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STUDY OF SINGLE-PHASE CONVECTION AND CONDENSATION IN THERMOPLATE HEAT EXCHANGER (PART II)

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ABSTRACT

The heat transfer and pressure drop in a thermoplate heat exchanger operated as condenser have been investigated experimentally. In order to separate the heat transfer resistances in the condensation process, first the single phase forced convection in the thermoplate using distilled water and Marlotherm oil has been studied and correlation for the Nusselt number and friction factor developed. For the condensation experiments an apparatus has been conceived comprising two identical condensers made of the same thermoplate type as employed in the single phase experiments. Isopropanol was used as test fluid at pressures below the atmospheric pressure. With the aid of the results obtained in the single phase studies the heat transfer resistances in the condensation experiments were separated and expressions for the condensation heat transfer and pressure drop developed.

1 INTRODUCTION

A thermoplate consists of two metallic sheets, which are spot-wellded according to an appropriate pattern, whereas the edges – except for the connecting tubes – are continuously seam-welded. By applying the hydro-form technique, a channel having a complex geometry is established between the sheets. One fluid is conducted through this channel, the other one through the channel bounded by two neighbouring thermoplates, Figure 1. Depending on the process conditions, for instance, a specified pressure drop of the external or inside fluid at the required thermal duty, thermoplates are usually assembled in parallel at an appropriate spacing thus making a heat exchanger. Such apparatus are encountered in several areas of cooling/heating technique and process technology e.g. as condensers or evaporators.

Despite the various use of thermoplate heat exchangers, their thermo-hydrodynamic characteristics still remain widely unknown. No relevant fluid flow and heat transfer investigations have been reported in the literature, neither with the inside nor the outside fluid. This causes substantial uncertainties regarding the construction and design of the apparatus which are commonly surmounted in the practice at the expenses of the original advantages of thermoplates in comparison to other constructions by oversizing and higher material usage.

In order to mitigate this unsatisfactory situation, experiments on heat transfer and pressure drop in thermoplate apparatus are unavoidable. Such experiments should include the transport processes in the inside and the outside fluid as well.

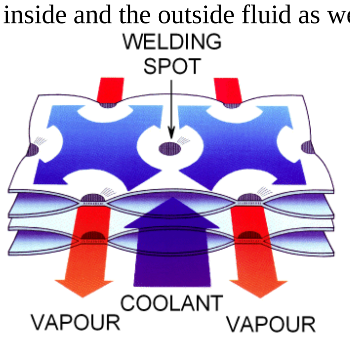

Figure 1: General view of fluid flow arrangement in a thermoplates apparatus.

In the present paper the results of a comprehensive experimental investigation of heat transfer and pressure drop in a thermoplate heat exchanger operated as condenser will be reported. The paper consists of two parts. Part I addresses the single phase forced convection heat transfer and pressure drop in a thermoplate. The experiments were performed with the Marlotherm oil and distilled water by electrically direct heating the thermoplate. Part II is devoted to vapour condensation in channels bounded by neighbouring thermoplates. It presents the heat transfer and pressure drop results obtained with the saturated isopropanol vapour at pressures below the atmospheric pressure. On the basis of the experiments correlations for the heat transfer and pressure drop of the cooling fluid as well as the condensing vapour are developed.

NOMENCLATURE

a, A	–	constant
A	m^2	heat transfer surface area
b	m	plate width
c, C, C_1	–	constant
c_p	$J/(kgK)$	constant pressure specific heat
d	m	channels width
d_h	m	hydraulic diameter
f	–	constant
g	–	constant
g	m/s^2	acceleration due to gravity
Δh	kJ/kg	condensation enthalpy
k	$W/(m^2K)$	overall heat transfer coefficient
K	–	constant
L	m	plate height
£	m	characteristic length
$\dot{m}$	$kg/(m^2s)$	mass flow density
m	–	exponent
$\dot{M}$	kg/s	mass flow rate
p	bar	pressure
$\dot{q}$	kW/m^2	heat flux
$\dot{Q}$	W	heat flow rate
s	m	sheet thickness
R	Ω	electric resistance
T	K, °C	temperature
V	m^3	volume
w	m/s	vapour velocity
x	–	vapour quality
X	–	degree of condensation
α	$W/(m^2K)$	heat transfer coefficient
δ	m	condense film thickness
κ	m^2/s	thermal diffusivity
κ	–	exponent
λ	$W/(mK)$	heat conductivity
μ	Ns/m^2	dynamic viscosity
ν	m^2/s	kinematic viscosity
ρ	kg/m^3	density
ζ	–	friction factor
τ	N/m^2	shear stress

Dimensionless quantities

Bo	Boiling number
Ka	Kapitza number
Nu	Nusselt number
Re	Reynonds number
Pr	Prandtl number

2 PART II

CONDENSATION HEAT TRANSFER AND PRESSURE DROP IN THE THERMOPLATE

2.1 Brief review of literature

Regarding the thermo-hydrodynamic behaviour of thermoplate apparatus operated as condensers the situation is much the same as with single phase convection considered so far. There are no papers in the literature dealing with the condensation heat transfer in a thermoplate channel. However, many papers are devoted to condensation in conduits having plain walls [1] or in vertical tubes [2]. Nusselt [3] was the first to investigate the condensation heat transfer on the basis of model equations. His theory is confined to laminar condensate films. Some 40 year later Rohsenow [4] and Labuntsov [5] formulated models for turbulent film condensation. Later on a number of turbulence models have been proposed starting from the Prandtl ideas on turbulent eddies. Their validity is confined to plane cooling surfaces. A review of these models can be found e. g. by Mitrovic [6].

As noted in Introduction, Part II of the present paper reports on the experimental results of condensation heat transfer and pressure drop studies with isopropanol as test fluid in a thermoplate apparatus. The geometrical parameters of the thermoplates investigated are the same as those with single phase studies the results of which, reported in Part I, are used for separation of the thermal resistances in the condensation experiments.

2.2 Methods of investigations

The condensation heat transfer was experimentally examined through condensing the isopropanol vapour by pumping the cooling water inside the thermoplates. With such arrangements the heat transfer coefficients α_{CW} and α_{CON} of the cooling water and of the condensing vapour as well as the overall heat transfer coefficient k determine heat flow rate $\dot{Q}$ by the well-known expressions:

$$\frac{1}{k} = \frac{1}{\alpha_{CW}} + \frac{s}{\lambda} + \frac{1}{\alpha_{CON}} \qquad (1)$$

$$\dot{Q} = k\,A\,\Delta T, \qquad (2)$$

where A denotes the heat transfer surface area, s the thickness of the metallic plate sheets and ΔT the driving temperature difference.

Experimentally attainable are the temperature difference ΔT and the heat flow rate $\dot{Q}$. The heat transfer coefficient α_{CW} can be calculated by eq. (12) reported in Part I, so that the quantities k and α_{CON} can be obtained by eqs. (1) and (2).

2.3 Experimental apparatus

The experimental apparatus for the condensation studies is shown in Figure 2. It consists mainly of a loop for the test fluid and a further loop for the cooling water. The loop for the test fluid itself consists of an evaporator 1, a partial condenser 2 and a total condenser 3. In each condenser three commercial thermoplates are assembled verticaly and parallel to each other at an appropriate spacing (10 mm). This arrangement makes a thermoplate heat exchanger with two channels for the vapour stream and three flow channels for the cooling fluid, Figure 1.

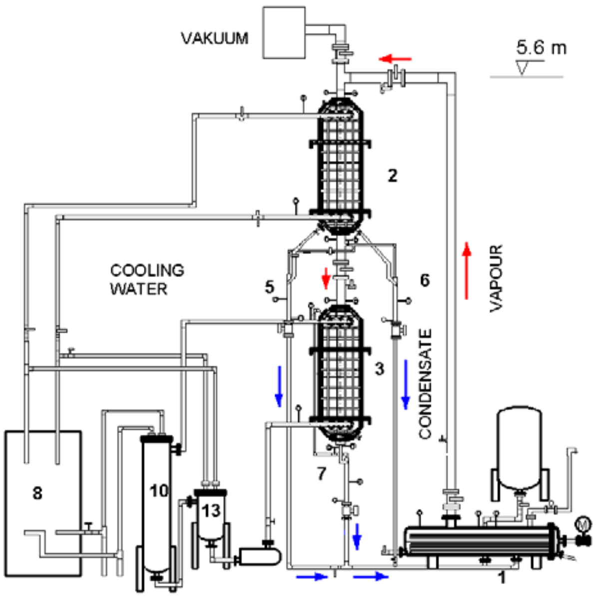

Figure 2: Apparatus for the condensation measurements.

Furthermore, the test fluid loop contains three condensate flow meters 5, 6 and 7. The flow meters 5 and 6 serve for the measurements of the condensate leaving the partial condenser 2, the flow meter 7 measures the condensate at the total condenser 3. The loop for the cooling water consists of a refrigerator 8, two heat exchangers 10 and 13 and a centrifugal pump 11. By means of the refrigerator 8 the thermoplates of the partial condenser 2 are cooled; the condensation temperature in the total condenser 3 is regulated by means of the heat-exchanger 13. The heat exchanger 10 release condensation heat from the total condenser to the main cooling water loop.

2.4 Measurement system and uncertainty

In order to seize the measuring data reliably, all measuring instruments were calibrated. The pressure transducers were calibrated by the manufacturer (Emerson). The uncertainty of these instruments is less than ± 0.075 %. The instrument for the measurement of the cooling water flow rates (Micro Motion elite CMF 050) exhibits an uncertainty of ± 0.15 % in a range of 500 kg/h to 4000 kg/h. The uncertainty of the calibrated thermocouples is less than ± 0.15 K. For the determination of the condense flow rates (devices 5, 6 and 7) a simple novel measurement method is used [7].
The acquisition and processing of the measuring signals were converted by two data acquisition units (Agilent 34970 A) and transmitted trough a GPIB interface to a PC. The signals from the Coriolis flow meters for the cooling fluid of the middle plate were conducted by an integrated transmitter directly to the computer. By means of an integrated multi-function card 34907A in the second Agilent unit, all valves, as pneumatic valves for condensate measurements, regulating valves for controlling of the evaporator power and pumps were operated by a computer. The measuring data were monitored and processed with the software lab VIEW.

2.5 Experimental procedures and evaluation

To determine the condensation heat transfer coefficient α_{CON} and the overall heat transfer coefficient k the apparatus in Figure 2 was operated as follows. The isopropanol vapour generated in evaporator 1 flows in the test condenser 2. Here the vapour is condensed partially and the rest vapour is condensed completely in the total condenser 3. The condensate flows are led afterwards into the evaporator 1. The condensate streams at the test condenser 2 were measured separately, the condensate stream from the middle thermoplate at the flow meters 5 and from the two lateral plates at the flow meter 6. The condensate stream in the total condenser was determined at the flow meter 7.
For the cooling of the test condenser 2 and stabilising the cooling temperature of the total condenser 3 the flow of the cooling water from the refrigerator 8 is split into two currents, the larger current flows through the test condenser and the smaller one through the heat exchanger 13. This measure was necessary to achieve a defined condensation temperature in the system. The experimental conditions (pressure, inlet steam velocity and condensation degree) have been varied by energy input in evaporator 1 and by variation of the cooling water parameters. After reaching the steady-state condition all the data were saved and processed accordingly.
The vapour velocity at the inlet of the test condenser 2 can be calculated by:

$$w_V = \frac{\dot{M}_V}{\rho_V A_S}, \tag{3}$$

where $\dot{M}_V$ is the vapour stream at the inlet of the partial condenser, A_S the cross-sectional flow area of the vapour channels in this condenser and ρ_V is the vapour density. The vapour flow rate was obtained by measuring the condensate flow rates $\dot{M}_5$ of the middle plate, of the lateral plates $\dot{M}_6$ and of the total condenser $\dot{M}_7$, $\dot{M}_V = \dot{M}_5 + \dot{M}_6 + \dot{M}_7$, where the indices refer to the measuring position.
The heat flow rate $\dot{Q}$ and the temperature difference ΔT in eq. (2) were obtained in common way:

$$\dot{Q} = \dot{M}_{CW} \cdot c_{pCW}\left(T_{CW\,OUT} - T_{CW\,IN}\right), \tag{4}$$

$$\Delta T = \left(\Delta T_{OUT} - \Delta T_{IN}\right)/\ln\left(\Delta T_{OUT}/\Delta T_{IN}\right), \tag{5}$$

$$\Delta T_{IN} = T_{CW\,OUT} - T_{V\,IN}, \tag{6}$$

$$\Delta T_{OUT} = T_{CW\,IN} - T_{V\,OUT}, \tag{7}$$

The symbols have the usual meanings, the indices OUT, IN, CW, CON and V refer to the outlet, inlet, cooling water, condensation, and vapour, respectively. All the physical properties were taken at the average temperature of the fluids.
With the mass flow of the condensate at the middle plate of the partial condenser and the total circulating isopropanol stream a condensation degree X is defined:

$$X = \frac{\dot{M}_5}{\dot{M}_V} = \frac{\dot{M}_5}{\dot{M}_5 + \dot{M}_6 + \dot{M}_7}. \tag{8}$$

This quantity measures the mass of condensate leaving the test plate in terms of the vapour flow rate at the inlet in the condenser 2, Figure 2.

The Reynolds- and the Nusselt numbers are given as:

$$Re_{CON} = \frac{w\delta}{\nu_L} = \frac{\dot{M}_5}{2b\,\eta_L} \qquad (9)$$

$$Nu_{CON} = \frac{\alpha_{CON} \cdot £}{\lambda_L} \qquad (10)$$

$$£ = \left(\frac{\rho_L}{\rho_L - \rho_V} \frac{\nu_L^2}{g} \right)^{1/3}, \qquad (11)$$

where b is the plate width.

The measured total pressure drop consists of three main contributions, the friction pressure drop Δp_F, the momentum charge due to condensation Δp_{CON}, and the hydrostatic pressure difference Δp_{HYD},

$$\Delta p = \Delta p_F - \Delta p_{HYD} - \Delta p_{CON} \,. \qquad (12)$$

To obtain the friction pressure drop Δp_F, the quantities Δp_{CON} and Δp_{HYD} are estimated by the homogeneous model for a two phase flow [8],

$$\Delta p_{CON} = \left(\frac{1}{\rho_V} - \frac{1}{\rho_L} \right) (x_{IN} - x_{OUT})\dot{m}^2, \quad \dot{m} = \frac{\dot{M}_V}{b \cdot d_h} \qquad (13)$$

$$\Delta p_{HYD} = \rho_m \, g L, \qquad (14)$$

$$\frac{1}{\rho_m} = \frac{x_m}{\rho_V} + \frac{(1 - x_m)}{\rho_L}, \qquad (15)$$

$$x_m = \frac{x_{IN} + x_{OUT}}{2}, \qquad (16)$$

$$x_{OUT} = \frac{\dot{M}_7}{\dot{M}_5 + \dot{M}_6 + \dot{M}_7}, \qquad x_{IN} = 1. \qquad (17)$$

In eq. (13) $\dot{M}_V$ denotes the total (both phases) mass flow.

With the pressure drop Δp according eq. (12) and the contributions stated in eqs. (13) and (14) the friction pressure drop Δp_F can be calculated and the friction factor ζ defined as:

$$\zeta = \frac{1}{2} \frac{\Delta p_F}{\dot{m}^2 / \rho_m} \frac{d_h}{L}. \qquad (18)$$

From the friction pressure drop Δp_F also the shear stress τ_δ at the condense film surface can be calculated by

$$\tau_\delta = \frac{\Delta p_F \cdot d}{2 \cdot L}, \qquad (19)$$

where L is the plate length and d the width of the condensation channel.

2.6 Illustration and discussion of the results

The heat transfer and pressure drop are investigated at the condensation temperatures of 60 °C and 70°C (vapour pressure of isopropanol 400 mbar and 640 mbar), the mass flow rates between 200 kg/h to 600 kg/h and the heat fluxes of 12 kW/m^2 to 30 kW/m^2. The measured condensation heat transfer coefficient and friction pressure drop are presented in dependences of the condensation degree in the partial condenser. The experimental heat transfer data are compared with the ones obtained by the Nusselt theory as well as from calculation procedures by Claus [10] and Blangetti [11]. Finally, correlation equations basing on the present data are proposed.

2.6.1 Heat transfer

The effects of the condensation degree X and inlet vapour velocity on the overall heat transfer coefficient k are shown in Figure 3.

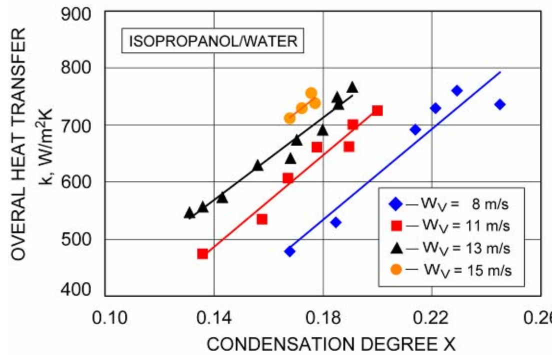

Figure 3: Overall heat transfer coefficients of isopropanol at different inlet vapour velocities as a function of condensation quality, system pressure $p = 400$ mbar.

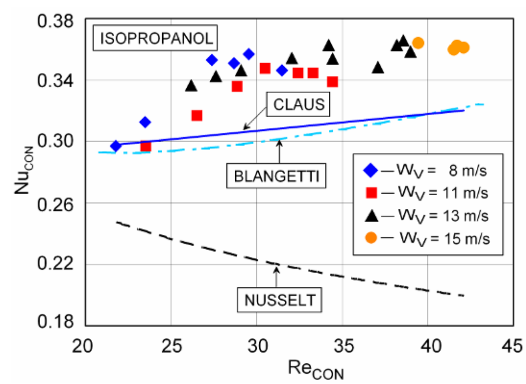

Figure 4: Comparison of the experimental Nusselt numbers with calculations from literature.

The quantity k increases almost linearly with the condensation degree X. A higher vapour velocity results in a larger shear stress in the condense film thus decreasing its thickness which leads to a better heat transfer from the film surface to the thermoplate wall.

Figure 4 compares the experimental data of isopropanol with those calculated by the equations proposed by Blangetti [10] and Claus [11]. In the correlation by Claus, the turbulence properties reach at the film surface their maximal values, while the correlation by Blangetti takes into account the effect of vapour shear at the interface. Both are deduced from condensation experiments with plain cooling surfaces. The experimental data with the thermoplate lie above the calculated ones which is primarily caused by the thermoplate structure.

2.6.2 Pressure Drop

The total pressure drop of the condensing isopropanol is presented in Figure 5. The pressure drop is observed to decrease with the condensation degree X. At a low condensation degree the vapour friction dominates the pressure drop. An increase of the condensation degree thereafter results a reduction of friction pressure drop, while the momentum change due to phase transition increases. The summation of the total pressure drop (eq. 12) at higher condensation degree is therefore smaller. Moreover, the shear stress due to vapour velocity leads to a larger total pressure drop. In general, the maximum pressure drop in the thermoplate condenser channels does not exceed 4 mbar, which is in comparison with corrugated plate heat exchanger considerably lower [12].

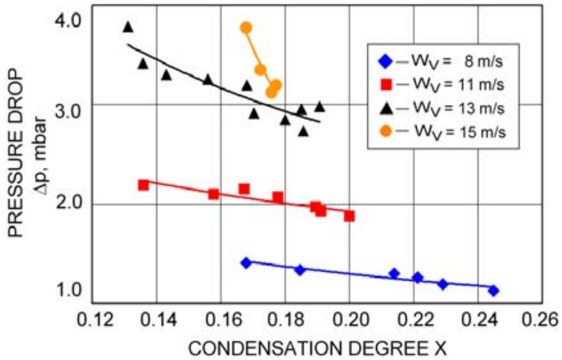

Figure 5: Pressure drop of isopropanol in thermoplate as a function of condensation degree at different vapour velocity; system pressure 400 mbar.

2.6.3 Correlations for heat transfer and pressure drop

In the literature, there are several correlations to calculate the condensation heat transfer [6]. These are based on simple equation of the form

$$Nu_{CON} = f(Re_{CON}, Re_V, Pr_L). \qquad (20)$$

and obtained by correlation of experimental data, e.g. Claus [11], or theoretically by model calculations [5, 6]. Some correlations follow from analogy considerations to two-phase flows in pipes, e.g. Yan [12], Shah [9].

With condensation of flowing vapours Blangetti [30] recommended the expression

$$Nu_{CON} = K\cdot Re_{CON}^{m_1}\cdot Pr_L^{m_2}(1+f\cdot\tau_\delta^*)^{m_3}, \qquad (21)$$

where the parameters K, m_1 to m_3 were determined from experimental data. The shear stress τ_δ^* at the phase boundary is to seize the influence of the vapour flow on the heat transfer.

The existing correlations do not satisfactory describe our experimental data with thermoplates. For this reason we develop a new correlation for the Nusselt number defined as follows:

$$Nu = \frac{\alpha_{CON}\cdot\delta}{\lambda_F} \qquad (22)$$

$$\delta = \frac{\tau_W}{(\rho_L-\rho_V)g} \qquad (23)$$

This definition of the Nusselt number considers the shear stress τ_δ at the film surface ($\tau_W = \Delta\rho\delta g + \tau_\delta$) which may play an important role in the thermoplate.

Although the dependence of the Nusselt number on the fluid properties in eq. (20) can be acquired in the Prandtl number as suggested e. g. by Blangetti [10], in the present considerations the Kapitza number Ka is introduced as a further parameter. This quantity should take into account the effects surface tension on the turbulence in the interface film region and the interface roughness (dynamic roughness) on the shear stress. The waviness of the cooling surface should also be considered when developing a correlation for the thermoplate, and instead of eq. (20) the following ansatz is adopted:

$$Nu = C\cdot\left(\frac{\theta}{\delta_N}\right)^{n_1}\cdot Pr_L^{n_2}\cdot\left(\tau_W^*\right)^{n_3}\cdot Ka^{n_4}, \qquad (24)$$

where θ is the wave amplitude of the thermoplate surface ($\theta = 1.7\ mm$) and δ_N accounts for the film thickness as obtained (qualitatively) by Nusselt,

$$\delta_N = \left(\frac{\nu_L^2}{g}\cdot Re_{CON}\right)^{1/3}, \qquad (25)$$

$$Ka = \frac{\sigma}{\rho_L g^{1/3}\nu_L^{4/3}}, \qquad (26)$$

$$\tau_W^* = \frac{\tau_W}{£(\rho_L-\rho_V)g}. \qquad (27)$$

Because of the small film thickness, τ_W may be set equal to the interfacial shear stress τ_δ.

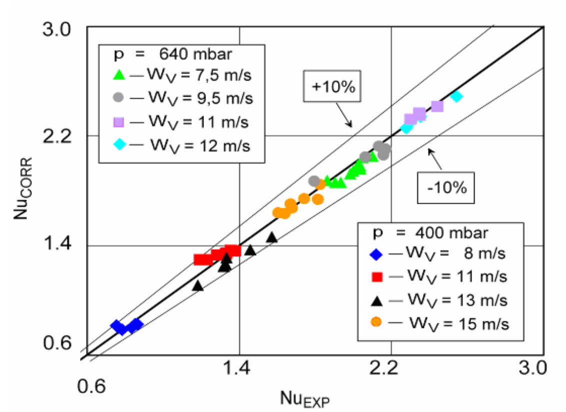

Figure 6: Comparison of the proposed correlation (Eq. (24)) for the heat transfer with the present experimental data the condensing isopropanol in thermoplate.

By adjustment of the experimental data with isopropanol at $\tau_W \approx \tau_\delta$ the constants C, $n_1 - n_4$ in eq. (24) were determined as follows: $C = 0.208$; $n_1 = -0.5977$; $n_2 = 0.45348$; $n_3 = 0.98599$; $n_4 = 0.03897$

With these values eq. (24) reproduces the measured Nusselt numbers with a deviation smaller than ± 10 %, Figure 6. This correlation applies only to the geometry of the examined thermoplate and the medium isopropanol at Reynolds number Re_{CON} between 10 and 70.

The pressure drop correlation proposed in the present study in terms of the friction factors ζ is based on an analogy to the two-phase flow [12] and has the shape

$$\zeta = C_1\cdot Re_{eq}^{\kappa 1}\cdot Bo^{\kappa 2}\cdot Ka^{\kappa 3} \qquad (28)$$

where Re_{eq} is the Reynolds number of the two-phase flow,

$$Re_{eq} = \frac{\dot m_{eq}\cdot d_h}{\mu_L}, \qquad (29)$$

$$\dot m_{eq} = \dot m\cdot\left((1-x_m)+x_m\cdot(\rho_L/\rho_V)^{0,5}\right), \qquad (30)$$

$\dot{m}$ being the mass flow density (both phases) and x_m the average vapour quality in the condensation channel.

The Kapitza number Ka is given in eq. (26), while the quantity Bo is to be calculate by:

$$Bo = \frac{\dot{q}}{\dot{m} \cdot \Delta h} \tag{31}$$

The ansatz (28) considers the effects of the film waviness affected by the surface tension (Ka) and the vapour suction due to condensation (Bo) on the pressure drop.

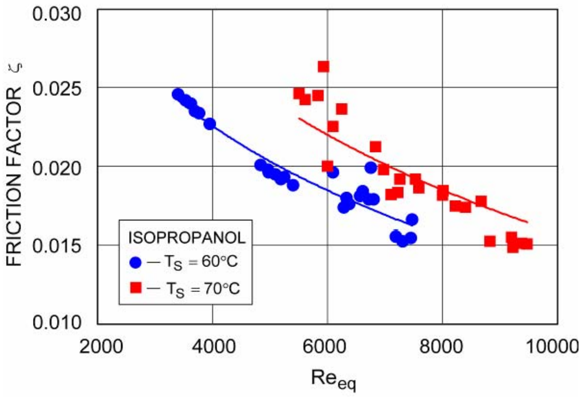

Figure 7: Condensation friction factor of isopropanol in the thermoplate at different temperatures.

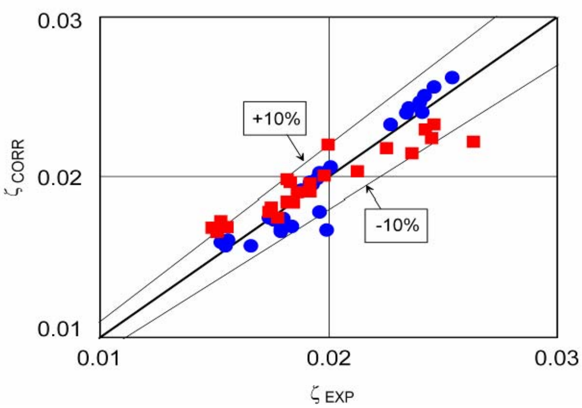

Figure 8: Comparison of the proposed correlation (Eq. (28)) for the friction factor with the experimental data for the condensation of isopropanol in the thermoplate.

The constants C_1, κ_1, κ_2, κ_3 in eq. (28) were determined from the experimental data as follows $C_1 = 0.01204$; $\kappa_1 = -0.61705$; $\kappa_2 = 0.00728$; $\kappa_3 = 0.84292$.

Figure 7 shows the friction factor ζ at different condensation temperatures, and Figure 8 compares the measured and the calculated friction factor data. The deviation of the two is less than ± 10 %.

3.7 Conclusions

Condensation heat transfer and pressure drop in the thermoplate with isopropanol as test fluid has been investigated at different vapour velocities, condensation degrees, system pressures and the film Reynolds numbers $20 < Re_{CON} < 70$. The heat transfer and pressure drop correlations were determined. The major results can be summarized as follows:

(a) In comparison with condensation on a vertical tube or plane plate, the thermoplate shows a better heat transfer due to the 3-D surface structure.
(b) The pressure drop in the condensation channel decreases due to the phase change, but the total pressure drop increase with increasing vapour velocity.
(c) Correlation equations for the condensation heat transfer and the friction factor proposed are valid for isopropanol and the examined thermoplate.

This study also includes the heat transfer and pressure drop of the inside fluid that were obtained on an apparatus specially devised for these requirements.

The expressions for the heat transfer and pressure drop of the inside (cooling) fluid and the outside condensation reported in this paper provide a more reliable basis for the calculations in the practice. These expressions contain dimensionless quantities which are considered to decisively affect the condensation kinetics. Despite the general form of the correlations, their reliable application is confined to the parameter ranges covered by the experiments.

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