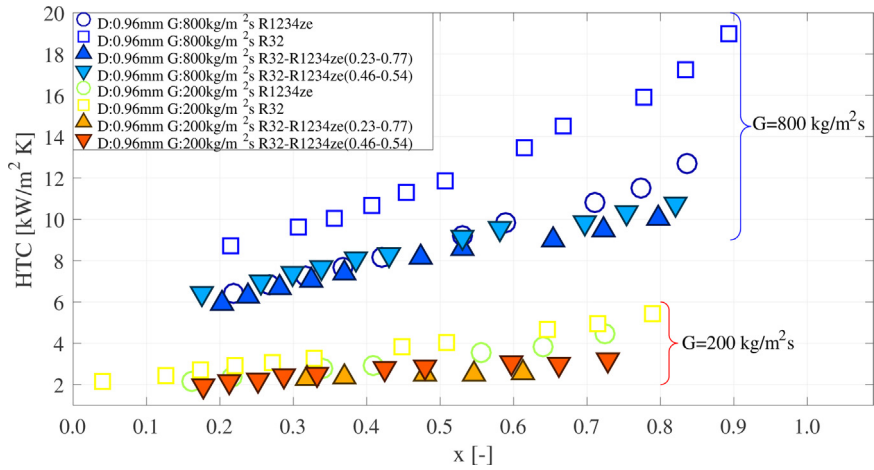


Fig. 1. Experiments for R32 and R1234ze and its mixtures for two selected mass fluxes.

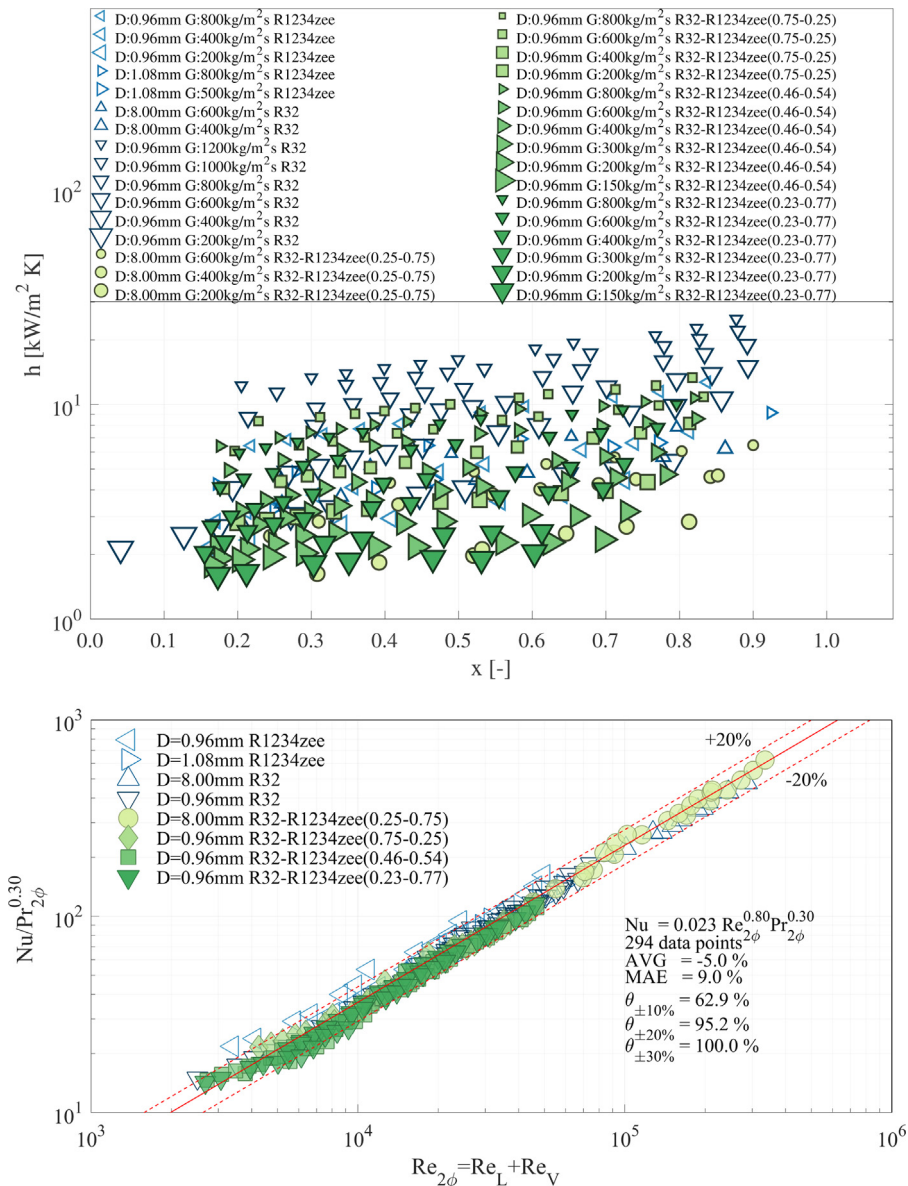


Fig. 2. Experiments for R32 and R1234ze and their mixtures. The data points are properly scaled by $Re_{2\phi}$.

Table 1
Condensation heat transfer database for mini/micro-channel flows.

		[mm]	[kg/m ⁻² s]	[°C]	
Aprea et al. (2003)	R407c	20.00	115	38.8–38.8	11
McDonald (2015)	C2-C3(0.33-0.67)	14.45	150-450	30-60	81
Mitre (2005)	R410a	9.40	200-800	61.1–66.6	236
Jung et al.(2004)	R410a	8.82	100-300	40-40	26
Smit (2002)	R22-R142b(0.50-0.50)	8.11	100-600	85.6–85.6	34
Smit (2002)	R22-R142b(0.60-0.40)	8.11	150-600	81-81	38
Smit (2002)	R22-R142b(0.70-0.30)	8.11	150-600	76-76	38
Smit (2002)	R22-R142b(0.80-0.20)	8.11	150-600	71-71	36
Smit (2002)	R22-R142b(0.90-0.10)	8.11	150-600	66-66	35
Cavallini (2000)	R125-R236ea(0.28-0.72)	8.00	100-750	57-57	23
Cavallini (2000)	R125-R236ea(0.46-0.54)	8.00	400-750	56-57	23
Cavallini (2000)	R125-R236ea(0.63-0.37)	8.00	100-750	56-56	29
Cavallini et al.(2001)	R410a	8.00	200-750	40-40	35
Han et al. (2016)	R32-R1234zee(0.25-0.75)	8.00	200-600	30-30	22
McDonald (2015)	C2-C3(0.67-0.33)	7.75	150-450	25-45	81
Afroz et al. (2008)	CO2-DME(0.21-0.79)	4.35	200-500	40-40	40
Afroz et al. (2008)	CO2-DME(0.39-0.61)	4.35	200-500	40-40	21
Dobson et al.(1998)	R32-R125(0.40-0.60)	3.14	300-800	35-35	34
Maraak (2009)	C1-C2(0.83-0.17)	1.00	490-690	-98–62.5	43
Maraak (2009)	C1-C2(0.59-0.41)	1.00	490-690	-76.6–36.2	28
Maraak (2009)	N2-C1(0.16-0.84)	1.00	195-690	-134–91	48
Del Col et al. (2015)	R32-R1234zee(0.46-0.54)	0.96	150-800	40-40	62
Del Col et al. (2015)	R32-R1234zee(0.23-0.77)	0.96	150-800	40-40	52
Azzolin et al. (2021)	R32-R1234zee(0.75-0.25)	0.96	200-800	41.5–41.5	43
Maraak (2009)	C1-C2(0.59-0.41)	0.50	490-690	-46.7–46.7	8
Total					1127

Table 2
Condensation heat transfer database for mini/micro-channel flows for fluids R32, R1234 and its mixtures.

		[mm]	[kg/m ⁻² s]	[°C]	
Del Col et al.(2015)	R1234zee	0.96	200-800	40-40	27
Liu et al.(2016)	R1234zee	1.08	500-800	50-50	11
Cavallini et al.(2013)	R32	8.00	400-600	40-40	11
Matkovic(2009)	R32	0.96	200-1200	40-40	66
Han et al. (2016)	R32-R1234zee(0.25-0.75)	8.00	200-600	30-30	22
Azzolin et al. (2021)	R32-R1234zee(0.75-0.25)	0.96	200-800	41.5–41.5	43
Del Col et al. (2015)	R32-R1234zee(0.46-0.54)	0.96	150-800	40-40	62
Del Col et al. (2015)	R32-R1234zee(0.23-0.77)	0.96	150-800	40-40	52
Total					294

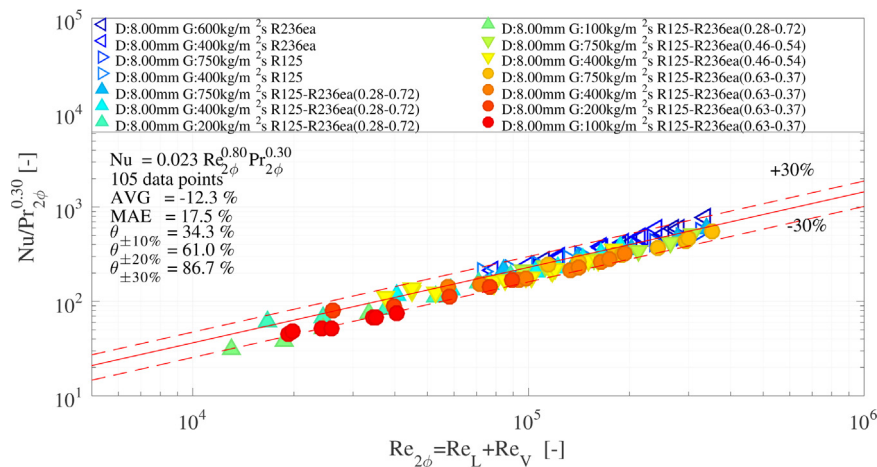


Fig. 3. Dependency of the $Nu/Pr_{2\phi}^{0.30}$ versus $Re_{2\phi}$ for R236ea and R125 and its mixtures.

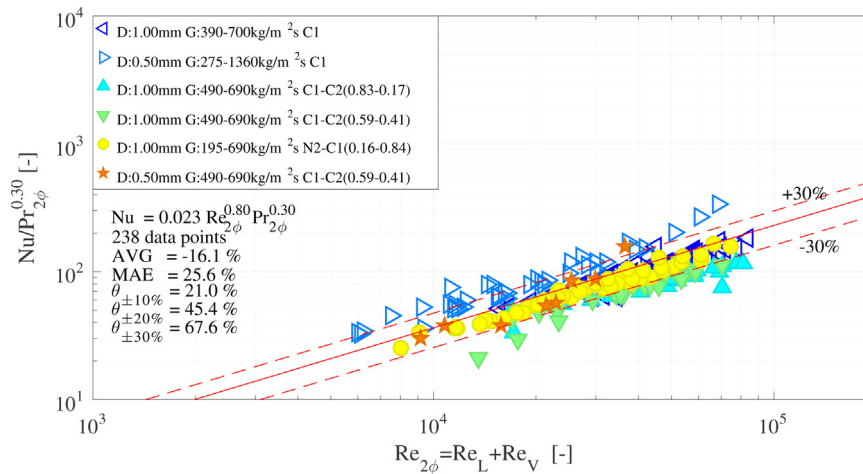
proposed to combine the model from Colburn and Drew [16] and his model for flow condensation [24]. Del Col et al. [25] extended the model from Thome et al. [26] to zeotropic mixtures by adding the heat transfer resistance to both the convective and the film regions in the model of Bell and Ghaly. In addition, the effect of the interfacial roughness on the vapour heat transfer coefficient acting on the convective film was included. The model was

evaluated against different fluid mixtures with temperature glides ranging from 3.5°C to 22°C and pipe diameters of 7.5 and 8.0 mm. Deng [27] suggested a simplified heat transfer model based on the heat transfer model proposed by Cavallini et al. [28] but following the approaches suggested by Del Col et al. [25] to include the Bell and Ghaly resistances. Macdonald and Garimella [9] experimentally studied the heat transfer coefficient during condensation of hy-

Table 3

Condensation heat transfer database for mini/micro-channel flows for fluids R236ea, R125 and its mixtures.

		[mm]	[kg/m ⁻² s]	[°C]	
Cavallini et al.(2013)	R236ea	8.00	400-600	40-40	19
Cavallini et al.(2001)	R125	8.00	400-750	40-40	11
Cavallini (2000)	R125-R236ea(0.28-0.72)	8.00	100-750	57-57	23
Cavallini (2000)	R125-R236ea(0.46-0.54)	8.00	400-750	56-57	23
Cavallini (2000)	R125-R236ea(0.63-0.37)	8.00	100-750	56-56	29
Total					105

**Fig. 4.** Dependency of the $Nu/Pr_{2\phi}^{0.30}$ versus $Re_{2\phi}$ for C1 and mixtures C1-C2, N2-C1.**Table 4**

Condensation heat transfer database for mini/micro-channel flows for fluids C1, C1-C2, and N2-C1.

		[mm]	[kg/m ⁻² s]	[°C]	
Maraak (2009)	C1	1.00	390-700	-132-85.7	73
Maraak (2009)	C1	0.50	275-1360	-131.4-86.3	38
Maraak (2009)	C1-C2(0.83-0.17)	1.00	490-690	-98-62.5	43
Maraak (2009)	C1-C2(0.59-0.41)	1.00	490-690	-76.6-36.2	28
Maraak (2009)	N2-C1(0.16-0.84)	1.00	195-690	-134-91	48
Maraak (2009)	C1-C2(0.59-0.41)	0.50	490-690	-47	8
Total					238

drocarbon mixtures inside horizontal smooth pipes. The data was analysed by extending the condensation heat transfer coefficient model for single component [29] with the Silver Bell Ghaly and Colburn and Drew models. In particular, the model proposed for single component [29] depends strongly on the liquid film thickness.

On the other hand, other models have not considered the models from Colburn and Drew [16] or the one by Bell and Ghaly [20] in their derivations. For example, Tandon et al. [30] studied the effect of mixture composition on the heat transfer coefficient during forced convection condensation of R22 and R12 mixtures in a horizontal pipe. It was observed that the binary mixture results in heat transfer coefficients lower than the pure R22 and higher than the pure R12 but the increase not being proportional to the increase in R22 mass concentration. A correlation for the three mixtures compositions studied was proposed considering the dependency of the heat transfer coefficient in the vapour mass velocity. An equivalent Reynolds number was defined in terms of the vapour velocity, the viscosity of only the liquid phase multiplied by the product of the liquid and vapour density ratio. The model also includes a Jakob number and the liquid Prandtl number, making in all up to 3 dimensionless groups and 4 adjusted parameters. Koyama et al. [31] investigated the condensation of R22/R114 in-

side a horizontal tube with internal spiral fins. A model was suggested considering the model from Fujii and Nagata (1973) for pure organic substances. The model considers the velocity of the vapour phase for defining an equivalent Reynolds number.

In summary, the modelling approach proposed by Colburn and Drew [16] and the one by Bell and Ghaly [20] have been incorporated in models during the past decade. A particular strong assumption in these models is the establishment of a concentration profile at the interface which leads to a mass resistance that dominates the heat transfer process particularly beyond the original assumptions of a laminar liquid film and low heat fluxes. However, in the particular case of an annular two-phase vapor-liquid flow, a thin, wavy liquid film is driven along the wall by the shear force exerted by the vapor phase in the core. The liquid film is traversed by high velocity waves known as disturbance waves. It has been shown that the disturbance waves lead to an increase level of turbulence [32,33], thus enhancing the mixing in the liquid film. Further, the liquid-vapour interface is also affected by a complex droplet entrainment/deposition process occurring at a faster time scale than the molecular diffusion process, and also influencing spatial length scales which are larger than the corresponding ones to the diffusion process. This fact has been recently highlighted when the liquid film thickness has been measured us-

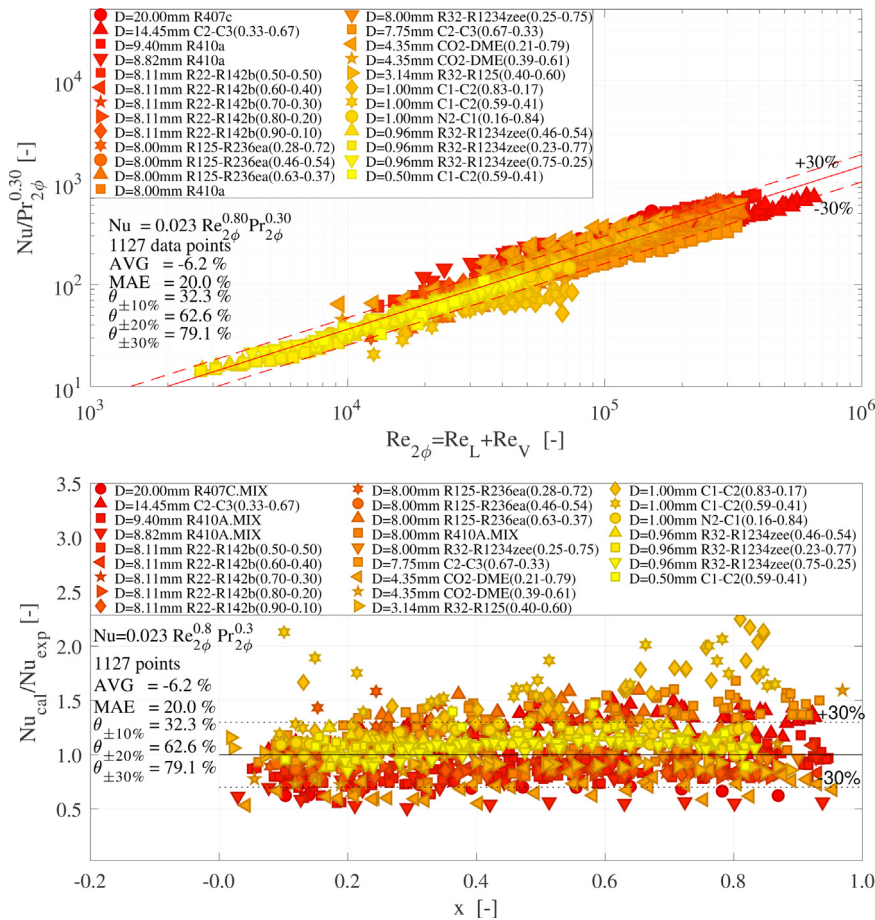


Fig. 5. Experimental data points from the literature scaled in terms of $Re_{2\phi}$. The error in terms of the equilibrium thermodynamic quality shows an independence of flow pattern.

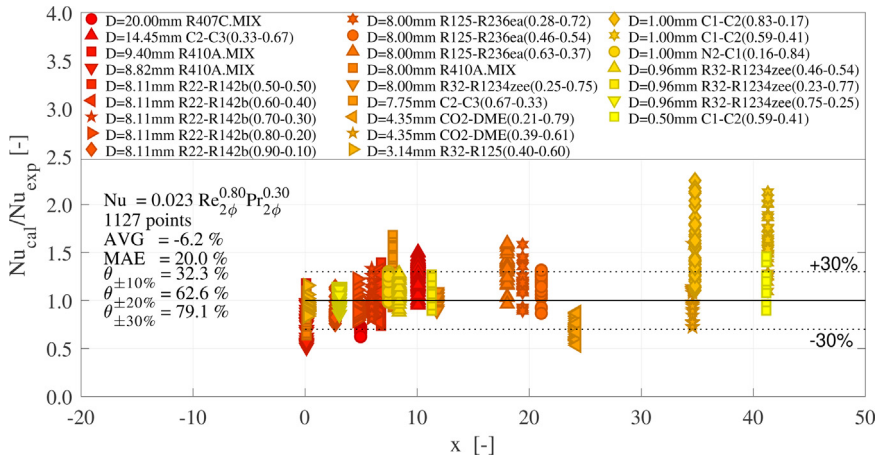


Fig. 6. The error shows an independence of the temperature glide.

ing a chromatic confocal sensor [34], showing that the magnitude of the thickness of the liquid film can vary more than 100% with dominant wave frequencies above 40Hz. These two mentioned processes, namely higher turbulence in the film due to disturbance waves and the entrainment/deposition processes, can prevent the existence of a sharp concentration profile at the liquid-vapour interface. Hence the validity of the assumption of a mass resistance that dominates the heat transfer process at high flow rate becomes questionable. It has also been discussed that some models have not considered the mass transfer resistance. In this work,

it will be shown that condensation of a binary mixture can be well predicted assuming a well mixed condition, and thus disregarding the contribution of a dominant mass resistance at the interface.

2. Method

In this study, experimental data from the literature have been gathered covering a large range of fluids and pipe diameters. The experimental data considered are summarised in Table 1.

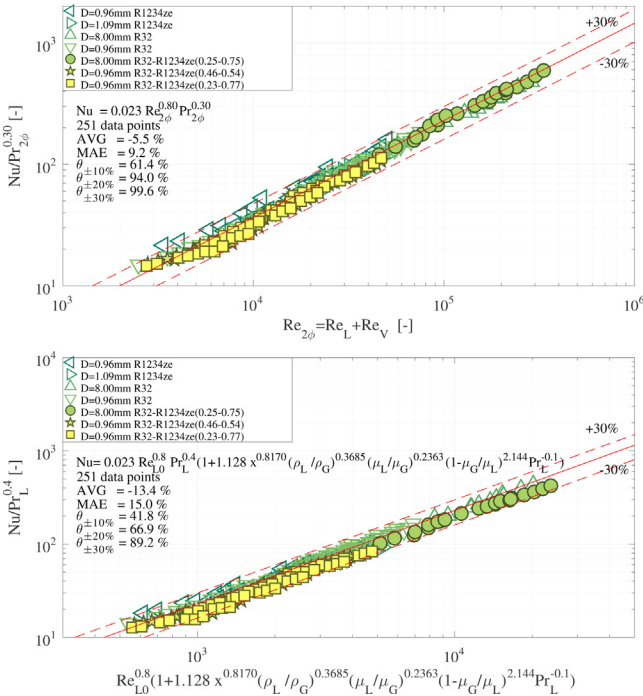


Fig. 7. Dependency of $Nu/Pr_{2\phi}^{0.3}$ in terms of $Re_{2\phi}$ for experimental data points with mass fluxes higher than 200 kg/m^2 .

In all the cases, only data points related to mass fluxes above 200 kg/m^2 s were considered for avoiding cases with flow stratification that can induce a different behaviour compared to high mass fluxes. The fluid properties are calculated with the software REFPROP version 9.1 [35] at the saturation conditions of the fluid. The data sets considered in this work are the ones in which a single-phase flow heat transfer coefficient has been reported and the data sets showing a lower dispersion.

In the next section the accuracy of the models is evaluated by $\theta_{\pm 10\%}$, $\theta_{\pm 20\%}$ and $\theta_{\pm 30\%}$, the percentage of data points predicted within $\pm 10\%$, $\pm 20\%$ and $\pm 30\%$ respectively, and the average and mean absolute error, defined as

$$AVG = \frac{1}{N} \sum \frac{Nu_{pred} - Nu_{exp}}{Nu_{exp}} \times 100 \quad (1)$$

$$MAE = \frac{1}{N} \sum \frac{|Nu_{pred} - Nu_{exp}|}{Nu_{exp}} \times 100 \quad (2)$$

2.1. Multicomponent heat transfer coefficient model

A model for the heat transfer coefficient for single component fluids inside tubes for high mass fluxes ($G > 200 \text{ kg/m}^2$ s) has been proposed in [36] by assuming an equivalence between the single-phase flow and flow condensation. The assumption consists in that the heat transfer coefficient model for single-phase flow can be used for flow condensation just by replacing the single-phase flow Re and Pr numbers for equivalent ones related to the two-phase flow case. For instance, the single-phase flow model by Dittus-Boelter [37] can be re-written as

$$Nu_L = 0.023 Re_{2\phi}^{0.8} Pr_{2\phi}^{0.3} \quad (3)$$

with

$$Re_{2\phi} = Re_L + Re_V = \frac{GxD}{\mu_V} + \frac{G(1-x)D}{\mu_L} \quad \text{for } 0 < x < 1 \quad (4)$$

a two-phase flow equivalent Reynolds number defined in terms of the sum of the superficial vapour and liquid Reynolds number, and

$$Pr_{2\phi} = Pr_L(1-x) + Pr_V x \quad \text{for } 0 < x < 1 \quad (5)$$

an equivalent two-phase flow Prandtl number.

The coefficients in the model correspond to the ones from the correlation of Dittus-Boelter for the single-phase flow heat transfer coefficient. Therefore, no best fitting of the coefficients of the model has been done. The model assumes that the dominant thermal resistance is located in the conductive sublayer which implies that neither the flow pattern, liquid film thickness nor a mass transfer resistance close to the liquid-vapour interface influence the heat transfer coefficient. In particular, it is considered that the commonly used assumption that the heat is transferred mainly by conduction across the film with a temperature at the surface of the film to be the saturation temperature of the vapour is not valid as the thickness of the conductive sublayer is thinner than the liquid film. In particular, a recent experimental study measuring the film thickness during flow condensation has shown that by assuming mainly conduction across the film the heat transfer coefficient is largely under-predicted [34]. This measurements support the assumption of a dominant thermal resistance thinner than the liquid film.

3. Results and discussions

3.1. Heat transfer coefficient deterioration

Figure 1 illustrates an example corresponding to the heat transfer coefficient during flow condensation of R32 and R1234ze and their mixtures from the literature [38]. The figure shows that for the case of the mixtures, the heat transfer coefficient is lower than the corresponding one to the pure fluid.

The observed trend has been attributed to the additional mass transfer resistance effect based on the assumption that the liquid composition and the vapour composition at the interface have values very close to those of equilibrium, as suggested originally by Colburn and Drew. While this description might be considered appropriate under stagnant conditions and laminar flow, as studied in early works [17,18], its validity can be questionable for highly turbulent flow conditions and high level of mixing due to the presence of bubbles, waves and entrainment and deposition phenomena.

For a fixed mass flux and quality, the heat transfer coefficient of the mixture shows lower values than the corresponding to the pure fluid, but note that the fluid properties are also different for a binary mixture. For example, the viscosity of a binary liquid mixture can show no maximum or minimum, one maximum or minimum, or both a maximum and a minimum in the whole composition range [39]. For this reason, considering the Nu number instead of the heat transfer coefficient should be more appropriate in order to take into account the variations of the fluid properties. In particular, it has been suggested that the thermal conductivity plays a major role in the observed trend [40].

To illustrate the mentioned effect, experimental data points from the literature for R32 and R1234ze and their mixtures for different pipe diameters and conditions are shown in (Fig. 2) and Table 2. The same data are shown in terms of $Nu/Pr_{2\phi}^{0.3}$ versus $Re_{2\phi}$. It is not possible to differentiate between the data corresponding to the single component and to the binary mixtures. The $Re_{2\phi}$ scales properly the heat transfer coefficient in terms of the Nusselt number. This trend suggests that if the heat transfer is affected by a mass resistance at the liquid-vapour interface in the binary mixture cases, its contribution is not noticeable. The corresponding AVG and MAE values are low and comparable to expected experimental errors and dispersion of the data. These estimators should

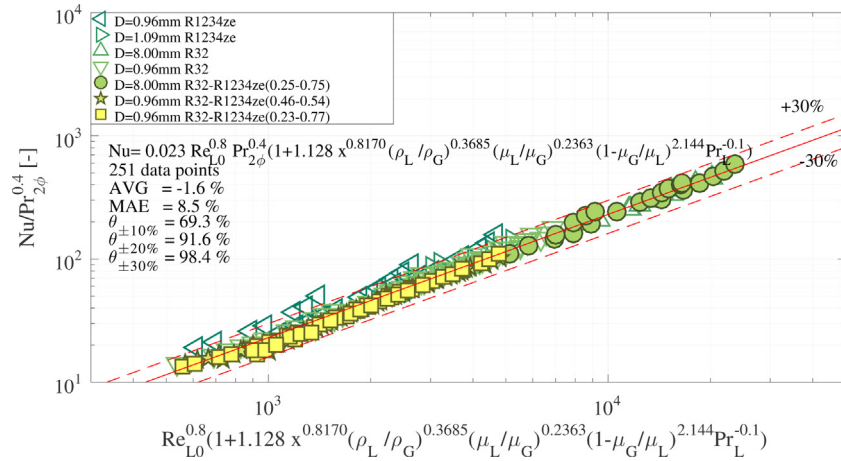


Fig. 8. Effect of replacing Pr_L by $Pr_{2\phi}$ in the model of Cavallini et al. (2006) for experimental data points with mass fluxes higher than 200 kg/m²s.

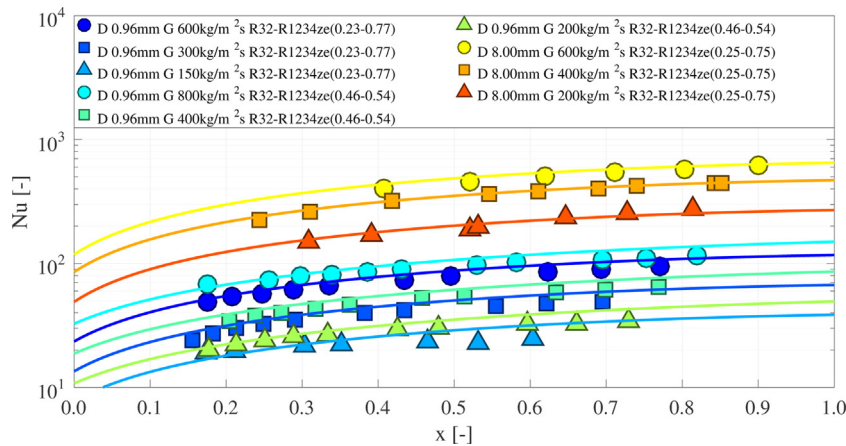


Fig. 9. Comparison between the prediction by Eq. (3) and experimental data points for binary mixtures of R22 and R1234ze.

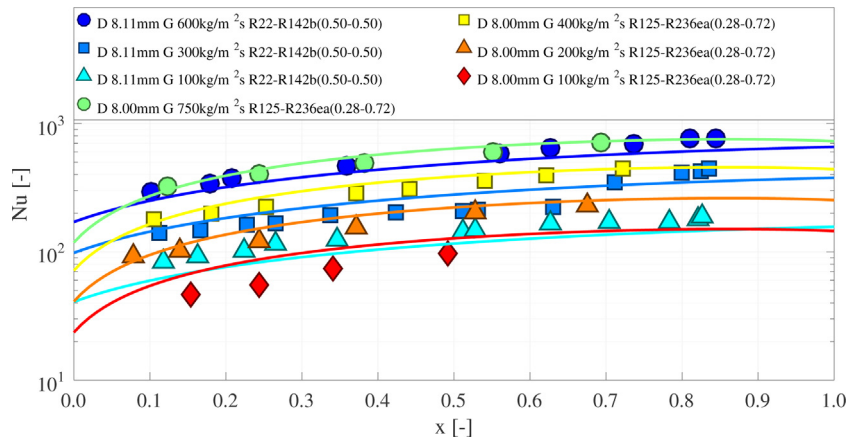


Fig. 10. Comparison between the prediction by Eq. (3) and experimental data points for binary mixtures of R22 and R142b.

be used as an indication of the deviation of the model compared to the experimental data more than as a goal on itself.

3.2. Scaling law of the heat transfer coefficient

A similar trend as the one in the previous section is observed in (Fig. 3) corresponding to experimental data points of mixtures of R236ea and R125 and corresponding pure fluids (Table 3), and in (Fig. 4) corresponding to methane and mixtures of methane/ethane and methane/nitrogen (Table 4). In this last case, the dispersion

of the data is higher, which has been attributed to the larger uncertainty of the experimental due to the challenging conditions. Finally, a larger data base is considered with a broad experimental data set covering a range of pipe diameters from 14.45 mm to 0.5 mm and a temperature glide of up to 40K in (Fig. 5). The trend is well captured by $Re_{2\phi}$ and the dispersion of the data is in same order than the reported experiments. The same figure shows the deviation of the model in terms of Nu_{cal}/Nu_{exp} versus the thermodynamic quality, while (Fig. 6) shows the deviation of the model in terms of Nu_{cal}/Nu_{exp} as a function of the temperature glide of the

mixtures. It is seen that the trend is properly captured. Some deviation are observed, namely for the mixtures containing methane, but it has been also acknowledged that these data can present high uncertainties due to the challenging conditions of the experiments.

3.3. Prediction of the heat transfer coefficient

Figure 7 shows a comparison of the prediction by the model given in Eq. (3) and the model from Cavallini et al. (2006) [28] for the ΔT -independent regime, i.e. $J_G > J_G^T$.

$$Nu = 0.023 Pr_L^{0.4} \frac{k_L}{D} \times Re_{L0}^{0.8} \left[1 + 1.128 x^{0.8170} \left(\frac{\rho_L}{\rho_G} \right)^{0.3685} \left(\frac{\mu_L}{\mu_G} \right)^{0.2363} \left(1 - \frac{\mu_G}{\mu_L} \right)^{2.144} Pr_L^{-0.1} \right] \quad (6)$$

where the transition boundary for the two regions is given as

$$J_G^T = \left[\left(\frac{7.5}{4.3 X_{tt}^{1.111} + 1} \right)^{-3} + C_T^{-3} \right]^{-1/3} \quad (7)$$

where C_T is 1.6 for hydrocarbons and 2.6 for other refrigerants.

The experimental data set corresponds to R32 and R1234ze and its mixtures [27,38]. The model from Cavallini et al. (2006) [28] does not include the correction due to the mass resistance at the liquid-vapour interface. However, the model is capturing the trend of the data quite good. This trend suggests that the contribution of the mass resistance at the liquid-vapour interface that may exist for the binary mixture cases might not be dominant. In particular, Azzolin and Bortolin (2021) [41] have shown that by adding the Silver-Bell-Ghaly correction to the model from Cavallini et al. (2006) for predicting mixtures of R32 and R1234ze(E) the effect is limited to 4%. Hence, the effect of the mass resistance correction compared to the additional complexity added to the model does not necessarily justify its use.

In (Fig 8) the Pr_L is replaced by the $Pr_{2\phi}$ for the model from Cavallini et al. (2006). On the one hand, the prediction for the binary mixture looks to improve but the corresponding to single components (open symbols) looks to be slightly worse. The similar prediction between the model from Cavallini et al. (2006) and Eq. (3) has been attributed to the fact that the hydrodynamic component of the model from Cavallini et al. (2006) is equivalent to $Re_{2\phi}$ [42], i.e.

$$Re_{L0}^{0.8} \left[1 + 1.128 x^{0.8170} \left(\frac{\rho_L}{\rho_G} \right)^{0.3685} \left(\frac{\mu_L}{\mu_G} \right)^{0.2363} \left(1 - \frac{\mu_G}{\mu_L} \right)^{2.144} Pr_L^{-0.1} \right] \approx Re_{2\phi}^{0.8} \quad (8)$$

This equivalence suggests that the hydrodynamic component of the model from Cavallini et al. (2006) is approximating the scaling law of the single-phase flow heat transfer coefficient. In order to compare the model from Cavallini et al. (2006) and Eq. (3) in a more similar condition, the Pr_L in the model from Cavallini et al. (2006) is replaced by $Pr_{2\phi}$ as shown in Fig. (8). With this change, both models provide equivalent errors.

Finally, a comparison between some selected experimental data points and Eq. (3) is shown in (Fig. 9) for a mixture of R32-R1234ze and in (Fig. 10) for a mixture of R22-R142b and mixtures of R125-R236ea. In all, the trend is well predicted both qualitatively and quantitatively by Eq. (3).

4. Conclusions

In this work, we have showed that the heat transfer coefficient of single and binary component fluids follows a similar scaling law in terms of a two-phase flow Reynolds number. This similarity is attributed to an equivalent heat transfer mechanisms between

them and the single-phase flow case. In these three situations, the dominant heat transfer resistance is attributed to be located in the conductive sublayer and thus unaffected by either the flow pattern or the processes taking place close to the liquid-vapour interface.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

CRedit authorship contribution statement

Carlos A. Dorao: Conceptualization, Methodology, Formal analysis, Writing – original draft. **Maria Fernandino:** Formal analysis, Writing – review & editing.

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