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Horizontal flow boiling of R22, R134a and their mixtures with oil in smooth and enhanced tubes

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Abstract The experimental stand and procedure for flow boiling investigations are described. Experimental data for pure R 22, R134a and their mixtures with oil in two smooth tubes and two enhanced tubes are also presented. The performance benefits of the micro-fin tube and corrugated tube are quantified and discussed. During tests inlet vapour quality was set 0±0.05 and outlet quality 0.7±0.01. Mass flux density varied from about 250 to 500 kg/m^{2}s. The experiments have been conducted for average saturation temperature 0^{o}C. The ability of selected models to predict the boiling heat transfer coefficient is evaluated by comparison with the experimental data obtained in smooth tubes for pure R 22 and R134a.

Keywords: Flow boiling; Enhanced surfaces; Refrigerants; Refrigerant/oil mixtures

Nomenclature

A	–	tube cross-sectional area, m^2
Bo	–	boiling number
c	–	specific heat, kJ/(kg K)
Co	–	convection number
d	–	inside diameter, mm
D	–	outside diameter, mm
$\dot{E}_p$	–	pumping power, W
EF	–	enhancement factor
G	–	mass flux density, kg/(m^{2}s)
h	–	heat transfer coefficient, W/(m^{2}K)

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H	–	fin height, mm
k	–	thermal conductivity, W/(m K)
L	–	active length of tube, m
$\dot{m}$	–	mass flow rate, kg/s
Nu	–	Nusselt number
P	–	absolute pressure, Pa
Δp	–	pressure drop, kPa
PF	–	penalty factor
Pr	–	Prandtl number
QF	–	thermal-hydraulic enhancement factor
Re	–	Reynolds number
T	–	temperature, K
U_L	–	overall heat transfer coefficient for 1 m of tube length, W/(m K)
ΔT_m	–	logarithmic mean temperature difference, K

Subscripts

av	–	average
C	–	convective boiling
cr	–	critical
en	–	enhanced tube
lo	–	liquid only
N	–	nucleate boiling
NPB	–	nucleate pool boiling
sm	–	smooth tube
vo	–	vapour only
w	–	water
1	–	inlet
2	–	outlet
$2F$	–	two phase

1 Introduction

International regulations, reducing application of the traditional refrigerants like R22, have initiated searching of new working fluids. One of them is chlorine-free, recently widely used R134a [1-9]. Under real working conditions, in evaporators of compressor refrigerating systems, boiling of a mixture of refrigerant and lubricant occurs. Amount of oil in the blend depends on tear and wear of the compressor and other system elements. Available experimental data show, that irrespective of the refrigerant and oil type, the presence of lubricant always increases pressure drop during flow boiling [10-17]. This effect is more distinct for more dense and viscous oils. The influence on heat transfer rate is different – the presence of small amount of oil may cause an enhancement of heat transfer coefficient, but higher lubricant concentrations always inhibit heat transfer and the maximum value of the heat transfer coefficient is shifted to lower vapour quality [11-17]. In order to intensify heat transfer during flow boiling in evaporators of refrigerating systems, extended surface tubes, for example micro-fin or porous coated tubes are used more and more often [18-28]. Experimental data are available only for small range of the test conditions and for very few combinations of refrigerant, oil and internal tube surface types. So further tests of flow boiling with use of new working fluids and their lubricant mixtures in smooth and extended surface tubes are needed.

2 Experimental

2.1 Test stand

The test stand consists of four main systems: test section, refrigerant loop, heating water loop and cooling loop. The test facility is capable of determining in-tube average heat transfer coefficient and pressure drop of refrigerant over the length of a test tube. A schematic diagram of the test stand is shown in Fig. 1.

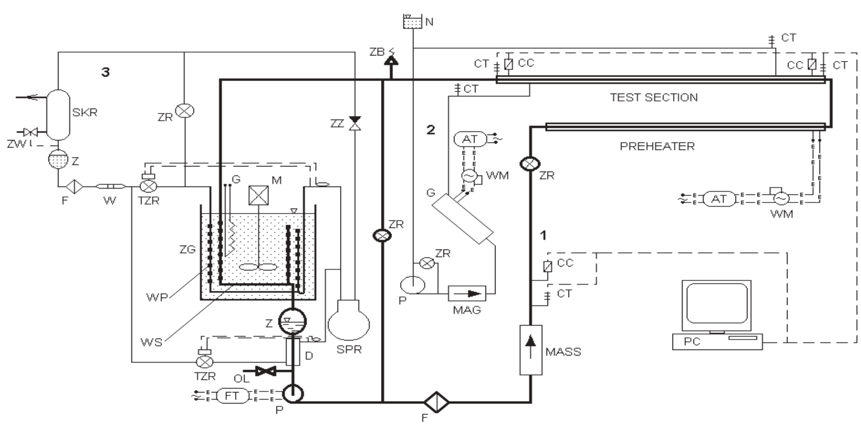


Figure 1. Test stand scheme: 1 – test loop, 2 – heating loop, 3 – cooling loop, P – pump, F – filter, Z – receiver, OL – oil valve, MASS, MAG – flowmeter, ZR – regulating valve, ZB – safety valve, WS – condenser coil, WP – evaporator coil, ZG – glycol tank, M – stirrer, G – electric heater, SPR – compressor, ZZ – check valve, SKR – condenser, ZW – water regulating valve, W – sight glass, TZR – thermostatic expansion valve, N – expansion tank, D – subcooler, CC – pressure sensor, CT – temperature sensor, PC – microcomputer –aided data acquisition system, WM – wattmeter, AT – autotransformer, FT – inverter.

2.2 Test section

The test section (Fig. 2) consists of tube-in-tube heat exchanger, sight-glasses and sensors for temperature and pressure measurement on the inlet and outlet of the section. The inner tube, i.e. the test tube, is a 2 m long horizontal tube with an outer diameter of about 10 mm. The tested tubes are specified in Table 1. The water in the annulus of 2 mm gap, surrounding the test tube, flows counter to the refrigerant flow and is used to heat the refrigerant, evaporating in the inner tube. Inlet and outlet water temperatures are measured with single thermocouples installed in thermometer wells.

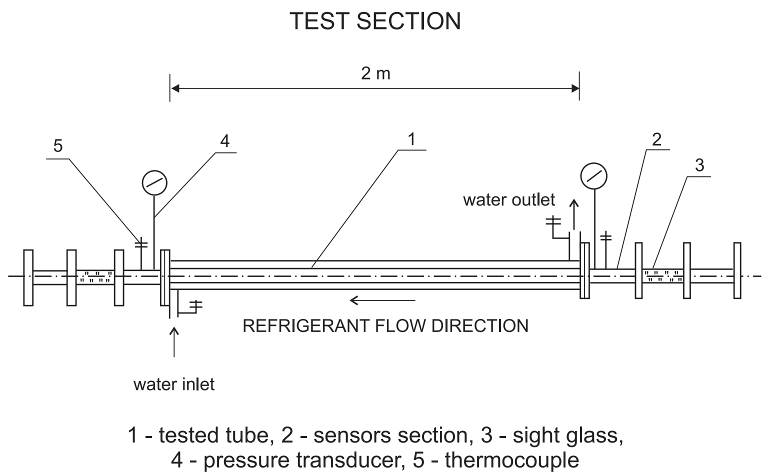


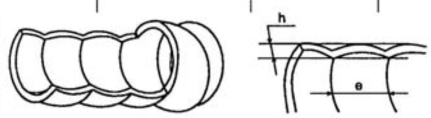
Figure 2. A detailed schematic of the test section: 1 – heat exchanger, 2 – sensors section, 3 – sight glass, 4 – pressure sensor, 5 – thermocouple.

Temperatures of the tested fluid flow entering and exiting the test section are measured with thermocouples accurate to ±0.3°C. The refrigerant pressure at the inlet and outlet of the section is measured by Trafag NA25.0V pressure transducers accurate to ±0.3%.

2.3 Refrigerant loop

Refrigerant is supplied to the test section at specific conditions (temperature, flow rate, quality) by the refrigerant loop. This loop contains a condenser, a subcooler, a pump, a filter dryer, a flowmeter, a regulating valve and a preheater.

Table 1. Tested tube.

Tube description	Diameter to base of enhancement	Thickness of enhancement or fin height	Material	Other characteristics
smooth	8 mm	–	Cu	seamless
smooth	8.8 mm	–	stainless steel	seamed
corrugated	8.8 mm	$h = 0.45$ mm	stainless steel	seamed, pitch $e = 6mm$
				
micro-fin	8.92 mm	$h = 0.2$ mm	Cu	60 fins, seamless, twist angle $\beta = 18°C$, fin angle $\alpha = 48°C$
				

Prior to entering the test section, the refrigerant temperature and quality is set in the preheater, having the form of 2.5 m long copper tube heated electrically. Mass flow rate of the tested refrigerant is measured by the Coriolis-effect flowmeter Danfoss MASS 2100 having an accuracy of ±0.15% of the actual flow rate.

2.4 Heating water loop

Water is supplied to the annulus side of the test section by the water loop. This loop contains a pump, an electrical heater and a flowmeter. Water flow rate is controlled by a by-pass line and is measured by the magnetic flowmeter Danfoss MAG 3100 accurate to ±0.25%.

2.5 Cooling loop

The purpose of the cooling loop is to condense and subcool the tested refrigerant circulating in the refrigerant loop. The system contains a semi-hermetic compressor, a condenser cooled by tap water and two evaporators: a copper coil immersed in the glycol tank and the subcooler in the refrigerant loop. The working fluid in the cooling loop is R22 and its flow rate in each evaporator is controlled automatically by thermostatic expansion valve.

2.6 Oil injection and sampling

The oil is injected in a batch process. A known amount of lubricant is withdrawn from the sampling cylinder by the flow of charged refrigerant.

The same cylinder is used for sampling refrigerant-oil mixture. After sampling, the refrigerant is removed from the cylinder by slowly bleeding its vapour. By knowing the empty weight of the cylinder, the weight immediately after sampling and the weight after bleeding off the refrigerant, the mass fraction of the lubricant in the mixture can be calculated.

2.7 Experimental procedure

During tests inlet vapour quality was set as 0 ± 0.05 and outlet quality 0.7 ± 0.01. Mass flux density varied from about 250 to 500 kg/m^{2}s. The experiments have been conducted for average saturation temperature 0°C. This temperature is inferred from the average saturation pressure in the tube, calculated as the arithmetic mean value of the inlet and outlet pressure P_C and P_D, respectively, Fig. 3.

The monitoring of the temperature and pressure readings is facilitated by use of the PC-aided data acquisition system. For each tested tube and refrigerant, the experimental points were collected for two or three days. A good repeatability of the test results was observed.

3 Data reduction

3.1 Experimental results

In-tube heat transfer coefficient $h_{2F,av}$ is calculated from the expression for overall thermal resistance:

$$\frac{1}{h_{2F,av}} = \frac{\pi}{U_L} - \frac{1}{h_w D} - \frac{1}{2k} \ln \frac{D}{d}. \tag{1}$$

Annulus-side heat transfer coefficient h_w is calculated from the Dittus-Boelter correlation, according to suggestion of [29]. Overall heat transfer coefficient per 1 m of tube length in the test section is equal to:

$$U_L = \frac{c_w \dot{m}_w (T_{w1} - T_{w2})}{L \Delta T_m} \tag{2}$$

The heat flux transferred to the boiling refrigerant in the test section is calculated from an energy balance on the water side – Fig. 3. Heat loss, inferred from temperatures monitored in various locations inside the insulation of the test section, did not exceed 0.08% of the transferred heat flux and is assumed as negligible.

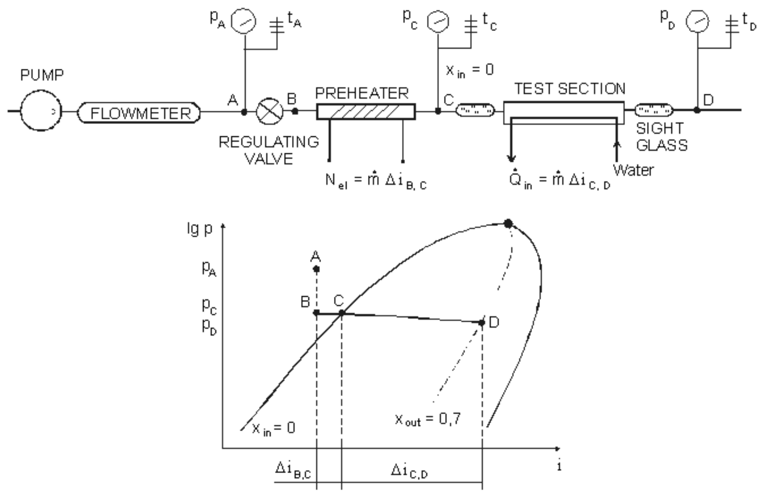


Figure 3. Pressure-enthalpy diagram representation of characteristic points.

Pressure drop over the test section is calculated as the difference between the inlet and outlet pressure of the tested refrigerant.

3.2 Heat transfer coefficient prediction

The experimental results have been compared with heat transfer coefficients calculated from correlations proposed by Kandlikar [30] and Witczak [31]. According to Kandlikar [30], convective and nucleate boiling heat transfer coefficients inside smooth tubes are correlated respectively as:

$$h_{2F,C} = 1.136\mathrm{Co}^{-0.9}(1-x)^{0.8}h_{lo} + 667.2\mathrm{Bo}^{0.7}(1-x)^{0.8}F_{fl}h_{lo} \tag{3}$$

and

$$h_{2F,N} = 0.6683\mathrm{Co}^{-0.2}(1-x)^{0.8}h_{lo} + 1058\mathrm{Bo}^{0.7}(1-x)^{0.8}F_{fl}h_{lo}. \tag{4}$$

For enhanced tubes the correlations are modified:

$$h_{2F,C} = 1.136\mathrm{Co}^{-0.9}(1-x)^{0.8}\mathrm{Re}_{lo}^{n}\mathrm{Pr}_{l}^{0.4}E'_{CB} + 667.2\mathrm{Bo}^{0.7}(1-x)^{0.8}F_{fl}\mathrm{Re}_{lo}^{n}\mathrm{Pr}_{l}^{0.4}E'_{NB} \tag{5}$$

and

$$h_{2F,N} = 0.6683\text{Co}^{-0.2}(1–x)^{0.8}\text{Re}_{lo}^{n}\text{Pr}_{l}^{0.4}E'_{CB} + 1058\text{Bo}^{0.7}(1–x)^{0.8}F_{fl}\text{Re}_{lo}^{n}\text{Pr}_{l}^{0.4}E'_{NB} \quad (6)$$

where empirically determined constants for the micro-fin tube and R 22 are: $E'_{CB} = 82$, $E'_{NB} = 72$, $n = 0.4$, [30].

Witczak [31] proposed correlation for convective boiling heat transfer coefficient inside smooth tubes in the form:

$$h_{2F,C} = h_{2F,E}(1 + \Psi_{2F}) \quad (7)$$

where equivalent coefficient is:

$$h_{2F,E} = \frac{1}{\frac{x}{h_{vo}} + \frac{1-x}{h_{lo}}}. \quad (8)$$

For nucleate boiling heat transfer coefficient Witczak proposed correlation:

$$h_{2F,N} = h_{2F,C}\ 1.28\ K_{2F}^{0.22}(1-x)^{0.08}\left(\frac{h_{NPB}}{h_{2F,C}}\right)^{0.74}. \quad (9)$$

Coefficient of heat transfer to boiling refrigerant h_{2F} is the maximum of coefficients for convective and nucleate boiling.

Correlations of Kandlikar and Witczak predict local evaporation heat transfer coefficient, i.e. for a specified boiling regime described by vapour quality. Average values of evaporation heat transfer coefficient in the tested range of vapour quality from 0 to 0.7 were calculated as follows:

$$h_{2F,av} = \frac{\int\limits_{x_1}^{x_2} h_{2F}dx}{x_2 - x_1}. \quad (10)$$

Equation (10) has been integrated numerically. For this purpose, the range of vapour quality is divided into 15 regions and local heat transfer coefficient was calculated for each of them, according to Kandlikar as well as Witczak correlations.

3.3 Heat transfer enhancement efficiency

For each tube and for each refrigerant, heat transfer enhancement factor and pressure drop penalty factor have been calculated. Heat transfer enhancement factor (EF) is defined as the ratio of average evaporation heat transfer coefficient for the enhanced tube to the heat transfer coefficient for the smooth tube:

$$EF = \frac{h_{en}}{h_{sm}}. \tag{11}$$

Pressure drop penalty factor (PF) is calculated as the ratio of pressure drop for enhanced and smooth tubes:

$$PF = \frac{\Delta p_{en}}{\Delta p_{sm}}. \tag{12}$$

The ratio of these two factors EF/PF can be treated as a measure of heat transfer enhancement efficiency [11,12,16,21].

Because the ratio EF/PF has no physical interpretation, own thermal-hydraulic enhancement factor QF has been proposed as a measure of heat transfer enhancement efficiency. Factor QF is defined as the ratio of average evaporation heat transfer coefficient for the enhanced tube to the heat transfer coefficient for the smooth tube, for the same pumping power:

$$QF = \left.\frac{h_{en}}{h_{sm}}\right|_{\dot{E}_p} . \tag{13}$$

The demand of equal pumping power for smooth and enhanced tube:

$$\dot{E}_{p,sm} = \dot{E}_{p,en} \tag{14}$$

for the same refrigerant can be expressed by equation:

$$G_{sm}\Delta p_{sm} = G_{en}\Delta p_{en}\frac{A_{o,en}}{A_{o,sm}}. \tag{15}$$

So, heat transfer coefficient for enhanced tube in Eq. (13) should be determined for the expression from the right-hand side of Eq. (15):

$$h_{en} = f\left(G_{en}\Delta p_{en}\frac{A_{o,en}}{A_{o.sm}}\right) \tag{16}$$

while heat transfer coefficient for smooth tube should be determined according to the left-hand side of Eq. (15):

$$h_{sm} = f\left(G_{sm}\Delta p_{sm}\right). \tag{17}$$

4 Results

4.1 Commissioning of test facility

The apparatus was commissioned by testing a plain copper tube as well as micro-fin tube for evaporation of pure R22 and R134a. Figures 4 and 5 shows the comparison of test results for smooth copper tube, and results of predictions by Kandlikar and Witczak correlations, and the test results obtained by other researchers. The comparison shown in Fig. 4 shows that test apparatus used in present study provides data in substantial agreement with the data published in literature.

Figure 6 displays present test results for micro-fin tube, and compares them with test results obtained by other researchers. It results from Fig. 6 that present data are in agreement particularly with manufacturer's data for lower mass flux density and in acceptable agreement with data obtained by Nidegger et al. [14]. The difference between present data and data obtained by Eckels and Pate [21] may results from different boiling temperatures. Eckels and Pate conducted their experiments at higher saturation temperature, i.e. 5^oC.

4.2 Pure refrigerants evaporation data

The heat transfer coefficients and pressure drop test results for all tested tubes are plotted in Figs. 7 and 8, respectively. A linear regression analysis using the least squares method was used to determine the best-fitting straight line for what were taken to be the steady-state results. The regression equations are presented in Tab. 2. For R22 and the micro-finned tube average heat transfer coefficients are higher than in the smooth copper tube by about 30% for high mass flux density to over 130% for low mass flux density. Values recorded for R22 and stainless steel corrugated tube are higher than for the smooth steel one, by over 10% for high mass flux density to under 50% for low mass flux density. Heat transfer coefficient for the corrugated tube is almost constant over the range of tested mass flux density.

A multidimensional regression analysis using the least squares method was used to determine equations for smooth and enhanced tubes. The regression equation for copper and stainless steel smooth tubes was found as:

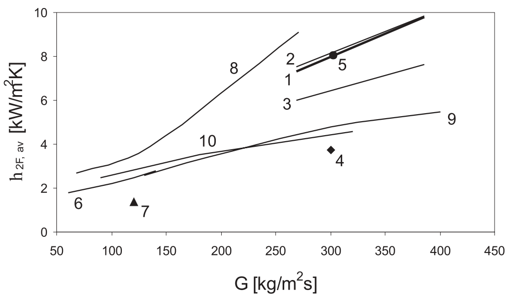


Figure 4. Heat transfer coefficient for R22 and smooth copper tube: 1 – present study, 2 – prediction of Kandlikar's correlation, 3 – prediction of Witczak's correlation, 4 – Haberschill et al. [33], 5 – Yu et al. [34], 6 – Ikeuchi i in. [23], 7 – Gorin [35], 8 – Kubanek, Miletti [36], 9 – Schlager et al. [16], 10 – Blaszewski [37].

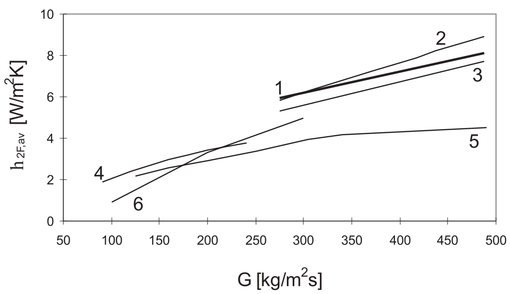


Figure 5. Heat transfer coefficient for R134a and smooth copper tube: 1-present study, 2-prediction of Kandlikar's correlation, 3-prediction of Witczak's correlation, 4-manufacturer's data [31], 5-Eckel and Pate [21], 6-Nidegger et al. [15].

$$\mathrm{Nu} = 0.0019\,\mathrm{Re}_{lo}^{0.28}\mathrm{Bo}^{-1.43}\left(\frac{P}{P_{cr}}\right). \tag{18}$$

The equation for micro-fin and corrugated tubes has the form:

$$\mathrm{Nu} = 62.33\,\mathrm{Re}_{lo}^{0.14}\mathrm{Bo}^{-0.09}\left(\frac{P}{P_{cr}}\right)^{0.42}\left(\frac{H}{d}\right)^{-0.5}. \tag{19}$$

Both regression equations are valid for R22 and R134a, for saturation temperature 0°C, for inlet vapour quality 0 and outlet vapour quality 0.7 and for mass flux density from 200 to 500 kg/m^{2}s. For about 96% of experimental points the discrepancy between experimental data and value calculated from equations 18 or 19 is lower than ±30%.

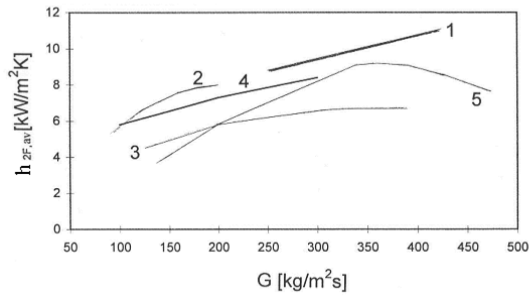


Figure 6. Heat transfer coefficient for R134a and micro-fin tube: 1 – present study, 2 – manufacturer's data [31], 3 – Eckel and Pate [21], 4 – Nidegger et al. [14], 5 – Liu [6].

Simultaneously higher pressure drops for enhanced tubes have been recorded (Fig. 8). Pressure drop for R22 in the micro-finned tube is higher than in the smooth copper one by upto 30%. In the case of stainless steel tubes, pressure drop for R22 is higher in the corrugated tube than in the smooth one, by over 30% for low mass flux density to over 50% for high mass flux density. The average boiling heat transfer coefficient for R134a in the micro-finned tube is about 50% higher, than in the smooth copper one (Fig. 7). The heat transfer coefficient in the corrugated stainless steel tube is higher, than in the smooth stainless steel one, by about 5% for higher mass flux density to about 40% for lower mass flux density.

Pressure drop during boiling of R134a in the micro-finned copper tube is higher, than in the smooth copper one, by over 10% for low mass flux density to under 20% for high mass flux density (Fig. 8). In the case of stainless steel tubes, pressure drop in the enhanced tube is higher by over 30% to 50% respectively.

Enhancement factor and penalty factor for R22 and the copper micro-finned tube refered to the smooth copper tube are plotted in Fig. 9. For R22 enhancement factor decreases with mass flux density increase, while penalty factor slightly increases. As a result, the ratio EF/PF decreases and is higher than 1 in the whole tested range of mass flux density. For the stainless steel corrugated tube referred to the smooth steel one enhancement factor for R22 decreases and penalty factor increases with the increase of mass flux density (Fig. 10). Only for low mass flux density, the ratio EF/PF is higher than 1.

For R134a and micro-fin tube enhancement factor slightly decreases and

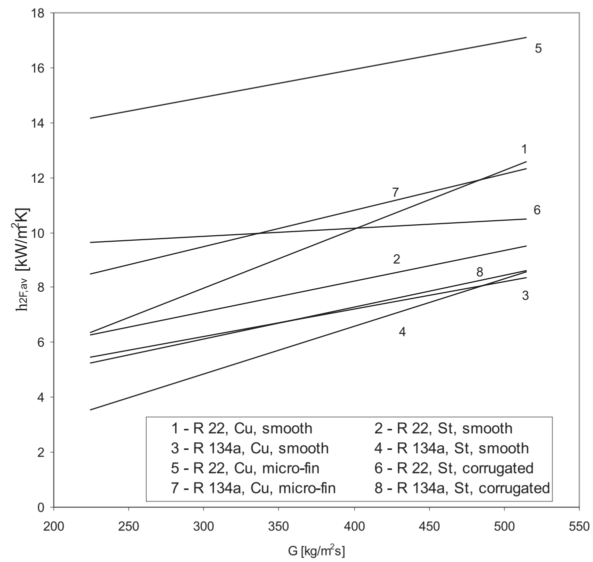


Figure 7. Average heat transfer coefficient versus mass flux density.

penalty factor increases, with mass flux density increase (Fig. 11). The ratio EF/PF decreases then, but in the whole range is higher than 1. Enhancement factor and penalty factor for the stainless steel corrugated tube referred to the smooth stainless steel tube are plotted in Fig. 12. Enhancement factor decreases and penalty factor increases with the increase of mass flux density. The ratio EF/PF is lower than 1 in the whole range of tested mass flux density.

Thermal-hydraulic enhancement factor QF for R22 is plotted in Fig. 13. For the micro-fin tube as well as for the corrugated tube QF factor is higher than 1. Thermal-hydraulic enhancement factor QF for R134a is plotted in Fig. 14. For the micro-fin tube QF factor is higher than 1, while for the corrugated tube, for higher mass flux density is lower than 1. Factor QF as well as the ratio EF/PF show low heat transfer enhancement efficiency in the case of stainless steel corrugated tube. This may contest advantages of this tube. In the case of smooth tubes, the experimental evaporation heat transfer coefficients were compared with values calculated from the correlations of Kandlikar and Witczak. For R22 and copper tube (Fig. 4) the experimental data are in very good agreement with predictions of Kandlikar and exceed predictions of Witczak by over 20%. In the case of R134a and

Table 2. Regression equations for average heat transfer coefficient $h_{2F,av}$ [kW/m^{2}K] as a function of mass flux density G [kg/m^{2}s].

Line number according to Fig. 7	Regression equation
1	$h_{2F,av}$ = 0.0215 G + 1.5157
2	$h_{2F,av}$ = 0.011 G + 3.8208
3	$h_{2F,av}$ = 0.0101 G + 3.1901
4	$h_{2F,av}$ = 0.0173 G - 0.3496
5	$h_{2F,av}$ = 0.0102 G + 11.871
6	$h_{2F,av}$ = 0.0029 G + 8.9883
7	$h_{2F,av}$ = 0.0132 G + 5.5245
8	$h_{2F,av}$ = 0.0115 G + 2.6694

copper tube (Fig. 5) the experimental data for high mass flux density are lower by about 10% than predictions of Kandlikar and exceed predictions of Witczak by about 5% for high mass flux density to over 10% for low mass flux density.

For R134a boiling in smooth stainless steel tube (Fig. 15) the biggest discrepancy between calculated and measured values of heat transfer coefficient has been recorded. Kandlikar's correlation overpredicts experimental data by about 25%.

Values calculated from the modified Kandlikar's correlation (Eqs. 5 and 6) for micro-finned tube (Fig. 16) are overpredicted by almost two and a half times. However, the constants used in the correlation have been determined experimentaly for different test conditions, i.e. higher boiling pressure and electrically heated test section [30]. Unfortunately, in the literature no correlations for boiling in corrugated tubes have been found.

4.3 Refrigerant/oil mixtures evaporation data

Figures 17 and 18 display the influence of oil concentration on R134a heat transfer coefficient and pressure drop for smooth copper tube, respectively. Generally, presence of oil in mixture during boiling inside smooth tubes inhibits heat transfer and increases pressure drop. For higher oil concentration investigated, i.e. 5%, the influence on heat transfer rate is different depending on mass flux density. For smaller mass flux density heat transfer

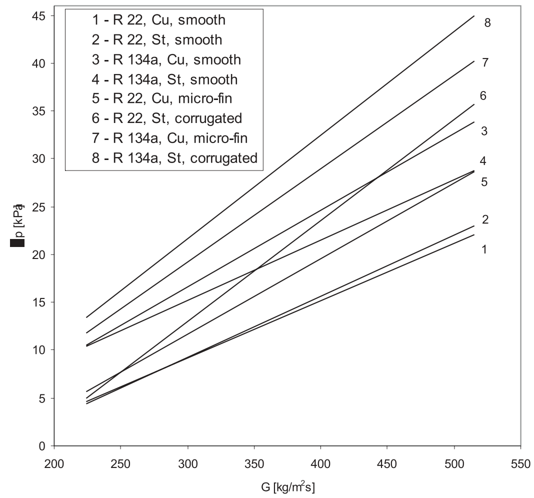


Figure 8. Pressure drop versus mass flux density.

coefficient is almost the same as for pure refrigerant, but for higher mass flux density dramatic heat transfer degradation simultaneously distinct pressure drop decrease have been observed, Figs. 17 and 18, respectively. Moreover, heat transfer hysteresis has been recorded for increasing mass flux density. The same phenomenon has been observed for R134a/oil mixture during boiling in smooth stainless steel tube as well as for R22/oil mixture and both smooth tubes. Figure 19 shows heat transfer coefficients for R22/oil mixtures boiling in the copper micro-fin tube. The presence of oil decreases heat transfer coefficient. The heat transfer coefficients test results for micro-

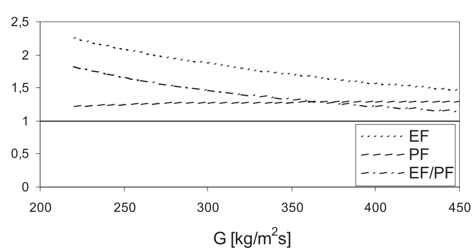


Figure 9. Enhancement and penalty factors for micro-finned tube and R22.

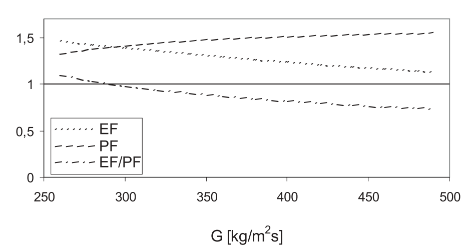


Figure 10. Enhancement and penalty factors for corrugated tube and R22.

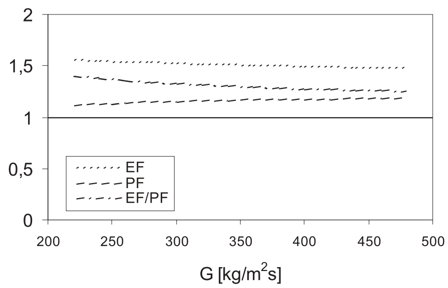


Figure 11. Enhancement and penalty factors for micro-finned tube and R134a.

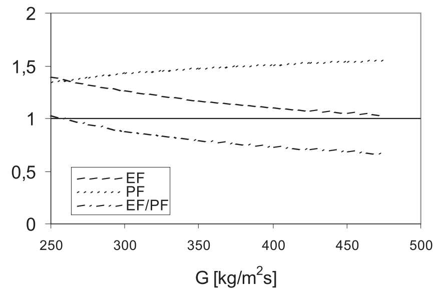


Figure 12. Enhancement and penalty factors for corrugated tube and R134a.

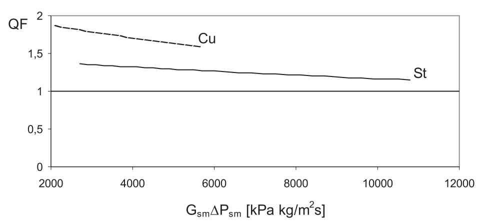


Figure 13. Thermal-hydraulic enhancement factor QF for R22: Cu – micro-fin tube referred to copper smooth tube, St – corrugated tube referred to stainless steel smooth tube.

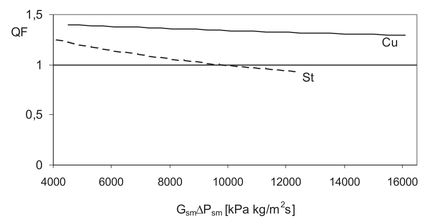


Figure 14. Thermal-hydraulic enhancement factor QF for R134a: Cu – micro-fin tube referred to copper smooth tube, St – corrugated tube referred to stainless steel smooth tube.

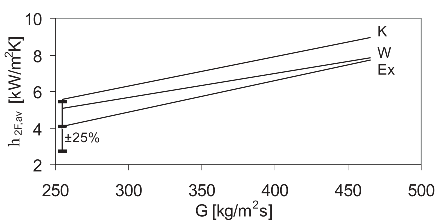


Figure 15. Comparison of experimental R134a heat transfer coefficient data for smooth stainless steel tube (EX) with the predictions of Kandlikar (K) and Witczak (W).

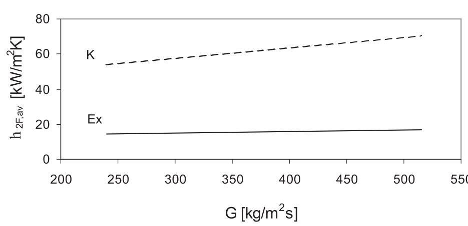


Figure 16. Comparison of experimental R22 heat transfer coefficient data for micro-finned tube (EX) with the predictions of Kandlikar (K).

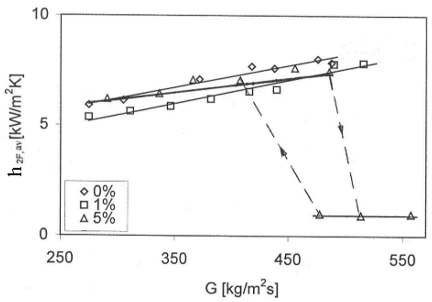


Figure 17. Influence of oil concentration on average heat transfer coefficient for R134a and smooth copper tube.

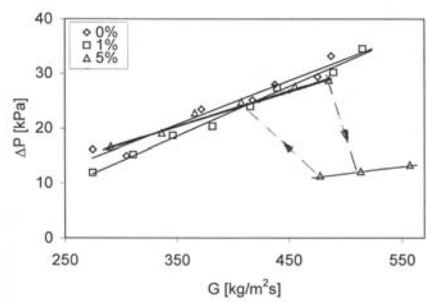


Figure 18. Influence of oil concentration on pressure drop for R134a in smooth copper tube.

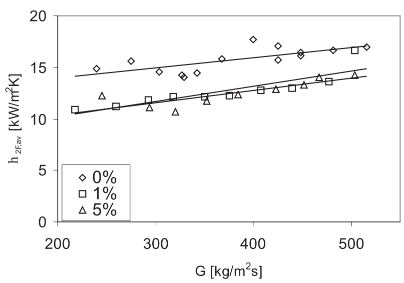


Figure 19. Average heat transfer coefficient versus mass flux density for R22 and micro-fin tube.

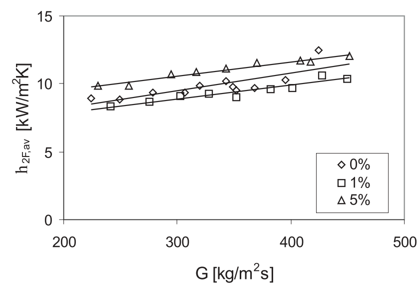


Figure 20. Average heat transfer coefficient versus mass flux density for R134a in micro-fin tube.

fin tube and R134a are plotted in Fig. 20. The presence of oil causes an enhancement of heat transfer coefficient – the lubricant concentration the higher the heat transfer coefficient, particularly in the case of corrugated tube. No heat transfer hysteresis has been recorded for both refrigerants boiling in enhanced tubes.

Influence of oil concentration on pressure drop (Fig. 21) was negligible for both refrigerants in both enhanced tubes investigated.

The heat transfer coefficients test results for R134a and all tested tubes for oil concentration 1% and 5% are plotted in Fig. 22 and Fig. 23, respectively. Heat transfer coefficients in enhanced tubes are higher than for the smooth ones, like for pure refrigerants.

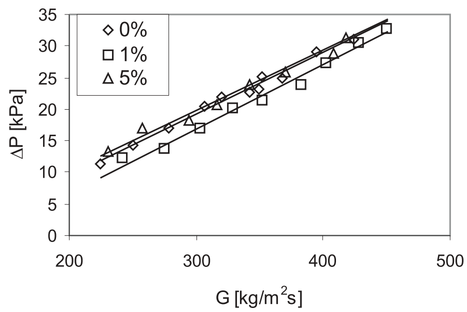


Figure 21. Pressure drop versus mass flux density for R134a and corrugated tube.

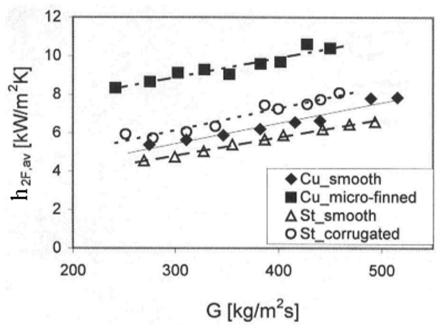


Figure 22. Average heat transfer coefficient for R134a and oil concentration 1%.

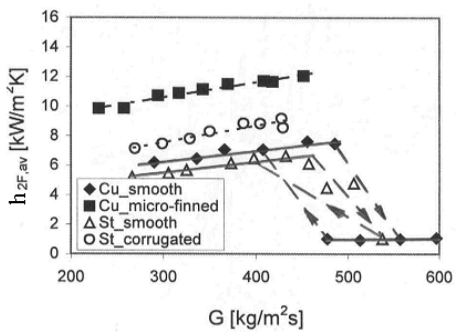


Figure 23. Average heat transfer coefficient for R134a and oil concentration 5%.

4.4 Uncertainty estimation

The uncertainties of the measured and calculated parameters are estimated by following the procedures described in Sections 2 and 3. The error analysis is done for the minimal and maximal refrigerant flow rate. Mean-square error for the average evaporation heat transfer coefficient is equal to:

$$\Delta h_{2F,av} = \left\{ \left(\frac{\partial h_{2F,sr}}{\partial U_L} \Delta U_L \right)^2 + \left(\frac{\partial h_{2F,sr}}{\partial h_w} \Delta h_w \right)^2 + \right.$$
$$\left. + \left(\frac{\partial h_{2F,sr}}{\partial d} \Delta d \right)^2 + \left(\frac{\partial h_{2F,sr}}{\partial D} \Delta D \right)^2 + \left(\frac{\partial h_{2F,sr}}{\partial k} \Delta k \right)^2 \right\}^{0.5} . \tag{20}$$

Relative errors for each tube and each refrigerant are specified in Tab. 3.

Table 3. Relative errors of average evaporation heat transfer coefficient determination.

Tube \ Refrigerant	R 22	R 134a	R 407C
smooth, copper	$17 \div 23$ %	$15 \div 18$ %	$18 \div 19$ %
micro-fin (copper)	$38 \div 45$ %	$28 \div 33$ %	$30 \div 36$ %
smooth, stainless steel	$19 \div 22$ %	$13 \div 19$ %	$12 \div 15$ %
corrugated (stainless steel)	$21 \div 24$ %	$16 \div 18$ %	$16 \div 23$ %

5 Conclusions

The results obtained for smooth tubes are in agreement with predictions obtained from correlations proposed by Kandlikar and Witczak. Distinctly higher average evaporation heat transfer coefficients for enhanced tubes have been observed, particularly for the micro-fin tube. Simultaneously higher pressure drops for enhanced tubes have been recorded as well. Enhancement factor EF decreases and simultaneously penalty factor PF increases with increase of mass flux density during boiling in the micro-fin tube. The ratio EF/PF and QF factor for the copper micro-finned tube are higher than 1 over the range of tested mass flux density. In the case of R134a and the stainless steel corrugated tube, the ratio EF/PF is lower than 1 and QF factor is higher then 1 only for low mass flux density. Heat transfer coefficient is higher during boiling in copper tubes, compared to stainless steel ones. Refrigerant R22 has an evident superiority over R134a for each tested type of tube.

Generally, presence of oil in mixture during boiling inside smooth tubes inhibits heat transfer and increases pressure drop. Heat transfer hysteresis has been recorded during boiling of refrigerant/oil mixtures with oil concentration ca. 5%, for higher mass flux density. For the same range of mass flux density heat transfer hysteresis was not observed in both enhanced tubes investigated.

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