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A method to combine local heat transfer and flow visualization of flow boiling in frame-and-plate heat exchanger

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ABSTRACT

This paper presents the development of a novel method to combine measurement of local heat transfer coefficient and flow visualization in frame-and-plate heat exchanger (FPHE). A special plate was instrumented using two original chevron plates and clay as thermal infills. Thermocouples were deployed on the inner surfaces of the two plates. With the known thickness and thermal conductivity of the clay, the instrumented plate functions as a heat flux sensor. After calibration, this plate was installed to measure fluid heat transfer from one side, while leaving visual access from the other. Preliminary results of water-water single phase heat transfer and R245fa-water flow boiling were presented. Combined with visualization, the method provides a viable solution to relate local heat transfer with flow regime within a plate channel, while preserving the real geometry and operating condition.

1. INTRODUCTION

Frame-and-plate heat exchanger (FPHE) is commonly used for their ease of cleaning, simple adjustment of heat transfer area, compactness and excellent thermal-hydraulic performance [1]. Early applications of FPHE are mainly for liquid-liquid heat transfer in the lower pressure range (usually below 1.6 MPa), including dairy, pulp and paper industries for their hygiene requirements [1]. It has also been adopted by two-phase applications such as steam generator and petrochemical boiler. With the introduction of brazed plate heat exchanger (BPHE), such plates could withstand higher pressure and later on found its increasing application as condenser and evaporator in air-conditioning and refrigeration systems.

Plate heat exchanger (PHE, including FPHE and BPHE) essentially consists of multiple thin metal plates that are stamped with a wavy chevron or herringbone pattern. Fluid channels are formed by pressing the plates with opposite chevron direction together. The high performance of PHE owe not only to the enlarged surface area, but also to the geometry of the plates such as the corrugation depth, wavelength and angle. The contact points between crests and troughs of two adjacent plates subdivide the fluid path into an array of interconnected unitary cells, which turbulate the flow and enhance heat transfer.

To the best of our knowledge there is no general correlation that accounts for the effects of all geometrical parameters, working fluids and operating conditions. Various corrugation designs have been developed mainly on an empirical basis. Recent development of generalized correlations, taking theoretical [1-3], semi-empirical [4] or empirical [5-7] approaches to predict single-phase heat transfer, has shown moderate success in accounting geometries such as surface enlargement factor, chevron angle and aspect ratio (wave depth/ wavelength) for low Pr number fluids. However, the understanding of two-phase flow in FPHE is still far from satisfactory. Numerous studies in this topic were found as summarized in review articles [8-10]. Each study has developed different correlations based on their own experiments with various fluids and plate geometries. However, comparison of these correlations at the same condition yields drastic difference, as pointed out by [10]. There is still no general correlation that accounts for each separate effects on two-phase flow regime, which, from the experience of that in round tube, heat transfer has been successfully
correlated with. Only a few works in literature focused on developing flow maps and studying flow regimes and their local heat transfer characteristics in PHE, as will be reviewed in the next section.

2. LITERATURE REVIEW

Convective heat transfer has shown to be greatly dependent on flow pattern, which is revealed through various visualization techniques. For single-phase flow, visualization provides information on the magnitude and direction of the local velocity field, the dynamic behavior of the flow, the transition to turbulence and flow recirculation zones. For two-phase flow, the respective distribution of the liquid and vapor also takes on some commonly observed structures, which the prediction of in-tube heat transfer in recent development has shown great success by phenomenologically relating to. Yet the flow visualization was rendered more difficult in PHE due to its complex geometry. A limited number visualization studies have been found in literature. Their techniques and some key findings are reviewed as below. For the scope of this paper, discussion is only limited to two-phase evaporation, which is also interchangeably referred to as flow boiling.

In 1987 Xu and Carey [11] combined measurement of local heat transfer and flow visualization of methanol and n-butanol in a cross-ribbed channel that resembles the geometry of PHE. Heat was provided through conduction on one side, and measured by thermocouple placed at distances with known thermal resistance. The adiabatic side provides visual access, which showed a churn or annular flow regime for vertical upward flow and wavy or annular for horizontal flow. Virtually no nucleate boiling was observed. The authors also described a method of correlating heat transfer for annular film flow.

Lin’s group [12-13] combined heat transfer and visualization with a different scheme. R134a at various mass flux and vapor quality was split into two streams, one for heat transfer measurement (3-channel) and the other for visualization (2-channel). In the visualization part the end plate was substituted with transparent acrylic plate machined and polished with the same geometry. In saturated boiling [12], liquid film was relatively thin and flow was dominated by evaporation at liquid-vapor interface. Intense nucleate boiling was observed near the inlet port. In subcooled boiling [13], bubbles were somewhat suppressed by raising the mass flux and inlet subcooling and heat flux show large effects on the bubble population, coalesce and generation frequency. Although the inlet condition was the same, the visualization only represented the flow with one side heat transfer. The difference was not investigated.

Jassim et al. [14] presented their investigation of adiabatic pressure drop and flow visualization in three different PHE geometries. The experiments were carried out with R134a in vertical upward flow arrangement. Chevron geometry was machined on a clear PVC plate and bolted together to form a PHE-similar flow passage. Light was provided by the reflection of stroboscope from a white background for uniformity. As a result, four flow regimes (bubbly, rough annular, smooth annular and mist) were clearly observed and mapped out on a mass flux versus quality basis for each geometry. Pressure drop was found to have a strong linear relationship with the kinetic energy per unit volume.

The test section presented by Grabenstein and Kabelac [15] provided a way to preserve the original geometry during visualization. The transparent part was casted by existing industrial stainless steel plate using polyurethane. Local heat transfer was measured by temperature oscillation IR thermography technique. The visualization was carried out in one channel, with R365mfc (normal boiling temperature 40.1 °C) as working fluid. Film, bubbly and slug flow were observed and heat transfer and pressure drop mechanism were developed in each flow regime.

Solotych et al. [16] applied IR thermography in measuring local heat transfer coefficient of both single-phase and flow boiling of HFE7100. The test section was composed of an IR transparent calcium fluoride (CaF$_2$) plate and a polycarbonate plate. Both were machined in a representative shape of a PHE. Electrical heating was provided through a thin layer of carbon dispersed polyimide to simulate a constant heat flux. The wall temperature was measured by IR camera and fluid temperature was obtained by linear interpolation of five thermocouple readings along the plate. Visualization was carried out adiabatically to link flow pattern with the measured local heat transfer. The comparison indicated the need for new prediction methods for local thermal-hydraulic performance over a wide range of operating conditions.

Other than low pressure refrigerant, gas liquid mixture was also found to be a common working fluid. Bai and Newell [17] investigated the pressure drop and flow visualization of air/alkylbenzene oil mixture in chevron plate heat exchanger. Results showed that liquid tended to follow the groove under certain conditions. Tribbe and Muller-Steinhagen [18] used air/water mixture in their visualization and pressure drop study in a 1-channel setup. Three chevron angles were investigated. Regular bubbly, irregular bubbly, churn, film and partial film flow patterns were
identified and flow map was constructed. Pressure drop was correlated with two-phase multiplier and Lockhart-Martinelli parameters. Vlasogiannis et al. [19] also used air/water mixture for heat transfer and visualization, which was made possible by replacing the end plate with a lithographically embossed plexiglass plate. Flow map was constructed with 5 flow patterns, each measured with heat transfer enhancement over single-phase flow at various superficial liquid velocities. Nilpueng and Wongwisess [20] reported visualization with air/water mixture in asymmetric chevron plates (55° and 10°), identifying bubbly, bubble recirculation and annular-liquid bridge pattern in upward flow and slug, annular-liquid bridge and annular-liquid bridge/ air-alone pattern in downward flow. Two-phase multiplier of all regimes was correlated as a function of Lockhart-Martinelli number in this study.

The aforementioned two-phase flow visualization were all carried out by using high speed camera to capture the liquid vapor interface. Nevertheless, visualization using neutron radiology was reported by Asano et al. [21] as a viable alternative. It was used to investigate the flow regime and void fraction of air/water mixture in both a 1-channel and 18-channel PHE. The liquid content was shown to be proportional to the image brightness. The result suggested that liquid distribution depended on the inlet liquid flow rate, while effect of the gas flow was little. The method was later used to combine heat transfer with flow visualization, as reported by Baba et al. [22] in 2009. The test section had 3 channel, with HCFC-142b (CH₃CFCl₂) evaporating in the center and fluoro-carbon FC3283 without hydrogen in side channels as heating medium for its low neutron attenuation. Their results showed downward boiling produced higher heat transfer coefficient, contradicting the conventional acceptance of superiority of the vertical upward flow for boiling.

From literature review, it is concluded that there hasn’t been a good way to simultaneously couple local heat transfer with flow visualization. Yet it is desirable to visualize how flow boiling changes its flow regime in real geometry and operating condition. This paper is to present the development of such experimental method in frame-and-plate heat exchanger (FPHE).

3. EXPERIMENTAL METHODOLOGY

3.1 Design and manufacturing of the test section

![Figure 1](image-url)

*Figure 1:* (a) Schematics of heat exchanger plate; (b) Side view of test section

The experimental methodology is presented in Figure 1(b). The test section consisted of 3 channel, with refrigerant flowing upward and hot water flowing downward. Heat transfer is measured to the left side of refrigerant channel by a specially instrumented plate which functions as a heat flux sensor. It was made of two original stainless steel (s.s.) plates, with thermocouples deployed in the inner surface for temperature measurement and infill material for conducting heat. The geometry of the plate and coordinate system are shown in Figure 1 and the values are summarized in Table 1. 32 type T thermocouples were soldered on to each plate respectively across the chevron area with the same (x,y) coordinates indicated by the black dots in Figure 6. In these 32 thermocouples, 26 were placed at the same z
coordinate at the half height of the sinusoidal corrugation, which were used to interpolate the x-y 2D temperature distribution of the plate. The remaining 6 were placed at 3 crests and 3 troughs in the upper region, reserved for measurement of a potential 3D temperature distribution in one unitary cell. For simplicity, these 6 thermocouples are not shown since the flow regime in the unitary cell is not discussed in this paper.

To the right of refrigerant channel in Figure 1 (b), all plates are made of transparent material to provide visual access. The first transparent plate was made from a flat polycarbonate sheet (Lexan 9034) with a thickness of 0.76 mm (0.03 inch) and thermal conductivity of 0.195 W/m-K. The sheet was formed into the exact replica of the original plate by heated pressure forming. The second plate was a replica by epoxy resin, also cast using original plate as mold for the inner surface. The outer surface was machined flat and polished for better clarity. The original plate, formed polycarbonate plate and molded resin plate are displayed in Figure 2.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>FPHE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chevron angle, φ</td>
<td>60°</td>
</tr>
<tr>
<td>Corrugation depth, b</td>
<td>2.20 mm</td>
</tr>
<tr>
<td>Corrugation pitch, Pc</td>
<td>10.0 mm</td>
</tr>
<tr>
<td>Plate thickness, t</td>
<td>0.60 mm</td>
</tr>
<tr>
<td>Port length, Lp</td>
<td>495 mm</td>
</tr>
<tr>
<td>Total length, Lv</td>
<td>578 mm</td>
</tr>
<tr>
<td>Port width, Lh</td>
<td>140 mm</td>
</tr>
<tr>
<td>Total width, Lw</td>
<td>210 mm</td>
</tr>
<tr>
<td>Heat transfer area, Aplate</td>
<td>0.1017 m²</td>
</tr>
<tr>
<td>Port diameter, Dp</td>
<td>35.0 mm</td>
</tr>
</tbody>
</table>

Table 1: Plate Geometry of FPHE

The thickness of the infill material is determined by matching the conductive thermal resistance of the polycarbonate (p.c.) plate through Equation (1), such that the heat flux measured from the sensor plate side could also represent the visualization plate side. In theory, any material with conformal contact with the plate corrugation could be used. Yet a realistic thickness for the ease of fabrication and installation should be taken into consideration. In addition, infill with larger thickness also tends to increase the axial conduction [23]. As a result, a sulfur free (non-corrosive) sculpture clay was selected. The thermal conductivity (k) of the clay is 0.6 W/m-K [24] and the resulting clay thickness (t) is 4.76 mm.

Figure 3: Heat flux plate and test section assembly
The clay is also a non-drying type that melting and solidifying process will not alter its properties. To fill in the clay, one plate was pinned onto an aluminum base (10.6 mm thick) heated to a temperature around 120 °C. Four pins/holes pairs with a diameter of 2.5mm at four corners were used for positioning. Pin holes were made at identical locations of the other plate so when pressed together both plates were in exact parallel phase. 6 aluminum strips with a thickness of 4.76mm were placed in the groove around the chevron area to support the plate while confining the fluidic melted clay to its desired area and thickness. 4 openings, each 3 cm wide, were left at the corners for thermocouple wires. 4 circular channels, also with a length of 4.76mm, were made and placed in the round groove surrounding the ports to direct the flow. Each channel was machined with groove for use of rubber O-ring to prevent leaking. Melted clay was carefully spread on to the chevron surfaces of each plate, with the amount slightly more than that to fill the volume. The plates were then clamped together with their aluminum base and the excess were squeezed out. The schematics picture of assembling is displayed in Figure 3.

3.2 Calibration of the instrumented plate

Inevitably, the thermal resistance could not be perfectly uniform across the heat flux plate, due to the embedded thermocouple wires and thickness variation. Therefore the heat flux plate was calibrated before installed in the real system. The calibration was carried out by comparing the uniform electrical heating in the bottom and resulting measured heat flux. As shown in Figure 4, heater was placed inside an insulated box. Instead of directly placing the plate on to the heater, as initial trial suggested that the non-conformal contact with the corrugation surface would cause non-uniform heating, a 5 cm layer of sand was placed between the plate and the heater. The sand layer not only provided conformal contact, but also diluted the difference between the heater to the crest and to the trough of the surface.

To make sure the electrical heating was uniform, IR camera (FLIR 7300) is used to measure the surface temperature when the sand is covered with an original plate. The plate was painted grey for higher emissivity (0.95). The measured surface temperature is presented in Figure 4. The analytical tool provided by FLIR showed that the surface temperature spanned from 33.2 °C to 35.6 °C, with a surface average of 34.8 °C. The core temperature of the heater was 54.8 °C. Therefore, the uncertainty (U) of heating uniformity determined from Equation (2) was ±4% ~ 8%.

\[
U_{\text{low}} = \frac{\Delta T_{\text{avg}}}{\Delta T_{\text{avg}}} = 4\%, \quad U_{\text{up}} = \frac{\Delta T_{\text{max}} - \Delta T_{\text{avg}}}{\Delta T_{\text{avg}}} = 8\% \tag{2}
\]

\[
\Delta T_{\text{max,avg,min}} = T_{\text{core}} - T_{\text{avg,core,max}}
\]

Figure 4: Calibration box and IR image
Figure 5: Temperature transients during calibration

The electrical power was provided by a variable DC power supply and determined by measuring the voltage and the resistance of the heater. The resistance is temperature insensitive, as verified at each calibration run. Calibration was

\[
\frac{T_{\text{p,c}}}{k_{\text{p,c}}} = 2 \times \frac{T_{\text{s,s}}}{k_{\text{s,s}}} + \frac{T_{\text{clay}}}{k_{\text{clay}}} \tag{1}
\]
repeated at 4 different voltages. A typical transient temperature response during calibration is plotted in Figure 5, which indicates a desired 14-hour interval before steady state is established. During steady state, temperature was recorded for 10 minutes and the time-averaged value is used for heat flux calculation.

![Figure 5: Transient temperature response during calibration](image1.png)

Figure 6: Thermocouple reading and contour by interpolation, at the heat input of 54.74 W

The 52 temperatures and their coordinates, as shown by the block dots in Figure 6, were used as inputs of a thermal conduction model in ANSYS. The corrugation surface was simplified into its hexagon projection, since only the 2D distribution of z-direction heat flux was of interest. Mesh sensitivity analysis was carried out at a power input of 54.74 W (DC input 13.44 V). The result was plotted in Figure 7. Changing from element number from 27696 to 45185, the total heat from surface integral varied less than 0.1 W (0.14%). A node number of 27696 was used in this study.

Initial result in Figure 7 indicated an over-prediction of 30.7%, a result caused by the simplifying geometry and/or possible inaccurate thermal resistance estimation. It turned out that both effect could be lumped into a correction factor to the averaged thermal resistance. For simplicity, the thermal conductivity of the clay was adjusted to 0.4 W/m-K. The correction extended well to the rest of the calibration data, as shown by Figure 8.

![Figure 7: Mesh sensitivity](image2.png)

Figure 7: Mesh sensitivity

![Figure 8: Results with updated average thermal resistance](image3.png)

Figure 8: Results with updated average thermal resistance

The calibration tests output a distribution of conductive thermal resistance. Normalized against its surface average, the contour is plotted in Figure 9. When heat flux plate is installed in real system, the measured value would be used in the same conduction model. The result is a combination of two effects: the heat flux distribution caused by fluid
flow and caused by non-uniform conduction. Taking the normalized conductive resistance from calibration as a correction factor at the same nodal location, the effect of non-uniformity would then be excluded.

Table 2: Measurement uncertainty

<table>
<thead>
<tr>
<th>Measured Parameter</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature (T type)</td>
<td>0.1 (°C)</td>
</tr>
<tr>
<td>Absolute pressure (0-2067 kPa)</td>
<td>0.25% (full scale)</td>
</tr>
<tr>
<td>Differential pressure (0-10.0 kPa)</td>
<td>0.25% (full scale)</td>
</tr>
<tr>
<td>Differential pressure (0-37.4 kPa)</td>
<td>0.25% (full scale)</td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>0.1% (reading)</td>
</tr>
</tbody>
</table>

Figure 9: Distribution of normalized conductive thermal resistance (i.e. correction factor)

4. EXPERIMENTAL APPARATUS AND PROCEDURE

The schematics of the experimental apparatus is shown in Figure 10. It consists of three independent loops: water loop (right), refrigerant loop (center) and water-glycol loop (left). Two magnetic driven pumps with variable frequency drives are used to circulate the refrigerant and water, respectively. Expansion tank is placed at the highest location of the water loop, which is held vacuum until fully charged with distilled water so that no pocket of air is trapped inside. Micromotion flow meter, absolute and differential pressure transducers, and type T (copper-constantan) thermocouples are installed at locations as indicated in Figure 10. Their range and uncertainty after calibration are listed in Table 2.

Figure 10: Schematics of experimental system
National Instrument SCXI1000 chassis is used for data acquisition. It is connected to a desktop computer through PCI-MIO-16e-1 and used in conjunction with LabVIEW software. The modules and terminal blocks used in the data logger are SCXI1102-SCXI1303 for input measurement and SCXI1124-SCXI1325 for output control. All data are obtained under steady state conditions for about 20 minutes. The temperature was used in the same model in ANSYS as in calibration for heat flux calculation. High speed camera was used to obtain image that captured liquid-vapor interface.

5. PRIMARY RESULTS AND DISCUSSION

The method was first experimented with water-water single-phase heat transfer in a 2-channel setup without the visualization part. The test section was insulated so that energy balance (measured heat load between hot and cold stream) is within ±5%. The results of wall temperature and measured heat flux of counter flow arrangement are plotted in Figure 11, with the arrows indicating the flow directions. The mass flow rate of both stream is 30 g/s. The heat load calculated from cold and hot streams respectively are 640.8 W and 666.4 W. The low conductivity of the plate is compensated by the large fluid temperature difference. The measured heat flux appeared rather uniform.

However, the measured data for water-water is still not sufficient to discern the distribution of heat transfer coefficient, as the distribution of fluid temperature is unknown. It would require a fluid distribution model and calculate by marching from the inlet. As for a two-phase flow boiling, the saturation temperature could be used. For R245fa flow boiling, some preliminary results are shown in Figure 12. The saturation temperature of R245fa is 15.8 °C and the fluid enters the test section with a subcooling of 5.8 °C. The mass flow rate is 10.7 g/s (mass flux 23.16 kg/m²-s). Heat load is determined from water side, which gives the exit quality of 0.4. The saturation temperature is linearly interpolated from the pressure drop. As a result, the heat transfer coefficient is estimated in Figure 13 (b).

Figure 11: Water-water counter flow, wall temperature and measured heat flux

Figure 12: R245fa flow boiling, wall temperature and measured heat flux
Note the presented data is only preliminary results and should only be used qualitatively. The accuracy of heat transfer coefficient needs further investigation. However, from visualization, the upper center region where much lower heat transfer coefficient was observed, has much liquid content. It seemed to be a stagnant zone that vapor bypassed. In the bottom right, as shown by Figure 13 (c), bubbles generated near the entrance quickly became elongated. It seemed to prefer the path near the edge, as also indicated by the higher heat transfer coefficient. In general, the visualization suggested that the flow regime in a real operating condition is in continuous changing. Instead of a steady regime as shown in adiabatic flow map, it also interactively vary with the heat flux, which may suggest a potential optimized design to match the water side operating condition.

**Figure 13:** Preliminary visualization and heat transfer (a) overview; (b) heat transfer coefficient; (c) 0-140 mm center right; (d) 140-270 mm center right; (e) 270-400 mm center right

### 6. CONCLUSIONS

The development and validation of a novel method to combine measurement of local heat transfer coefficient and flow visualization in frame-and-plate heat exchanger (FPHE) is presented. The instrumented plate, essentially a heat flux sensor that preserves original geometry, seemed promising for such investigation. Although the accuracy still needs further validation, the preliminary results of visualization and its simultaneous heat transfer measurement of R245fa flow boiling proved their self-consistency. As a result, the visualization suggested that the flow regime in a real operating condition is in continuous changing, interactively vary with the heat flux. It may suggest a potential optimized design to match the water side operating condition, which was seldom reported during correlation development.

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