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Review

Intensive literature review of condensation inside smooth and enhanced tubes

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ABSTRACT

This paper presents a detailed review of research work on in-tube condensation in the literature due to its significance in refrigeration, air conditioning and heat pump applications. The heat transfer performance of heat exchangers can be improved by heat transfer enhancement techniques, such as active and passive techniques. Passive techniques requiring fluid additives or special surface geometries are mentioned in depth, by comparison with active techniques requiring external forces, e.g. electrical field, acoustic or surface vibration, etc., in the paper due to their common usage in condensation applications. In addition, the importance of usage of hydrocarbons instead of fluorocarbons is emphasised. This paper can not only be used as the starting point for the researcher interested in in-tube condensation process, but it also includes new investigations on condensation inside tubes.

Contents

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Nomenclature

A	area, m^2
B	fin root distance, m
d	inner diameter, or average inner diameter of tube, m
EHD	Electrohydrodynamic
G	mass flux, kg m^{-2} s^{-1}
g	gravitational acceleration, m s^{-2}
h	heat transfer coefficient, W m^{-2} K^{-1}
L	length of test tube, m
l	fin height, m
n	number of fins
R	radius, m
P	pressure, N m^{-2}
p	fin pitch, m
Re	Reynolds number
S	slip ratio
t	fin tip thickness, m
u	axial velocity, m s^{-1}
v	radial velocity, m s^{-1}
w	tube wall thickness, m
We	Weber number
x	average vapor quality
X	Lockhart Martinelli parameter
y	wall coordinate, m
z	axial coordinate, m
ΔP	pressure drop, Pa

Greek symbols

γ	apex angle, °
α	void fraction
τ_i	interfacial shear stress, N m^{-2}
δ	film thickness, m
β	helix angle, °
σ	surface tension, N m^{-1}
ρ	density, kg m^{-3}
μ	dynamic viscosity, Pa s
η	kinematic viscosity, m^2 s^{-1}
Θ	angle of the condensate film layer, °

Subscripts

f	frictional term
g	gas/vapor
l	liquid
mf	micro-fin
o	outer
s	smooth

1. Introduction

Heat exchangers are devices that are commonly used to transfer heat between two or more fluids at different temperatures. They are used in a wide variety of applications, e.g. refrigeration and air conditioning systems, power engineering and other thermal processing plants.

One of the major contributors to the depletion of the ozone layer is hydro chlorofluorocarbon refrigerants used in the refrigeration and air conditioning industry. More compact equipment with higher system operating efficiency for air conditioning equipment has been investigated following the changes in efficiency standards. Refrigerant mixtures with enhanced surfaces have been developed as an alternative solution to replace hydro chlorofluorocarbon refrigerants. Accurate methods for the determination of the thermal and fluid-dynamic behaviour of new refrigerants need to be researched in order to improve the efficiency of heat exchangers. To the design and develop of new equipment, the usage of a numerical simulation can be an alternative technique besides experimental investigation. Because of the multidimensionality of the two-phase flow, analytical and numerical methods present rather limited solutions, while on the other hand, two-phase flow through tubes can be treated assuming a one-dimensional flow. One-dimensional analysis involves empirical knowledge of the shear stress, heat flux, and two-phase flow structure. Determination of the heat transfer coefficient is a significant value to obtain for accurate solutions.

In this paper, studies on in-tube condensation using smooth and enhanced tubes are intensively reviewed since two-phase flow in tubes is the most challenging phenomenon in the heat exchanger systems. All effective possible research subjects of in-tube condensation were classified generally according to the tube orientation (horizontal, vertical, and inclined tubes) and tube geometry (smooth and enhanced tubes). Detailed information on the in-tube condensation studies of heat transfer, pressure drop, flow pattern, void fraction, and refrigerants in the literature were given. This paper mentions not only the new enhancement techniques of heat transfer, but also includes some information on the new refrigerants. Finally, it is expected to be the pioneer source as an intensive literature review for in-tube condensation processes.

2. Condensation heat transfer inside tubes

Heat exchangers using in-tube condensation have great significance in the refrigeration, automotive and process industries. Effective heat exchangers have been rapidly developed due to the demand for more compact systems, higher energy efficiency, lower material costs and other economic incentives.

Enhanced surfaces, displaced enhancement devices, swirl-flow devices and surface tension devices improve the heat transfer coefficients in these heat exchangers.

2.1. Tube orientation

2.1.1. Horizontal tubes

Condensation inside horizontal tubes is important in the chemical process and power industries. Shell side condensation is rarely preferred to tube side condensation when the coolant is air or a process gas, or when the condensing refrigerant is at high pressure, dirty or corrosive. For tube side condensers, the horizontal orientation is most commonly applied.

Dobson and Chato [1] investigated condensation of zeotropic refrigerants over the wide range of mass flux in horizontal tubes. They stated that heat transfer coefficient increases with increasing the mass flux and quality in annular flow due to increased shear stress and thinner liquid film than in other flow regimes. They used a two-phase multiplier approach for annular flow. Sweeney and Chato [2] extended their model for R407C, using mass flux based modification.

Cavallini et al. [3] presented a theoretical analysis of the condensation process and a critical review of a number of correlations for predicting the heat transfer coefficients and pressure drops for refrigerants condensing inside various commercially manufactured tubes with enhanced surfaces. Recently, Cavallini et al. [4] reviewed the most recent work in open literature on the condensation inside and outside smooth and enhanced tubes.

Recently, Wongwises and Polsongkram [5] compared the average heat transfer coefficient of condensation process in the helically coiled concentric tube-in-tube heat exchanger with that in the straight concentric tube-in-tube heat exchanger at the same condition and found that the average heat transfer coefficient in the helically coiled concentric tube-in-tube heat exchanger is 33–53% higher, while the pressure drop is 29–46% higher.

2.1.2. Vertical–Inclined tubes

Condensation inside vertical tubes occurs usually in nuclear reactors. In the design of passive containment cooling systems (PCCS), existence of non-condensable gases inside condensed steam is an important technical problem. These kinds of condensers have almost 50 mm diameter vertical stainless steel tubes immersed in a tank of water under atmospheric-pressure conditions outside the containment.

Experimental studies on the heat transfer coefficient of reflux condensation and gravity controlled cocurrent flow in vertical tubes were compared with each other in the ESDU data item [6]. Correlations of gravity controlled cocurrent flow can be used for the reflux situation due to similarity of the data in the ESDU data item [6] according to the different ranges of the condensate film Reynolds number such as Nusselt [7] (Re: 67.5), Kutateladze [8] (7.5 < Re < 400), and Labuntsov [9] (Re > 400), respectively.

Under the assumptions of Nusselt's theory [7], Hassan and Jakob [10] conducted analytical analysis on the laminar film condensation of saturated vapours on the outside of inclined circular cylinders. They used experimental results of the heat transfer coefficient for the cocurrent condensation of steam inside an inclined tube to compare with their analytical results. The analytical results were 28–100% lower than the experimental results due to the rippling of the condensate film, which is not evaluated in their model. Besides this, Fieg [11] developed an analytical solution which includes surface tension effects to obtain the local film thickness on the outside of an inclined elliptical tube using Nusselt-type condensation.

2.1.2.1. Cocurrent downward flow. Heat transfer and pressure drop characteristics of refrigerants have been studied by a large number of researchers, both experimentally and analytically, mostly in a horizontal straight tube. The study of the heat transfer and pressure drop of CFCs inside a small diameter vertical tube for downward condensation has received comparatively little attention in the literature. In addition to this, there are few studies on condensation of R134a during downward flow in vertical micro-fin tubes. Briggs et al. [12,13] have used large diameter tubes of approximately 20.8 mm with CFC113. Shah's correlation [14] has been compared by researchers commonly for turbulent condensation conditions, and is considered to be the most comparative condensation model for the annular flow regime in a tube.

Nusselt [7] proposed the first theoretical solution for predicting heat transfer coefficients. He assumed a linear temperature profile through a laminar film flowing downwards without entrainment on a vertical plate. Waves and an interfacial shear effect between the phases were not considered. Under these conditions, it is possible for the Nusselt-type analysis to be used for convective condensation in round tubes.

Moreover, Rohsenov [15] and Dukler [16] developed a model which successfully predicts momentum transfer for turbulent film flow. In addition to this, Levich's model [17] and Blangetti's method [18] have been used in the literature to estimate the local heat transfer coefficients of the film in the high mass flux region.

Rohsenov [15] and Dukler [16] developed a model to predict momentum transfer for turbulent film flow. Additionally, Levich [17] and Blangetti et al. [18] used a model to estimate the local heat transfer coefficients of the film. An empirical relationship between the Fanning friction factor and vapour Reynolds number for an annular flow regime was proposed by Bergelin et al. [19]. They studied the pressure drop of air water and several organic vapours for the downward turbulent flow through a vertical 25.4 mm i.d. tube. Their diagram was used by several researchers, such as Blangetti et al. [18] and Maheshwari et al. [20], for various refrigerants. The diagram takes account of the effect of mass transfer by including a correction factor developed by Bird et al. [21], considering the effect of suction in condensation. Krebs and Schlunder [22] investigated mass transfer coefficients in the turbulent gas and film flow of a vertical condenser tube in the presence of non-condensing gases. Kuhn et al. [23] and Peterson et al. [24] investigated local heat transfer from condensation in the presence of non-condensable gases inside a vertical tube connecting a passive containment cooling system (PCCS). They studied the degradation factor method, diffusion layer theory and mass transfer conductance model. Recently, Maheshwari et al. [20] followed the same path and adopted the model, using the analogy between heat and mass transfer, to the PCCS in nuclear reactors. In their investigation they considered non-condensable gas with a wide range of Reynolds numbers. The film waviness effect on the gas/vapour boundary layer, the suction effect due to condensation, and the developing flow and property variation of the gas were also considered in their study. Local film heat transfer coefficients were multiplied by a factor of 1.28 for the wave effect between the phases in the high mass flux region. In addition to this, Oh and Revankar [25] studied the vertical passive condenser (PCCS) for complete condensation in nuclear reactors. They used a similar analysis model as [15–24]. Their model includes a modified Nusselt theory with a McAdams correction factor of 1.2 [26], a modified Blangetti model [18] and an interfacial shear effect from the Couette flow analysis [27] for small film Reynolds numbers and small interfacial shear conditions.

Carey [28] studied on the solution of convective condensation in round tubes during an annular flow regime. He applied Nusselt's theory [7] with interfacial shear stress, added new simplified equations, and offered an iterative technique for the computation of interfacial shear and the determination of the local heat transfer coefficients at the end. He assumed constant thermo physical properties of the refrigerant for condensation, and noted that the pressure drop along the test tube was small. Besides this, he stated that his analysis was not appropriate for full or partial turbulent film flow.

Wongwises et al. [29] used Carey's [28] theoretical model for 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 the interfacial shear stress, along with the modified Nusselt model with McAdams correction factor [26] and the classical Nusselt model [7], were compared with the experimental data. Comparisons with laminar flow at low mass flux data 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 affects the condensation process of R134a in a vertical smooth tube. The classical Nusselt theory and the McAdams heat transfer coefficient, which includes the wave effect between the phases, overestimate the data.

2.1.2.2. Reflux flow. Reflux condensation occurs when the vapour phase of refrigerant enters the condenser tube at the bottom and flows upward, while the condensate of refrigerant flows downward countercurrently to the vapour of the refrigerant by means of gravitational force. The phenomenon of flooding, in other words, the onset of flooding or the flooding point, limits the reflux condensation, and it occurs when some part of the condensate is carried upward by the maximum vapour velocity, which is known as the flooding vapour velocity. Maximum flooding vapour velocity and the maximum liquid mass flux occur at the vapour inlet in the event of reflux condensation. Most of the experimental flooding studies have been performed under adiabatic conditions, which can not be made directly analogous with those under reflux condensation conditions such as the air water combination in relatively large vertical tube diameters.

Mouza et al. [30] studied flooding and used Wallis correlation [31] for the mixture of air water in a 7 mm i.d. vertical and inclined 0.8 m long glass tube.

English et al. [32] proposed a correlation which is well known in process industry to predict the flooding point in reflux condensers.

Wang and Ma [33] offered semi-empirical correlation for vertical and inclined thermosyphons. They emphasised the independency of optimum inclination angle where the maximum heat transfer coefficient was obtained from working conditions.

2.2. Tube geometry

2.2.1. Smooth tube

Performance of smooth tubes has been determined by many researchers with pure refrigerants as operating fluids. Generally, empirical methods have been offered to compute the condensation heat transfer coefficients in horizontal smooth tubes. Most of these proposed models are modifications of the Dittus and Boelter's single-phase forced convection correlation [34], and were modified in most of proposed models by for instance Akers et al. [35], Cavallini and Zecchin [36], and Shah [14].

Eckels and Pate [37] made comparison with R134a and R12 on the condensation heat transfer in an 8 mm i.d. smooth tube. According to their experimental analysis, the condensation heat transfer coefficient for R12 was 25–35% lower than that for R134a. Additionally, it should be noted that the condensation heat transfer coefficients decrease with decreasing the saturation temperature but increase with the mass flux of the refrigerants.

Torikoshi and Ebisu [38] condensed R134a, R32, and R134a/R32 mixture for the comparison with R22 in an 8.7 mm i.d. smooth tube. The condensation heat transfer coefficients of R134a and R32 were 65–10 higher than for R22 respectively. In addition to this, the pressure drop of R134a was higher than that for R22, while on the contrary, the pressure drop of R32 was lower than that for R22. The condensation heat transfer coefficient for the mixture of R134a and R32 was lower than that for R22, while the pressure drop was higher than that for R22.

2.2.2. Enhanced tubes

The engineering cognisance of the need to increase the thermal performance of heat exchanger, thereby effecting energy, material and cost savings, as well as a consequential mitigation of environmental degradation, has led to the development and use of many heat transfer enhancement technique. In general, enhancement techniques can be divided into two groups: namely active and passive techniques. The active techniques require external forces, e.g. electrical field, acoustic or surface vibration. A heat transfer enhancement technique utilising "electrohydrodynamic" (EHD) can be achieved by utilising the interaction between the electrical field and fluid flow in a dielectric fluid medium. This interaction can result in an increase of fluid motion, which leads to a higher heat transfer coefficient. Electro-convection is a phenomenon in which a previously quiescent fluid will start moving in a certain direction when a strong electric field is applied to the dielectric permittivity of the fluid. Heat transfer enhancement using the EHD technique, especially during in-tube condensation, has received little attention. The 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, the term is named as compound enhancement.

Improvements on condensation heat transfer in horizontal tubes have been the subject of significant concern in the design and operation of air conditioning and refrigeration systems. Royal and Bergles [39] and Luu and Bergles [40] have studied several enhancement techniques, such as rough surfaces and twisted-tape inserts, while on the other hand, micro-fin tubes have recently been used intensively because of their high condensation heat transfer performance and moderate pressure drop.

2.2.2.1. Micro-fin tubes. 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 the heat transfer in both single-phase and two-phase applications, and are one of the most efficient and common heat transfer enhancement mechanism for the heat exchangers due to their superior heat transfer performance.

The heat transfer performance of the tubes is increased in an effective manner by the presence of the micro-fins on the internal wall surface of the horizontal tubes. Table 1 shows the tube geometries according to researchers [41–71], fin type, average tube diameter (d), fin pitch (p), fin height (h), fin space (b), fin tip thickness (t), apex angle (γ), helix angle (β), number of fins (n), tube thickness (w), and augmentation ratio (A_{mf}/A_s). The detail of Miyara et al.'s [42] test section is shown in Fig. 1.

Many experimental investigations have been performed to decide the effects of fin geometry, tube diameter, refrigerant, etc. on the condensation heat transfer and pressure drop performance of the micro-fin tubes. The presence of the micro-fins inside the tube enhances the heat transfer providing improved and 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.

This enhancement has been denoted by many researchers, for instance Cavallini et al. [72], Yu and Koyama [73], and Kedziersky and Goncalves [74]. Cavallini et al. [72] stated that heat transfer enhancement from 80% to 140% is achieved, with an increase in pressure drop from 20% to 80% due to micro-fin tubes in comparison with equivalent smooth tubes under the same operating conditions. Analogously, Yu and Koyama [73] pointed out that the heat transfer enhancement in a horizontal micro-fin tube is twice that of a smooth tube with the same inner diameter due to the increase in heat transfer area. Kedziersky and Goncalves [74] tested R134a, R410A, R125, and R32, and demonstrated heat transfer enhancement using their correlation due to the fins behaving as a surface roughness in micro-fin tubes.

The most common passive heat transfer enhancement technique nowadays for condensers is the use of helical micro-fin tubes. Liebenberg et al. [75], Liebenberg and Meyer [76] conducted experiments using 8.9 mm i.d. helical micro-fin tubes, and found that heat transfer coefficient of micro-fin tubes twice that of a smooth tube.

The herringbone tube, consisting of a double chevron-shape with embossed micro-fins was developed as a new generation of micro-fin tube in the 1990s. Olivier et al. [77] showed the difference in liquid distribution with helical micro-fin tubes. According to their experimental results, the distribution of liquid at the top inside the tube is lower than that at the bottom, due to gravity, especially at low velocities. Miyara et al. [42] found a heat transfer enhancement, as shown in Fig. 2, up to 350%, and showed the

Table 1
Tube geometries for condensation inside micro-fin tubes made of copper.

Researchers	Fin type	d_o (mm)	p (mm)	l (mm)	b (mm)	t (mm)	γ (°)	β (°)	n	w (mm)	A_{mf}/A_s
Miyara et al. [41,42]	Helical	7	0.41	0.21	0.2	0.07	41	18	50	0.25	1.83
Islam and Miyara [43]	Herringbone	7	0.34	0.22	0.22	0.05	18	16	60	0.25	2.19
	Helical	7	0.41	0.21	0.2	0.07	41	18	50	0.25	1.81
	Herringbone	7	0.35	0.22	0.23	0.05	18	16	58	0.25	2.15
	Herringbone	7	0.35	0.18	0.15	0.09	33	14	59	0.27	1.83
	Herringbone	7	0.33	0.17	0.18	0.09	20	8	62	0.22	1.93
	Herringbone	7	0.35	0.16	0.16	0.1	33	14	58	0.27	1.74
	Herringbone	7	0.4	0.2	0.25	0.1	15	28	50	0.28	2.02
	Herringbone	7	0.33	0.14	0.17	0.12	17	8	61	0.28	1.78
Cho and Tae [44]	–	9.52	8.53	–	0.2	–	–	18	60	–	1.51
Eckels and Pate [45]	–	9.52	–	0.2	–	–	50	15	60	0.4	–
Eckels et al. [46,47]	–	9.52	–	0.2	–	–	50	18	60	0.3	–
Eckels and Tesene [48]	–	12.7	–	0.2	–	–	50	17	60	0.4	–
	–	9.52	–	0.2	–	–	50	17	60	0.3	–
	–	9.53	–	0.203	–	–	51	18	60	0.305	–
	–	15.88	–	0.305	–	–	45	27	60	0.605	–
	–	7.94	–	0.203	–	–	57	18	50	0.3	–
Nualboonrueng et al. [49]	–	9.52	–	0.2	–	–	52.45	18	60	0.7	–
Uchida et al. [50]	–	7	–	0.163	–	–	40	18	60	0.3	–
Goto et al. [51–53]	Helical	8	–	0.28	–	–	32	37	–	–	–
	Cross micro-fin	8	–	0.28	–	–	29.6	23.17	–	–	–
	Helical	8.01	–	0.17	–	–	55	18	55	0.27	1.498
	3d micro-fin	8.01	–	0.2	–	–	55	18	–	–	–
	Herringbone	8	–	0.24	–	–	18	18	60	–	2.032
	–	9.52		0.17	–	–	50	25	–	–	–
	–	6.35		0.14	–	–	50	16	–	–	–
Shinohara and Tobe [54]	–	9.52	–	0.12	–	–	35	7	60	0.3	–
	–	9.52	–	0.2	–	–	50	25	60	0.3	–
	–	9.52	–	0.2	–	–	90	25	60	0.3	–
	–	9.52	–	0.12	–	–	35	7	65	0.3	–
	–	9.52	–	0.2	–	–	50	25	65	0.3	–
Hitachi [55]	–	9.5	–	0.2	–	–	40	17	60	0.28	–
	–	9.52	–	0.21	–	–	53	18	60	0.29	–
Graham et al. [56]	Helical	9.52	–	0.18	–	–	–	18	60	–	1.62
	Axial grooved	9.52	–	0.18	–	–	–	0	60	–	1.62
Bogart and Thors [57]	–	9.53	–	0.203	–	–	50	18	60	0.33	–
Ebisu and Torikoshi [58]	–	7	–	0.18	–	–	40	18	50	0.3	–
Wang and Honda [59]	–	9.5	0.53	0.24	–	0.004	30.5	20	47	–	1.49
Wang et al. [60]	–	10	0.42	0.16	–	0.027	19.9	18	60	–	1.52
	–	9.5	0.44	0.19	–	0.015	22.3	18.7	60	–	1.51
	–	7	0.39	0.21	–	0.019	19.1	18	50	–	1.71
	–	7	0.34	0.19	–	0.018	13.1	18	60	–	1.78
	–	7	0.4	0.22	–	0.037	12.7	12	50	–	1.83
	–	7	0.39	0.15	–	0.017	22.1	15	50	–	1.48
	Helical	7	0.4	0.24	–	0.021	15.6	13	50	–	1.87
	Helical	9.5	0.45	0.2	–	0.01	23.7	18	60	–	1.52
Tang et al. [61]	–	9.52	–	0.2	–	–	15	0	60	0.36	–
	–	9.52	–	0.2	–	–	40	18	72	0.36	–
Schlager et al. [62,63]	–	9.52	0.44	0.2	–	–	50	18	60	0.4	1.5
	–	9.52	1.21	0.38	–	–	10	30	21	0.5	1.8
	–	9.52	–	0.2	–	–	40	18	60	0.3	–
	–	9.52	–	0.15	–	–	40	25	60	0.3	–
Yasuda et al. [64]	–	9.52	–	0.2	–	–	40	18	60	0.3	
	–	9.52	–	0.25	–	–	40	18	60	0.3	
	–	7.94	–	0.2	–	–	30	18	50	0.3	
	–	7.94	–	0.25	–	–	30	18	50	0.3	
Muzzio et al. [65,66]	–	9.52	–	0.15	–	–	40	18	54	0.3	–
	–	9.52	–	0.23	–	–	90	25	65	0.34	–
Chamra et al. [67]	–	15.9	–	0.2	–	–	–	15	60	–	–
	–	15.88	–	0.35	–	–	30	15	74	0.5	–
	–	15.88	–	0.35	–	–	30	27	80	0.5	–
Nozu and Honda [68]	–	9.53	–	0.24	–	–	63	20.05	47	0.2	1.49
	–	9.5	–	0.16	–	–	87	24	65	0.19	1.32
	–	10	0.42	0.35	–	0.05	27.5	18	60	–	–
Koyama et al. [69,70]	–	7.52	–	0.19	–	–	–	19	50	–	1.62
	–	9.52	–	0.18	–	–	25	25	70	0.33	1.67
	–	9.52	–	0.15	–	–	–	30	60	0.6	–

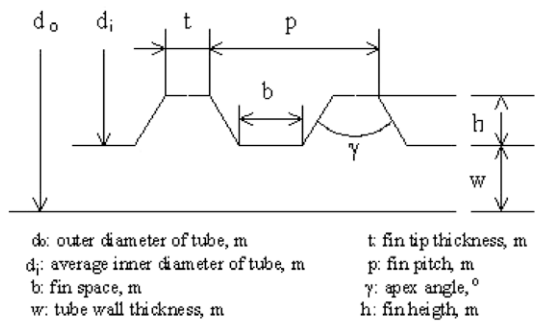

Fig. 1. Cross-section of fin [From Miyara et al. [42], with permission from Elsevier.]

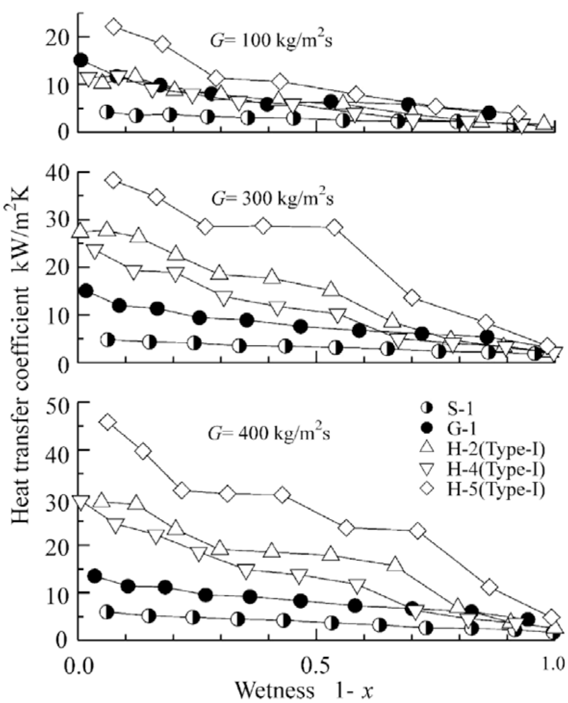

Fig. 2. Effect of helix angle on heat transfer coefficients. [From Miyara et al. [42], with permission from Elsevier.]

minor effect of tube orientation on the heat transfer and pressure drop and reported that the effect of fin height and helix angle are important. They noted that heat transfer and pressure drop increase with increasing helix angle. According to their analysis, heat transfer and pressure drop increase using micro-fin tubes whose fin heights are up to 0.18 mm, on the other hand, they found negligible heat transfer enhancement with increased pressure drop using micro-fin tubes whose fin heights are greater than 0.18 mm.

Palen et al. [78] focused on the complexity of flow characteristics of liquid in horizontal micro-fin tubes due to the effect of surface tension, gravitational force and vapour shear force. Combined surface tension and gravitational forces help the draining of condensate that has occurred on a fin surface into the spiral groove, then the vapour shear force assists in driving the condensate through the groove in the downstream direction. In a high mass flux region, the effect of gravitational force may not be important, on the other hand, in a low mass flux region the gravitational force can cause stratification of condensate in the lower part of the tube.

Khanpara et al. [79] tested CFC113 in a total of eight kinds of 9.5 mm o.d. micro-fin tubes, and the best performing tubes according to their studies had 65 fins. Schlager et al. [80] conducted experimental studies with three 9.52 mm o.d. and three 12.7 mm o.d. micro-fin tubes, and showed the best performing tubes. Hori and Shinohara [81] used 20 micro-fin tubes with different diameters ranging from 4 to 12.7 mm o.d. to decide the best performing tube for each diameter. Haraguchi [82] tested three different refrigerants to show the axial variation of the heat transfer enhancement factor in a micro-fin tube. He also derived a correlation on the local frictional pressure gradient.

Nualboonrueng et al. [83] investigated the two-phase heat transfer coefficients of pure HFC-134a condensing inside horizontal smooth and micro-fin tubes experimentally. According to their results, the average heat transfer coefficient for the 9.52 o.d. micro-fin tube is 10–85% higher than that for the smooth tube.

2.2.2.2. Micro-channel tubes. In view of the growing trend in the industry for better heat transfer performance, compact heat exchangers have been developed not only to reduce their size using micro-channel tubes, but also untapped applications such as high ambient air conditioning, hazardous duty, portable personal cooling devices and medical devices will benefit from a fundamental understanding of condensation at the micro-scales.

Feng and Serizawa [84] noted the importance of surface forces over those of body forces, surface characteristics and the interactions between the fluid and the wall due to increase in surface area using micro-channels. In addition to this, gravitational forces are dominated by surface tension and viscous forces in micro-channels. It should be noted that, according to observations in the literature, two-phase flow phenomena and flow regime characteristics in micro-channels are quite different from in the conventional larger diameter tubes. This is due to changes in the relative magnitudes of gravity, shear, and surface tension forces. The flow regime is determined by surface tension forces according to the combination of liquid and vapour-phase velocities. For these reasons, the correlations of large round tubes such as in Lockhart and Martinelli [85], Chisholm [86], Traviss et al. [87], and Shah [14] to determine the heat transfer coefficient and pressure drop may not be suitable for micro scale condensation with their large error bands. Besides this, heat transfer coefficient and pressure drop depend indirectly on local vapour quality. The relation between two-phase flow pattern and its effect on heat transfer and pressure drop should be known in order to achieve accurate design of heat exchangers for condensation. Determination of flow regime based correlation is important for that reason.

Recently, Park et al. [88] reported that R290 has a higher condensation heat transfer coefficient than R22 and R134a in an aluminium multi-channel flat tube. Guo and Anand [89,90] studied the prediction of condensation heat transfer coefficient related with two-phase flow regimes inside a rectangular channel using R410A in a continuing study.

2.2.2.3. Corrugated tubes. The heat transfer coefficient can be increased by means of corrugation inside the tube in turbulent flow by mixing the flow boundary layer, and also by increasing the turbulence level of the fluid flow. Corrugated tubes may be used in the production of shell-and-tube industrial heat exchangers. The usage of corrugated tubes instead of smooth tubes can reduce the size of these heat exchangers. Several researchers used corrugated tubes for heat transfer enhancement in the literature. Mimura and Isozaki [91] investigated the heat transfer and pressure drop characteristics of corrugation in terms of the shapes of corrugation with different relative depths and relative pitches. Ganeshan and Rao [92] focused on the heat transfer and friction factor characteristics of single and multi-start spirally corrugated tubes. Asako and Nakamura [93] analysed the heat transfer and pressure drop characteristics of corrugated ducts with rounded corners in terms of three corrugation tube angles and four aspect ratios numerically.

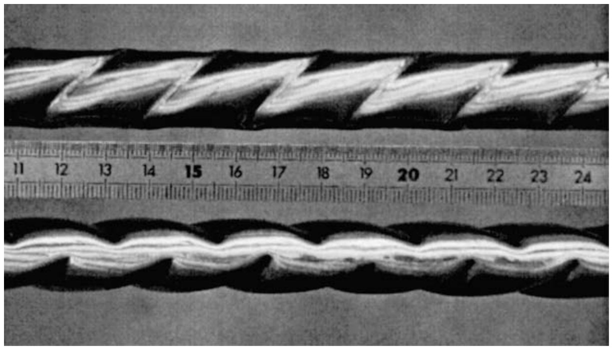

Fig. 3. Front and side view of the corrugated tube. [From Barba et al. [95], with permission from Elsevier.]

Dong et al. [94] investigated the turbulent friction and heat transfer characteristics of four spirally corrugated tubes using various geometrical parameters. They found that the spirally corrugated ribs enhance the heat transfer, but not as greatly as the increases in friction. Barba et al. [95] studied the heat transfer and pressure drop in a corrugated tube experimentally, as shown in Fig. 3, during single-phase flow at moderate Reynolds numbers ($100 < Re < 800$) using ethylene glycol as a testing fluid. The Nusselt number of the corrugated tube increased in comparison to smooth tube, while the friction factor increased by a factor of 2.45–1.83. Rainieri and Pagliarini [96] investigated the thermal performances of axial symmetrical and helical corrugated tubes with different pitch values for the enhancement of the convective heat transfer. The Reynolds number's range was between 90–800. They found that the helical corrugation causes important swirl components, to which, however, an equally important heat transfer enhancement is not associated.

Zimparov [97] performed the most productive studies on the extended performance evaluation criteria for enhanced heat transfer surfaces. Zimparov [98,99] investigated heat transfer and isothermal friction pressure drop results of spirally corrugated tubes combined with five twisted tape inserts with different relative pitches, and found that the friction factor and heat transfer coefficients belonging to these tubes were higher than those from the smooth tube. Zimparov [100,101] developed a simple mathematical model to predict the friction factor and heat transfer coefficient during fully developed turbulent flow using a spirally corrugated twisted tape inserted tube. Vicente et al. [102,103] performed experiments on the mixed convection heat transfer and isothermal pressure drop in corrugated tubes in cases of laminar, transition and turbulent flows. According to their results, the Nusselt numbers of these enhanced tubes are 30% higher than those for the smooth tube at high Rayleigh numbers, however, the friction factors were 5–25% higher than those for the smooth tube.

2.2.2.4. Tube with wire insert. Heat transfer enhancement techniques can improve the heat transfer duty or thermal performance of heat exchangers. Passive enhancement techniques for in-tube enhancement during two-phase and single-phase flows are currently the majority of commercially available ones in several heat transfer applications, for example, heat recovery processes, air conditioning and refrigeration systems, chemical reactors, and food and dairy processes. Helical wire inserts, coiled wire inserts, internal threads, corrugated tubes, and twisted tape inserts cause turbulence and/or swirl flow as rough surfaces which can incur significant pressure drop penalties. These rough surfaces, apart from the wire inserts, are not common applications in refrigeration systems due to their pressure drop penalties which can be greater than the heat transfer enhancements.

High-profile fins, helical micro-fins, annular offset strip ribbon fins, and inter-secting fins are commonly used as extended surfaces, and these techniques are more preferred in the refrigeration industry than the passive techniques mentioned above.

Prasad and Shen [104,105] tested 12 different wire-coil inserts during turbulent flow and offered a new criterion for the determination of passive heat transfer enhancement. Ravigururajan and Bergles [106] proposed some correlations to obtain friction factor and heat transfer coefficient during single-phase turbulent flow in internally enhanced tubes. Kang et al. [107] studied flooding in a fluted tube with a twisted tape insert and proposed a correlation

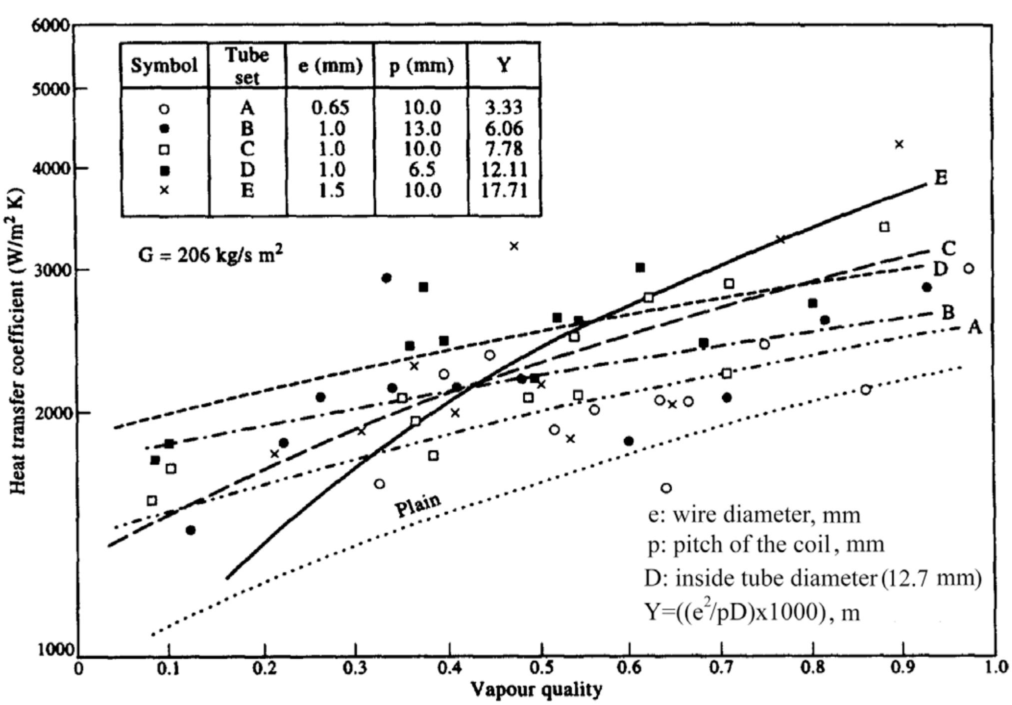

Fig. 4. Comparison of heat transfer coefficients for tubes with-coiled wire inserts. [From Agrawal et al. [109], with permission from Elsevier].

for the effects of the twisted tape insert and the angle of inclination. Al-Fahed et al. [108] used plain, micro-fin, and twisted tape insert tubes to compare the pressure drop and heat transfer coefficients. Agrawal et al. [109] tested forced convection condensation of R-22 experimentally to study the heat transfer enhancement inside horizontal coiled wire inserted tubes shown in Fig. 4. Liao and Xin [110] focused on the heat transfer and friction characteristics of water, ethylene glycol, and ISO VG46 turbine oil flowing inside passive enhanced tubes. Kim et al. [111] conducted visualisation experiments for the determination of the flow pattern, void fraction and slug rise velocity during counter-current two-phase flow in a vertical round tube with wire coil inserts. Wang and Sund [112] examined the heat transfer enhancement techniques in heat exchangers. Rahai and Wong [113] performed experiments to examine the turbulent jets in coil inserted round tubes. Zimparov [100,101] developed a simple mathematical model to determine heat transfer coefficients and friction factors, and validated it with experimental data in a spirally corrugated tube combined with a twisted tape insert during fully developed turbulent flow. Ozceyhan [114] used a finite-difference scheme to solve the energy and governing flow equations on the conjugate heat transfer and thermal stress in a wire coil inserted tube.

3. Flow pattern of condensation

Observed two-phase flow patterns in a condensation process which tends to wet the top of the tube wall in all flow regimes are slightly different from adiabatic or evaporating conditions.

The orientation and interaction of the liquid and vapour phases inside the tubes is one of the most significant characteristics of two-phase flow. This phenomenon is related to flow regime and flow pattern. Different flow patterns may occur depending on the tube position, the geometry of the tube, flow rates and physical properties of the two phases. Generally, a set of phases includes bubble flow, slug flow, churn flow, annular flow, and droplet flow for most of the significant liquid vapour flow regimes.

Annular flow conditions along the tube length include convective condensation which occurs for many applications inside tubes. Annular two-phase flow is one of the most important flow regimes, and is characterised by a phase interface separating a thin liquid film from the gas flow in the core region. Two-phase annular flow occurs widely in film heating and cooling processes, particularly in power generation, and especially in nuclear reactors. This flow regime has received the most attention, both analytically and experimentally, because of its practical importance and the relative ease with which analytic treatment may be applied. In addition to this, condensate distribution inside the tube wall is almost symmetric, and there is high velocity vapour flow in the core during annular flow.

Stratified flow occurs at very low vapour velocities in a horizontal tube, in other words, when the mass flux of refrigerant decreases. In this situation, the condensate is seen on the upper portion of the tube wall and driven downward by gravity, and collects at the bottom of the tube.

There are other flow patterns, such as annular-mist flow with a mixture of vapour and mist in the core flow; slug flow exists when interfacial waves grow sufficiently to block the entire cross-section at some transversal sections; and wavy flow occurs when the waves affect the vapour and exists on portion of the tube wall near the interface between the liquid pool and the vapour. Also, there are some subcategories of these flow patterns in relation to the transition between phases. Investigations of flow regime maps for condensation inside tubes have been conducted by many researchers, including Dobson et al. [115], Baker [116], Traviss and Rohsenow [117], Mandhane et al. [118], Taitel and Dukler [119], Palen et al. [78], Breber et al. [120], and Soliman [121,122]. Moreover, visualization of flow transitions is done by Liebenberg and Meyer [76] and their study can be seen from Fig. 5.

Although most of these flow regime maps were worked up for adiabatic two-phase flows, they are often used for the diabatic processes of evaporation and condensation. For that reason, reliable results may not be produced in some applications. Taitel and Dukler [119] provided a good review for a description of flow regime transition theory.

3.1. Horizontal tubes

It is possible to divide flow patterns into two groups in horizontal tubes: those that appear at high void fractions ($\alpha > 0.5$), and those that appear only at low void fractions ($\alpha < 0.5$). The first group has five flow patterns: stratified flow, wavy flow, wavy-annular flow, annular flow, annular-mist flow. The second group has three flow patterns: slug, plug, and bubbly flow. The five flow patterns shown in Fig. 6 occur gradually with increase in the vapour velocity. The three flow patterns in the second group appear with an increase in the liquid inventory (or a decrease in α).

There are numerous proposed flow pattern maps in the literature to predict two-phase flow pattern transitions in horizontal

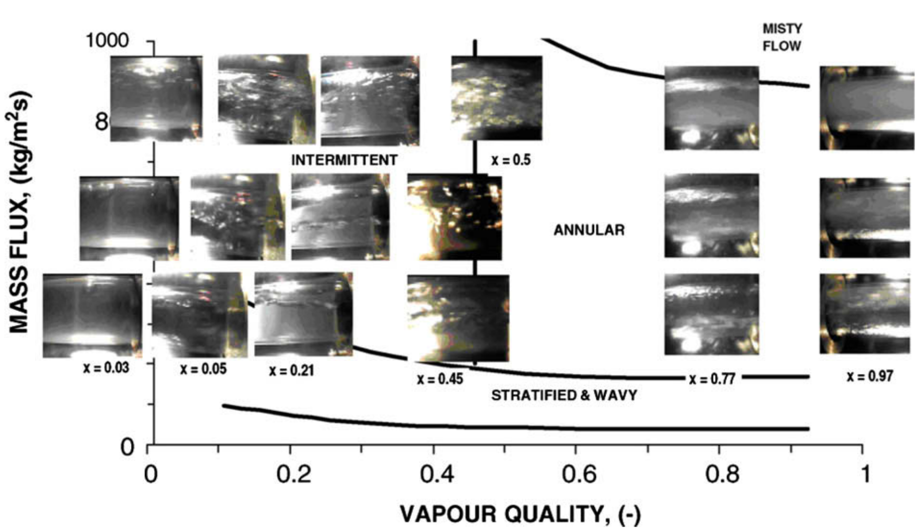

Fig. 5. Video images of condensing R134a at various mass fluxes in a smooth tube. [From Liebenberg and Meyer [76], with permission from Elsevier].

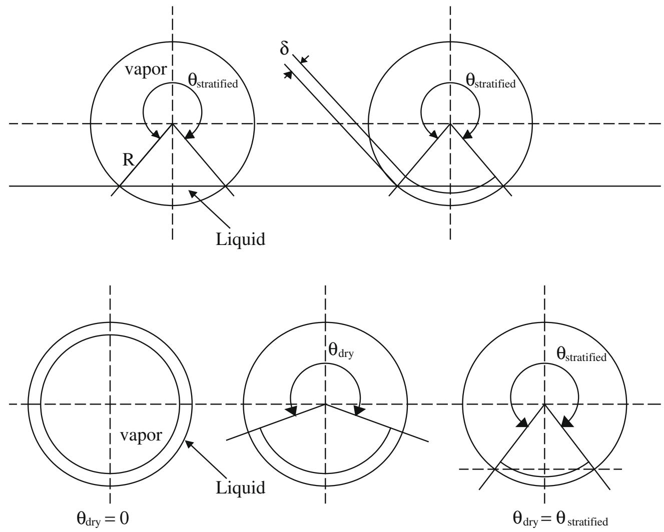

Fig. 6. Simplified flow structures for two-phase flow patterns. [From Thome [128], with permission from Elsevier].

tubes under adiabatic and diabatic conditions, and several flow regime based heat transfer models proposed by Shao and Granryd [123] and Cavallini et al. [124]. El Hajal et al. [125] adapted Kattan et al.'s [126] flow-boiling two-phase flow pattern map for condensation inside horizontal tubes. El Hajal et al. [125] and Collier and Thome [127] studied the prediction of void fractions by a new method based on flow regime at pressures between the atmosphere and near the critical pressure, and presented a new heat transfer model using HFC and HFC fluids. Thome [128] modified Steiner's [129] map, which in turn benefited from Taitel and Dukler's [119] map.

3.2. Vertical tubes

Several recognisable flow structures are seen through the distribution of the liquid gas phases for cocurrent upflow of the phases in a vertical tube such as bubbly flow, slug flow, churn flow, annular flow, wispy annular flow, mist flow.

Vertical upward gas–liquid two-phase annular flow is encountered in several industrial applications, including the flow of refrigerants in air conditioning and refrigerating systems, the flow of oil and gas in petroleum industries, and the flow of steam in power plants, e.g. in emergency core cooling (ECC) systems of a nuclear reactor during the postulated loss of coolant accidents (LOCA). Annular two-phase flow is one of the most important flow regimes, and is characterised by a phase interface separating a thin film from the gas flow in the core region. Because of its practical importance and the relative ease with which analytic treatment may be applied, this flow regime has received the most attention both analytically and experimentally. In this flow regime, it is generally true that due to the break up of the disturbance wave, part of a liquid phase is entrained as droplets into the gas core. It is also accepted that mass, momentum and energy transfers are strongly affected by entrainment of the droplets to the gas core.

The most widely recommended flow pattern maps for vertical tubes are Fair [130] and Hewitt and Roberts [131].

4. Refrigerants

Since the depletion of the earth's ozone layer has been discovered, many corporations have been forced to find alternative chemicals to chlorofluorocarbons (CFCs). Because the thermophysical properties of HFC-134a are very similar to those of CFC-12, refrigerant HFC-134a has been receiving support from the refrigeration and air-conditioning industry as a potential replacement for CFC-12 since the Kyoto protocol [132] in 1997. However, even though the difference in properties between the two refrigerants is small, it may result in significant differences in the overall system performance. Therefore, the properties of HFC-134a should be studied in detail before it is applied.

The use of natural refrigerants such as hydrocarbons is one of the possible solutions to avoid CFCs, HCFCs and HFCs. In spite of the prohibition of flammable hydrocarbon refrigerants few decades ago in normal refrigeration and air-conditioning applications due to a safety concerns, according to Kruse [133] and Jung et al. [134]some of the flammable refrigerants have been used in the certain applications. Recently, isobutane (R600a), propane (R290) and propylene (R1270) have been used for such heat transfer applications as in refrigerators, freezers, and heat pumps. Besides this, R429A and R432A including dimethyl ether (DME, RE170), are proposed as alternatives for R134a and R22.

Koyama et al. [69] developed a correlation for the condensation heat transfer coefficient using a new zeotropic mixture of HCFC22/CFC114 in a horizontal micro-fin tube.

Cavallini et al. [135] focused on heat transfer degradation due to new refrigerant mixtures emphasising the importance of mass transfer diffusion, sensible heating effects, and non-equilibrium effects in two-phase flow. They condensed R134a, R236ea and R410A

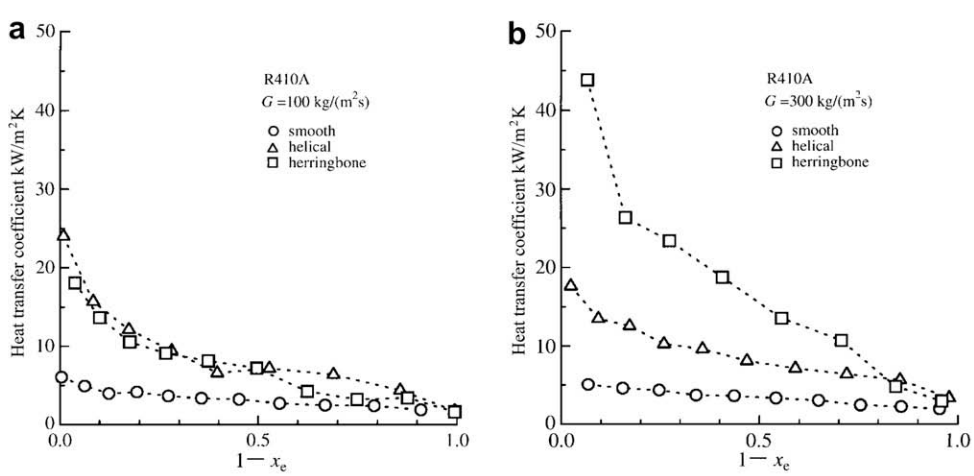

Fig. 7. Varation of heat transfer coefficient of R410A with wetness. [From Miyara et al. [41], with permission from Elsevier].

Table 2
Void fraction models and correlations.

Void fraction model/correlation	Model/correlation
Homogeneous model	$\alpha_H = \dfrac{1}{1+\left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_l}\right)S} \quad S=1$
Zivi's model [149]	$\alpha = \dfrac{1}{1+\left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_l}\right)^{2/3}}$
Rigot's correlation [150]	$\alpha = \dfrac{1}{1+\left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_l}\right)S} \quad S=2$
Smith's model [151]	$\alpha = \left\{1+\left(\dfrac{\rho_g}{\rho_l}\right)K\left(\dfrac{1}{x}-1\right)+\left(\dfrac{\rho_g}{\rho_l}\right)(1-K)\left(\dfrac{1}{x}-1\right)\left[\dfrac{\left(\frac{\rho_l}{\rho_g}\right)+K\left(\frac{1}{x}-1\right)}{1+K\left(\frac{1}{x}-1\right)}\right]\right\}^{-1} \quad K=0.4$
Levy's correlation [152]	$x = \dfrac{\alpha(1-\alpha)+\alpha\sqrt{(1-2\alpha)^2+\alpha\left[2\left(\frac{\rho_l}{\rho_g}\right)(1-\alpha)^2+\alpha(1-2\alpha)\right]}}{\left[2\left(\frac{\rho_l}{\rho_g}\right)(1-\alpha)^2+\alpha(1-2\alpha)\right]}$
Fauske's correlation [153]	$\alpha = \left(1+\left(\dfrac{1-x}{x}\right)\left(\dfrac{\rho_g}{\rho_l}\right)^{0.5}\right)^{-1}$
Thom's correlation [154]	$\alpha = \dfrac{\gamma x}{1+x(\gamma-1)}$ $\gamma = Z^{1.6} \quad Z = \left(\dfrac{\rho_l}{\rho_g}\right)^{0.555}\left(\dfrac{\mu_g}{\mu_l}\right)^{0.111}$
Baroczy's model [155]	$\alpha = \left[1+\left(\dfrac{1-x}{x}\right)^{0.74}\left(\dfrac{\rho_g}{\rho_l}\right)^{0.65}\left(\dfrac{\eta_l}{\eta_g}\right)^{0.13}\right]^{-1}$
Chisholm's correlation [86]	$\alpha = \dfrac{1}{1+\left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_l}\right)S}$ $S = \left(1-x+\dfrac{x\rho_l}{\rho_g}\right)^{1/2}$
Madsen's correlation [156]	$\alpha = \left(1+\left(\dfrac{1-x}{x}\right)^{b}\left(\dfrac{\rho_g}{\rho_l}\right)^{-0.5}\right)^{-1}$ $b = 1+\log\left(\dfrac{\rho_l}{\rho_g}\right)\left(\log\left(\dfrac{1-x}{x}\right)\right)^{-1}$
Ahren's correlation [157]	$P.I.2 = \left(\dfrac{\mu_l}{\mu_g}\right)^{0.2}\left(\dfrac{\rho_g}{\rho_l}\right)$ P.I.2 numbers versus slip ratio for Ahren correlation can be obtained from their article

Table 2 (*continued*)

Void fraction model/correlation	Model/correlation
Spedding and Spence correlation [158]	$\alpha = \left(1 + 2.22\left(\frac{1-x}{x}\right)^{0.65}\left(\frac{\rho_g}{\rho_l}\right)^{0.65}\right)^{-1}$
Chen's correlation [159]	$\alpha = \left(1 + 0.18\left(\frac{1-x}{x}\right)^{0.6}\left(\frac{\rho_g}{\rho_l}\right)^{0.33}\left(\frac{\mu_l}{\mu_g}\right)^{0.07}\right)^{-1}$
Lockhart and Martinelli correlation [85]	$\frac{1-\alpha}{\alpha} = 0.28\left(\frac{1-x}{x}\right)^{0.64}\left(\frac{\rho_g}{\rho_l}\right)^{0.36}\left(\frac{\mu_l}{\mu_g}\right)^{0.07}$
El Hajal's correlation [125]	$\alpha = \frac{\alpha_H - \alpha_{steiner}}{\ln\left(\frac{\alpha_H}{\alpha_{steiner}}\right)}$
Turner and Wallis two-cylinder model [160]	$\frac{1-\alpha}{\alpha} = A\left(\frac{1-x}{x}\right)^{p}\left(\frac{\rho_g}{\rho_l}\right)^{q}\left(\frac{\mu_l}{\mu_g}\right)^{r}$ $\begin{array}{lllll} \text{Correlation or model} & \text{A} & \text{p} & \text{q} & \text{r} \\ \hline \text{Homogeneous} & 1 & 1 & 1 & 0 \\ \text{Zivi} & 1 & 1 & 0.67 & 0 \\ \text{Turner-Wallis} & 1 & 0.72 & 0.4 & 0.08 \\ \text{Lockhart-Martinelli} & 0.28 & 0.64 & 0.36 & 0.07 \\ \text{Thom} & 1 & 1 & 0.89 & 0.18 \\ \text{Baroczy} & 1 & 0.74 & 0.65 & 0.13 \end{array}$
Wallis correlation [31]	$\alpha = \left(1 + X^{0.8}\right)^{-0.378} \quad X = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1}$
Domanski and Didion correlation [161]	$\alpha = 0.823 - 0.157\ln(X) \quad X = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1}$
Yashar et al.'s correlation [162]	$\alpha = (1 + F_t^{-1} + X)^{-0.321} \quad F_t = \left(\frac{x^3G^2}{\rho_g^2 gd(1-x)}\right)^{0.5} \quad X = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1}$
Wilson et al.'s correlation [163]	$\alpha = (1 + a/F_t + bX)^n \quad F_t = \left(\frac{x^3G^2}{\rho_g^2 gd(1-x)}\right)^{0.5}$ $\begin{array}{lllll} \text{Tube} & X_{tt}+1/F_t & a & b & n \\ \hline \text{Smooth} & <2.00 & 1.84 & 3.11 & -0.21 \\ & >2.00 & 0.5 & 1.2 & -0.35 \\ 18^\circ\ \text{Helix} & <2.00 & 5.80 & 8.60 & -0.16 \\ & >2.00 & 1.50 & 2.70 & -0.31 \\ 0^\circ\ \text{Helix} & <2.00 & 1.38 & 3.30 & -0.26 \\ & >2.00 & 2.26 & 2.50 & -0.26 \end{array}$
Premoli et al.'s model [164]	$\alpha = \frac{1}{1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_1}\right)S} \quad S = 1 + K1\left(\frac{Y}{1+CY} - CY\right)^{1/2} \quad K1 = 1.578 + Re_1^{-0.19}\left(\frac{\rho_1}{\rho_g}\right)^{0.22}$ $C = 0.0273 We_1 Re_1^{-0.51}\left(\frac{\rho_1}{\rho_g}\right)^{-0.08} \quad Y = \frac{\beta^*}{1-\beta^*}$ $We_1 = \frac{G^2 d}{\sigma\rho_1 g} \quad Re_1 = \frac{Gd}{\mu_1} \quad \beta^* = \frac{1}{1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_1}\right)}$
Kawahara et al.'s correlation [165]	$\alpha = \frac{C_1\alpha_h^{0.5}}{1 - C_2\alpha_h^{0.5}} \quad C_1 = \begin{cases} 0.03 & \text{for } d = 100\ \mu\text{m} \\ 0.02 & \text{for } d = 50\ \mu\text{m} \end{cases}$ $C_2 = \begin{cases} 0.97 & \text{for } d = 100\ \mu\text{m} \\ 0.98 & \text{for } d = 50\ \mu\text{m} \end{cases} \quad d > 250\ \mu\text{m} \quad \alpha = \alpha_H$
Graham et al.'s correlation [56]	$F_t > 0.01032 \quad \alpha = 1 - \exp(-1 - 0.3\ln(F_t) - 0.0328(\ln(F_t))^2)$ $F_t < 0.01032 \quad \alpha = 0 \quad F_t = \left(\frac{x^3G^2}{\rho_g^2 gd(1-x)}\right)^{1/2}$
Hughmark's model [166]	$\alpha = \frac{K_H}{1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_l}\right)} \quad K_H = 0.71 + 0.0001P$
Hamersma and Hart's correlation [167]	$\alpha = \left(1 + 0.26\left(\frac{1-x}{x}\right)^{0.67}\left(\frac{\rho_g}{\rho_l}\right)^{0.33}\right)^{-1}$
Bangkoff's correlation [168]	$\alpha = [0.71 + (0.0145)P]\alpha_H$

(*continued on next page*)

Table 2 (*continued*)

Void fraction model/correlation	Model/correlation
Czop et al.'s correlation [169]	$\alpha = -0.285 + 1.097\alpha_H$
Tandon et al.'s model [170]	$50 < Re_l < 1125 \quad \alpha = 1 - 1.928\dfrac{Re_l^{-0.315}}{F(X)} + 0.9293\dfrac{Re_l^{-0.63}}{F(X)^2}$ $Re_l > 1125 \quad \alpha = 1 - 0.38\dfrac{Re_l^{-0.088}}{F(X)} + 0.0361\dfrac{Re_l^{-0.176}}{F(X)^2}$ $Re_l = \dfrac{Gd}{\mu_l}$ $X = \left(\dfrac{1-x}{x}\right)^{0.9}\left(\dfrac{\rho_g}{\rho_l}\right)^{0.5}\left(\dfrac{\mu_l}{\mu_g}\right)^{0.1}$ $F(X) = 0.015\left(\dfrac{1}{X} + \dfrac{2.85}{X^{0.476}}\right)$
Huq and Loth's correlation [171]	$\alpha = 1 - \dfrac{2(1-x)^2}{1 - 2x + \left[1 + 4x(1-x)\left(\frac{\rho_l}{\rho_g} - 1\right)\right]^{0.5}}$
Armand and Massina's correlation [172]	$\alpha = (0.833 + 0.167x)\alpha_H$
Chisholm and Laird's correlation [173]	$\alpha = 1 + \left[0.8\Big/\left(1 + \dfrac{21}{X} + \dfrac{1}{X^2}\right)\right]^{1.75}$
Steiner's correlation [174]	$\alpha = \dfrac{x}{\rho_g}\left([1 + 0.12(1-x)]\left[\dfrac{x}{\rho_g} + \dfrac{1-x}{\rho_l}\right] + \dfrac{1.18(1-x)[g\sigma(\rho_l - \rho_g)]^{0.25}}{G\rho_l^{0.5}}\right)^{-1}$
Harms et al.'s model [175]	$\alpha = \left[1 - 10.06Re_l^{-0.875}(1.74 + 0.104Re_l^{0.5})^2\left(1.376 + \dfrac{7.242}{X^{1.655}}\right)^{-1/2}\right]^2$
Chisholm and Armand's correlation [176]	$\alpha = \dfrac{1}{\alpha_H + (1-\alpha_H)^{0.5}}\alpha_H$
Armand and Treschev's correlation [177]	$\alpha = 0.833\alpha_H$

in a multiport mini-channel tube, and reported that the pressure drop of R236ea had the largest value among the tested refrigerants, while the pressure drop of R-410A was significantly lower than those of R-134a and R236ea under the same conditions.

Sami and Poirier [136] condensed and evaporated the refrigerants of R-410A, R-410B, R-507 and the mixture of R-32/125/143a/134a to determine the two-phase heat transfer coefficients and pressure drops inside enhanced-surface tubings.

Ebisu and Torikoshi [137] measured the condensation heat transfer coefficients using R-410A, R-407C and R-22 in a horizontal smooth tube, and noted that the condensation heat transfer coefficient of R-22 was slightly higher than that of R-410A, while on the other hand, pressure drop of R410A was about 30% lower than that for R-22 due to the differences in vapour density of the two refrigerants.

Wijaya and Spatz [138] tested R-22 and R-410A to compare the heat transfer characteristics of these refrigerants flowing inside horizontal smooth copper tube.

Miyara et al. [41] studied the comparison of condensation heat transfer shown in Fig. 7, and pressure drop of R-410A and R-22 using a herringbone-type and helical-type micro-fin tube, and found that both of the refrigerants' condensation heat transfer coefficient and pressure drop in the helical micro-fin tube were lower than in the herringbone-type micro-fin tube in the higher mass velocity region.

Chitti and Anand [139] investigated the condensation heat transfer coefficients of R410A and R22, and showed that the condensation heat transfer coefficient of R410A was about 15–20% higher than that of R22 flowing inside smooth horizontal tubes.

4.1. Effects of oil

Numbers of researchers have investigated the effect of refrigeration oils on condensation heat transfer. Degradation of the condensation heat transfer coefficient has been reported due to the presence of refrigeration oil. The mixture of small amounts of compressor lubricant in the refrigerant is a typical problem in vapour compression refrigeration systems. This mixture affects the performance of the condenser and influences the all system performance for that reason.

Chato [140] investigated the influence of oil on condensation using refrigerants CFC12 and HCFC22, and reported a reduction in heat transfer coefficient due to the mixture of oil-refrigerant. Tichy et al. [141] noted 10% and 23% degradation in the heat transfer of CFC 12, testing 2% and 5% 300 SUS napthenic based oil concentrations, respectively. Schlager et al. [80] stated a 13% reduction in the heat transfer with 5% lubricant and 150 SUS napthenic oil in the mixture for the condensation of HCFC 22. Eckels and Pate [37] reported a drop of approximately 10% in the heat transfer for CFC12 at 5% 150 SUS napthenic oil mixture, but on the other hand, no important effect on the heat transfer coefficient of HFC134a at 165 SUS PAG oil. Shao et al. [142–144] reported that he added ester based oil to the refrigerant at the concentration of 2% and %5 respectively. As a result of his studies, value of the condensation heat transfer coefficient of R134a decreased 10–20% due to usage of ester based oil with the refrigerant inside the condenser. Boissieux et al. [145] and Meyer and Dunn [146] focused on the oil effects on condensation heat transfer of R404A in horizontal smooth and micro-fin tubes. They reported heat transfer enhancements for R404A up to oil concentrations of 3%. Sur and Azer [147] reviewed some correlations to predict the effect of oil on the heat transfer performance of the refrigerant in smooth and micro-fin tubes. Cavallini et al. [148] condensed refrigerant-oil mixtures inside tubes, and noted that the heat transfer coefficient decreases with increasing oil concentration in all the geometries tested. Dobson and Chato [1] proposed a correlation for the increase in pressure drop in smooth tubes due to the existence of oil in the system.

5. Void fraction

Void fraction, defined as the cross-sectional area occupied by the vapour in relation to the area of the flow channel, is an important parameter and is always used to determine the flow pattern transition, heat transfer coefficient and two-phase pressure drop. Two-phase separated flow is commonly analysed using the slip flow model. In this model, it is assumed that the seperated phases have different uniform velocities. By contrast, the homogeneous model is defined as an ideal case, as it assumes a homogeneous mixture providing uniform velocities for both phases, for that reason, it is the simplest method for the determination of void fraction.

Over the years, many studies have been conducted on the modelling of void fractions, and can be divided into several groups: slip ratio void fraction models and correlations [86,149–159], Lockhart and Martinelli parameter based void fraction models and correlations [85,125,160,31,161–165], flow regime based void fraction models and correlations [56,166,167], $K\varepsilon_H$ parameter based void fraction and correlations [168–170], and general void fraction models and correlations [171–177]. A brief and comprehensive overview of these works is presented in Table 2.

However, there is lack of void fraction correlations on the condensation inside micro-fin tubes in the literature. Determination of void fraction in a micro-fin tube is an important design and operating parameter for the heat exchanger, and it is necessary to calculate the amount of refrigerant charge in the evaporator and condenser. Therefore, it still preserves its importance and it is a worthwhile subject to investigate, but there have been limited investigations in open literature until now. Yashar et al. [162] performed experimental studies on condensation and evaporation inside smooth and micro-fin tubes, and proposed a correlation to predict void fraction of R134a and R410A.

Dalkilic et al. [178,179] benefited from Table 2 to investigate the effect of void fraction models on the condensation friction factor and film thickness of R134a in vertical downward flow at high mass flux in a smooth tube.

6. Condensation pressure drop inside tubes

The two-phase pressure drop is a significant design parameter in many engineering applications, such as in the chemical process industry, nuclear industry, petroleum industry, refrigeration and air conditioning applications, and space applications. There have been a number of investigations into this subject in the literature due to its importance. The frictional, acceleration, and gravitational components form the two-phase total pressure drop in tubes. Determination of void fraction is necessary for computing the acceleration and gravitational components, and in a similar way, determination of either the two-phase friction factor or the two-phase frictional multiplier is necessary for computing the frictional component of pressure drop.

The Lockhart and Martinelli [85], Chisholm [86], and Friedel [180] correlations are generally used for the determination of pressure drop in conventional channels. Some modifications to account for the specific geometry or flow conditions are made in these correlations. In spite of their large deviations from the data for small channels in the condensation process, they are still used as the basis for many of the recent correlations.

6.1. Smooth tube

6.1.1. Vertical tubes

Heat transfer and pressure drop characteristics of refrigerants have been studied by a large number of researchers, mostly in horizontal tubes. Generally, empirical methods are most often used to compute the pressure drop during condensation by using pure refrigerants as working fluids. The study of the pressure drop of refrigerants during downward condensation in small diameter vertical tubes has received comparatively little attention in the literature. A brief summary of pressure drop studies of downward condensation is given as follows:

Goodykoontz and Dorsch [181] investigated the local condensation heat transfer coefficients and pressure distribution of R113 for the mass fluxes of 21–455 $kg\,m^{-2}\,s^{-1}$ in 7.4–15.9 mm i.d. vertical tube. Kim and No [182] developed a turbulent film condensation model including pressure drop for high pressure steam in 46 mm i.d. vertical tube. Ma et al. [183] studied the two-phase friction factors of downward R113 flow for mass fluxes from 400 to 800 $kg\,m^{-2}\,s^{-1}$ in 20.8 mm i.d. smooth and finned tube.

Akers et al. [35] developed a two-phase multiplier which assumes that two-phase flows are similar to single-phase flows. Their correlation predicts frictional two-phase pressure drop by means of a multiplying factor, which is the same rationale as the Lockhart-Martinelli multiplier [85]. Their model is known as 'equivalent Reynolds number model'. It can be used for an annular flow regime which can be replaced by an equivalent all liquid flow. According to this model, the equivalent all liquid flow produces the same wall shear stress as that of the two-phase flow. Several latest researchers have used this model, for example, Ma et al. [183] whose study can be seen from Fig. 8 and Moser et al. [184].

Dalkilic et al. [185] discussed and investigated the effects of various relevant parameters on the condensation pressure drop such as condensing temperature, condensation temperature difference, vapour quality and mass flux. They developed a new correlation of the two-phase friction factor, determined using the equivalent Reynolds number model from a large amount of data.

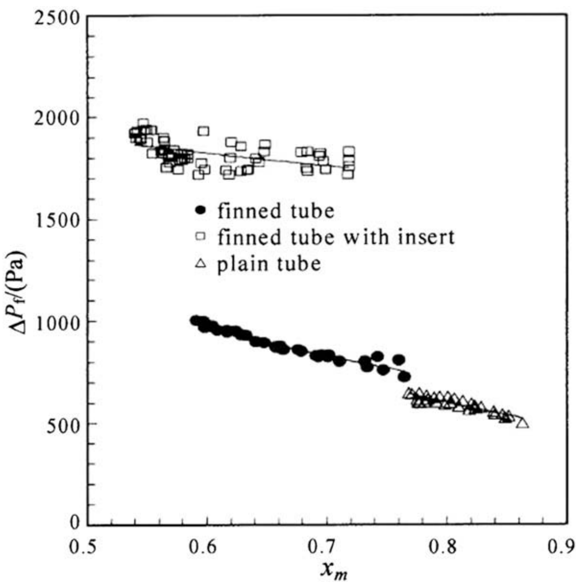

Fig. 8. Friction pressure drop for various tube configuration. [From Ma et al. [183], with permission from Elsevier].

6.1.2. Inclined tubes

According to the ESDU Data Item [5], pressure recovery can be ignored in calculations of the total pressure drop in the event of reflux condensation. There are not enough studies on pressure drop for the case of reflux condensation in inclined, small diameter tubes during reflux condensation in the literature. Russell [186] investigated the pressure drop of steam in a 5 m long, 19.8 mm i.d. tube at a slope of 57° to the horizontal for reflux condensation. Schoenfeld and Kröger [187] reported a study on the pressure drop of steam during reflux condensation, using 7 m long elliptical inclined tube at 60° to the horizontal. Stephan [188] condensed R12 for the pressure drop of an adiabatic falling film flow during reflux condensation in a vertical rectangular channel. Thumm [189] focused on the friction factor for the reflux flow of steam in saturated and adiabatic conditions using 28.2 mm i.d. vertical tube. Chen et al. [190] proposed a better correlation than the Lockhart–Martinelli correlation [86] for their closed two-phase thermosyphon system. Abdelmessih et al. [191] used Chen et al.'s [190] correlation for the prediction of pressure drop in a vertical shell-and-tube condenser. Brauer [192] determined the friction factor of a falling film proposing a calculation method during reflux flow. Thumm [189] used Brauer's [192] correlation to predict the friction factor during reflux flow, and pointed out that, according to his data, it is more suitable than correlations proposed by Hadley [193] or Andreussi [194].

6.1.3. Horizontal tubes

Condensation of refrigerants R12, and R22 with a small fraction of lubricant inside smooth tubes has been used in automobiles and residential air-conditioners for almost 60 years. For that reason, the pressure drop of R12 and R22 inside smooth tubes has been investigated commonly. Nowadays, new condensers have been designed to work with alternative refrigerants to replace R12 and R22 inside enhanced tubes due to ozone crises and efficiency requirements.

The pressure drop in smooth tubes during condensation of refrigerants has been studied by a large number of researchers as a comparative value with enhanced ones mentioned several times in the paper.

6.2. Enhanced tubes

The usage of helical horizontal micro-fin tubes is the most common passive enhancement device for condensers in use nowadays due to their high heat transfer performance and moderate increase in pressure drop. Determinations of the heat transfer and pressure drop have major significance in design practice. Inaccurate calculation of condenser pressure drop can affect not only pumping power consumption, but also importantly the heat transfer performance, due to the relationship with the local condensing temperature and pressure of refrigerant.

Numerous researches have been conducted on condensation in micro-fin tubes, as comprehensively reviewed by Newell and Shah [195], Cavallini et al. [148], Liebenberg et al. [75], Haraguchi et al. [196], Kedzierski and Goncalves [197], Nozu et al. [198], Goto et al. [52], and Choi et al. [199]. They proposed correlations on the basis of different experimental conditions for prediction of pressure drop. Liebenberg and Meyer [76] presented an increase of about 200% on heat transfer coefficient in an 8.9 mm i.d. helical micro-fin tube, compared to that of a smooth tube. However, a large increase in pressure drop of about 100% compared to a smooth tube was observed due to increased vapour velocities; in other words, increased turbulence inside the tube compared to a smooth tube. Haraguchi et al. [196] investigated pressure drop of R123, R134a and R22 in a micro-fin tube, and proposed a correlation based on their experimental data. Kedzierski and Goncalves [197] condensed R32, R125, R134a and R410A in a micro-fin tube and modified Pierre's [200] correlation to develop friction factor to take account of the fin effect on the flow. Cavallini et al. [148] modified Friedel's [177] correlation using an equivalent roughness to take account of the effect of micro-fins. Nozu et al. [169] proposed an annular flow model based on three micro-fin tubes and four refrigerants, R11, R123, R22 and R134a, to take account for the shear stress at the condensate surface and geometrical parameters. Newell and Shah [195] modified Souza and Pimenta's [201] correlation to develop a method for pressure drop using the pressure drop penalty factor. Goto et al. [52] condensed and evaporated refrigerants R22 and R410A in a helical micro-fin tube and a herringbone micro-fin tube and proposed two correlations. Choi et al. [199] modified Pierre's [200] correlation to determine pressure drop for condensation of R32, R125, R134a and R410A and for evaporation of R32, R125, R134a, R410A, R22, R407C, and R32/R134a in a micro-fin tube. They proposed a correlation that is applicable to evaporation and condensation in smooth and micro-fin tubes for lubricant-free refrigerants and refrigerant/lubricant mixtures.

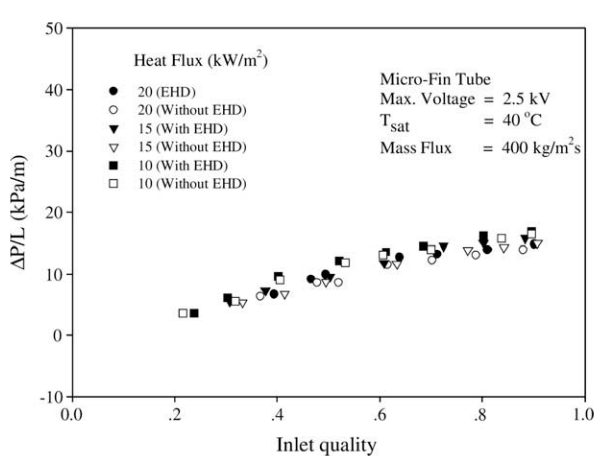

Fig. 9. Friction pressure drop for various tube configuration. [From Ma et al. [183], with permission from Elsevier].

Recently, Laohalertdecha et al. [202,204,205], Nualboonrueng and Wongwises [203] investigated the heat transfer enhancement using EHD technique and pressure drop in smooth and micro-fin tubes shown in Fig. 9. Their results show that the maximum heat transfer enhancement and pressure drop are about 1.15% and 50%, respectively.

7. Conclusions

This review has considered heat transfer and pressure drop investigations during in-tube condensation. Almost all possible research subjects have been summarised on the case in the literature, such as condensation heat transfer and pressure drop studies according to the tube orientation (horizontal, vertical, inclined tubes) and tube geometry (smooth and enhanced tubes), flow pattern studies of condensation, void fraction studies, and refrigerants with the effect of oil.

In-tube condensation of refrigerants is a crucial event in many applications, such as condensers used in air conditioning, refrigeration, and heat pumps. Struggles with depletion of the stratospheric ozone layer and global warming caused by chemical compounds commenced relatively recently. To this end, condenser producers have been trying to change working fluids and to use enhanced geometries in the process of considering energy efficiency. These kinds of improvements will continue to take attention and will have an important impact on the HVAC industry in the future.

The review indicates that numerous works have reported the heat transfer characteristics for both vertical, inclined and horizontal surfaces. Authors strongly believe that the study of condensation heat transfer mechanism is still unlimited. Researchers who are willing to commence the study of condensation in refrigeration and heat transfer systems will benefit greatly from this paper.

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