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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.

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.

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

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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

Non-dimensional numbers

Bo	boiling number, $\mathrm{Bo} = \frac{\dot{q}}{\mathrm{Gr}}$
Nu	Nusselt number, $\mathrm{Nu} = \frac{\alpha b}{\lambda}$
Pr	Prandtl number, $\mathrm{Pr} = \frac{c\mu}{\lambda}$
Re	Reynolds number, $\mathrm{Re} = \frac{\mathrm{Gb}}{\mu}$

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 five times better than for plain-tube performance for exit qualities between 70% and 95%. 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 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 and 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 (90 × 100 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 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 pressure – 6.9–17.6 MPa and mass velocity – 500–2500 kg/m^{2}s. Two geometries of test channel have been