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

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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

Originally, thermoplates were conceived as heat transfer devices for food and paper industry. Since the beginning of their development, some twenty years ago, thermoplate heat exchangers comprising about $2\cdot10^5\,m^2$ surface area are installed in various production and energy conversion processes. The chemical industry with more then $65\cdot10^3\mathrm{m}^2$ holds not only the top position, but also shows the highest annual increment rate [1].

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.

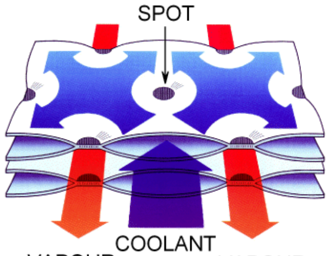

Figure 1: General view of fluid flow arrangement in a thermoplates apparatus.

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	m^2	heat transfer surface area
c_p	$J/(kgK)$	constant pressure specific heat
d_h	m	hydraulic diameter
g	m/s^2	acceleration due to gravity
I	A	electric current
k	$W/(m^2K)$	overall heat transfer coefficient
L	m	plate height
$\dot{M}$	kg/s	mass flow rate
n	-	exponent
p	bar	pressure
P	W	energy
Δp	N/m^2	pressure drop
$\dot{q}$	kW/m^2	heat flux
$\dot{Q}$	W	heat flow rate
s	m	sheet thickness
R	Ω	electric resistance
T	K, °C	temperature
u	m/s	fluid velocity
U	V	voltage
V	m^3	volume
α	$W/(m^2K)$	heat transfer coefficient
κ	m^2/s	thermal diffusivity
λ	$W/(mK)$	heat conductivity
ν	m^2/s	kinematic viscosity
ρ	kg/m^3	density
ζ	–	friction factor

Dimensionless quantities

Nu	Nusselt number
Re	Reynolds number
Pr	Prandtl number

Subscripts

Corr	correlation
CW	cooling water
F	film, fluid, friction
FIN	fluid at inlet
FOUT	fluid at outlet
h	hydraulic
IN	inlet
i	inside
L	liquid
lam	laminar
S	cross-sectional
W	wall

2 PART I

SINGLE PHASE FORCED CONVECTION IN A THERMOPLATE

2.1 Previous studies

Following a literature review, thermoplates are almost entirely unknown as heat transfer devices. A paper by Mitrovic and Maletic [2] appears to be the only one that is immediately devoted to thermoplates. A publication by Witry et al. [3] may be mentioned in this context in so far as they used an automotive aluminium plate radiator provided with staggered, equidistantly arranged, dimples in their fluid flow and heat transfer investigations. Even though this construction belongs basically to the family of thermoplate apparatus, the results achieved cannot be directly applied to thermoplates of the original design. On the other hand, there are several papers dealing with the fluid flow and heat transfer in flat channels [4, 5], corrugated plate heat exchangers [6, 10], or conduits with sine-like shaped walls, see e.g. Gradeck et al. [7], Wang and Chen [8], and Wang and Vanka [9]. Also in these cases are the results obtained not necessarily expected with thermoplates.

Correlations for the heat transfer in apparatus of a complex geometry, like corrugated plate heat exchangers, are mostly borrowed from the boundary layer theory and correspondingly modified. This makes the Wilson-plot method applicable for separation of the thermal resistances. However, this method is not recommendable in case of complex flow arrangements, like in a thermoplate, if a higher reliability of the experimental results is required. These facts demand for other experimental methods of heat transfer investigations with thermoplates.

2.2 Experimental apparatus

Figure 2 shows the flow sheet of the apparatus used in the single phase experiments on a thermoplate. The test fluid is circulated by the pump 1; the heat exchanger 2 adjusts the fluid temperature at the inlet of the thermoplate 3. The plate flowed by the fluid is switched as a resistance heater into an electric circuit and the electrical energy dissipated in the plate wall is taken up by the fluid as thermal energy. The plate is electrically disconnected from the test rig by means of POM (Polyoxymethlylen) inserts at the inlet and outlet. The electrical energy provided by a DC-power supply 6 is transported to the plate by massive copper strips that are tightly connected to the plate along its whole width. To maintain steady states, the test fluid is cooled in a heat exchanger 4 and conducted into the storage tank 5 that is connected to the pump 1. Heat losses to the surroundings are almost entirely suppressed by thermal insulation.

The therrmoplate made of 0.8 mm thick stainless steel sheets is vertically installed; its dimensions (width x height x thickness) are 300mm x 1000mm x 5mm, Figure 3. The plate is provided with staggered circular welding joints, 10 mm in diameter, which are arranged at the corners of an isoscale triangle 42 mm apart. The cross – sectional fluid flow area amounts to $725 \times 10^{-6}\ m^2$.

The connection tubes (Ø 27 mm) for the fluid inlet and outlet are positioned in the middle of the plate width.

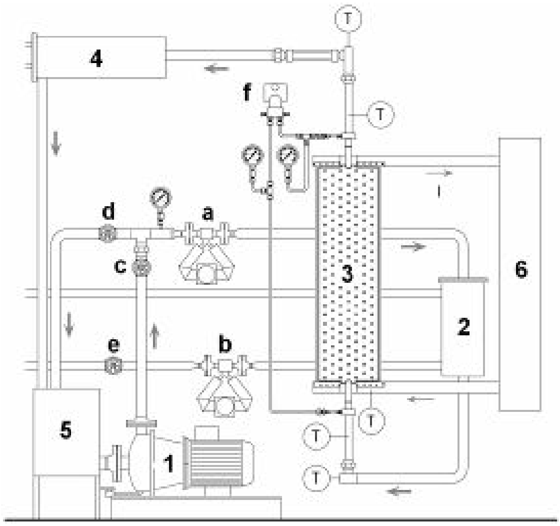

Figure 2: Single phase experiments on an electrically heated thermoplate as electrical resistance for the electrical heating.

The flow rate of the test fluid was measured by a Coriolis flow meter (a), the pressure drop by using a pressure drop transducers (Rosemount 3051CD4), the temperatures by calibrated thermocouples and/or Pt-resistance thermometers. The positions of the measuring sensors may be taken from Figure 2.

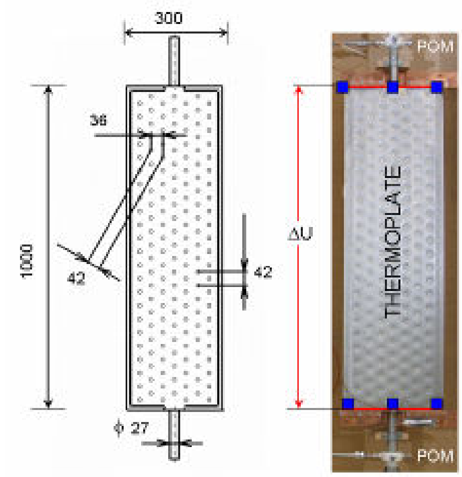

Figure 3: Geometry of the thermoplate and the measurement positions of the electrical quantities.

The single-phase pressure drop measurements are relatively simple; on the contrary, those of the heat transfer are incomparably more complex. Local heat transfer measurements are hardly possible because of the thermoplate structure that does not permit a reliable determination of the wall temperature by common methods. The wall temperature changes not only along the fluid flow direction but also in the transversal direction. For this reason, the test plate is switched as a resistance heater into an electric circuit and the measured electrical resistance of the plate is used to determinate its wall temperature.

2.3 Experimental procedures and evaluation

In the experiments, first the electrical plate resistance R was determined under isothermal conditions and correlated as a function of the wall temperature T_W,

$$R = R(T_W)\,. \qquad (1)$$

In these experiments, the fluid flow rate was adjusted as high as possible (approximately 4000 kg/h) and the heat input across the plate due to measurements of its resistance was kept as small as possible. Under such conditions, the change of the fluid temperature in the plate was neglected and the fluid temperature was set equal to the average wall temperature of the thermoplate.

With the aid of eq. (1) the heat transfer in the thermoplate can be quantified. The electrical energy P dissipated in the plate wall is obtained from

$$P = \Delta U \cdot I = R \cdot I^2 \qquad (2)$$

where R is the electrical resistance, ΔU the voltage drop over the plate and I the electrical current.

Taking into account possible heat losses $\dot{Q}_V$ to the surroundings, the energy $\dot{Q}$ taken up by the fluid, the net heat flow rate, becomes

$$\dot{Q} = P - \dot{Q}_V = \dot{M} c_p \left(T_{FOUT} - T_{FIN}\right) \qquad (3)$$

The average heat transfer coefficient α_{IN} of the fluid inside the thermoplate is defined by

$$\alpha_{IN} = \frac{\dot{Q}}{A\,\Delta T}, \qquad (4)$$

where A is the heat transfer surface area (both sides of the plate) and ΔT is the average driving temperature difference,

$$\Delta T = \left(T_{FOUT} - T_{FIN}\right) \Big/ \ln\frac{T_W - T_{FIN}}{T_W - T_{FOUT}}, \qquad (5)$$

the indices FOUT, FIN and W referring to the fluid outlet, inlet, and the wall, respectively, so that eqs. (3) to (5) give

$$\alpha_{IN} = \frac{\dot{M} c_p}{A} \ln\frac{T_W - T_{FIN}}{T_W - T_{FOUT}}\,. \qquad (6)$$

By this equation, the mass flow rate $\dot{M}$, the fluid inlet temperature T_{FIN}, and the wall temperature T_W can be varied independently from each other. In the experiments the flow rate and the inlet temperature of the fluid were selected as primary parameters.

To develop correlations for the heat transfer and pressure drop the dimensionless quantities are introduced:

$$Nu = \frac{\alpha_{IN} d_h}{\lambda_L}, \quad Pr = \frac{\nu}{\kappa}, \quad Re = \frac{u d_h}{\nu}\,. \qquad (7)$$

$$\zeta = \Delta p \Big/ \left(\frac{L}{d_h} \frac{\rho u^2}{2} \right) \quad (8)$$

ζ being the friction factor and Δp the frictional pressure drop.

2.4 Measurements technique and ranges of parameters

Distilled water and Marlotherm oil were used as test fluids. The electrical resistance of these fluids is high enough to allow for the application of the measurements method described above. The fluid inlet temperature was varied from 20°C to 85 °C.

The fluid flow measuring instrument (Micro Motion elite CMF 050) exhibits an accuracy of 0.15 % in the range of 1000 kg/h to 4000 kg/h. For the measurements of the fluid temperatures calibrated 4-wires-systems (Pt 100 resistance thermometer) were installed in the tubes connecting the thermoplate to the cylindrical POM-inserts, Figure 3. The uncertainty of the measurement was less than ± 0.15 K. Furthermore, the temperatures of the fluid in the storage tank, at the inlet and outlet of the heat exchangers as well as on the electrical connections (copper strips) were recorded.

For the determination of the heat generated in the plate by direct electrical heating and the plate temperature, the voltage drop ΔU over the plate and the amperage I were measured, Figure 2. The voltage drop was obtained at six positions as shown in Figure 3. The electrical current was determined from the measured voltage drop ΔU_S on a high precision resistance (60 µΩ, class 0.5) installed within the DC-Power supply. The voltage was measured with a micro volt/nano-ohm meter (Agilent 34420 A) at an accuracy of approx. ± 0.005 %. The quantities ΔU and I deliver the electric resistance R and by eq.(1) the wall temperature T_W. All temperature signals were processed by a precision thermometer (Prema 3040) of an accuracy of ± 0.04 °C.

For the measurements of the pressure drop in the thermoplate a differential pressure transducer (Rosemount 3051) with an accuracy of ± 0.075 % was used. The connections for the pressure transducer were positioned in the POM adapter approx. 200 mm upstream and downstream the plate inlet and outlet.

2.5 Illustration and discussion of the results

2.5.1 Electric resistance of the thermoplate

The first step in the experiments was to reliably determine the electrical plate resistance under isothermal conditions. For this purpose the thermoplate was flowed by the test fluid at a constant inlet temperature. To achieve a uniform temperature in the whole plate the electrical quantities required for the measurements were kept as small as possible, which was demanded by the measurement accuracy.

The amperage I required for a reliable determination of the electrical resistance R was smaller than 200A and the heat generation in the plate below 60W. The heating up of the test fluid (oil) at a flow rate of 4000 kg/h due to this energy input resulted in a temperature rise of approx. 0.04 K, which was within the uncertainty range of the temperature sensors used.

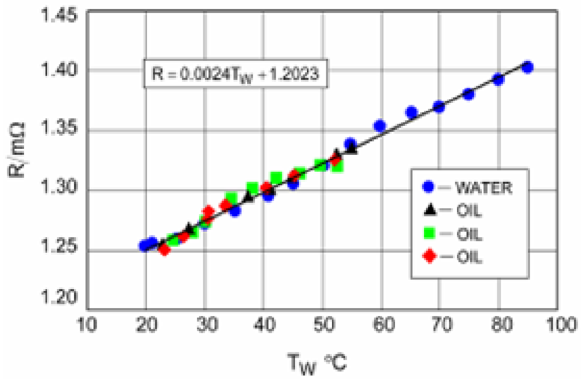

Figure 4: Electrical resistance R of the thermoplate as a function of the wall temperature T_W.

Figure 4 shows the measured electrical plate resistance R as a function of the wall temperature T_W. Three experimental runs with oil at different flow rates and one run with water were undertaken. The data can be approximated by the following straight line:

$$R = 1.2023 + 2.4 \cdot 10^{-3} \cdot T_W \;, T_W \text{ in °C} \;, R \text{ in m}\Omega. \quad (9)$$

The temperature range (20°C to 80°C) covered by this expression permits a sufficient variation of the fluid properties in the experiments on heat transfer.

2.5.2 Heat transfer

The heat transfer coefficients α_{IN} were determined according to eq. (6) at different mass flow rates and different temperatures of the test fluid. The corresponding experimental Nusselt numbers Nu are shown in Figure 5 as function of the Reynolds number Re at different average oil temperatures. In the experiments, the wall temperature was kept as constant as possible and the mass flow rate was varied. As expected, the Nusselt number increases with rising Reynolds number; thereby a higher oil temperature corresponds to a larger Nusselt number.
Considering the dependence of the Nusselt number on the Reynolds number, the empirical relationship

$$Nu \sim Re^{0.444} \quad (10)$$

works well. A similar dependency is also observed in the experiments with water, Figure 6.
A summary of all of the heat transfer results is illustrated in Figure 7. The Nusselt number is smaller at lower fluid temperature (larger Prandtl number), a behaviour that may appear questionable at first sight. As is commonly found in the literature, a larger Prandtl number corresponds to a larger Nusselt number and for simple fluid flow situations (e.g. boundary layer) the relationship

$$\mathrm{Nu} \sim \mathrm{Pr}^n, \; n > 0 \qquad (11)$$

is usually adopted.

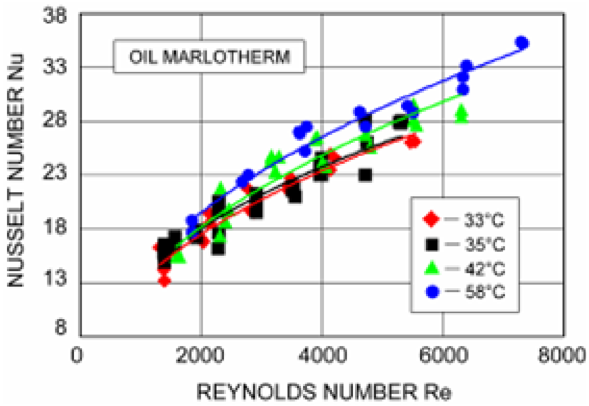

Figure 5: Nusselt number Nu as a function of the Reynolds number Re at different oil temperatures.

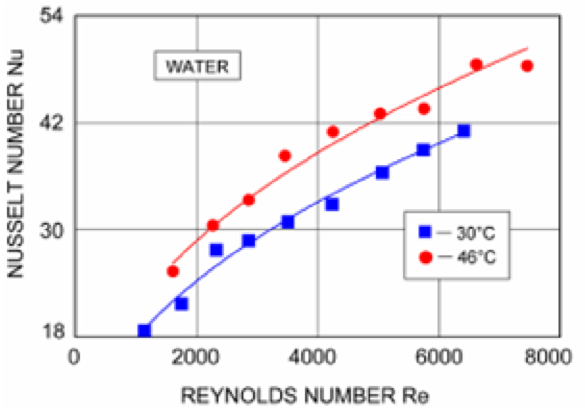

Figure 6: Nusselt number Nu as a function of the Reynolds number Re at different water temperatures.

The expression (11) is also recommended for plane or corrugated plate heat exchangers [6, 10], irrespective of the Prandtl number and the flow structures. However, due to the welding joints the boundary layer model cannot be adopted in the present case, the flow characteristic of the fluid in the thermoplate is highly 3-dimensional. In general, the Validity of the relationship (11) should be considered in dependence of the geometrical features of the heat transfer surfaces.

In order to examine the influence of the Prandtl number on the heat transfer in the thermoplates, experiments with different Prandtl numbers are inevitable. Our own measurements with the Malrotherm oil and distilled water show the Nusselt number to decrease with increasing Prandtl number. With these two fluids the following correlation is obtained:

$$\mathrm{Nu}_{IN} = 1.869 \cdot \mathrm{Re}^{0.444} \, \mathrm{Pr}^{-0.484} \qquad (12)$$

It predicts the experimental Nusselt numbers with a deviation smaller then $\pm 10\%$, Figure 8.

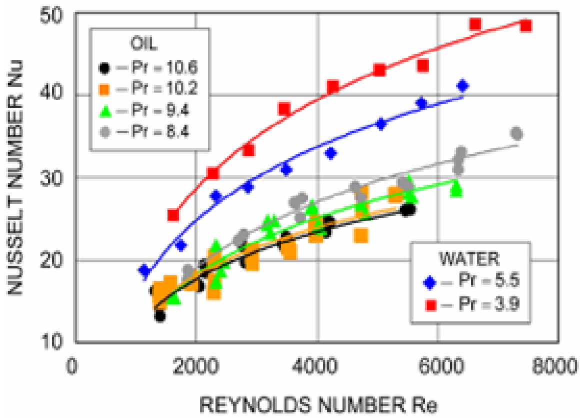

Figure 7: Nusselt number Nu as a function of the Reynolds number Re at difference Prandtl numbers.

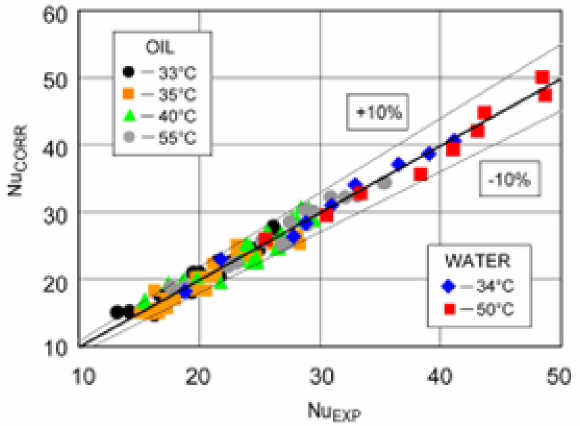

Figure 8: Comparison of the proposed heat transfer correlation with the experimental date.

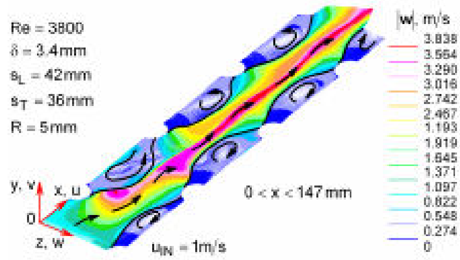

Figure 9: Flow field in the thermoplate obtained in numerical experiments [2].

According to eq. (12), the convection heat transfer depends strongly on the viscosity and heat conductivity of the fluid. This behaviour is most probably caused by the complex velocity field in the thermoplate. In the proximity of the welding joints the flow is laminar, whereas in the middle region between the sheet joints it might be turbulent, Figure 9. This may justify the numerical value of the exponent of the Reynolds number that lies between those of laminar and turbulent flows. As is obvious from this Figure, the flow field is strongly different from a boundary layer flow and the fluid domain is penetrated by well-established secondary flows with very sharp recirculation regions [2].

2.5.3 Pressure drop

The pressure drop in the thermoplate was determined at different flow rates of the test fluids. As Figure 10 shows, the friction pressure drop of water is slightly higher than of oil which can be attributed mainly to the larger density of water and different flow structures (Figure 9) associated with different fluid viscosities.

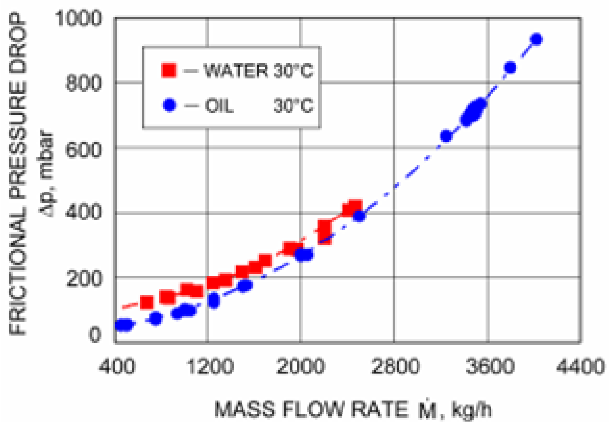

Figure 10: Pressure drop Δp as a function of flow rate $\dot{M}$.

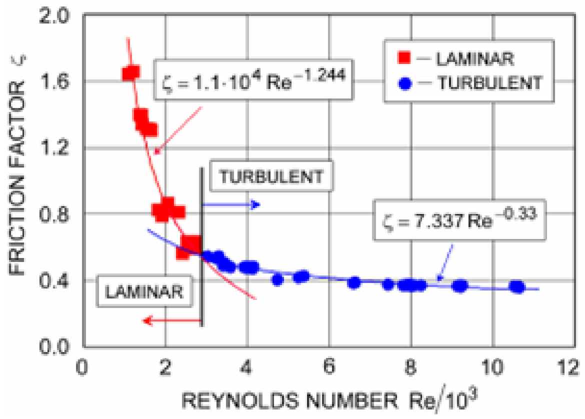

Figure 11: Friction factor ζ as a function of the Reynolds number Re.

The experimental friction factor data are shown in Figure 11 as function of the Reynolds number. The laminar flow region lies below the Reynolds numbers of 2400, and the friction factor ζ largely obeys the expression,

$$\zeta = 1.1 \cdot 10^4 \ \mathrm{Re}^{-1.244} \tag{13}$$

while in the turbulent region we have

$$\zeta = 7.34 \ \mathrm{Re}^{-0.33}. \tag{14}$$

Both correlations for the friction factor predict the experimental data with a deviation smaller than $\pm 10\%$.

2.6 Conclusions

The convective heat transfer and pressure drop in a thermoplate have been investigated with the Malrotherm oil and distilled water as test fluids in the Reynolds number range $1000 < \mathrm{Re} < 7500$. In the heat transfer experiments the electrical direct heating method was applied and the average plate wall temperature was obtained from voltage-current measurements. Basing on the experimental data correlations for the heat transfer and pressure drop were developed. The heat transfer correlation deviates from the ones in the literature for plane channels not only quantitatively but also qualitatively. The reason for this behaviour lies mainly in the complex flow filed in the thermoplate dictated by welding joints. As could be shown in an accompanied numerical study [2], strong secondary flows and re-circulation flow regions develop behind the welding spots.

The empirical expression for the friction factor follows qualitatively the behaviour known from the literature for other channel geometries. In the laminar flow region the decrease of the friction factor with increasing Reynolds number is stronger in comparison to the well-know Hagen-Poiseuille formula. Similar trend is also observed in the turbulent region in comparison with the Blasius expression.

The correlations proposed above are reliable within the parameter ranges covered by the experiments. However, they are incomplete with respect to a variation of the geometrical plate parameters.

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