International Journal of Heat and Fluid Flow 21 (2000) 449–455

International Journal of HEAT AND FLUID FLOW

www.elsevier.com/locate/ijhff

Bubbly boiling of environment-friendly refrigerating media

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Received 15 October 1999; accepted 29 February 2000

Abstract

The paper describes results of investigations of heat transfer and pressure drop during bubbly boiling of new environment-friendly refrigerating media. Taking advantage of the experimental results, an analytical model of heat transfer and pressure drop is worked out. This continuum model draws on conservation equations for energy, momentum and mass of the two-phase one-component (liquid–vapour) mixture. In the light of engineering applications, a one-dimensional model is used, where physical quantities are cross-section averaged. Numerical calculations performed with the aid of an experiment-based correlation allow us to obtain the continuous distribution, along the channel, of quantities that characterise the process of bubbly boiling. The comparison of computational results and experimental data, own and from other authors, shows that the proposed method of determination of heat transfer coefficients and pressure drop can be applied to a number of refrigerating media in a wide range of heat and flow parameters, assuring accuracy to ±20%. Therefore, the method can be serviceable in the design of heat exchangers for refrigeration units as well as complete refrigeration units.

Keywords: Bubbly boiling; Heat transfer; Pressure drop; Experimental investigations; Environment-friendly refrigerant; Analytical model

1. Introduction

As it is widely known, refrigerating media from the group of CFCs and HCFCs have a destructive influence on the ozone layer and essentially contribute to the extension of the greenhouse effect. Decisions reached in the Montreal Protocol in 1987 and during subsequent meetings of the involved parties up to the VII Conference in Vienna in 1995 have found their way to legislation of the EC countries and also in Poland. CFC compounds are practically eliminated from the equipment newly manufactured. The elimination of HCFC compounds will proceed in a longer time horizon. Among substitutes for freon R12, a refrigerating medium R134a is getting popularity. Freon R22 is replaced by R404A, whereas R502 is superseded by R507.

The correlations, well established and widely recognised in the world literature relevant to the subject, to calculate the heat transfer coefficients and pressure drop have been validated experimentally, in general to good effect, for refrigerating media from the group of CFCs and HCFCs used so far. However, further research works are in demand to investigate the heat and flow characteristics for new ecological refrigerants (Bohdal, 1997; Bohdal et al., 1998; Eckeis et al., 1994; Hambraeus, 1995).

The aim of the presented fundamental investigations of new environment-friendly refrigerants is to acquire information concerning heat transfer and pressure drop during saturated bubbly boiling in channel flows through horizontal and vertical channels. The obtained information should allow the construction of experiment-based correlations and a semi-empirical model for the above-mentioned range of boiling, and should also enable the comparison with correlations worked out by other investigators.

2. Experimental investigations

Fundamental experimental investigations were carried out at a specially designed test facility erected at the Department of Thermomechanics and Refrigerating Engineering of the Technical University of Koszalin (Bohdal, 1997, 1998; Bohdal et al., 1999).

The test facility consists of the following main elements:

1. investigation system, comprising test sections and instrumentation,
2. supply system to feed the installation with the refrigerant,
3. electrical installation,
4. installation for cooling water, and
5. measuring and control apparatus with computer recording and processing of data.

A schematic of the test facility is presented in Fig. 1.

There are two test sections with instrumentation. The test sections are made in the form of one vertical and one horizontal straight channel – pipe stretches of length 0.66 m each,

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0142-727X/00/$ - see front matter
PII: S0142-727X(00)00026-6

Notation

A	channel cross-sectional area, coefficient
Bo	boiling number
C	flooding circumference
D	channel diameter
f	flow number
Fr	Froude number
g	acceleration of gravity
h	specific enthalpy
Ku	Kutateladze number
L	channel length
$\dot{m}$	mass flow rate
m	coefficient
n	normal, coefficient
Nu	Nusselt number
p	pressure
q	heat flux density
r	heat of evaporation
Re	Reynolds number
S	slip
t	time
T	temperature
x	dryness fraction
$w\rho$	mass flux density
w	velocity
z	channel axial coordinate

Greeks

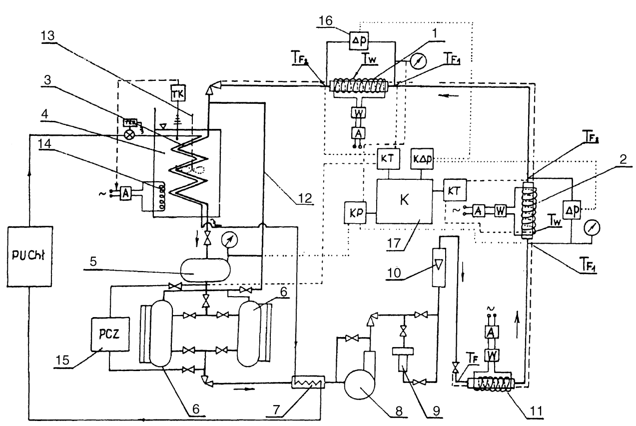
α	heat transfer coefficient
β	pipe slope with respect to the horizontal plane
φ	static void fraction
λ	heat conductivity
μ	dynamic viscosity
ν	kinematic viscosity
ρ	density
σ	surface tension

Subscripts

cr	critical
exp	experimental
f	fluid
h	constant enthalpy
p	constant pressure
R	equilibrium
s	saturation
th	theoretical
w	wall
1	inlet
2	outlet

Superscripts

$'$	liquid
$''$	vapour



Symbols:
PUChl auxiliary refrigeration unit,
W watt meter
A autotransformer
Pcz refrigerant flow rate meter
Δp pressure difference transducer
KP pressure transducer interface
KΔp pressure difference transducer interface
KT temperature transducer interface
K computer with accesories

Fig. 1. A schematic of the test facility: 1. horizontal test section with instrumentation, 2. vertical test section with instrumentation, 3. condenser, 4. coolant vessel, 5. refrigerant vessel, 6. measuring vessel, 7. subcooler, 8. pump, 9. filter, 10. rotameter, 11. preheater, 12. pressure equalising pipe, 13. stirrer, 14. heating element, 15. electronic flow meter of *Sonoflo* type, 16. electronic pressure difference transducer, 17. computer connected with transducers for measurements of temperature, pressure and pressure difference.

with a circular cross-section of inner diameter 13 mm. Pipes are made of copper with the wall thickness equal to 1.5 mm. The measuring sections are electrically heated. The inlet and outlet of the experimental channel are made transparent in the form of glass pipes of length 0.25 m and inner diameter 13 mm to enable observations of the structure of the flowing two-phase mixture.

The feed station is equipped with the following elements: refrigerant condenser 3, coolant vessel 4, refrigerant vessel 5, measuring vessels 6, subcooler 7, pump 8, filter 9, rotameter 10, refrigerant preheater 11. The flow of the refrigerant in the cycle is driven by a gear pump. The maximum volumetric flow rate of the refrigerant is equal to 440 l/h (122×10^{-3} m^3/h). The mass flux of the refrigerant flowing through the measuring sections is controlled by throttling at the delivery side of the pump and partly by extraction of the refrigerant onto the suction side of the pump.

The circulating pump sucks the refrigerant, subcooled by a few Kelvin, from the measuring vessels to which the refrigerant flows gravitationally down from the condenser through a pressure vessel (auxiliary pipes 12 are used to equalise the pressure in the system of gravitational flow of the refrigerant). The condenser is made in the form of tube coils submerged in a vessel filled with the solution of ethylene glycol. The heat is absorbed by a compressor refrigeration system whose coil evaporator is placed in the same vessel where the coils of the condenser are submerged. The installation of the compressor refrigeration system consists of: a compressor-condensing aggregate for freon R22 with water cooling, condenser, thermostatic expansion valve, liquid subcooler, elements of refrigeration automatics. The coolant vessel is equipped with a stirrer 13 and heating elements 14 to assure the required temperature of the coolant. The change of heating power of the elements 14 is achieved by means of an autotransformer and contact thermometer. The possibility of control of the coolant temperature in the vessel of the condenser enables the stabilisation of the condensation temperature of the refrigerant as well as keeping the pressure in the condenser and in the test section constant. In order to cool the refrigerant a subcooler 7 is used. The preheater 11 serves to achieve the required temperature of the refrigerant at the inlet to the test section. The refrigerant passes through a filter and humidity absorber 9.

An electronic flow meter Sonoflo 15 (manufactured by Danfoss) calibrated in preliminary measurements is applied to measure the flow rate of the refrigerant. The flow rate of the medium can also be measured by means of the calibrated measuring vessels 16. The rotameter 10 plays the role of indicator of flow stabilisation. Pressure and pressure difference measurements at characteristic stations of the test sections are made by means of tensometric sensors. The wall temperature at the heated stretch of the flow channel as well as the temperature of the medium are measured with the aid of copper–constantan thermoelements, with thermoelectrodes of diameter 0.35 mm and length 2 m. Thermoelectric sensors for temperature measurements of the medium are submerged in the liquid spanning on 0.7 of the channel inner diameter. Weld joints of the thermoelements for wall temperature measurements are mounted at a depth of 0.5 mm from the outer surface of the channel.

All quantities obtained from the sensors of temperature, flow rate and pressure are converted into voltage signals and fed into the computer. The software enables recording and processing of the acquired data. The system of computer-aided experimental investigations facilitates planning of measurements, their control and processing of the obtained results.

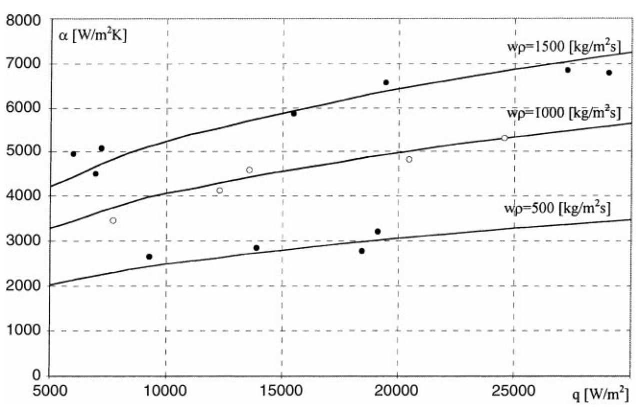


Fig. 2. The results of measurements of heat transfer coefficient in the form $\alpha = f(q)$ for $(w\rho) = \text{const}$; $T_s = -20°C$, $= 0.05$, vertical channel, R507.

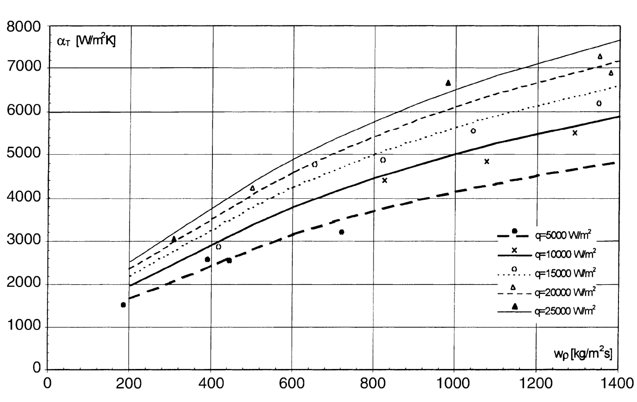


Fig. 3. The results of measurements of heat transfer coefficient in the form $\alpha = f(w\rho)$ for $q = \text{const}$; $T_s = -10°C$, $x = 0.1$, horizontal channel, R404A.

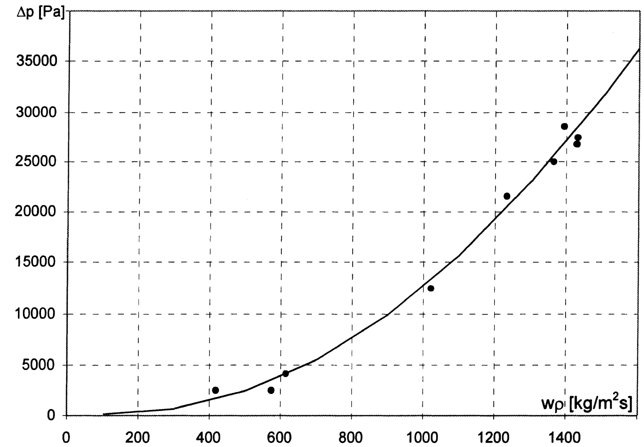


Fig. 4. The results of measurements of pressure drop in the form $\Delta p = f(w\rho)$ for $q = 15$ kW/m^2, $T_s = -20°C$, $x = 0.1$, horizontal channel, R507.

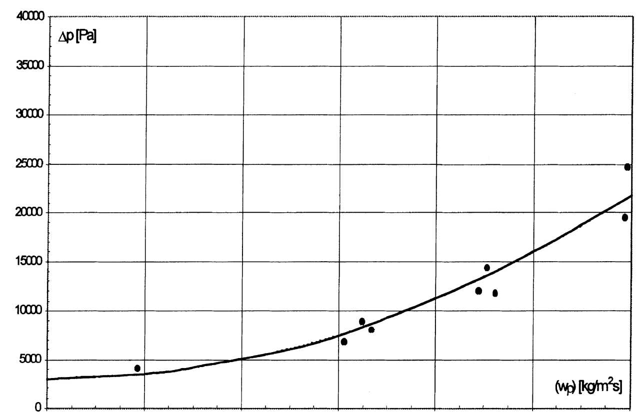


Fig. 5. The results of measurements of pressure drop in the form $\Delta p = (w\rho)$; $q = 20$ kW/m^2, $T_s = 0°C$, $x = 0.1$, vertical channel, R404A.

Experimental investigations are carried out for refrigerating media having a certificate of liquid purity from the manufacturer. A filter 9 is installed upstream of the test sections so as to remove possible foreign bodies and humidity from the refrigerant. The experiments are conducted under constant thermal and flow parameters of the medium at the inlet to the test sections. New refrigerating media: R134a, R404A and R507 are investigated. The sample results of measurements of the heat transfer coefficient and pressure drop are presented in Figs. 2–5.

Local values of the heat transfer coefficient α and pressure drop Δp are charged with some systematic and random errors. Due to a relatively small number of measurements performed at the same conditions, random errors will not be estimated here. However, the systematic errors following from the measuring and computational method can be evaluated as they do not depend on the number of measurements. The mean-square error has been estimated. In the case under consideration, the mean-square errors for the heat transfer coefficient α and pressure drop Δp, determined based on the measurements, depend mainly on the mass flux density $(w\rho)$ and heat flux density q. Bearing in mind the above, the estimation of the mean-square error has been carried out in four variants, taking into account the maximum and minimum values of the mass flux density $(w\rho)$ and heat flux density q. It has been found from the calculations that the experimental data are charged with the mean-square error of 9–10%.

3. Analysis of experimental results

The heat transfer coefficient α can be found from the Newton law

$$\alpha = \frac{q}{\Delta T}, \tag{1}$$

where q is the heat flux density at the inner wall of the channel, and ΔT is the temperature difference between the wall and liquid (in the flow core).

The temperature difference ΔT is a result of temperature gradient $(\partial T/\partial n)$ in the laminar boundary sublayer. It has been established from the review of the literature sources that the temperature gradient follows from the phenomena listed below (Bilicki, 1979; Bohdal, 1997; Chawla, 1967; Madejski, 1973; Takamatsu et al., 1993):

- the laminar boundary sublayer that forms a main part of pressure drop in the process of heat transfer;
- the effect of heat flux density q transferred from the inner surface of the channel wall to the fluid.

With the increasing distance from the inner wall, there is a change in contribution of individual mechanisms of heat transfer. Nearest to the wall, convective heat transfer prevails, and then decreases with the increasing distance, giving way to internal heat sources and the effect of turbulence in the fluid layer.

The effect of creation of vapour bubbles on the magnitude of the temperature gradient is comprised in the boiling number (Bo) as

$$\left(\frac{\partial T}{\partial n}\right) \sim Bo = \frac{ql}{\rho'' r \gamma'}, \tag{2}$$

where l is the characteristic dimension of the vapour bubble

$$l = \sqrt{\frac{\sigma}{g(\rho' - \rho'')}}. \tag{3}$$

The Kutateladze number (Ku) describes the rate of creation of the gas phase related to the actual fluid velocity along the channel axis

$$Ku = \frac{q(1-\varphi)}{r\rho'' w'(1-x)}, \tag{4}$$

where φ is the void fraction, and x is the dryness fraction.

The thickness of the laminar boundary sublayer δ depends on the Reynolds number (Re):

$$\delta \sim \frac{1}{Re} = \frac{\gamma'(1-\varphi)}{w'(1-x)D}. \tag{5}$$

The Froude number (Fr) encapsulates the effect of intensity of heat transfer during bubbly boiling in flow with respect to that of forced convection

$$Fr = \frac{(w\rho)^2}{gD\rho'^2}. \tag{6}$$

Based on the theory of flow similarity for developed bubbly boiling in flow, one can write

$$Nu = ARe^{n_1} Fr^{n_2} Ku^{n_3} Bo^{n_4}. \tag{7}$$

The exponents and the constant that appear in the above equation were found from the analysis of over 200 measuring points. The following values were established for the refrigerants R134a and R404A: $n_1 = 2.42$; $n_2 = -0.35$; $n_3 = 1.38$; $n_4 = -1.08$ and $A = 5.1 \times 10^{-5}$ for a horizontal channel and $A = 4.5 \times 10^{-5}$ for a vertical channel. The empirical correlation (12) was determined for the range of parameters as follows:

- evaporation temperature $T_S = -30 - +10°C$,
- heat flux density $q = 3000–30{,}000$ W/m^2,
- mass flux density $(w\rho) = 100–1600$ kg/m^{2}s,
- dryness fraction $x = 0–0.15$,
- void fraction $\varphi = 0–0.95$,
- criterial numbers: Ku=0.0005–0.02, Bo=5–60, Re=10,000–100,000, Fr=0.1–12.

In deriving correlation (7), it has been assumed that the dryness fraction x and void fraction φ can be linked by a relationship

$$\varphi = \left(1 + \frac{1-x}{x}\frac{\rho''}{\rho'}S\right)^{-1}, \tag{8}$$

where S is the slip found from the correlation of Huhn (1992)

$$S = 1 + 0.27\left(\frac{x}{1-x}\frac{\rho'}{\rho''}\right)^{n}\left(\frac{\rho'}{\rho''}\right)^{0.12}\left(1 - \frac{\rho''}{\rho'}\right)^{6} \times \left(1 + \frac{5}{0,1 + Fr}\right)^{m}, \tag{9}$$

$$n = 0.32\left(\frac{\rho'}{\rho''}\right)^{0.125}, \tag{10}$$

$$m = 6.7\frac{p}{p_{cr}}\left(1 - \frac{p}{p_{cr}}\right)^{4}, \tag{11}$$

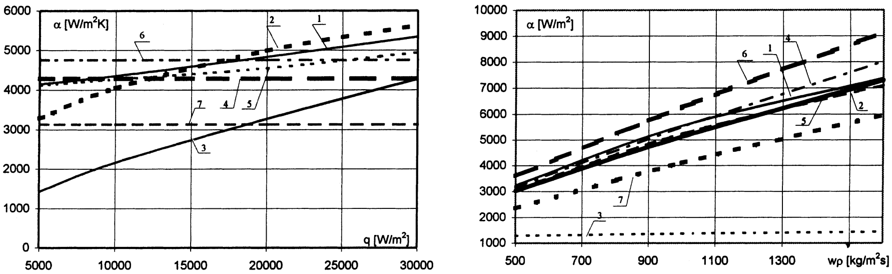
$$\left(\frac{\rho'}{\rho''}\right)^{0.42} \geqslant S \geqslant 1 + 0.05\left(\frac{\rho'}{\rho''}\right)^{0.5}. \tag{12}$$

The results of calculations from correlation (7) were compared with the experimental data showing a reasonable agreement, with discrepancies not exceeding ±20%. The comparison was also made with results of calculations based on correlations from other authors. A comparative collection of graphs obtained from calculations based on correlations of different authors for R134a and R404A is given in Fig. 6

It is worthwhile to note that the mass flux appears twice in correlation (7), as it is included in the Reynolds number Re and Froude number Fr. The Reynolds number additionally takes into account an increase of the velocity of the two-phase mixture by accounting for the increase in volume of the boiling refrigerant. This follows from the fact that the quality x and void fraction φ are introduced into the Reynolds number. In the process of elaboration of the experimental data, the author tried to work out a correlation (7) not including the Froude number. However, there were difficulties in determination of empirical coefficients (A, n_1–n_4) with a satisfactory accuracy, mainly due to the large range of variation of the quality x and void fraction φ. Another reason is that forced convection has an effect on heat transfer during under-developed bubbly boiling in channel flow. The introduction of the Froude number brought an agreement between the results of calculations and experimental data within a ± 20% accuracy margin.

4. Analytical model

I here propose a simplified description of the process of bubbly boiling in channel flow through a straight pipe. The



$w\rho$ = 1000 kg/m²s, T_s = -20 °C, x = 0.05 vertical channel

q = 15000 Wm², T_s = 0 °C, x= 0.1 horizontal channel

Fig. 6. The comparison of computational heat transfer coefficient α calculated from correlations of different authors for R507: 1. Bilicki (1979), 2. Bohdal, 3. Chawla (1967), 4. Dengler i Addoms (Madejski, 1973), 5. Mikielewicz (1974), 6. Troniewski (1977), 7. Schrock and Grossman (Madejski, 1973).

two-phase one-component (liquid–vapour) system is treated as a continuum governed by the laws of conservation of energy, momentum and mass in the form of differential equations. The continuum is characterised by parameters that describe the two-phase system, such as density of the two-phase mixture, void fraction or equilibrium dryness fraction. In the light of engineering applications, a one-dimensional model is used where physical quantities are cross-section averaged. This way the average velocity, pressure, temperature, and so on, are introduced.

4.1. Foundations of the model

1. A two-phase one-component mixture under saturation temperature T_s flows down a straight pipe inclined at a certain angle β to the horizontal plane.
2. The flow is turbulent. The mass flux density is equal to $w\rho$.
3. The mixture is characterised by the dryness fraction x and void fraction φ.
4. Heat is supplied to the channel from the surroundings. The heat flux density is constant along the pipe and equal to q_w.
5. The process of boiling with generation of vapour bubble takes place in the channel.
6. The process takes place under steady-state conditions,

$$\frac{\partial T}{\partial t} = 0.$$

4.2. Theoretical model

As mentioned in the introduction, the conservation equations to describe the boiling process in channel flow will be written in a simplified engineering-oriented one-dimensional representation (Bilicki, 1983; Bilicki et al., 1996)

- energy conservation equation

$$\rho\frac{\partial h}{\partial t} - \frac{\partial p}{\partial t} + \rho w\frac{\partial h}{\partial z} - w\frac{\partial p}{\partial z} = \frac{\tau_w w C}{A} + \frac{q_w C}{A}, \tag{13}$$

- momentum conservation equation

$$\rho\frac{\partial w}{\partial t} + \rho w\frac{\partial w}{\partial z} = \rho g\cos\beta - \frac{\tau_w C}{A} - \frac{\partial p}{\partial z}, \tag{14}$$

- mass conservation equation

$$\frac{\partial \rho}{\partial t} + w\frac{\partial \rho}{\partial z} + \rho\frac{\partial w}{\partial \rho} = -\rho w\frac{1}{A}\frac{\partial A}{\partial z}. \tag{15}$$

After introduction of the state equation in the form

$$\rho = \rho(h,p) \tag{16}$$

and subsequent transformation, the set of Eqs. (13)–(15) can be rewritten as follows:

$$\rho\frac{\partial h}{\partial t} - \frac{\partial p}{\partial t} + \rho w\frac{\partial h}{\partial z} - w\frac{\partial p}{\partial z} = \frac{\tau_w w C}{A} + \frac{q_w C}{A}, \tag{17}$$

$$\rho\frac{\partial w}{\partial t} + \rho w\frac{\partial w}{\partial z} + \frac{\partial p}{\partial z} = \rho g\cos\beta - \frac{\tau_w C}{A}, \tag{18}$$

$$D_1\frac{\partial p}{\partial t} + D_2\frac{\partial h}{\partial t} + wD_1\frac{\partial p}{\partial z} + wD_2\frac{\partial h}{\partial z} + \rho\frac{\partial w}{\partial z} = -\rho w\frac{1}{A}\frac{\partial A}{\partial z}, \tag{19}$$

where

$$D_1 = \left(\frac{\partial \rho}{\partial p}\right)_h, \qquad D_2 = \left(\frac{\partial \rho}{\partial h}\right)_p. \tag{20}$$

For steady-state conditions, the mass conservation Eq. (19) can be expressed in an integral form

$$w\rho A = m = \text{const.} \tag{21}$$

Placing Eq. (21) into Eqs. (17) and (18) gives

$$-\frac{dp}{dz} + \rho\frac{dh}{dz} = \frac{\tau_w C}{A} + \frac{q_w \rho C}{\dot m}, \tag{22}$$

$$\left(1 - \frac{\dot m^2 D_1}{A^2\rho^2}\right)\frac{dp}{dz} - \frac{\dot m^2 D_2}{A^2\rho^2}\frac{dh}{dz} = \frac{\dot m^2}{A^3\rho}\frac{dA}{dz} + g\rho\cos\beta - \frac{\tau_w C}{A}. \tag{23}$$

Eqs. (22) and (23) make up a set of ordinary differential equations that incorporate the laws of conservation of energy, mass and momentum, as well as the state equation $\rho = \rho(h,p)$. The closure equations describing the heat flux density q_w and shear stress τ_w at the inner wall of the channel are as follows,

$$\tau_w = \frac{1}{2} f \frac{\dot m^2 (1-x)^2}{\rho A^2 (1-\varphi)^2}, \tag{24}$$

where

$$A = \frac{\pi D^2}{4}, \quad f = 0.164 Re^{-0.25}, \quad Re = \frac{\dot m D(1-x)}{\rho A v (1-\varphi)}, \tag{25}$$

q_w = const. (a given quantity).

Parameters that characterise the two-phase mixture are determined from the following relationships:

- equilibrium dryness fraction x_R

$$x_R = \frac{h - h'}{h'' - h'} \tag{26}$$

- void fraction φ

$$\varphi = x_R\frac{\rho}{\rho''}, \tag{27}$$

where ρ is the density of the two-phase mixture

$$\rho = \frac{\rho' \rho''}{\rho'' + x_R(\rho' - \rho'')}. \tag{28}$$

The heat transfer at the inner wall of the channel can be described by the Newton law

$$q_w = \alpha(T_w - T_f), \tag{29}$$

where the fluid temperature T_f is equal to the saturation temperature T_s corresponding to the saturation pressure. Given the heat transfer coefficient α, one can find the temperature at the inner wall of the heated channel

$$T_w = T_s + \frac{q_w}{\alpha}. \tag{30}$$

The following quantities have to be assumed to get started with the calculations:
- geometrical dimensions of the channel,
- heat flux density at the inner wall,
- mass flux density of the boiling medium,
- a correlation that enables the determination of the heat transfer coefficient α,
- boundary conditions at the inlet section of the channel, that is the inlet specific enthalpy $h = h_1$ and pressure $p = p_1$ of the two-phase mixture.

4.3. Results of calculations

The numerical calculations were carried out based on the above-described theoretical model of bubbly boiling. In order to evaluate the heat transfer coefficient α, the empirical correlation (7) was used. As a result of calculations, a number of parameters that characterise the process of bubbly boiling in channel flow were evaluated, including the heat transfer coefficients and pressure drop.

The comparison of computational results based on the presented theoretical model and experimental data is shown in Figs. 7 and 8. The obtained results concern heat transfer expressed in terms of the Nusselt number Nu, and pressure drop in the pipe channel of inner diameter $D = 13$ mm and length $L = 0.6$ m. The calculations and experiments were carried out for new environment-friendly refrigerants R134a, R404A and R507. It was found from the comparison that the discrepancy between the computational results and experimental data does not exceed ±20% – for 95% of the collected results concerning heat transfer, and for all results referring to the pressure drop.

The presented theoretical model was also applied to calculate the flow of refrigerants R12 and R22 widely used so far. The obtained results of bubbly boiling are compared with experimental data of Schlünder, Chawla, Bankoff and Lockhart-Martinelli (Schlünder, 1975a,b). Figs. 9 and 10 exhibit a good agreement between theoretical calculations and experimental data, which implies that the presented theoretical model can also be recommended for application to other refrigerating media.

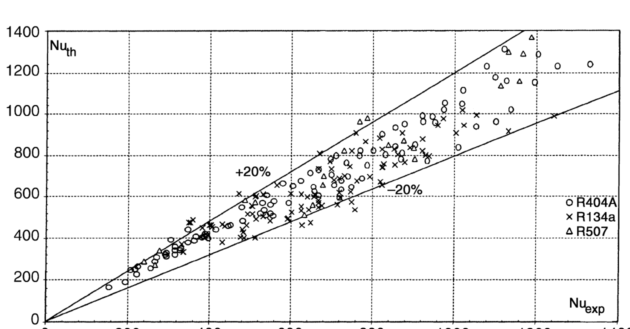


Fig. 7. The comparison of Nusselt number obtained from theoretical calculations, Nu_{th}, and determined from own experimental investigations, Nu_{exp}.

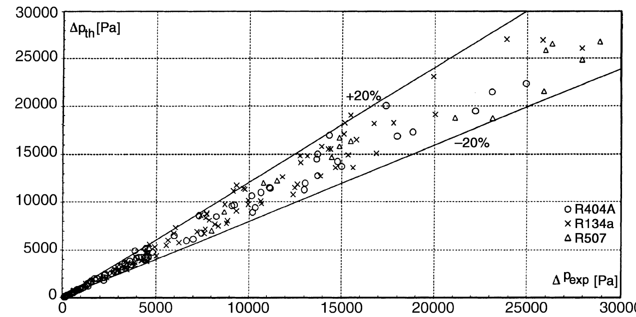


Fig. 8. The comparison of pressure drop in the pipe channel obtained from theoretical calculations, Δp_{th}, and own experimental investigations, Δp_{exp}.

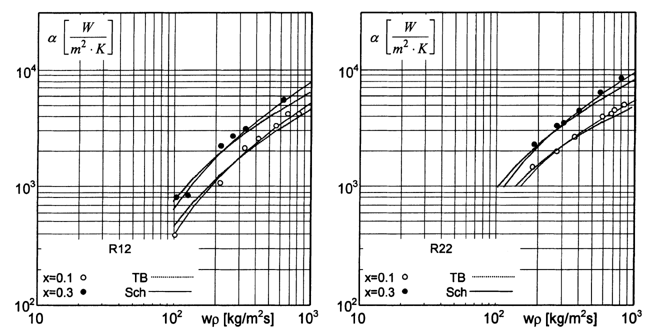


Fig. 9. The comparison of heat transfer coefficients α obtained from the proposed calculation model and experimental investigations of Schlünder (1975a,b).

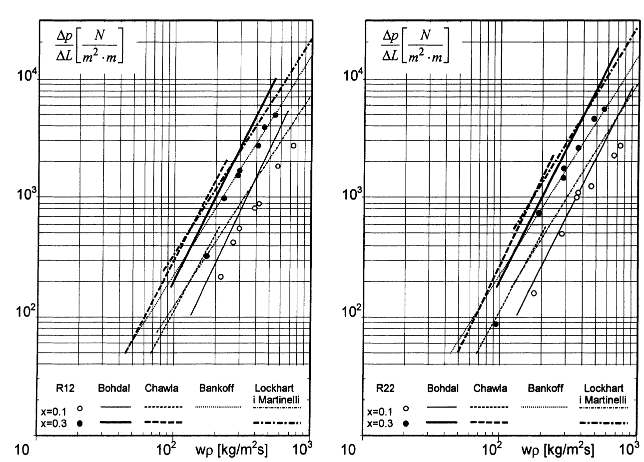


Fig. 10. The comparison of pressure drop $\Delta p/\Delta l$ obtained from the proposed theoretical model, correlations of Chawla, Bankoff, Lockhart-Martinelli and experimental investigations of Schlünder (1975a,b).

5. Conclusions

(1) Experimental investigations of subcooled and developed bubbly boiling of refrigerants R134a, 404a and R507 in horizontal and vertical pipe channels were carried out in the following range of parameters $T_s = -30°C - +10°C$, $(w\rho) = 100$ –1600 kg/m^2 s, $q = 3000$–30,000 W/m^2, $x = 0$–0.15. The results of investigations enable the elaboration of correlations to calculate the heat transfer coefficient.

(2) The presented theoretical model that draws on conservation equations for energy, momentum and mass of a one-dimensional continuum, although containing a number of simplifications, is capable of accurately describing the process of bubbly boiling of refrigerating media in channel flows. Numerical calculations performed with the aid of an experiment-based correlation (7) enable us to obtain the continuous distribution, along the channel, of quantities that characterise the process of bubbly boiling.

(3) The comparison of computational results and experimental data, own and from other authors, speaks in favour of the proposed method of determination of heat transfer coefficients α and pressure drop. The method can be applied to several refrigerating media in a wide range of heat and flow parameters, assuring accuracy to $\pm 20\%$, and therefore can be serviceable in design of heat exchangers for refrigeration units as well as complete refrigeration units.

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