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Pressure drop during evaporation of 1,1,1,2-tetrafluoroethane (R-134a) in a plate heat exchanger

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Abstract: Experimental results for the pressure drop during the evaporation of the refrigerant 1,1,1,2-tetrafluoroethane (R-134a) in a vertical plate heat exchanger are presented in this paper. The influences of mass flux, heat flux and vapor quality on the two-phase pressure drop are specially analyzed and compared with previously published experimental data and literature correlations. All results are given in graphical form as the dependency of the frictional pressure drop on the mean vapor quality.

Keywords: plate heat exchanger, evaporation, 1,1,1,2-tetrafluoroethane, pressure drop.

INTRODUCTION

The advantages of plate and frame (or gasketed) plate heat exchangers over shell and tube type heat exchangers may be summarized as:

a) better thermal performance,
b) lower space requirements,
c) easy accessibility to all areas and
d) lower capital and operating costs.

These are the reasons for the expanded application of plate heat exchangers (PHE) in recent years.

The use of plate heat exchangers in food, pharmaceutical and utility industries is known since the 1960s[1] but the application in the process industry had to wait for a further 20 years. Today, in refineries and petrochemical plants, PHE are applied to many hydrocarbon processes, including catalytic reforming, desulphurization, isomerization, aromatic recoveries, sour water treatment, gas separation, *etc.*[2]

Plate heat exchangers were also found to be useful for two-phase applications as evaporators or condensers in refrigeration and air conditioning systems, and in district heating systems with steam condensation.

On the other hand, one limitation of their application is the demand for higher allowable pressure drops. Since heat transfer and the pressure drop in PHEs

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are closely connected to each other and dependant on the plate geometry, they should be studied together; the characteristics of the pressure drop have to be known in order to predict thermal behavior.

A detailed literature survey of the current state of investigations concerning the evaporation heat transfer coefficient and two-phase pressure drop in PHEs was given in a previous study.[3]

A comparison of the heat transfer coefficients (or *j* factors) of a typical plate heat exchanger with a mild corrugation angle and shell-and-tube (S&T) heat exchanger showed that with the same allowable pressure drop, a PHE can give 1.5–2.5 times higher heat transfer rates.[2] On the contrary, a similar comparison of frictional factors, shown in Fig. 1, suggests that for the same *Re* number, this parameter can reach ten times higher values in a plate heat exchanger than in a shell-and-tube heat exchanger.

It was also confirmed[2] that the pressure drop in a PHE is very sensitive to the corrugation angle, much more than to the heat transfer characteristics. It can be seen in Fig. 2 that a change in the corrugation angle from 40 to 55º more than doubles the friction factor but increases the *j* factor by 50 %.

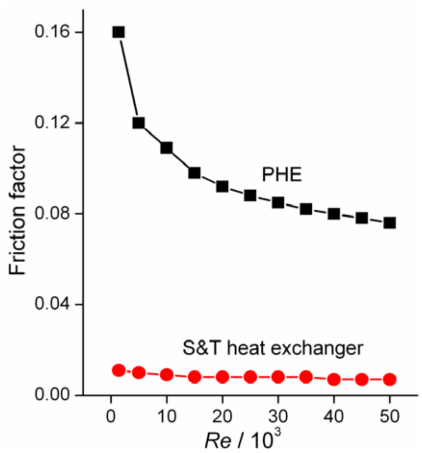

Fig. 1. Comparison of the frictional factors for PHE and S&T heat exchangers.

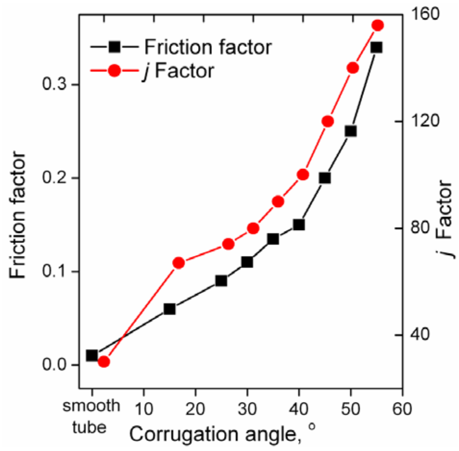

Fig. 2. Influence of plate corrugation angle on the *j* factor and the frictional factor.

In the study presented here, the focus was directed to the experimental investigation of the pressure drop during the evaporation process of R-134a in a plate heat exchanger and its dependency of mass flux, heat flux and vapor quality.

EXPERIMENTAL

A detailed description of the experimental setup used for investigation of the evaporation of the refrigerant R-134a in a vertical plate heat exchanger was given in the previous paper.[3] However, it should be noted here that it consists of two main loops, a refrigerant loop and a water–glycol loop, and a data acquisition unit.

The refrigerant includes several main elements: an evaporator, a separation vessel, an expansion valve, an inner heat exchanger, a compressor, two oil separators, a condenser, a refrigerant collector with level indicator, two sight glasses and two volume flowmeters, one at the evaporator inlet and the other just before the expansion valve. A vertical plate and frame heat exchanger are used as the evaporator and the condenser, respectively.

The water–glycol loop is constructed from two sub-cycles – each connected with one of the plate heat exchangers – the evaporator or the condenser. One of the characteristics of the water–glycol loop, in the evaporator sub-cycle, is a four-way valve, which enables the change of the water–glycol flow direction from concurrent to countercurrent and, consequently, the investigation of the influence of flow direction on the heat transfer and pressure drop.

The data acquisition system consists of the following elements: a recorder, a power supply and a personal computer. The experiment was monitored and controlled and a preliminary balance check was performed by a routine written in the LabVIEW program. The main screen is shown in Fig. 3, which includes a simplified schematic representation of the refrigerant loop with all the temperature, pressure and flow rate measuring points connected to the acquisition system. A detailed description of the measuring instrumentation and equipment, including the measuring accuracy, was given in the previous paper.[3]

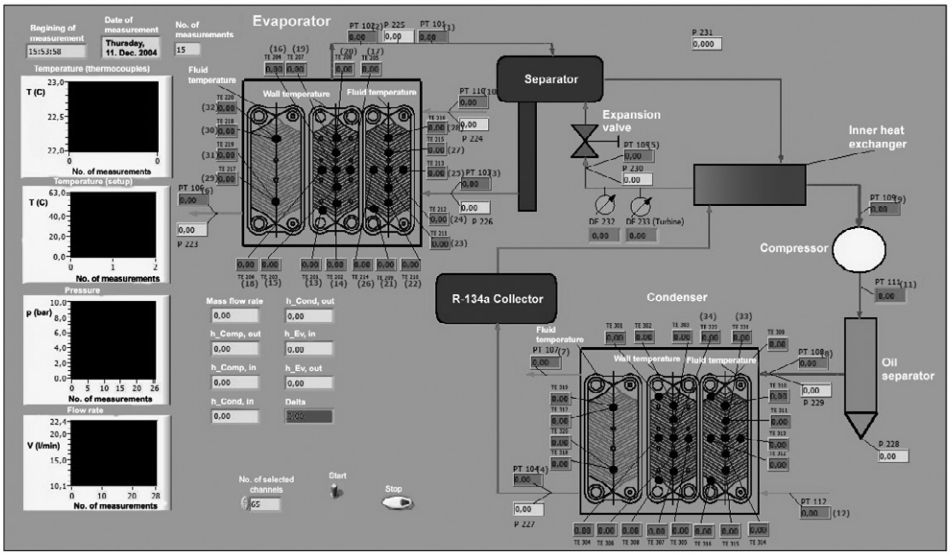

Fig. 3. Refrigerant cycle.

All the measurements were performed in a stationary state regime and the time dependency of the measured process parameters could be followed in the diagrams on the left side of the screen. Results of the preliminary balance check were also shown on the main screen. A second and more accurate balance check was performed after the experiment as a part of a data reduction procedure. Only then could the further calculation of the heat transfer coefficient and pressure drop be undertaken.

Details of the calculations of the heat transfer coefficient were described in our previous work.[3]

For the vertically upward refrigerant flow, the frictional pressure drop can be calculated from the Equation:

$$\Delta p_{\mathrm{f}} = \Delta p_{\mathrm{exp}} - \Delta p_{\mathrm{man}} - \Delta p_{\mathrm{acc}} - \Delta p_{\mathrm{ele}} \tag{1}$$

The acceleration and elevation pressure drops were estimated by the homogenous model for two-phase flow:[4]

$$\Delta p_{acc} = m_{\text{flux,r}}^2 \Delta v_{\text{lg}} \Delta x \quad (2)$$

$$\Delta p_{\text{ele}} = \frac{gL_{\text{p}}}{v_{\text{m}}} \quad (3)$$

The mean specific volume of a homogeneously mixed, vapor-liquid flow v_{m} can be determined as:

$$v_{\text{m}} = xv_{\text{g}} + (1-x)v_{\text{l}} = v_{\text{l}} + x\Delta v_{\text{lg}} \quad (4)$$

where Δv_{lg} is the difference between the specific volumes of the vapor and liquid:

$$\Delta v_{\text{lg}} = v_{\text{g}} - v_{\text{l}} \quad (5)$$

The pressure drops in the inlet and outlet manifolds and ports can be calculated from the empirical correlation:[5]

$$\Delta p_{\text{man}} \approx 1.5\left(\frac{u_{\text{m}}^2}{2v_{\text{m}}}\right)_{i,o} \quad (6)$$

while the mean flow velocity u_{m} can be expressed as:

$$u_{\text{m}} = m_{\text{flux,r}} v_{\text{m}} \quad (7)$$

Finally from the definition of the friction factor, its value can be obtained as:

$$f_{\text{tp}} = -\frac{\Delta p_{\text{f}} D_{\text{h}}}{2m_{\text{flux,r}}^2 v_{\text{m}} L_{\text{p}}} \quad (8)$$

Mass flux of the refrigerant and the hydraulic diameter were calculated from the equations:

$$m_{\text{flux,r}} = \frac{m_{\text{r,ch}}}{2aB_{\text{p}}} \quad (9)$$

$$D_{\text{h}} \approx 4a \quad (10)$$

An uncertainty analysis was conducted using a formula proposed by Kline and McKlintock[6] and the evaluation results are presented in the previous work.[3]

RESULTS AND DISCUSSION

In the present investigation of R-134a evaporation in a vertical plate heat exchanger, series of experiments were conducted under different test conditions. The evaporation temperature was varied from –8.85 to 11.08 °C (saturation pressure from 0.21 to 0.43 MPa), the values of the refrigerant mass flux were between 40 and 90 kg m^{-2} s^{-1}) and the imposed heat flux was gradually increased from 9 to 15 kW m^{-2}. Thermophysical properties of R-134a are taken from the REFPROP database.[7] The calculated values of pressure drop are presented as a dependency on the mean vapor quality x_{m} in the plate heat exchanger. The mean vapor quality was defined and calculated as the arithmetic mean value between the inlet and the outlet vapor qualities.

Comparison of the current data with previous measurements involving evaporation of the refrigerant R-134a[8] showed satisfactory agreement, although the experiments were conducted using a plate heat exchanger of smaller size, different geometry, with a smaller number of single plates and at the room temperature (25–31 °C). The results presented in this study were obtained using a plate heat exchanger with a larger number of plates in order to approach real exploittation conditions, and at the lower temperatures. The characteristics and dimensions of the plates are summarized in Table I.

TABLE I. Plate dimensions

Length, L_p / mm	872
Width, B_p / mm	486
Amplitude, a / mm	1.6
Wave length, λ / mm	12
Plate thickness, δ_p / mm	0.6
Thermal conductivity, λ_p / W m^{-1} K^{-1}	15
Corrugation angle, ψ / °	63.26

The influences of the mass flux of the refrigerant, imposed heat flux and vapor quality in the two-phase pressure drop are now closely analyzed. Selected data shown in Fig. 4 represent the dependency of the frictional pressure drop on the vapor quality for three different mass fluxes. It can be seen that the pressure drop rises with increasing vapor quality, but less significantly than in the case of the heat transfer coefficient.[3] Higher mass fluxes also induces higher pressure drops, as can be seen in Fig. 4. This is the consequence of the fact that a higher mass flux also means a higher velocity of the two-phase flow and thus a higher pressure drop.

The effects of heat flux on the frictional pressure drop are presented in Fig. 5. Two heat fluxes are compared under the same conditions of mass flux and system pressure. It seems that the pressure drop is only slightly affected by increasing heat flux, less than in the previous case of the influence of mass flux. A similar behavior was noticed previously when the influences on the heat transfer coefficients were analyzed.[3]

In addition to the experimental results for the friction pressure drop Δp_f, the values calculated from correlations based on the heterogeneous pressure drop model and the Lockhard–Martinelli approach, as described in the literature,[9] are also presented in Fig. 5:

$$\Delta p_{\mathrm{lit}} = \Delta p_{\mathrm{1ph}} \left(1 + \frac{5}{X_{\mathrm{tt}}} + \frac{1}{X_{\mathrm{tt}}^2} \right) \tag{11}$$

where X_{tt} is the Martinelli parameter:

$$X_{\mathrm{tt}} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\mu_{\mathrm{r,l}}}{\mu_{\mathrm{r,g}}} \right)^{0.1} \left(\frac{\rho_{\mathrm{r,g}}}{\rho_{\mathrm{r,l}}} \right)^{0.5} \tag{12}$$

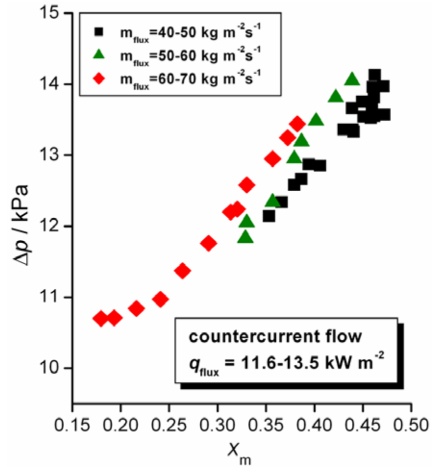

Fig. 4. Influence of mass flux on the pressure drop.

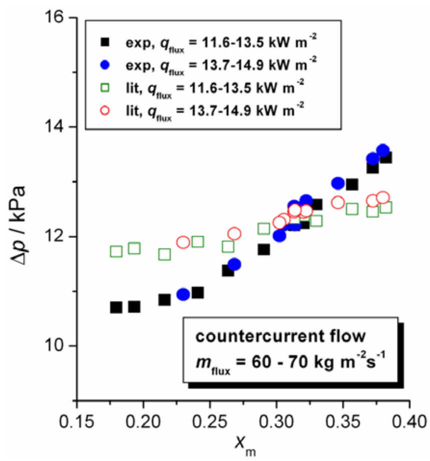

Fig. 5. Influence of heat flux on the pressure drop.

The agreement between the experimental and literature values is very good with a maximum deviation of approximately 10 % for all presented cases, as can be seen in Fig. 6.

The results of the experiments obtained under various test conditions are compared in Fig. 7 with previous measurements reported in the literature for the same refrigerant.[8] Although differences in plate geometry and working conditions exist, the amplitude, the wave length and the corrugation angle are similar, which gave a reasonable basis for comparison. It can be concluded from Fig. 7 that the agreement between the two series of measurements is fair.

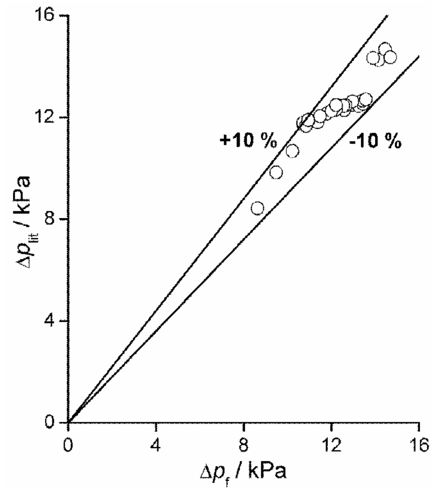

Fig. 6. Comparison of the current data with literature correlations.

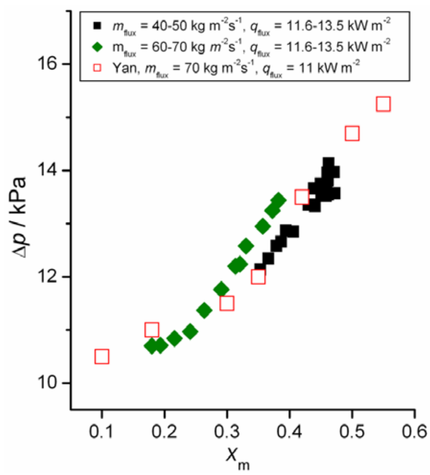

Fig. 7. Comparison with previous experimental data.[8]

CONCLUSIONS

The results presented in this paper show that both the mass flux and heat flux influence, to some extent, the frictional pressure drop during the evaporation process. The pressure drop is also a function of vapor quality, although the effect is not as significant as in the previously reported case of the heat transfer coefficient.[3] Comparison with previously reported measurements for the same refrigerant under different test conditions and with a different plate geometry[8] shows a good agreement, which opens the possibility for future successful and accurate predicttion of pressure drop in new processes involving this refrigerant.

NOMENCLATURE

A – Amplitude of plate corrugation, m
B – Width, m
D_h – Hydraulic diameter, m
f_{tp} – Friction factor
g – Gravitational acceleration, m s^{-2}
L – Length, m
mch – Mass flow rate through one of the channels, kg s^{-1}
m_{flux} – Mass flux, kg m^{-2}s^{-1}
q – Heat flux, W m^{-2}
u_m – Mean flow velocity, m s^{-1}
v_g – Vapor specific volume, m^3 kg^{-1}
v_l – Liquid specific volume, m^3 kg^{-1}
v_m – Specific volume of mixed vapor and liquid flow, m^3 kg^{-1}
x_m – Mean vapor quality
X_{tt} – Martinelli parameter

Greek letters

δ_p – Thickness of the plate, m
Δp_{acc} – Acceleration pressure drop, Pa
Δp_{ele} – Elevation pressure drop, Pa
Δp_{exp} – Experimental pressure drop, Pa
Δp_f – Friction pressure drop, Pa
Δp_{ma} – Pressure drop in ports and manifolds, Pa
Δx – Change of vapor quality between inlet and outlet
λ – Wavelength of plate corrugation, m
λ_p – Thermal conductivity of plate material, W/mK
μ – Viscosity, Pa s
ρ – Density, kg m^{-3}
ψ – Angle of plate corrugation, deg

Subscripts

ch – Channel
g – Gas
i – Inlet
l – Liquid
o – Outlet
p – Plate
r – Refrigerant

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ИЗВОД

ПАД ПРИТИСКА ПРИ ИСПАРАВАЊУ 1,1,1,2-ТЕТРАФЛУОРЕТАНА (R-134a) У ПЛОЧАСТОМ РАЗМЕЊИВАЧУ ТОПЛОТЕ

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У овом раду су представљени експериментални резултати за пад притиска током проеса испаравања расхладног флуида R-134a у вертикалном плочастом размењивачу топлоте. Посебно су анализирани утицаји масеног и топлотног флукса на пад притиска у двофазном току и упоређени са раније објављиваним експерименталним подацима и корелацијама из литературе. Сви резултати су представљени у графичком облику, као зависност фрикционог пада притиска од средњег степена сувоће.

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