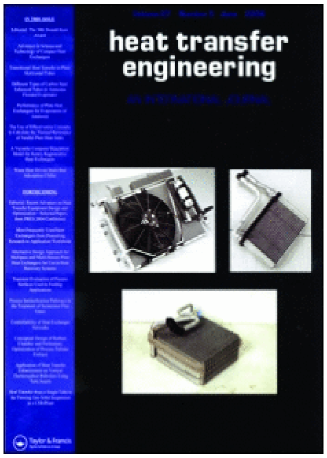

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Pressure Drop Analysis of Steam Condensation in a Plate Heat Exchanger

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In cooperation with Alfa Laval Thermal AB Company in Sweden, a steam condensation test rig of a plate heat exchanger (PHE) was set up. The heat transfer and pressure drop characteristics of two kinds of typical operation condition in a PHE were obtained: complete condensation and partial condensation. This article introduces the test rig, various sensors, and the data acquisition system used in the measurements. The process of steam condensation in a PHE is analyzed in detail. The two-phase frictional pressure drop along this wavy channel was obtained from the measured steam condensation pressure drop. The Lockhart-Martinelli model was extended to predict the pressure drop of steam condensation in a PHE and was verified by the experimental results. Based on the data processing of 109 experimental points, a correlation of frictional pressure drop of steam condensation in a PHE is suggested. This correlation is recommended for calculation of the steam condensation pressure drop in a PHE.

Plate heat exchangers have been widely used for nearly a century because of their high effectiveness, compactness, flexibility, and cost competitiveness. Originally, their main use was in single-phase applications. From 1970, PHEs were found to be suitable for two-phase applications [1], particularly in district heating systems via steam condensation. In China, many PHEs have already been used for steam condensation, for example, the Beicheng Thermal Factory in Beijing.

In steam applications of PHEs, the characteristic of pressure drop is very important. The reasons are as follows. (1) The condensation heat transfer and the pressure drop in PHEs are coupled to each other, and therefore the characteristic of pressure drop should be studied in order to predict the thermal features. (2) PHEs are used as only a part of the steam system in practical applications. Control regulation is related to the steam mass flow rate, the steam pressure, the internal condensate level, and the coordination with other facilities. This control depends mainly on the steam-side pressure drop. Hence, the investigation of steam condensation pressure drop in PHEs becomes very important.

Unfortunately, there are only a few publications related to the steam condensation in PHEs so far [1, 2].

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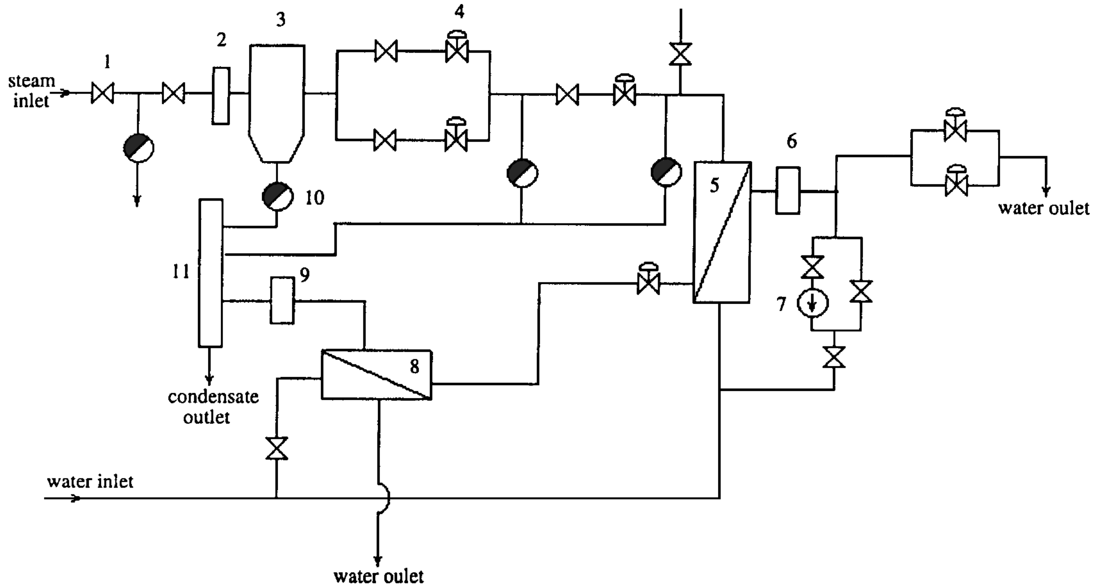

Figure 1 Schematic drawing of the test rig: 1, manual stop valve; 2, filter; 3, separator; 4, pressure control valve; 5, test PHE; 6, cooling-water flow meter; 7, pump; 8, secondary cooler; 9, condensate flow meter; 10, steam trap; 11, condensate collector.

Reference [3] studied only the water–air two-phase pressure drop features in PHEs. This is somewhat unrelated to the real steam condensation. Reference [4] dealt with the steam condensation pressure drop in PHEs, but did not suggest a correlation for calculation of the steam condensation pressure drop. Only these references related to the prediction of steam condensation pressure drop in PHEs can be gathered for the time being.

In cooperation with Alfa Laval Thermal AB Company in Sweden, a steam condensation plate heat exchanger test rig was set up to study the process of steam condensation in PHEs. First, the heat balance was checked based on the heat flow rate on the steam side and the cooling-water side, respectively. The deviation is within 5%. Then the isothermal steam and liquid pressure drop tests were carried out. The results are compared with the single-phase test results carried out by Alfa Laval Thermal AB previously, and they agreed satisfactorily. Finally, some two-phase steam condensation tests in a PHE were completed and a total of 109 effective experimental points were obtained. Based on a comprehensive analysis of the experimental results, a correlation for the frictional pressure drop during steam condensation in PHEs was developed. The Lockhart-Martinelli model, originally obtained from tests of isothermal two-phase, two-component flow in horizontal tubes, was extended to be applicable for steam condensation in a PHE. This correlation is recommended to be used in engineering design and performance calculations of PHEs.

TEST RIG FOR STEAM CONDENSATION IN A PHE

The test rig for steam condensation in a PHE is shown in Figure 1.

Pt-100 resistance thermometers are used to measure the steam inlet, condensate outlet, and cooling-water inlet and outlet temperatures. The temperature resolution is 0.1 °C. To improve the measurement accuracy, two Pt-100 meters are installed at every inlet and outlet point. If the deviation between the two readings is large, this measurement will be considered inaccurate and disregarded. If they are very close (the difference should be less than 0.3 °C), their average value will be taken as the true value. In fact, the difference is always within 0.2 °C.

The pressure sensor is manufactured by Alfa Laval Automation AB. Absolute pressure and differential pressure sensors are used because of the importance of steam pressure measurement. In the actual measurement process, there is always a difference between these two readings. The reason for this is that there is some condensate in the connecting pipe. The static pressure drop from the condensate level cannot be estimated precisely, which leads to a certain difference. In the tests, two absolute pressure sensors are installed to measure the inlet and outlet pressures. The pressure sensor and the measurement point are at the same level, which avoids the uncertain static pressure drop. For the water side, the differential pressure sensor is used to measure the pressure drop in order to check the results with those already available at Alfa Laval Thermal AB.

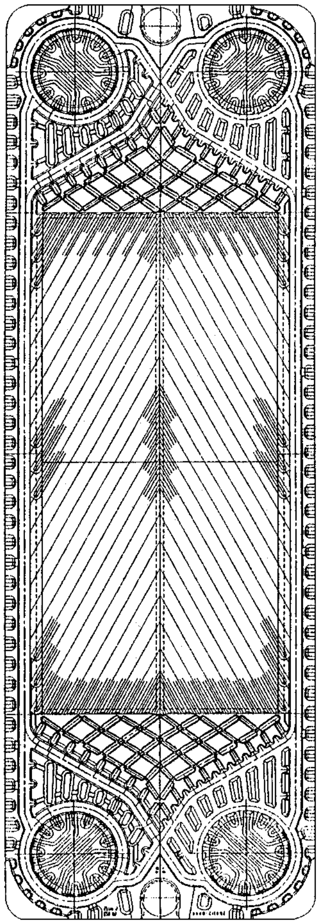

Figure 2 The investigated plate.

An electromagnetic-type meter manufactured by Danfoss is used to measure the cooling-water flow. The resolution is 0.1%. The maximum mass flow rate of condensate is only about 0.6 kg/s. The measurement error will be very large if the same type of meter is used. Therefore, a container and a stopwatch are used instead. When the operating condition is stable, the mass flow rate is measured twice successively and the average value is taken. If the difference between these two values is large, it is remeasured after the operating condition becomes stable again.

All experimental data, except the mass flow rate of condensate, are measured and collected by a data acquisition system, an AAC-2 provided by Swedish INTAB, which is connected to a computer. The output of the Pt-100 is a voltage signal, and the output of the pressure sensors as well as the water flow meter is a 4–20 mA electrical current signal. These sensors and the data acquisition system have high resolution and good calibration, giving reliable test results.

The plate heat exchanger (see Figure 2) investigated is manufactured by Alfa Laval Thermal AB. It is a gasket-and-frame type of PHE. There are 21 channels altogether, which include 10 steam channels and 11 cooling-water channels. The flow arranged for the tests is counterflow. The height and width are 750 mm and 250 mm, respectively. The thickness of each plate is 0.5 mm. The plate material is AISI 316 stainless steel. The plate angle of corrugation is 30°.

DATA PROCESSING OF TEST RESULTS

Steam condensation pressure drop in PHEs depends mainly on the following parameters: mass velocity in the channel, steam quality, pressure, and relative flow direction of steam to coolant. Figure 3 [Re_{gin} in the figure is the steam inlet Reynolds number, $Re_{gin} = (G_s \cdot D_e)/\mu_s$] shows the pressure drop as a function of steam flow rate and steam pressure for complete condensation. From this figure it is obvious that for a given steam mass flow rate the pressure drop decreases with the increase of steam pressure.

For the normal pressure drop permitted in PHEs, the steam velocity at the inlet is less than 80 m/s. Along the corrugated plate, the flow direction of a two-phase

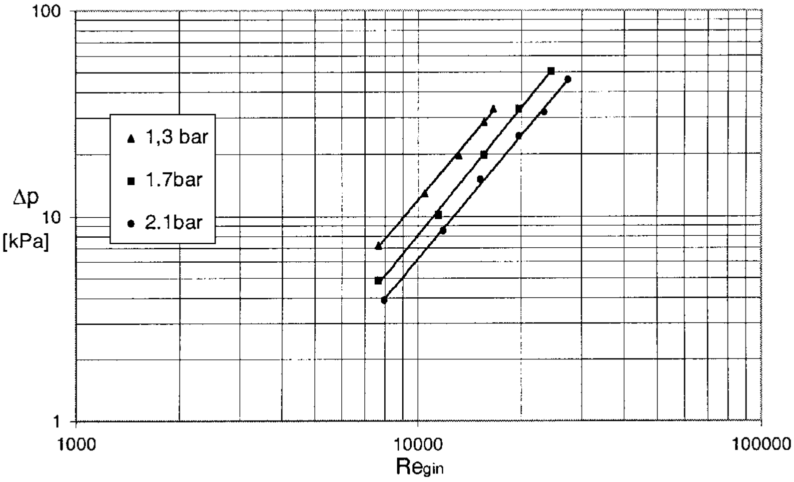

Figure 3 Pressure drop as a function of steam flow rate and steam pressure.

mixture always changes in the wavy channel. Near the steam inlet, the liquid film is very thin and the steam rotates continuously, with high velocity, in the wavy channel. The flow is probably annular flow or annular mist flow. The steam velocity becomes lower and lower with the decrease of steam quality. The two-phase flow changes to totally separated flow, and the condensate film changes to turbulent instead of laminar. The process of steam condensation in PHEs is a very complicated two-phase and three-dimensional flow. The prediction of pressure drop becomes very complicated and difficult.

The Lockhart-Martinelli model has been applied to estimate the steam condensation pressure drop in PHEs [2]. In fact, the Lockhart-Martinelli model was obtained from a series of tests of isothermal two-phase, two-component flow in horizontal tubes [5]. Its application in PHEs has not been verified by experiments yet. In addition, some constants are needed, as the Lockhart-Martinelli model is applied to estimate the pressure drop during steam condensation in PHEs. This shortcoming leads to some difference between the calculated and measured values. Based on the present investigation of heat transfer and pressure drop during steam condensation in PHEs, a correlation was obtained for the estimation of pressure drop. The deviation between the calculated and experimental values is within ± 12%.

Total Pressure Drop and Frictional Pressure Drop

The total steam condensation pressure drop ΔP_{TP} in a PHE was obtained directly in the tests. This pressure drop is written as follows:

$$\Delta P_{\mathrm{TP}} = \Delta P_f + \Delta P_g + \Delta P_a + \sum \Delta P_{\mathrm{Ni}} \tag{1}$$

where ΔP_f is the two-phase frictional pressure drop, ΔP_g is the gravity pressure drop, ΔP_a is the acceleration pressure drop, and $\sum \Delta P_{\mathrm{Ni}}$ is a sum of additional pressure drops.

The gravity pressure drop in a vertical channel of steam condensation is calculated from [6]

$$\Delta P_g = \int_0^L [\rho_v \alpha + \rho_l \cdot (1-\alpha)] \cdot g \cdot dz \tag{2}$$

where the void fraction α in the vertical channel can be estimated as [7]

$$\frac{1}{\alpha} = 1 + \frac{1-x}{x}\left(\frac{\rho_v}{\rho_l}\right)^{2/3} \tag{3}$$

The acceleration pressure drop of condensation is calculated from [6]

$$\Delta P_a = G^2 \left\{ \left[\frac{(1-x_2)^2}{\rho_l(1-\alpha_2)} + \frac{x_2^2}{\rho_v \alpha_2} \right] - \left[\frac{(1-x_1)^2}{\rho_l(1-\alpha_1)} + \frac{x_1^2}{\rho_v \alpha_1} \right] \right\} \tag{4}$$

where G is the steam mass velocity, α_1 and α_2 are the inlet and outlet void fractions respectively, and x_1 and x_2 are the inlet and outlet steam qualities respectively. The value of the acceleration pressure drop increases with the increase of steam mass velocity G and the difference between the inlet and outlet steam quality. In addition, this value is negative for steam condensation.

The additional pressure drops in PHEs mainly include: (1) the inlet and outlet connection pressure drop $\Delta P_{\mathrm{connin}}$, $\Delta P_{\mathrm{connout}}$ and (2), the inlet and outlet port pressure drop $\Delta P_{\mathrm{portin}}$, $\Delta P_{\mathrm{portout}}$. In this article, empirical formulas are used to estimate these additional pressure drops. The dynamic pressure VP is [8]

$$\mathrm{VP} = \frac{\rho v^2}{2} \tag{5}$$

where ρ and v are the density and velocity of the fluid, respectively. For two-phase flow at the outlet connection and port during partial condensation, the homogenous flow model is used to calculate the two-phase density. $\Delta P_{\mathrm{connin}}$ and $\Delta P_{\mathrm{connout}}$ are calculated from [8]

$$\begin{aligned} \Delta P(\mathrm{in}) &= X_i(c)(\mathrm{in}) \cdot \mathrm{VP}(\mathrm{in}) \\ \Delta P(\mathrm{out}) &= X_i(c)(\mathrm{out}) \cdot \mathrm{VP}(\mathrm{out}) \end{aligned} \tag{6}$$

$X_i(c)$ is a coefficient used to calculate the pressure drop for the connection and port. Its value depends on the connection and port diameters. $\Delta P_{\mathrm{portin}}$ and $\Delta P_{\mathrm{portout}}$ are calculated from [9]

$$\begin{aligned} \Delta P(\mathrm{in}) &= -\mathrm{VP}(\mathrm{in}) \cdot [0.8 - f \cdot 4L/(3 \cdot \phi)] \cdot R \\ \Delta P(\mathrm{out}) &= \mathrm{VP}(\mathrm{out}) \cdot [2.7 + f \cdot 4L/(3 \cdot \phi)] \cdot R \end{aligned} \tag{7}$$

where ϕ is the port diameter, f is the friction factor, L is the port length, and R is a correction factor.

From the experimental data and Eqs. (1)–(7), the two-phase frictional pressure drop ΔP_f is calculated from

$$\Delta P_f = \Delta P_{\mathrm{TP}} - \Delta P_g - \Delta P_a - \sum \Delta P_{\mathrm{Ni}} \tag{8}$$

According to the calculated results of all experimental points, gravity pressure drop, acceleration pressure drop, port pressure drop, and connection pressure drop are only 0.2%, 4%, 2%, and 0.1% of the total pressure drop, respectively. This indicates that the final results will not change significantly even if the estimations of void fraction and empirical formulas have some errors.

Experimental Program

To evaluate the experimental two-phase pressure drop data using the Lockhart-Martinelli model, the experiments for the single-phase pressure drop of liquid as well as steam should be carried out first. The liquid and steam mass flow rates and their corresponding inlet and outlet pressures are measured. Then the single-phase frictional pressure drop factor is obtained through the following formula:

$$f = \Delta P \frac{D_e}{L} \frac{2\rho}{G^2} \tag{9}$$

The single-phase heat transfer and frictional characteristics of different PHE models have already been determined by Alfa Laval Thermal AB [9]. The correlation for the friction factor f at turbulent flow for the considered PHE is

$$f = 0.56 * \mathrm{Re}^{-0.12} \tag{10}$$

where $\mathrm{Re} = (G \cdot D_e)/\mu$ is the fluid Reynolds number. The results of the single-phase pressure drop tests are compared with Eq. (10). The deviation is within ± 5%, which shows that the experimental results are reliable.

Next, the heat balance on both sides is checked using complete condensation runs with minor subcooling. The results of 33 points used show that heat balance error is within the permitted limit. In most cases, the heat balance error is within 3%. There are only two cases with errors exceeding 5%, which can be considered invalid. Therefore, the heat balance of the tests is within ± 5% and the measurements have sufficient accuracy.

Finally, the steam condensation heat transfer and pressure drop are analyzed for the chosen PHE. A total of 109 experimental points were obtained, including 27 of complete condensation and 82 of partial condensation. The steam inlet pressure varied from 1.3 bar to 3.3 bar absolute. The outlet steam quality varied from 0.3 to 0. For the partial condensation, the outlet steam quality is obtained based on the cooling-water heat load.

In order to guarantee that the full length of the plate is used for steam condensation, the condensate should leave the heat exchanger at saturation condition. This is always the case in partial condensation. However, in complete condensation, small subcooling at the outlet cannot be avoided in a real experimental process. This amount of subcooling should be as small as possible. In this test, the maximum subcooling is 6℃. Therefore, the maximum contribution of the subcooling to the total heat load is only 1.1%, which can be ignored within sufficient confidence.

Evaluation of Measurements

The Lockhart-Martinelli model [5] is used to correlate the experimental data. The Lockhart-Martinelli parameter X and the two-phase friction multiplier ϕ_l for liquid are expressed as

$$X^2 = \frac{\Delta P_l}{\Delta P_v} \qquad \phi_l^2 = \frac{\Delta P_f}{\Delta P_l} \tag{11}$$

ΔP_l is the liquid-phase frictional pressure drop calculated for the average steam quality, and ΔP_v is the calculated vapor-phase pressure drop.

$$\Delta P_l = f_l \frac{L}{D_e} \frac{G_s^2 (1-x)^2}{2\rho_l} \tag{12a}$$

$$\Delta P_v = f_v \frac{L}{D_e} \frac{G_s^2 x^2}{2\rho_v} \tag{12b}$$

where G_s is the two-phase total mass velocity, x is the average integral steam quality, f_l and f_v are friction factors for the liquid and vapor phases, respectively, and L and D_e are the channel length and the equivalent diameter, respectively. ρ_v is the steam density, and is determined according to the average value of the saturation pressure at the inlet and outlet. The equivalent diameter D_e is defined as $D_e = 2h$, where h is the average channel gap.

Cooling water is heated from the steam side and its temperature increases along the channel, while the average temperature difference and the heat load decrease gradually. Under complete condensation the inlet and outlet steam qualities are 1.0 and 0, respectively, and in the case of counterflow, the steam quality decreases slower at the beginning of the channel than at the end, which means that the average steam quality in the channel is greater than 0.5. On the other hand, it is less than 0.5 in co-current flow. The plate is divided into 100 segments to calculate the local steam qualities (by an iterative process), and the average steam quality is calculated. The result is very similar to the recommended value in [4]. Therefore, the average steam quality in complete condensation is taken as 0.54 during the data processing of the condensation pressure drop.

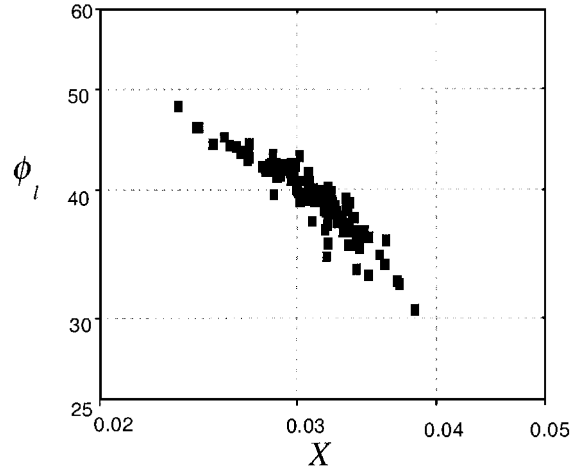

Figure 4 ϕ_l–X for steam condensation in a PHE.

When partial condensation is evaluated, the individual steam and liquid mass flow rate are calculated from

$$G_{ss} = 0.54 * [G_s * (1 - x_2)] + G_s * x_2 \tag{13a}$$

$$G_{sl} = G_s - G_{ss} \tag{13b}$$

where x_2 is the outlet steam quality in the PHE. This means that under partial condensation, G_{ss} and G_{sl} are used to replace $G_s x$ in Eq. (12*b*) and $G_s(1 - x)$ in Eq. (12*a*), respectively.

From the experimental data, the single-phase frictional pressure drops ΔP_l and ΔP_v, and the two-phase frictional pressure drop ΔP_f are calculated. Then the Lockhart-Martinelli parameter X and the two-phase friction multiplier ϕ_l are determined. Finally, Figure 4, showing ϕ_l–X can be obtained. This diagram shows a very good correlation between ϕ_l and X.

Correlation of Frictional Pressure Drop

Chisholm recommended the following correlation for the two-phase frictional pressure drop [5, 6]:

$$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{14}$$

The value of constant C should be chosen according to:

Liquid–vapor flow condition	C
Turbulent–turbulent, T-T	20
Viscous–turbulent, V-T	12
Turbulent–viscous, T-V	10
Viscous–viscous, V-V	5

From each experimental measurement point a corresponding value of C can be obtained. The averaged value of all of these data is calculated, and the result is $C = 16$ and hence lies between the T-T and the V-T flow models. This value is used to calculate the two-phase pressure drop of all the experimental data points. The differences between calculated and experimental values are shown in Figure 5. The error is within ±12%, which can be considered satisfactory in the prediction of two-phase frictional pressure drop.

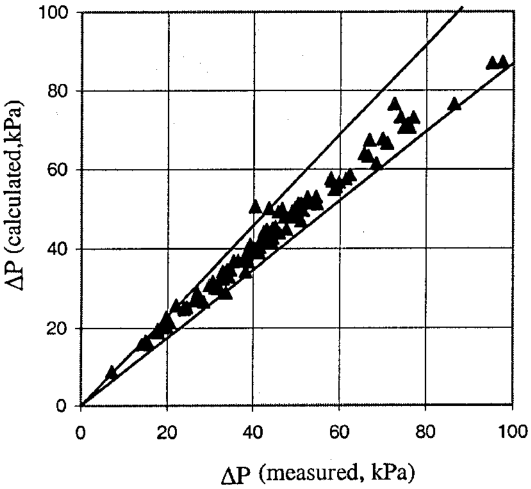

Figure 5 Comparison of calculated and experimental pressure drops.

As stated before, during the process of steam condensation in a PHE, the steam flow is always turbulent; condensate film flow is laminar at first and then changes to turbulent flow very quickly in the wavy channel. At the outlet, the Reynolds number of the condensate flow [$\mathrm{Re}_l = G_s \cdot (1 - x_2) \cdot D_e / \mu_l$] is up to 800–1,800 and the flow is totally turbulent. Therefore, $C = 16$ is a very reasonable value. The following correlation is thus recommended to calculate the two-phase multiplier for steam condensation in PHEs:

$$\phi_l^2 = 1 + \frac{16}{X} + \frac{1}{X^2} \tag{15}$$

CONCLUSION

1. Steam condensation in PHEs is a complicated two-phase flow process. The frictional pressure drop along the plate channel can be derived from the measured total pressure drop. The gravity pressure drop and the acceleration pressure drop can be calculated using heterogeneous models. Other additional pressure drops can be predicted through empirical formulas.
2. The Lockhart-Martinelli model is extended to calculate the steam condensation pressure drop in a PHE. This is justified by the test results. Based on the regression of experimental data, Eq. (15) is

recommended to calculate the two-phase multiplier for steam condensation in a PHE. The constant $C = 16$ shows that steam flow is always turbulent and the condensate film flow changes from laminar to turbulent. The deviation between the calculated values for steam condensation pressure drop and the experimental results is within $\pm 12\%$.

NOMENCLATURE

C	constant in Chisholm correlation
D_e	equivalent diameter, m
f	single-phase friction pressure drop factor
f_l, f_v	friction factors for liquid and vapor phase, respectively
G_s	two-phase total mass velocity, kg/m^2 s
L	length of channel, m
ΔP_a	acceleration pressure drop, Pa
ΔP_{connin}	inlet connection pressure drop, Pa
$\Delta P_{\text{connout}}$	outlet connection pressure drop, Pa
ΔP_f	two-phase frictional pressure drop, Pa
ΔP_g	gravity pressure drop, Pa
ΔP_l	frictional pressure drop of liquid phase, Pa
ΔP_{portin}	inlet port pressure drop, Pa
$\Delta P_{\text{portout}}$	outlet port pressure drop, Pa
ΔP_{TP}	total steam condensation pressure drop, Pa
ΔP_v	frictional pressure drop of vapor phase, Pa
$\sum \Delta P_{\text{Ni}}$	sum of additional pressure drops, Pa
Re	Reynolds number
Re_{gin}	inlet steam Reynolds number
Re_l	outlet liquid Reynolds number
v	velocity of the fluid, m/s
VP	local dynamic pressure, Pa
x	average integral steam quality
x_1, x_2	inlet and outlet steam quality
$X_i(c)$	pressure drop coefficient for the connection and port
X	Lockhart-Martinelli parameter
α_1	inlet void fraction
α_2	outlet void fraction
μ	fluid viscosity, kg/ms
ρ	fluid density, kg/m^3
ϕ_l	two-phase friction multiplier for liquid

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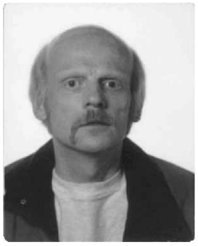

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