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DEVELOPMENT OF A TEST SECTION FOR DETERMINATION OF LOCAL HEAT TRANSFER COEFFICIENT DURING IN-TUBE FLOW

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Abstract. *This study presents the development of a test section for experimental determination of local heat transfer coefficient during single and two-phase flow inside tubes, focused on applications with secondary flow for heating or cooling. The non-intrusive experimental apparatus was developed using thermopiles to increase the sensitivity of the measurements. The heat transfer coefficient was estimated based on thermal resistance analysis and temperature measurement. The test section consists in a copper tube with 4.8 mm of internal diameter and 420 mm long, partially coated with polymeric material. Initially, the heat flux is imposed by an electrical resistor actuated by adjustable power supply. Experiments were performed for Reynolds between 982 and 4744 to determine the local heat flux and heat transfer coefficient. The results were compared with predictive methods widely adopted in the literature. Based on the experiments, the heat flux uncertainty was estimated as ± 6.2%. An uncertainty up to 9.1% has been obtained for heat transfer coefficient. Based on the developed prototype it is expected to perform experiments for condensation inside channels obtaining local heat transfer coefficients.*

Keywords: *convection, thermopile, heat flux*

1. INTRODUCTION

Several equipment and processes for industrial applications, including power generation and HVAC&R (*Heating Ventilation, Air Conditioning and Refrigeration*) are based on the convective heat transfer to cool and heat fluids and surfaces (Alvarez *et al.*, 2015). Despite the common use of heat exchangers, recent advances in manufacturing, as well as the current demand for more efficient and environmentally friendly equipment, require development and/or adjustment of predictive methods for heat transfer coefficient, which in turn relies on reliable experimental results.

In this context, the heat transfer coefficient can be determined based on the Newton's cooling law, as follows:

$$h = \frac{\dot{q}''}{T_{w,i} - T_f} \tag{1}$$

where $\dot{q}''$ refers to heat flux, and $T_{w,i}$ and T_f for surface and fluid reference temperature, respectively. Therefore, reliable approaches to measure temperature and heat flux are required in order to determine heat transfer coefficient with low uncertainty.

Kanizawa (2011) e Mogaji (2014) investigated the heat transfer process during internal flow in conventional diameters ducts, using circular tubes with thermocouples installed inside grooves machined on the external surface. The fluid temperature T_f was estimated from an energy balance along the main flow and, based on heat diffusion equation, it was possible to approximate the inner surface temperature of the tube $T_{w,i}$. Electric resistors were wrapped around the test section and controlled by adjustable power supply provided the flow heat input. Nevertheless, the measurement of surface temperature was not trivial and several considerations were necessary.

Similarly, Tibiriçá (2011) conducted experimental investigation of heat transfer coefficient in thin-walled microchannels. In this case, heat was provided by Joule effect, by directly imposing electrical current through the wall of the tubes. However, the materials that composed the channel wall must have high electrical resistivity, otherwise, excessively high electrical currents could lead to energy dissipation at unwanted regions of the system.

On the other hand, in case of experiments for flow inside channels under condensation, the heat flux is usually imposed via a secondary flow under controlled conditions. In this context, Cavallini *et al.* (2001) performed experiments for determination of heat transfer coefficient in a tube-in-tube heat exchanger configuration, with cooling water flowing through annulus region and condensing refrigerant inside. Type T thermocouples were positioned in four axial groves and the inner wall temperature $T_{w,i}$ was estimated based on heat diffusion analysis. The annular portion was divided into segments, each instrumented with a thermopile to evaluate enthalpy changes and, consequently, estimate the average heat flux per section. Finally, it was possible to calculate the mean heat transfer coefficient per segment. Likewise, Del Col *et al.* (2012) and Silva (2017) performed experiments to evaluate the heat transfer coefficient for propane and other refrigerants inside microchannels, focusing on refrigeration applications. However, the set of thermocouples used could act as fin, causing non-uniformity in heat transfer through the tube walls. Furthermore, positioning the thermocouples and thermopiles avoiding any disturbance in the flow is a significantly laborious process.

Alternatively, Wilson (1915) proposed a graphical method that avoids the measurement of the wall temperature and prevents effects of non-uniformity in heat transfer due to thermocouples. The technique was originally developed to evaluate the heat transfer coefficients in shell and tube condensers, with condensation occurring externally to the tubes and a cool liquid flowing inside. By neglecting fouling effect, assuming one-directional heat transfer, uniform properties and dominant thermal resistance corresponding to the internal flow convection, the thermal circuit of the shell and tube configuration could be written as:

$$R_{ov} = \frac{1}{C_t v_r^n} + C_c \tag{2}$$

where v_r is the cooling fluid velocity, R_{ov} is the overall thermal resistance determined from enthalpy change of the cooling liquid, n is an empirical velocity exponent and C_t e C_c are determined based on data regression. Observing Eq. (2), it is noticed by the authors that $1/C_t$ and C_c could be estimated by linear regression as reproduced in Figure 1 and the mean heat transfer coefficient based on these parameters.

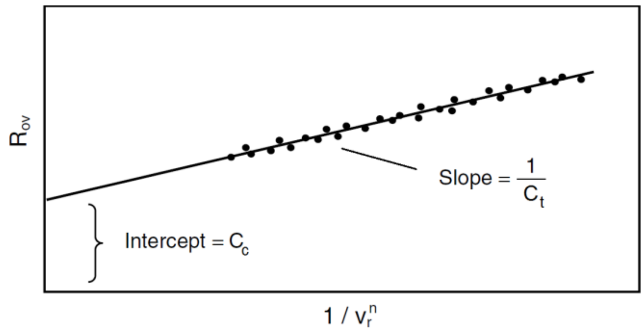

Figure 1. Wilson plot technique (Fernández-Seara *et al*, 2007).

In this sense, further modifications were proposed by Young and Wall (1957, apud Khartabil, 1987), Briggs and Young (1969, apud Khartabil, 1987) and Khartabil (1987), including modification of shell and tube side thermal resistances to increase accuracy. In this case, all the parameters are estimated from more than one linear regression procedure. This modified version is very useful when the system geometry is complex, such as for finned heat exchangers. However, Wilson technique and its modifications capture only mean heat transfer coefficient along a region, and are not capable of providing local information on heat flux and heat transfer coefficient.

Based on the discussion above, this study presents the development of a test section to determine the local heat transfer coefficient during single-phase and two-phase flow inside tubes with heating and cooling via secondary flow. The heat flux sensor is composed by thermopiles and thermocouples positioned inside the test section wall so that this arrangement is non-intrusive, minimizing any disturbances in the flow due to experimental devices. In order to validate the experimental apparatus, electrical resistor is tightly wrapped around the test section to validate the experimental apparatus. The results will be compared with classical methods from the literature.

2. METHODOLOGY

The present prototype was developed to determine local heat transfer coefficient. The apparatus is designed so that the main flow can be heated or cooled by means of secondary flow. In this sense, Figure 2 depicts the test section concept. Thin-walled copper tube is coated with epoxy resin of uniform thickness. The epoxy is appropriate due to the ability to withstand wide range of temperature, and to its relatively low thermal conductivity, allowing higher temperature difference measured by the sensors, improving the sensitivity of the method.

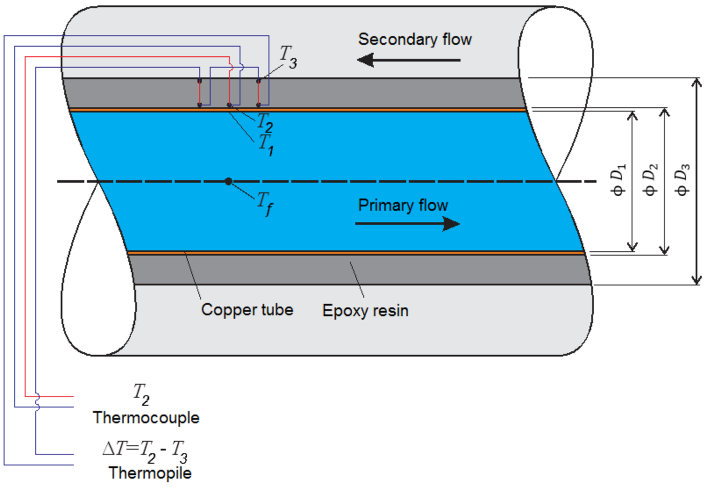

Figure 2. Schematics of the test section.

Type T thermocouples are placed on the inner and outer surfaces of the resin layer. According to procedures described in Doebelin (1990), a thermopile is configured by alternating copper and constantan wires to increase the electrical potential difference measured at the terminals. Therefore, the temperature difference measurement $\Delta T_{ep} = T_2 - T_3$ is more accurate than if each of the temperatures were measured individually.

Assuming steady-state conditions, the one-dimensional heat flux is estimated based on the thermal resistance of the epoxy layer, as follows:

$$\dot{q}''_{exp} = \frac{2\,k_{ep}\,(T_2 - T_3)}{\mathrm{D}_1 \ln(D_3/D_2)} \tag{3}$$

where k_{ep} is the thermal conductivity of the epoxy resin and the diameters are referenced in Figure 2.

The inner copper wall temperature is evaluated from the heat diffusion equation and temperature T_2. The fluid temperature T_f is evaluated based on energy balance along the flow. Once the temperatures and heat flux are known, the local heat transfer coefficient is calculated from Eq. (1).

2.1 Test section manufacturing

Figure 3 schematically depicts the test section setup. It corresponds to a horizontal copper tube with inner diameter D_1 of 4.8 mm and outer diameter D_2 of 6.4 mm, with a total length of 420 mm whereas 220 mm length refers to the segment of the copper tube covered by the epoxy layer, and subjected to heat transfer with secondary flow. The upstream region of 150 mm corresponds to the entrance region, which aims to promote flow development previous to entering the test section.

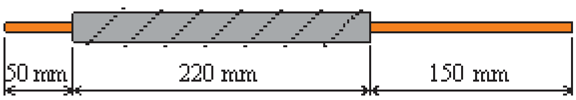

Figure 3. Test section measurements.

Table 1 presents the geometric parameters of the test section referred in the Figure 2.

Table 1. Geometric parameters of test section.

Parameter	**Length** (mm)
D_1	4.8
D_2	6.4
D_3	13.0

2.2 Experimental validation

In this step, the main flow is heated by electrical resistors wrapped around the heat flux sensor in order to allow imposing controlled heat flux and to validate the apparatus. Figure 4 depicts the experimental circuit to validate the method.

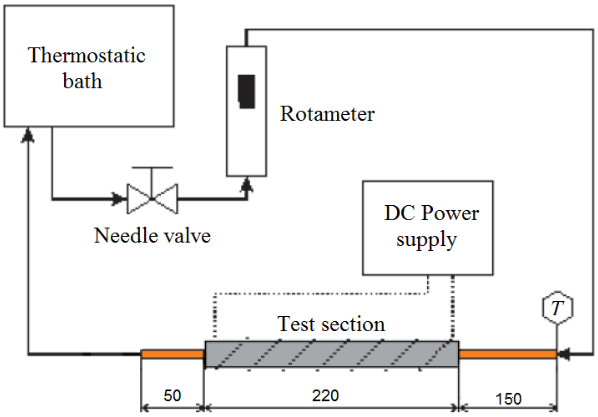

Figure 4. Test circuit for experimental validation, with dimensions in mm.

In the experimental validation circuit, distilled water flows from the thermostatic bath model Quimis Q-214M2. A control and measurement system composed by needle valve and rotameter adjusts the volumetric flow. Then, the flow is directed to the test section where it is heated by the electrical heater, and finally returns to the thermostatic bath. The complete circuit was thermally insulated by using elastomeric foam, except for the rotameter.

The volumetric flow measurement is performed by a rotameter model LZS DN15 with a measuring range from 6 to 60 L/h for water, and the flow control is performed manually through a needle valve. The heat flux was imposed by using flexible heater Omegalux FGS051-020 with nominal power of 105 W is helically arranged over the 220 mm section as indicated in Figure 4. These resistors are powered by a direct current power supply with adjustable voltage and current model PS-A303D. Finally, a thermopile was installed along the insulation layer to account for the heat gain/loss to the external ambient. Figure 5 depicts the experimental circuit developed by the research group at the Laboratory of Thermosciences at Universidade Federal Fluminense.

Figure 5. Test circuit at the Laboratory of Thermosciences at Universidade Federal Fluminense.

3. DATA REDUCTION AND PRELIMINARY RESULTS

Preliminary results were obtained for the experimental validation step. Part of the thermal energy supplied by the electric heaters $\dot{W}_e$ is dissipated in the external ambient ($\dot{q}_a$) and can be estimated based on the thermopile measurement of the elastomeric material, as follows:

$$\dot{q}_a = \frac{2\,\pi\,k_i\,L_{ef}\,\Delta T_{iso}}{\ln\left(\frac{D_4 + 2e}{D_4}\right)} \tag{4}$$

where k_i and e are respectively the thermal conductivity and elastomeric foam thickness, L_{ef} is the effective length of heat input given by the electrical resistor and ΔT_{iso} is the temperature difference between the external and internal surfaces of the thermal insulation measured by the thermopile.

Applying the First Law of Thermodynamics in a closed system containing the test section, the net heat power absorbed by the flow inside the tube can be determined by the following equation:

$$\dot{q}_{ref} = \dot{q}_a + \dot{W}_e \tag{5}$$

It is important to note that Eq. (5) will be taken as reference to validate the heat flux estimated by the test section.

3.1 Evaluation of the epoxy layer thermal resistance

The procedures adopted during the test section manufacturing could make its application unfeasible if the level of uncertainty of the heat flux sensor was significantly high. Therefore, to ensure that the experimental apparatus is validated, it is necessary to check eventual variation of the epoxy thermal resistance expressed by the following equation for various operating conditions:

$$R_{ep} = \frac{T_2 - T_3}{\dot{q}_{ref}} = \frac{\Delta T_2}{\dot{q}_{ref}} \tag{6}$$

Figure 6 depicts the epoxy thermal resistance calculated based on Eq. (6) for imposed heat flux in the range from 2.2 to 3.2 kW/m². The internal flow was maintained at a fixed flow rate of 54 L/h, corresponding to superficial velocity of 0.83 m/s, varying the inlet temperature in the test section from 15, 24 and 35°C.

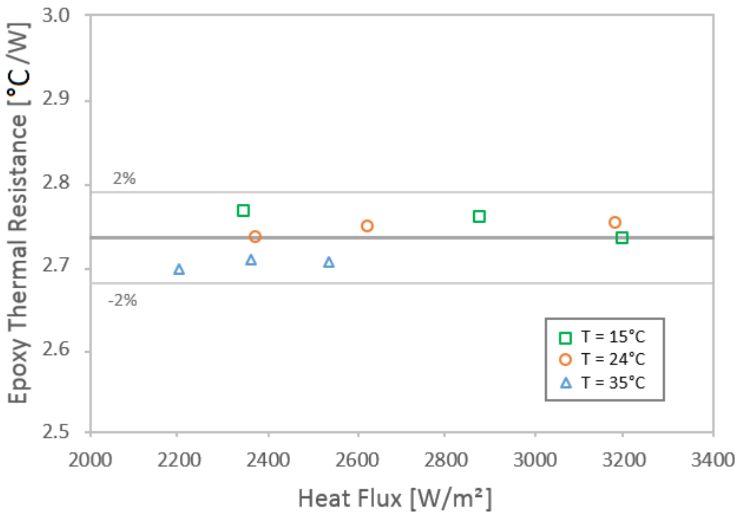

Figure 6. Experimental thermal resistance of epoxy layer.

As expected, the thermal resistance of the epoxy layer has a small variation around its average value of 2.73 °C/W. This indicates that this thermal resistance is independent of the operational temperature and heat flux.

3.2 Evaluation of heat flow in the experimental validation

The heat flux is estimated dividing the heat transfer rate to the fluid, given by Eq. (5), by the heat transfer area with the main fluid, as follows.

$$\dot{q}''_{ref} = \frac{\dot{q}_{ref}}{\pi\, D_1\, L_{ef}} \tag{7}$$

where L_{ef} is the length of epoxy layer.

Figure 7 represent the comparison between the heat flux obtained by Eq. (7) with the heat flux evaluated by the developed heat flux sensor, according to Eq. (3). The objective is to evaluate the deviations between the heat flux given by the testing section with the reference values evaluated based on the electrical power.

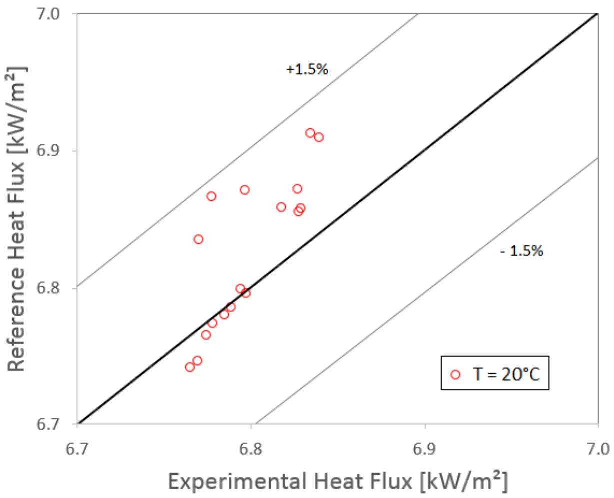

Figure 7. Comparison between reference and experimental heat flux.

According to this figure, it can be noticed satisfactory agreement, with deviations within ±1.5%.

3.3 Evaluation of the heat transfer coefficient for single-phase flow

The heat transfer coefficient is estimated from Eq. (1), as follows:

$$h_{exp} = \frac{\dot{q}''_{exp}}{T_1 - T_f} = \frac{\dot{q}''_{exp}}{\Delta T_1} \tag{8}$$

where $\dot{q}''_{exp}$ is given by Eq. (3).

The temperature T_1 can be determined by thermal resistance in the copper wall, as follows:

$$T_1 = \dot{q}''_{exp} \frac{D_1 \ln(D_3/D_2)}{2\,k_{cu}} + T_2 \tag{9}$$

where T_2 is measured by a thermocouple and k_{cu} is the thermal conductivity of copper.

The fluid temperature T_f is estimated by an energy balance between the entrance of the test section and the axial position of the thermocouple as follows:

$$T_f = T_e + \frac{2\dot{q}''_{exp} L_{ef}}{\rho \bar{v} D_1 c_p} \tag{10}$$

where ρ is the density of water, $\bar{v}$ is the mean flow velocity, c_p is the specific heat at constant pressure of and T_e is the fluid inlet temperature in the test section.

Figure 8 depicts the comparison between the heat transfer coefficients obtained from Eq. (8), using the facility described in section 2.2, and predictive methods available in the open literature, for Reynolds number between 982 and 4744. For laminar flow, it was considered a constant Nusselt number of 4.36. In the transitional and turbulent flow, the predicted heat transfer coefficient is obtained from Gnielinski (1976) correlation. The heat flux was fixed at 3360 W/m².

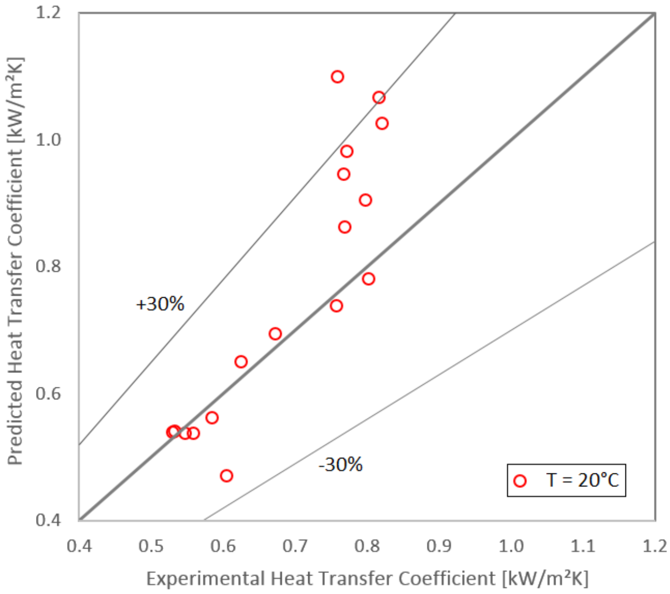

Figure 8. Comparison between experimental and reference heat transfer coefficient.

Figure 9 depicts the ratio of the experimental and predicted heat transfer coefficient values, and shows that 64.7% of the results are within ±25%. However, the deviations become more significant with the increase in volumetric flow rate, and the deviation might be attributed to the flow regime transition from laminar to turbulent. Nonetheless, based on these results, it can be concluded that the proposed procedure for estimating heat flux and temperatures, as well as section manufacturing procedure, are appropriate.

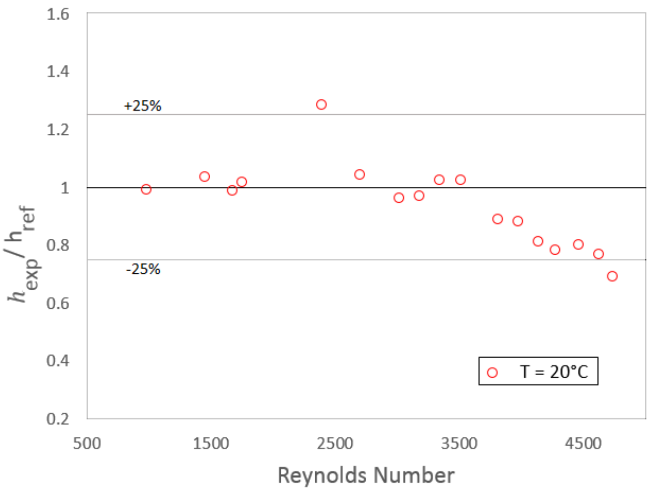

Figure 9. Comparison between the heat transfer coefficient ratio and the Reynolds number.

3.4 Error analysis

As the present study proposes a new experimental method, an uncertainty analysis of the parameters investigated must be a prior concern. Therefore, this section aims to present an uncertainty analysis of the proposed methodology for determination of heat transfer coefficient U_h. Based on the approach described by Moffat (1985) and Eq. (1), the U_h can be estimated as follows:

$$U_h = \sqrt{\left(\frac{U_{\dot{q}''_{exp}}}{\Delta T_1}\right)^2 + \left(\frac{U_{\Delta T_1}}{\Delta {T_1}^2}\right)^2} \tag{11}$$

where the heat flux uncertainty $U_{\dot{q}''_{exp}}$ is also estimated based on a propagation, however based on Eq. (7), and $U_{\Delta T_1}$ is the uncertainty of the temperature difference ΔT_1 which assumes the uncertainty for temperature measures of 0.2 °C, as described in Table 2, which was obtained based on a calibration procedure.

Nonetheless, the main concern in this analysis was to proper evaluate the uncertainty of the measured thermal resistance of the epoxy layer used in the calculation of the heat flux. Hence, as described in Abernethy and Thompson (1973), the uncertainty of the thermal resistance $U_{R_{ep}}$ can be estimated as follows:

$$U_{R_{ep}} = B + tS \tag{12}$$

where B is the bias error, t is the t-Student value and S is the standard deviation of the measurements present in Fig. 6. The bias can be expressed as the uncertainty of the reference value used to estimate the parameter of interest, by that:

$$B = \sqrt{\left(\frac{U_{\Delta T_2}}{\dot{q}_{ref}}\right)^2 + \left(-\frac{\Delta T_2 U_{\dot{q}_{ref}}}{\dot{q}_{ref}^{\ 2}}\right)^2} \tag{13}$$

where $U_{\Delta T_1}$ is the uncertainty of the temperature difference ΔT_2 listed in Table 2 and $U_{\dot{q}_{ref}}$ is the uncertainty of the reference heat flux, corresponding to the precision of the electrical power measurement.

Hence, Table 2 presents the list of uncertainties for the main parameters used in this study. According to this table, the heat transfer coefficient can be determined by this method with an uncertainty of ± 9.2% which is lower than the heat transfer coefficients reported in modified Wilson plot method by Khartabil (1987) ranging between 9.6 and 14.1%.

Table 2: Uncertainty for main parameters.

Parameter	**Uncertainty**
Temperature	± 0.2 ºC
R_{ep}	± 0.15 °C/W
Heat flux	± 6.2%
Heat transfer coefficient	± 9.2 %

4. CONCLUSIONS

This study proposes a new experimental procedure for local heat transfer coefficient determination for flows inside channels. Therefore, the following conclusions can be drawn:

- This method presented the development of a test section to determine the local heat transfer coefficient during single-phase and two-phase flow inside tubes, based on thermopile for estimation of heat flux;
- The thermal resistance of epoxy layer presented negligible deviation from its mean value, indicating its independence from operational temperature and heat flux;
- The heat transfer coefficient uncertainty of ± 9.2% obtained is lower than the modified Wilson plot method;
- It is expected to use the developed concept for experimental investigation of heat transfer coefficient for phase change problems, such as convective condensation, critical heat flux, among other conditions.

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