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HEAT TRANSFER AND PRESSURE DROP DURING FLOW BOILING OF PURE REFRIGERANTS AND REFRIGERANT/OIL MIXTURES IN TUBE WITH POROUS COATING

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Abstract

The experimental stand and procedure for flow boiling investigations are described. Experimental data for pure R22, R134a, R407C and their mixtures with polyester oil FUCHS Reniso/Triton SEZ 32 in a tube with porous coating and smooth, stainless steel reference tube are presented. Mass fraction of oil was equal to 1% or 5%. During the tests inlet vapour quality was set at 0 and outlet quality at 0.7. Mass velocity varied from about 250 to 500 kg/m^{2}s. The experiments have been conducted for average saturation temperature 0°C. In the case of flow boiling of pure refrigerants, the application of a porous coating on inner surface of a tube results in higher average heat transfer coefficient and simultaneously in lower pressure drop in comparison with the flow boiling in a smooth tube for the same mass velocity. Correlation equation for heat transfer coefficient calculation during the flow boiling of pure refrigerants inside a tube with porous coating has been proposed.

Nomenclature

b	- characteristic length [m]
c	- specific heat [J kg^{-1} K^{-1}]
C	- constant (Eq. 6)
d	- inside diameter [m]
D	- outside diameter [m]
EF	- heat transfer enhancement factor [-]
g	- acceleration due to gravity [m s^{-2}]
G	- mass velocity [kg m^{-2} s^{-1}]

k_L - overall heat transfer coefficient per unit length [W m^{-1} K^{-1}]

L - length [m]

$\dot{m}$ - mass flux [kg s^{-1}]

n - exponent (Eq. 6)

q - heat flux [W m^{-2}]

p - pressure [Pa]

P - correction factor (Eq. 7)

PF - pressure drop penalty factor

r - latent heat of evaporation [J kg^{-1}]

R_{M-S} - two-phase flow multiplier (Eq. 9)

t - temperature [°C]

x - quality

Greek letters

α - heat transfer coefficient [W m^{-2} K^{-1}]

μ - viscosity [Pa s]

λ - thermal conductivity [W m^{-1} K^{-1}]

σ - surface tension [N m^{-1}]

Subscripts

1,2 - inlet, outlet

av - average

en - porous coated

L - liquid

LMTD - log mean temperature difference

loc - local

PB - pool boiling

REF - reference

S - saturated

sm - smooth

TPB - two-phase boiling

v - vapor

w - water, water side

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Non-dimensional numbers:

Bo - boiling number, $Bo = \frac{\dot{q}}{G\ r}$

Nu - Nusselt number, $Nu = \frac{\alpha\ b}{\lambda}$

Pr - Prandtl number, $Pr = \frac{c\ \mu}{\lambda}$

Re - Reynolds number, $Re = \frac{G\ b}{\mu}$

1. INTRODUCTION

Application of enhanced tubes has become lately standard industrial practice in chemical engineering and refrigeration systems [1-4]. These surfaces have been designed in a number of forms, from simple low integral fins to more complicated doubly enhanced tubes or metallic porous coatings [5-10]. As pool boiling investigations show, heat transfer coefficient can be many times higher than for smooth tube when metallic porous coating is applied [11-14].

However, 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 workmanship and wear of the compressor and other system elements.

Published literature data for pool boiling of oil-refrigerant mixture on porous coated surfaces show that even small oil concentrations (1-3%) can cause significant reduction of heat transfer coefficient [15,16], although, Czikk et al. [17] found that oil concentrations up to 2% had very little effect on the performance of the R-11 chiller. Available experimental data for flow boiling inside smooth and selected enhanced tubes show, that irrespective of the refrigerant and oil type, the presence of lubricant always increases pressure drop. 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 (above 5%) always inhibit heat transfer and the maximum value of the heat transfer coefficient is shifted to lower vapour quality [18].

No data of flow boiling of oil-refrigerant mixture inside porous coated tube have been found in the literature, although refrigeration seems to be relevant area of application of porous coated channels.

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The main aim of the study was determination of average evaporation heat transfer coefficient and simultaneously pressure drop during evaporation of R22, R134a and R407C and their oil mixture inside smooth and porous coated tube.

2. LITERATURE REVIEW

Most of the published data for boiling on porous coated surfaces have been done for pool boiling conditions. Literature data for flow boiling in a tube with porous coating display that heat transfer coefficient is also higher in comparison with a smooth tube, however, data about flow boiling in a tube with porous coating are very scarce.

Czikk et al. [19] performed study using liquid oxygen, ammonia, and R22 inside a vertically oriented 18.7 mm diameter tube internally covered with the commercially available High Flux coating. They reported that the heat transfer coefficient for the porous-coated tube was insensitive to quality and mass flux and was typically an order of magnitude greater than that for smooth-tube data. Czikk et al. [19] also tested ammonia inside a horizontally oriented porous-coated tube with a 25 mm outside diameter. Ikeuchi et al. [20], carried out experiments with boiling R22 inside a 17.05 mm internal diameter tube with plated 0.115 mm diameter copper particles inside. The heat transfer coefficient was approximately 5 times better than for plain-tube performance for exit qualities between 70 and 95 percent. Khasanov et al. [21] studied boiling of distilled water inside electrically heated 2 m long and ID equal to 7. 78 mm tube with sintered porous coating of 0.22-0.28 mm thick and porosity 70-80% made from stainless steel particles of 60 μm in diameter. They established that the wall temperature in post-dry out region for a tube with porous coating was distinctly lower, although heat flux was about 25% higher, than for a smooth tube. Simultaneously, the level of temperature pulsations was four times smaller than for a smooth tube. Savkin et al. [22] conducted experiments with vertically oriented tube described by Khasanov et al. [21]. The investigation showed that in pre-CHF region average heat transfer coefficient was three times higher than for a smooth tube. The influence of porous coating increases with the increase of pressure. The higher was the pressure inside tube (0.1 – 6 MPa) the higher was the intensification ratio. In the transition region the temperature pulsation was five times smaller than for a smooth tube. After 500 performance hours of the tube, they did not observe the deterioration of heat transfer rate. Shklover and Kovalov [23], studied heat transfer mechanism from horizontal, flat surface coated with 2 mm sintered porous layer made from bronze particles 0.2-0.3 mm in diameter during the flow boiling of ethyl alcohol. During the

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tests inlet vapour quality was set at 0.1 or 0.3. It is interesting that for lower heat flux density investigated (below 100 kW/m^2), pool boiling heat transfer coefficient was higher than for flow boiling one. For higher heat flux density (up to 1 MW/m^2) – independently on inlet vapour quality, heat transfer coefficient was distinctly higher for porous coated surface. Shklover andd Kovalov claimed, that liquid movement along porous layer facilitates vapour outlet from the structure. This effect escalates with the increase of liquid velocity. Kovalov and Shklover [24] performed experiments with water boiling on flat surface (90x100 mm) placed in a rectangular channel 250 mm long and 1.0-14 mm high, coated with porous layers 1 or 2 mm thick, made from bronze particles 0.063-0.5 mm in diameter. Tests were conducted for subcooled water (ΔT=40 K) and two phase mixture with quality equal to 0.3 at atmospheric pressure. In case of subcooled flow boiling for whole investigated heat flux density range (200 – 3000 kW/m^2), heat transfer coefficient was 1.3 to 3 times better than for plain-tube performance. For heat flux density lower than 1.2 MW/m^2, thick coatings (2 mm) made from big diameter particles were more effective, and for heat flux density higher than 1.2 MW/m^2 thin, low thermal resistance coatings (1 mm) were better. Solov'ev and Shklover [25] compared the performance of the same sintered porous layers during pool and flow boiling. Porosity of the porous layers - made from bronze particles, were 15-64%, thickness 1 or 2 mm, and mean pore diameter 10-200 μm. Tests were conducted for subcooled water (ΔT=40 K) and ethanol. For heat flux density below 700 kW/m^2, heat transfer coefficient during pool boiling was higher than for flow boiling. For heat flux density higher than 700 kW/m^2 inverse situation was observed. A model – based on micro-heat pipe idea, for subcooled flow boiling outside porous coated surface was presented. Morozov et al. [26] conducted experiments with potassium boiling in vertical tubes covered with metal-fibre layer made from stainless steel fibres. Porous coating was applied to full length of the tube (ca. 1.3 m) and to half of the length of the tube – at the outlet region. In the tubes with porous coating along the full length the crisis develops nearby the boiling incipience cross-section, and in the tubes with porous coating along the half-top, nearby the exit. Morozov et al. established that for porous coated tube and mass velocity lower than 70 kg/m^{2}s, critical vapour quality is constant and equal to almost one. For mass velocity between 70 kg/m^{2}s and 120 kg/m^{2}s critical vapour quality decreases with the mass velocity increase, but is higher than for a smooth tube, and for mass velocity higher than 120 kg/m^{2}s critical vapour quality decreases with the mass velocity increase, too, but is lower than for a smooth tube. Kotov et al. [27] carried out experiments with water boiling inside vertical channels at wide range of

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pressure – 6.9-17.6 MPa and mass velocity – 500-2500 kg/m^{2}s. Two geometries of test channel have been examined: annular channel and circular tube. The annular channel was 1.1 m long, the annulus gap was 2 mm and mean diameter equaled 12 mm. Porous coating fabricated by spraying was applied to the outside surface of the inner tube and was made from chrome-nickel particles. The circular tube – 0.8 mm long, consisted of two parts: smooth (0.275 m) and porous coated (0.525 m). Sintered porous coating was applied to the inner surface of the tube and was made from stainless steel particles. Porosity of the porous layers was 30-45% and thickness ranged from 0.15 to 0.3 mm. Kotov et al. determined that critical heat flux density decreases with the increase of pressure and mass velocity. For higher pressure and mass velocity the influence of porous coating was more distinct. They determined the positive effect of the porous coating to start at pressure higher than 12 MPa. Andrianov et al. [28] conducted comprehensive study on pressure drop during one- and two phase flow inside vertical 8 mm in diameter porous coating tube. Test section was 1.19 m long, and porous coating was produced by sintering stainless steel particles of 63-100 μm in diameter. Porosity of the porous layer was estimated to be 40-50% and average thickness ca. 0.2 mm. During investigation, the pressure ranged from 3 to 16 MPa and mass velocity from 500 to 2000 kg/m^{2}s. Flow of subcooled water (ΔT=20 K), adiabatic flow of water-steam mixture and water-steam mixture flow in a heated tube were studied. Andrianov et al. established that friction factor for a tube with porous coating was 4-5 times higher than for smooth tube, and 3 times higher than for tube of technical roughness, and ca. 1.4 times higher than for sandblasted surface. According to Andrianov et al. the tube surface roughness resulting from the porous layer application is not a sole and prime reason of pressure drop increase during one- and two phase flow inside porous coated tube, but for hydraulic resistance increase, hydrodynamic processes which take place inside skeleton of the porous layer are responsible. It was established too, that heating of the test section has no influence on hydraulic resistance observed. Zuev and Malyshenko [29] studied water boiling crisis phenomena in vertical 1 m long tubes of 8 mm ID with porous coating applied to full length of the tube and tube with porous coating applied to outlet region 0.4 m long. Porous coating – 0.2 mm thick with porosity 40-50% was fabricated by vacuum sintering and consists of 63-100 μm stainless steel particles. They established that the application of a porous layer dramatically modifies the dynamics of initiation and evolution of forced boiling crisis. The time of the crisis evolution increases from 10 seconds to a few minutes in comparison with smooth tube at the same parameters. The boiling crisis begins with low frequency and fine

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scale (1 K) fluctuations of the wall temperature. Generally, thermal stability increase was observed in near- and post crisis modes. Nevertheless, the application of a porous layer results in substantial increase in pressure drop – three to four times in comparison with a smooth tube.

Wadekar [30] reported results of R113 flow boiling in a specially designed vertical tube of 19 mm ID tube. The test section is composed of three sections – a central smooth section and a coated section on either side; each being 1 m in length. The porous coating – 0.2-0.3 mm thick was made from 150 μm copper-phosphorous particles. An order of magnitude augmentation in boiling heat transfer coefficient was obtained due to porous coating on the tube wall. Wadekar suggested that the dry out on the porous coated surface is related to the vapour blanketing (high heat flux) rather than the drying out of the liquid film. Wadekar reported an upstream propagation of boiling front, too. Schröder-Richter et al. [31] studied CHF during water flow boiling in short (342 mm) vertical tube with porous coating made from Inconel-600. Sintered porous layer was 0.18 mm thick. Almost no effect of porous coating on CHF was observed for the range of parameters investigated: subcooling ΔT= 3-55K, mass velocity below 200 kg/m^{2}s and pressure 0.1-0.7 MPa. Malyshenko et al. [32] building on comprehensive experimental study of dry out phenomenon during wispy-annular flow boiling of water inside vertical tube with porous coating, concluded that the application of a porous layer at the wall tube leads to fundamental changes of a liquid film hydrodynamics on the tube wall as well as mass exchange processes between the vapour core and the liquid film. The flow of a liquid film and heat transfer in the boundary region depends not only on "standard" parameters, like vapour quality, mass velocity and pressure, but depends on porous layer characteristics, too. They stressed and discussed the following porous coating effects on mass exchange processes during dry out phenomenon: modification of liquid droplet spectrum in vapour core, considerable effect of "shooting through" droplets, effect of aeration - the role of aeration of a liquid film by bubbles becomes more important in wide range of vapour quality - up to critical vapour quality, capillary effects inside porous structure, intensification of rewetting effect in the dry out region. Yildiz [33] and Yildiz and Bartsch [34] examined high porosity (50-70%) porous coated tubes of 170 mm and 310 mm in length, made from Inconel-600 and stainless steel, respectively. They established that positive effect of porous coating on CHF is restricted to higher pressure studied, i.e. 0.7 MPa. Furthermore, the increase or decrease of CHF with porous coating depends on flow patterns – the increase of CHF is determined by DNB phenomenon. In addition, the pressure drop of the

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porous coated tube is higher than that of the smooth tube. Malyshenko [35], discussed three expected porous coating effects at stratified and intermittent two-phase flows in horizontal steam generating channels, which can lead to the decrease of the wall temperature non-uniformity. These are: coating friction effect, azimuthal heat pipe effect and drops-wetting effect. Ammerman and You [36] performed experiments with FC-87 boiling inside a single 2 mm square cross-section channel of 8 cm in length. The painting technique was applied in order to produce porous coating of a thickness of approx. 100 μm. The particles were 8-12 μm industrial diamond powder. The application of the microporous coating resulted in boiling incipience at lower wall superheats, considerable heat transfer augmentation, and enhancement of CHF. Furthermore, application of the porous layer leads to minor impact on pressure drop in the case of subcooled flow boiling. Rainey et al. [37] studied effects of fluid velocity and subcooling on the heat transfer performance from a microporous, flat (10x10 mm) surface mounted in the bottom of a square horizontal channel. The porous coating was fabricated by use of the painting technique and was ca. 50 μm thick. Microporous structure consisted of aluminium particles 1-20 μm in diameter and epoxy. The tests were conducted with FC-72 at atmospheric pressure. Rainey et al. determined that higher fluid velocities than for the smooth surface are required to provide additional enhancement of nucleate boiling heat transfer. In addition, the enhancement of CHF provided by the microporous coating over the smooth surface increases with increased fluid subcooling, however, compared to the smooth surface, the enhancement effectiveness of the coating decreases with increased velocity.

As the literature review presented, the application of a porous coating improves generally flow boiling performance. The increase in pressure drop as well as penalty-free heat transfer enhancement have been reported for porous coated channels. In addition, porous layer modifies dynamics of CHF.

3. EXPERIMENTAL FACILITY

3.1. 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.

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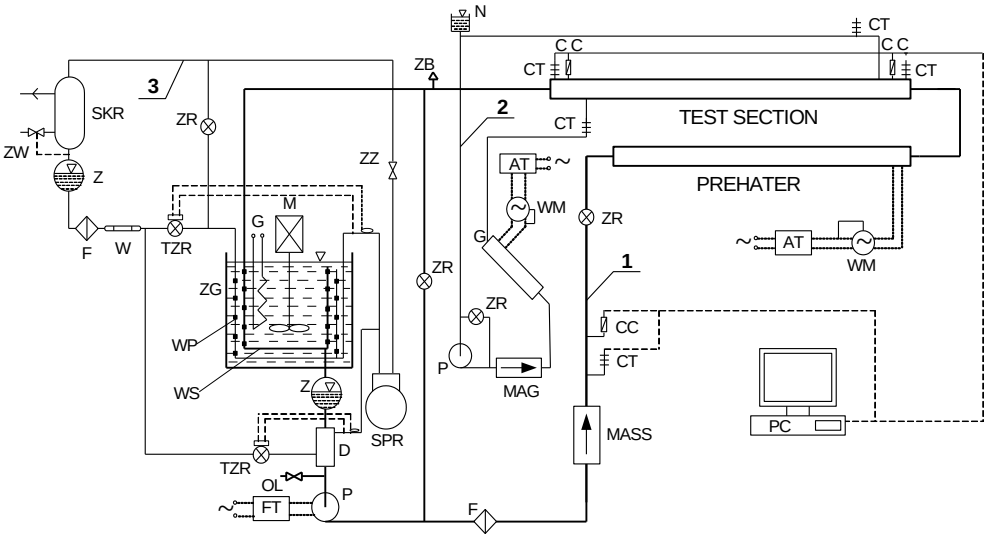

Fig. 1. Test stand scheme: 1- test loop, 2- heating loop, 3- cooling loop, P- pump, F- filter, Z- receiver, OL - oil valve, MASS, MAG - flowmeter, ZR - regulating valve, ZB - safety valve, WS - condenser coil, WP - evaporator coil, ZG - glycol tank, M - stirrer, G - electric heater, SPR - compressor, 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

3.2. Test section

The test section consists of tube-in-tube heat exchanger, sight-glasses and sensors for temperature and pressure measurement on the inlet and outlet of the section – Fig. 2. 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 inside wall tube was covered with metallic porous coating. The main parameters of the porous coating are: thickness 55 μm, porosity 18% and mean pore radius 1,45 μm. A smooth stainless steel tube served as a reference tube. The water in the annulus gap, surrounding the test tube, flows counter to the refrigerant flow and is used to heat the refrigerant, evaporating in the inner tube. The temperature of the fluid entering and exiting the test section is measured by 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%. The outside wall temperature of the tested tube is measured by thermocouples accurate to ±0,3°C inserted into the thermocouple pockets at six locations.

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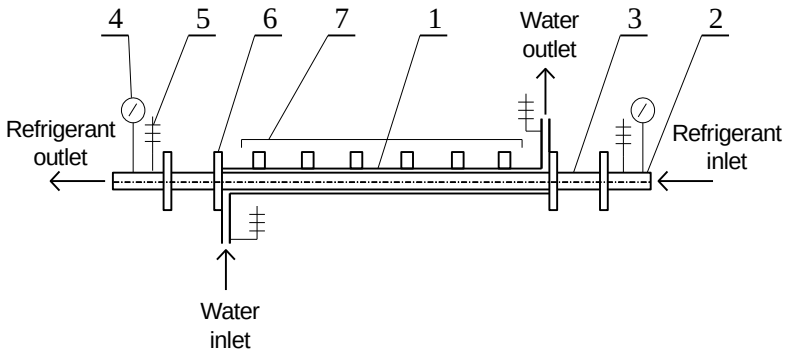
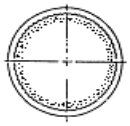
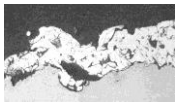


Fig. 2. Test section scheme: 1 - tested tube in water jacket, 2 - measuring section, 3 - sight-glasses, 4 - pressure transducer, 5 - thermocouple, 6 - flange, 7 – thermocouples

Tab. 1 Geometry of tested tubes

Tube	OD	ID	Material	Other
smooth	10 mm	8 mm	316L	Seamed, R_a = 0,40 μm
porous coated	10 mm	8.8 mm	316L	Al δ = 55 μm ε = 18 % a = 1.45 μm
 				

3.3. Refrigerant loop

The 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. 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.

3.4. 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 (Fig. 1). Water flow rate is controlled by a by-pass line and is measured by the magnetic flowmeter Danfoss MAG 3100 of accuracy ±0,25%.

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3.5. 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 (Fig. 1). The working fluid in the cooling loop is R22 and its flow rate in each evaporator is controlled automatically by thermostatic expansion valve.

3.6. 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 weight of the empty 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.

4. EXPERIMENTAL PROCEDURE

Tested refrigerants were R22, R134a and R407C and their mixtures with polyester oil FUCHS Reniso/Triton SEZ 32. Mass fraction of oil was equal to 1% or 5%. During the tests, inlet vapour quality was set at 0 and outlet quality at 0.7. Vapour quality was calculated from heat balance. Mass velocity varied from about 250 to 650 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. Average outside wall temperature is estimated as an arithmetic mean value of the six thermocouple readings along tested tube. Investigations have been conducted for constant, but dependent on refrigerant mass velocity outside surface temperature of a tested tube.

The test facility is capable of determining in-tube average heat transfer coefficient and pressure drop of a pure refrigerant or a refrigerant/oil mixture over the length of a test tube.

Detailed description of the experimental set-up and procedure is presented in [38].

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5. DATA REDUCTION

5.1. Average heat transfer coefficient

Average in-tube heat transfer coefficient α_{TPB} is calculated from the expression for overall thermal resistance:

$$\frac{1}{\alpha_{TPB}\ d} = \frac{\pi}{k_L} - \frac{1}{\alpha_w\ D} - \frac{1}{2\ \lambda}\ \ln\ \frac{D}{d} \tag{1}$$

Annulus-side heat transfer coefficient α_w is calculated from the Dirker-Meyer [39] correlation. Overall heat transfer coefficient per 1 m of tube length in the test section is equal to:

$$k_L = \frac{c_w\ \dot{m}_w\ \left(T_{w1} - T_{w2}\right)}{L\ \ \Delta T_{LMTD}} \tag{2}$$

where log mean temperature is given by

$$\Delta T_{LMTD} = \frac{\left(t_{w1} - t_{w2}\right) - \left(t_{w2} - t_S\right)}{\ln \dfrac{t_{w1} - t_S}{t_{w2} - t_S}} \tag{3}$$

Heat flux transferred to the boiling refrigerant in the test section is calculated from an energy balance on the water side. 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.

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

5.2. 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 a tube with porous coating to the heat transfer coefficient for a smooth tube

$$EF = \frac{\alpha_{en}}{\alpha_{sm}} \tag{4}$$

Pressure drop penalty factor (*PF*) is calculated as a ratio of pressure drop for a tube with porous coating and a pressure drop for a smooth tube:

$$PF = \frac{\Delta p_{en}}{\Delta p_{sm}} \tag{5}$$

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The ratio of these two factors *EF/PF* can be treated as a measure of heat transfer enhancement efficiency – Schlager et al. [40].

5.3. Uncertainty estimation

The uncertainties of the measured and calculated parameters are estimated by mean-square method. Uncertainty in pressure drop determination was estimated as ±4%. Average evaporation heat transfer coefficient measurement uncertainty was done for the minimal and maximal refrigerant flow rate. Relative errors for each refrigerant are specified in Tab. 2.

Table 2. Relative errors of average heat transfer coefficient determination

Tube \ Refrigerant	R 22	R 134a	R 407C
smooth, stainless steel	1,7 – 2 %	1,8 – 2,1 %	1,9 – 2,4 %
porous coated	10,5 – 18,5 %	7,1 – 10,2 %	6,8 – 11,6 %

6. RESULTS

6.1. Validation of the experimental rig and procedure

The apparatus was qualified by comparing of present results for flow boiling of pure R22 inside smooth stainless steel tube with test results obtained by other researchers and with predictions by Kandlikar [41] and D. Mikielewicz et al. [42] correlations – Fig. 3. Average heat transfer coefficient obtained was higher than experimental data recorded by Greco and Vanoli [43] and Jung et al. [44] which may result from different conditions of the conducted tests, i.e. different range of boiling temperature and vapor quality. Secondly, the tested tubes in Greco and Vanoli and Jung et al. experiments were heated electrically, while in present study water heating jacket was used. Present results lie above the theoretical predictions. D. Mikielewicz et al. correlation underpredicts heat transfer coefficient within the whole range of mass velocity of ca. 25%, while according to Kandlikar's correlation the difference between present data and predictions equals ca. 60% and 25% for lower and higher mass velocity, respectively. However, present data are in excellent agreement with experimental results obtained by Targańsk and Cieslinski [18] on the same experimental rig with the same experimental procedure and comparable test conditions.

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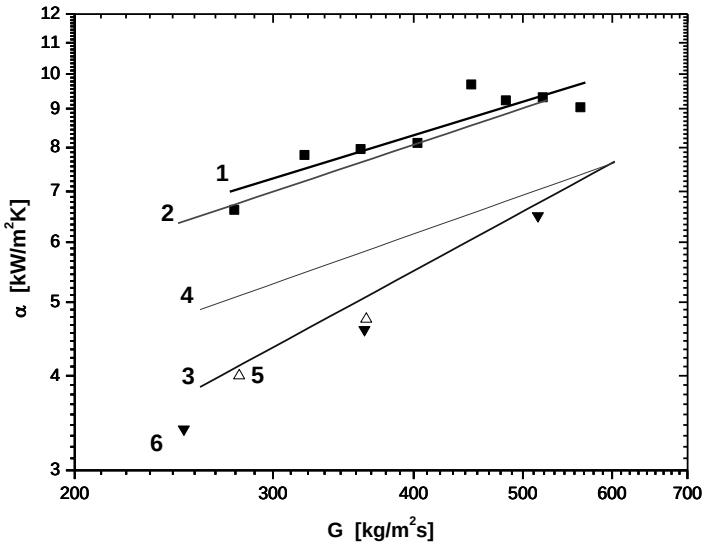

Fig. 3. Heat transfer coefficient for flow boiling of R22 in smooth stainless steel tube: 1 – ■ – present study, 2 – Cieśliński & Targański [41], 3 – Kandlikar correlation prediction [37], 4 – D. Mikielewicz et al. correlation prediction [38], 5 – □ – Greco & Vanoli [39, 42], 6 – □ – Jung et al. [40]

Figure 4 shows comparison of present results for flow boiling of pure R22 inside tube with porous coating with published data. Taking into account all differences in test tube design, i.e. diameter of the test tube, parameters and method of fabrication of the porous coating as well as different test conditions (vapor quality, mass velocity) present data are in reasonable agreement with experimental results obtained by Czikk et al. [19] and Ikeuchi et al. [20].

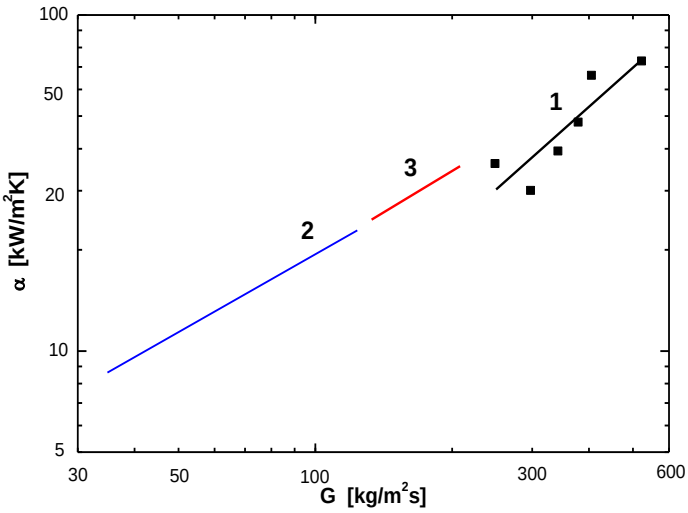

Fig. 4. Heat transfer coefficient for flow boiling of R22 in tube with porous coating: 1 – present study, 2 – Ikeuczi et al. [16], 3 – Czikk et al. [15]

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6.2. Pure refrigerant and refrigerant/oil mixture data

As an example, Fig. 5 and Fig. 6 show present results for flow boiling of pure R22 in smooth stainless steel tube and in tube with porous coating. A linear regression analysis using the least squares method was applied to determine the best-fitting straight line. Like in pool boiling, the application of a porous layer results in dramatic increase in heat transfer coefficient – Fig. 5. For all three tested pure refrigerants average heat transfer coefficient was 5 to 6 times higher than for a smooth tube for the same mass velocity. Simultaneously, lower pressure drop as compared with smooth tube for the same mass velocity and inlet/outlet vapour quality was recorded for the tube with porous coating - Fig. 6. That phenomenon can be explained by strong aeration of a thermal layer by vapour bubbles generated inside porous coating and as a result liquid core is separated from the rough surface of a porous layer.

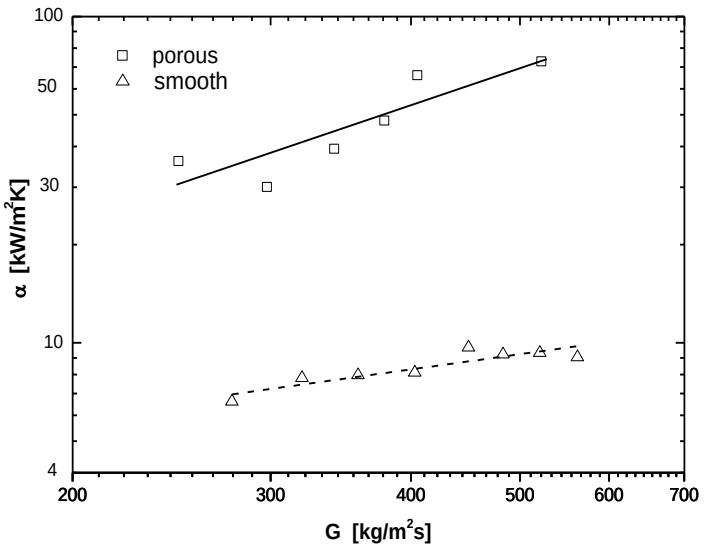

Fig. 5. Average heat transfer coefficient for flow boiling of pure R22 inside smooth tube and tube with porous coating

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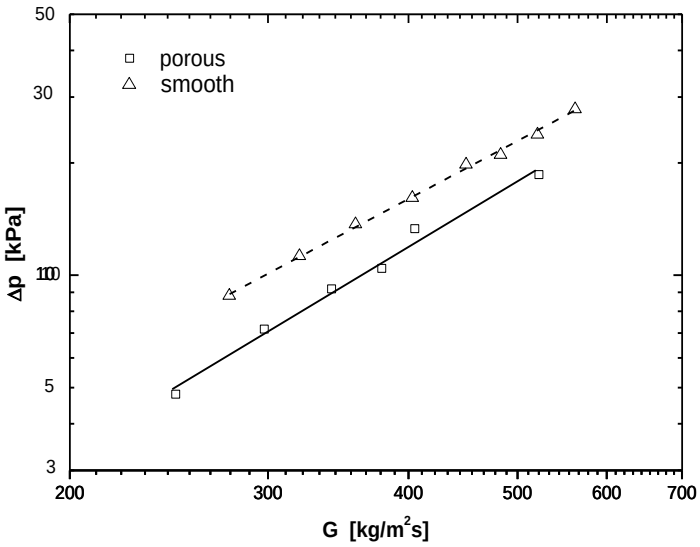

Fig. 6. Pressure drop for flow boiling of pure R22 inside smooth tube and tube with porous coating

Figure 7 and Fig. 8 display the influence of oil concentration on average heat transfer coefficient and pressure drop for R22 boiling in smooth stainless steel tube, respectively. For lean mixture (1% of oil concentration) heat transfer was slightly inhibited and simultaneously pressure drop has increased by about 35%. For rich mixture (5% of oil concentration) heat transfer rate and pressure drop have depended on mass velocity. For mass velocity below 400 $kgm^{-2}s^{-1}$ heat transfer coefficient increases with mass velocity increase, and pressure drop was almost the same as for pure refrigerant, but for higher mass velocity dramatic heat transfer degradation and simultaneously distinct pressure drop decrease have been observed – Fig. 7 and Fig. 8, respectively. Moreover, heat transfer hysteresis as well as pressure drop hysteresis have been recorded for increasing and decreasing mass velocity. The same phenomena have been observed for R134a/oil mixtures as well as R407C/oil mixtures during boiling in smooth tube.

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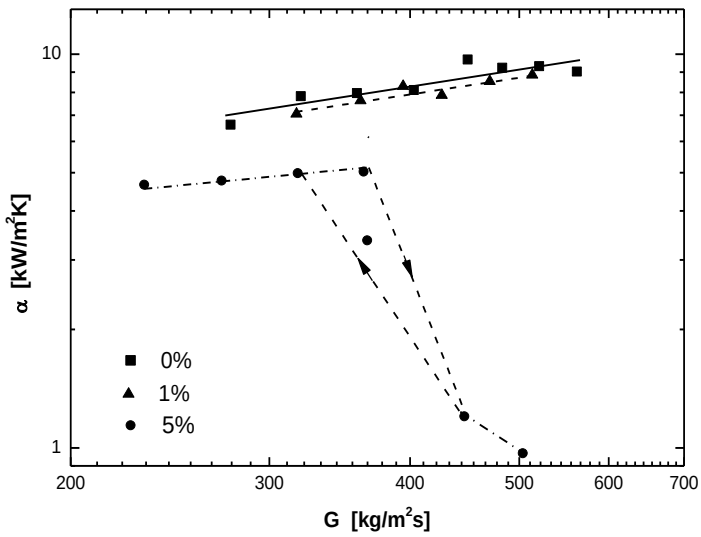

Fig. 7. Influence of oil concentration on average heat transfer coefficient for flow boiling of R22 in smooth stainless steel tube

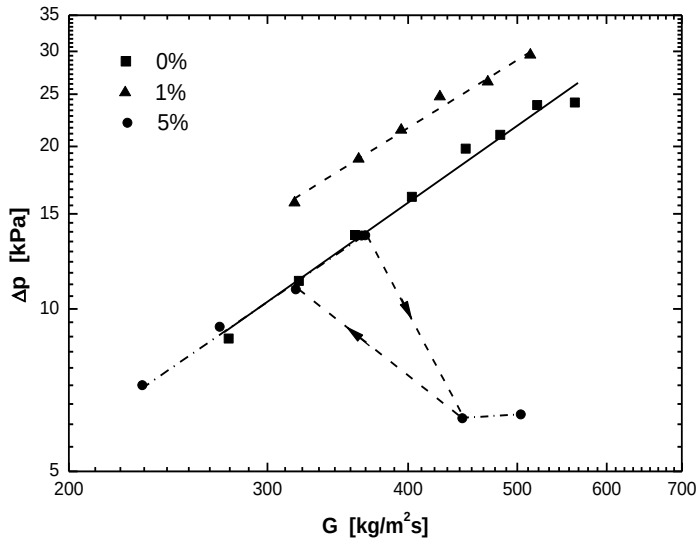

Fig. 8. Influence of oil concentration on pressure drop for flow boiling of R22 in smooth stainless steel tube

As an example, Fig. 9 and Fig. 10 illustrate the influence of oil concentration on average heat transfer coefficient and pressure drop for flow boiling of R407C and R407C/oil mixture in porous coated tube, respectively. Similarly as for smooth stainless steel tube, addition of even small amount of oil results in an evident heat transfer degradation in comparison with boiling of pure refrigerant. For lean mixture (1% of oil concentration) average heat transfer coefficient is two times lower than for boiling of pure R407C – Fig. 9 while pressure drop for lean mixture is almost the same as for boiling of pure refrigerant – Fig. 10. For rich mixture

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(5% of oil concentration) heat transfer coefficient decreases within the whole range of mass velocity investigated and similarly as for smooth stainless steel tube, for mass velocity above 400 $kgm^{-2}s^{-1}$ a kind of boiling crisis has been observed and with mass velocity increase heat transfer hysteresis has been recorded. Simultaneously, for mass velocity below 400 $kgm^{-2}s^{-1}$ pressure drop was ca. 35% higher than for boiling of pure R407C and for mass velocity above 400 $kgm^{-2}s^{-}1$ pressure drop exhibits hysteresis effect – Fig. 10.

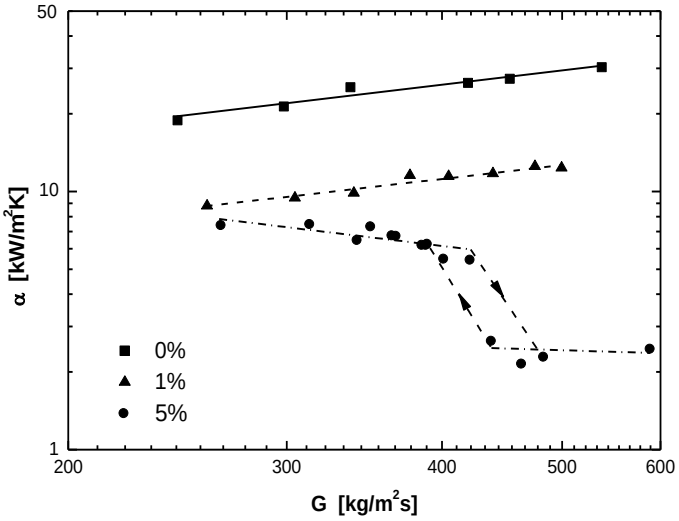

Fig. 9. Influence of oil concentration on average heat transfer coefficient for flow boiling of R407C in tube with porous coating

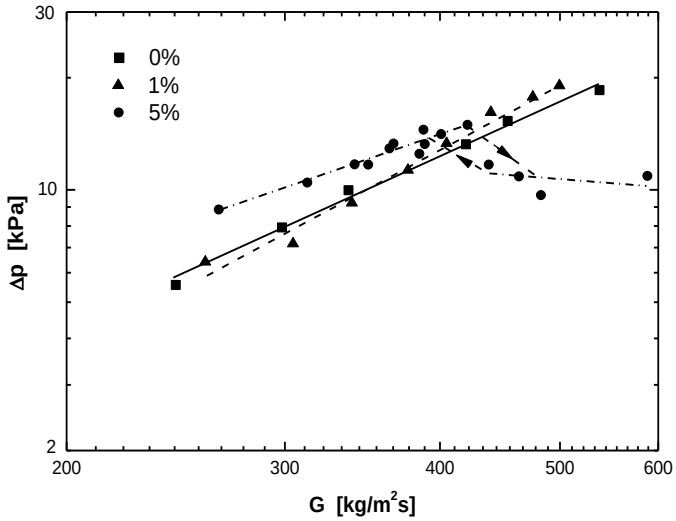

Fig. 10. Influence of oil concentration on pressure drop for flow boiling of R407C in tube with porous coating

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Figures 11 to Fig. 13 show average heat transfer coefficient for pure R22, R134a and R40C and their refrigerant/oil mixtures with 1% and 5% of oil concentration against mass velocity. Pure refrigerant R22 has an evident superiority over R134a and R407C while boiling inside tube with porous coating – Fig. 11. Average heat transfer coefficient of R22 and R134a lean mixtures (1% of oil concentration) is almost the same – Fig. 12, and for rich refrigerant/oil mixtures (5% oil concentration), average heat transfer coefficient was almost the same for all three refrigerant tested within the whole range of mass velocity investigated – Fig. 13.

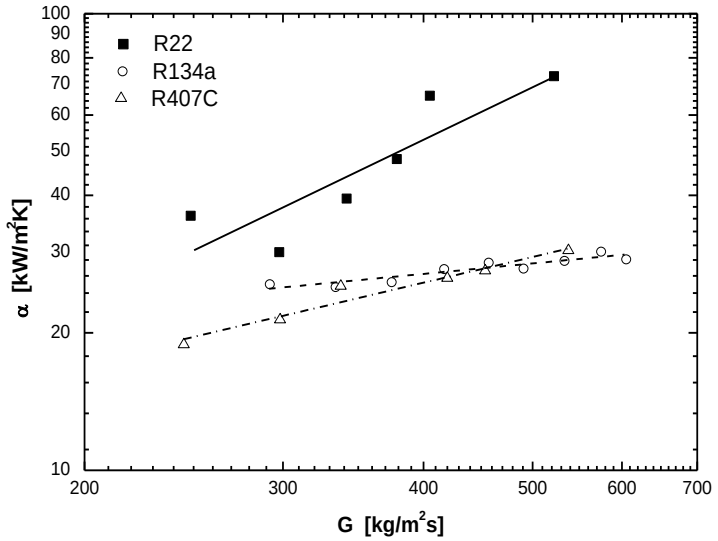

Fig. 11. Average heat transfer coefficient for flow boiling of pure refrigerant in tube with porous coating

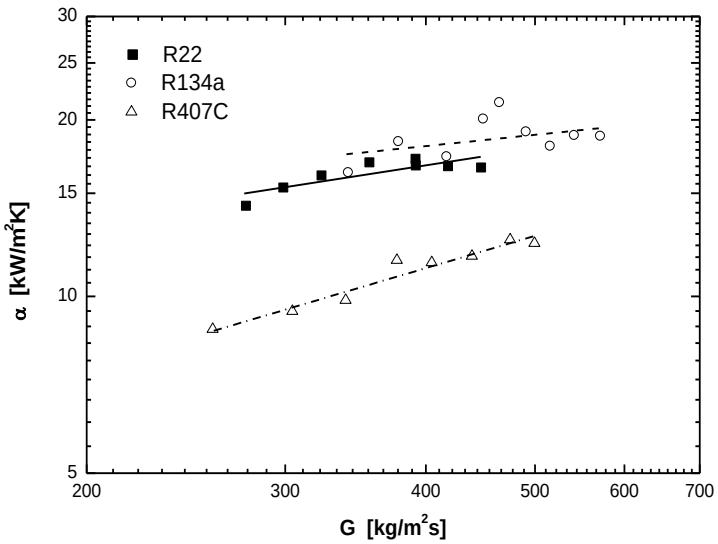

Fig. 12. Average heat transfer coefficient for flow boiling of refrigerant/oil mixture with 1% of oil concentration in tube with porous coating

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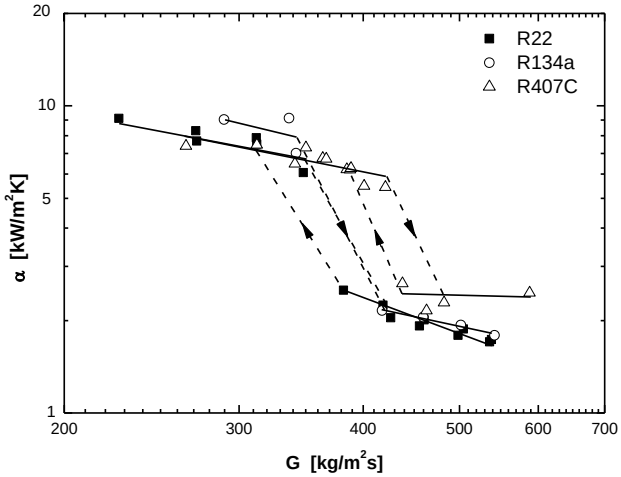

Fig. 13. Average heat transfer coefficient for flow boiling of refrigerant/oil mixture with 5% of oil concentration in tube with porous coating

Exemplarily, in Fig. 14 comparison of average heat transfer coefficient for boiling of R134a lean mixture (1% of oil concentration) inside smooth stainless steel tube and tube with porous coating is presented. As results from corresponding Fig. 15, enhancement factor EF is almost constant and equals ca. 2.8, penalty factor PF is lower than one and slightly increases with mass velocity increase. As a result heat transfer enhancement efficiency EF/PF is very high in the whole range of tested mass velocity, although slightly decreases with mass velocity increase.

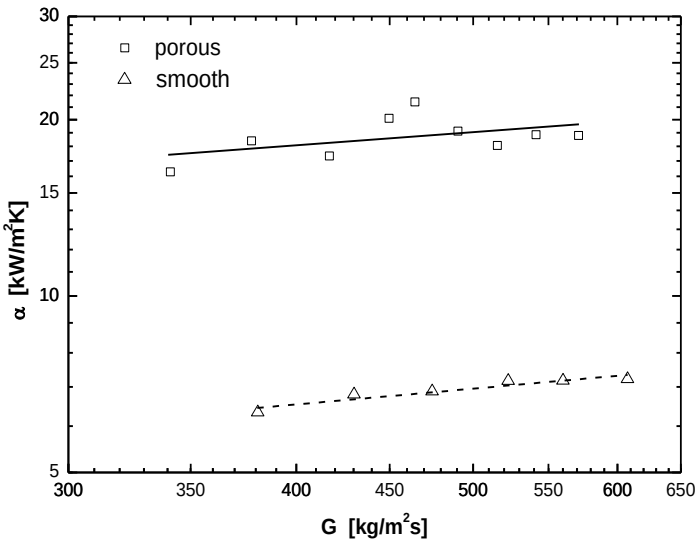

Fig. 14. Average heat transfer coefficient for flow boiling of R134a/oil mixture with 1% oil concentration in smooth stainless steel tube and in tube with porous coating

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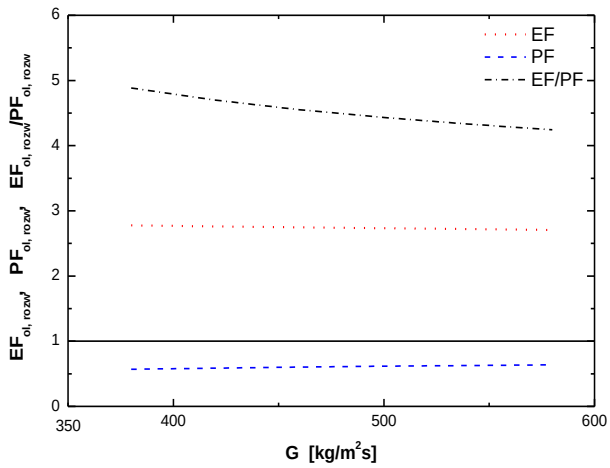

Fig. 15. Enhancement factor EF, penalty factor PF and heat transfer enhancement efficiency EF/PF for flow boiling of R134a/oil mixture with 1% oil concentration in tube with porous coating

6.3. Correlation equation for pure refrigerants

In order to generalize present data for flow boiling of pure refrigerants inside a tube with porous coating, correlation equation originally developed by D. Mikielewicz et al. [42] has been proposed.

Particularly, new correlation for pool boiling heat transfer coefficient on a porous coated surfaces has been introduced

$$\alpha_{PB}^{*} = C \cdot q^{n} \tag{6}$$

where constant C and exponent n are given in Tab. 3.

Tab. 3. Constant C and exponent n in Eq. 6

	C	***n***
R22	2.94	1
R134a	3.18	0.68
R407C	3.24	0.66

Moreover, the exponents in originally proposed correction factor P^{*} have been adjusted using present experiment data. A modified version of P^{*} reads

$$P^{*} = 2.53 \cdot 10^{-3} \left(R_{M-S} - 1\right)^{1} \mathrm{Re}^{1,17}\, Bo^{0,65} \tag{7}$$

Finally, proposed form of a correlation for heat transfer coefficient in flow boiling of pure refrigerants inside tubes with porous coating reads

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$$\frac{\alpha_{TPB}}{\alpha_{REF}} = \sqrt{R_{M-S}^{0.76} + \frac{1}{1+P^*}\left(\frac{\alpha_{PB}^*}{\alpha_{REF}}\right)^2} \tag{8}$$

where $R_{M\text{-}S}$ is a Muller-Steinhagen&Heck two-phase flow multiplier

$$R_{M-S} = \left[1+2\left(\frac{1}{f_1}-1\right)x\right]\cdot(1-x)^{1/3} + x^3\frac{1}{f_{1z}}, \tag{9}$$

where functions f_1 and f_{1z} take the form: $f_1 = \left(\frac{\mu_L}{\mu_v}\right)^{0,25}\cdot\frac{\rho_v}{\rho_L}$ and $f_{1z} = \frac{\mu_v}{\mu_L}\cdot\frac{c_L}{c_v}\cdot\left(\frac{\lambda_L}{\lambda_v}\right)^{1,5}$.

Reference heat transfer coefficient α_{REF} is equal to liquid only heat transfer coefficient α_L and is calculated as

$$\alpha_L = 0{,}023\left(\frac{\lambda_L}{d}\right)\cdot \mathrm{Re}_L^{0,8}\,\mathrm{Pr}_L^{1/3}. \tag{10}$$

The results of calculations using relation (8) have been presented in Fig. 16. As can be seen quite satisfactory consistency is obtained for the set of experimental data of R22, R134a and R407C. Over 80% of experimental points is described by the correlation within ±30% error margin.

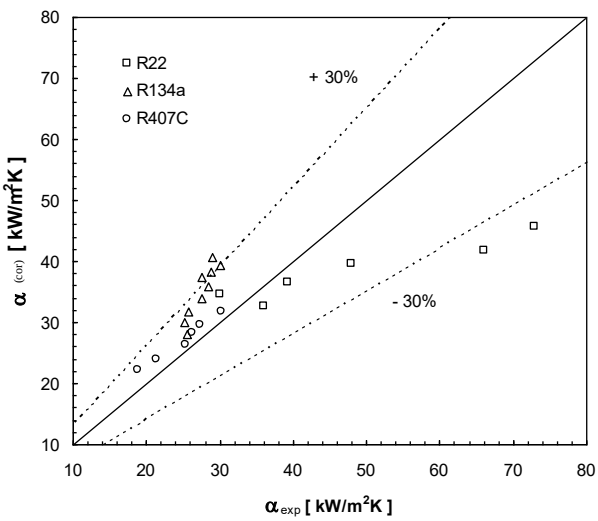

Fig. 16. Predicted vs. experimental data

7. CONCLUSIONS

The application of a porous coating on inside surface of a tube results in a higher – even 5 to 6 times, average heat transfer coefficient and simultaneously in lower pressure drop in comparison with smooth stainless steel tube for flow boiling of tested pure refrigerants R22,

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R134a and R407C.

Heat transfer deterioration for lean mixtures (1% of oil concentration) of all three refrigerants tested during flow boiling inside smooth stainless steel tube as well as in tube with porous coating has been observed, however average heat transfer coefficient degradation in tube with porous coating was more distinct. Nevertheless, absolute value of average heat transfer coefficient for lean mixtures flow boiling in a tube with porous coating is still ca. 2 times higher than for flow boiling of lean mixture in a smooth tube.

For rich mixtures (5% of oil concentration) heat transfer hysteresis and pressure drop hysteresis for both smooth tube and a tube with porous coating have been recorded. The nature of the phenomenon needs clarification. However, what is characteristic for both types of tested tubes is that hysteresis effect occurs within the mass velocity range 400-500 $kgm^{-2}s^{-1}$, which may indicate the hydrodynamic nature of the phenomenon.

Pure refrigerant R22 has an evident superiority over R134a and R407C while boiling inside smooth tube as well as in tube with porous coating.

Heat transfer enhancement efficiency EF/PF is well above one for boiling of pure refrigerants as well as lean mixtures (1% of oil concentration) in a tube with porous coating over the range of tested mass velocity, but decreases with mass velocity increase.

Correlation equation for heat transfer coefficient calculation during flow boiling of pure refrigerants inside a tube with porous coating has been proposed.

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