

## COB-2019-0549

# A NEW EXPERIMENTAL APPROACH FOR HEAT TRANSFER COEFFICIENT DETERMINATION DURING FLOW INSIDE CHANNELS

Matheus C. Constantino

Brunno Abreu da Fonseca Vargas

Fabio T. Kanizawa

Department of Mechanical Engineering, Universidade Federal Fluminense. Rua Passo da Pátria, 156, São Domingos, Niterói, RJ. CEP 24210-240

matheusconstantino@id.uff.br; brunnovargas@id.uff.br; fabiokanizawa@id.uff.br

**Abstract.** This paper reports a new experimental method for heat transfer coefficient determination during flow inside channels when a secondary flow is involved. The method is based in a cylindrical test section composed of a thin-walled copper tube covered with epoxy resin and built in heat flux sensor based on thermopile. The main purpose of this new methodology is to establish a proper way to estimate the local heat transfer coefficient of flows under cooling conditions, such as convective condensation, with reduced uncertainties in comparison with Wilson plot method. A validation test of the developed system and an error analysis is also presented.

**Keywords:** Heat transfer coefficient, Heat flux sensor, Thermopile

## 1. INTRODUCTION

Convective condensation inside channels is a common condition in heat exchangers widely used in important economic sectors, such as in refrigeration, heating and power generation industries. Condensers are key devices present in systems of common use, responsible, for example, to conserve food and to keep the temperature of an environment in comfort conditions. In this context, an accurate estimative of the heat transfer coefficient during condensation is necessary to allow engineers to design compact and efficient heat exchangers, which impacts the overall efficiency of the system. Even though some predictive methods have been developed during the recent decades for estimation of heat transfer coefficient and pressure drop during convective boiling and condensation, most of them rely on adjustments and validation by comparison with experimental results. In conditions of heat transfer during phase change, the complexity of the phenomena precludes the development of reliable models; hence engineers and researchers rely on experimentation to develop semi-empirical correlations for heat transfer coefficient.

The experimental results, however, cannot be directly applicable for conditions distinct than those of the experiments, including variation of working fluids, wall and saturation temperature, mass flux, etc., therefore, correlations have been developed and adjusted to predict the relevant parameters for distinct conditions. In this context, Silva (2017) compared the existing correlations for the heat transfer coefficients for in-tube convective condensation with database available in the open literature and observed significant deviations between experimental data and predictions, and among predictions. This conclusion can be partially explained by the fact that the phenomena involved in phase change flows are not totally understood. During in-tube convective condensation, the mass transfer occurring along the channel length implies in flow pattern transitions, as schematically depicted in Fig. 1, which impacts on the dominant mechanisms for heat and momentum transfer. Therefore, developing a reliable and general model that can be used in different circumstances than those in the experiment used for its development is challenging.

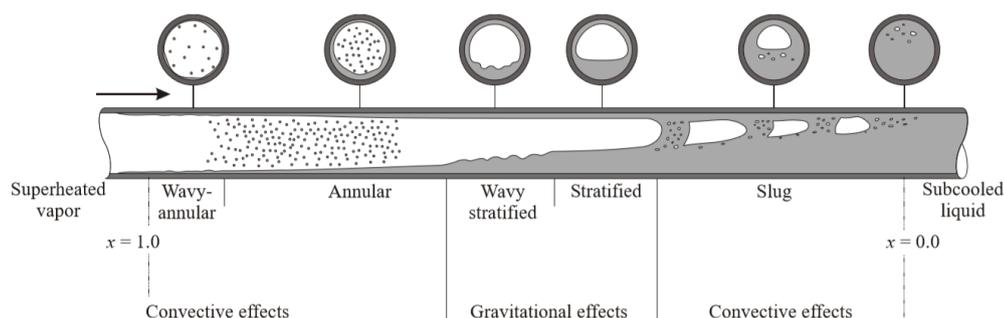


Figure 1. Schematics of convective flow condensation inside horizontal channel (Silva, 2017).

Based on these aspects, researchers developed precise experimental methods in controlled conditions to better understand the phenomena involved during heat transfer with phase change. In this context, the heat transfer coefficient  $h$  can be determined based on the Newton's cooling law, as follows:

$$h = \frac{q}{(T_s - T_f)} \quad (1)$$

where  $q$  corresponds to heat flux, and  $T_s$  and  $T_f$  for surface and fluid reference temperature, respectively, and all parameters should be determined precisely to obtain an accurate estimative of heat transfer coefficient.

For heating conditions, such as in studies involving convective boiling, the majority of the experimental methods for estimating the heat transfer coefficient are based in measure the surface temperature of the channel. In these conditions, the heat flux can be imposed in controlled and precise way via electrical heater; the local fluid temperature can be estimated by its saturation temperature, and the wall temperature by measurements via thermocouples and solution of the heat diffusion equation. Nevertheless, the measurement of surface temperature is not trivial, and depending on the experimental approach several considerations are necessary, such as discussed in the studies of convective flow boiling inside tubes presented by Kanizawa (2011), Mogaji (2014) and Tibiriça (2011).

Conversely, in case of experiments for flow inside channels under cooling situations, i.e. convective condensation, it is more difficult to impose heat removal from the working fluid in a precise way. In this context, the thermal energy is usually removed via a secondary flow with controlled temperature and flow rate, in a similar way to a tube-in-tube heat exchanger. However, measuring the surface temperature of the internal tube wall without perturbation of the external flow can be tricky in these applications, possibly introducing errors in the estimative of the heat transfer coefficient. Additionally, the heat flux depends on several parameters, including the internal and external flow conditions.

Cavallini *et al.* (2011), Silva (2017) and Rossato *et al.* (2017) used a similar approach to estimate the local heat transfer coefficient during condensation inside channels. A tube-in-tube condenser with refrigerant condensing in the inner tube and cooling water flowing in the segmented annulus region, as depicted in Fig. 2, was developed by Cavallini research group. By an energy balance on the cooling water side, the mean heat flux could be estimated in each segment, and by positioning type T thermocouples in four axial grooves, the inner surface temperature was estimated, so the heat transfer coefficient could be directly estimated by Eq. (1). However, the thermocouple installation can result in heat flux non-uniformity due the variation of thermal resistance that can occur on the grooves, and the sensors probes and wires can disturb the secondary flow, introducing errors on the heat transfer coefficient estimative. Moreover, in the case of investigation the heat transfer coefficient in tubes with reduced diameter, drilling the walls can compromise the integrity of the test section.

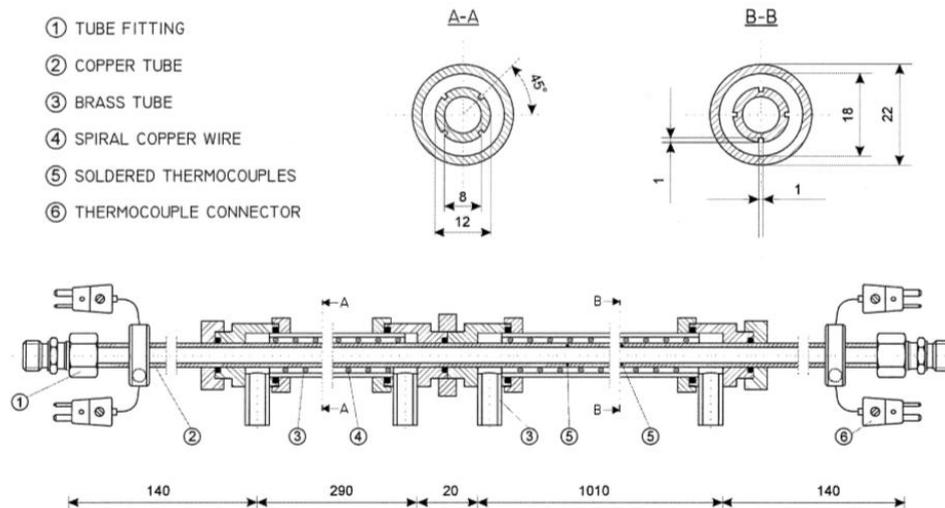


Figure 2. Drawing of the test section, with dimensions in mm (Cavallini *et al.*, 2011).

In this context, Wilson (1915) presented an alternative method that avoided the direct measurement of the wall temperature by a linear regression technique. Wilson originally developed the method for determination heat transfer coefficient during condensation external to horizontal tubes, but the method can be used for internal flow as well. Therefore, several studies were proposed in order to generalize Wilson's technique and to increase its accuracy, such as Khartabil (1987). However, a series of experiments is required for determination of a single experimental datum point according to the original proposal of the method. Wilson, therefore, was able to avoid all the possible perturbations on

the data due the wall temperature measurement, however the heat transfer coefficient estimative relies on mean values as the heat exchange measurement is based in energy balance of the secondary flow, which cannot account eventual transitions that can occur during two-phase flow or the particular phenomena involved in each flow pattern. As discussed above, Cavallini *et al* (2011) uses the same approach to estimate the heat flux imposed on the test section, which gives a mean value as the wall temperature varies along the flow path, hence, even by measuring the local wall temperature, the estimative of the heat transfer coefficient might not account certain phenomena in phase change flows. As presented by Rossato *et al* (2017), this problem can be minimized by dividing the channel of the secondary flow into several passes and making an energy balance in each section.

Based on the above discussion, this study aims to propose a new experimental approach for estimation of the local heat transfer coefficient in conditions where a secondary flow is involved. The technique is based on a surface temperature measurement, and heat flux estimation by built-in sensor based on thermopile. During the present stage, the system is calibrated and validated with imposed heat flux by electrical heating, and for single-phase flow. An error analysis is also presented, by means of analytical treatments for the uncertainties that can result in each parameter calculated by the method.

## 2. METHODOLOGY

The main purpose of the new developed technique is to properly estimate experimentally the local heat transfer coefficient for flows inside channels, with heating or cooling imposed by a secondary flow. Hence, a cylindrical test section was developed made from a thin-walled copper tube covered with epoxy resin, which consists of a polymer suitable for operation with intermediate temperatures. The inner surface of the copper is in contact with the working fluid, while the outer surface of the epoxy will be exposed to the secondary flow. The thermocouple joints were positioned in the external surface of the copper and of the resin, in order to allow measurement of the copper tube outer surface temperature ( $T_2$ ) and to form a thermopile for heat flux estimation. Fig. 3 schematically depicts the test section with copper and constantan wires represented by orange and blue lines, respectively.

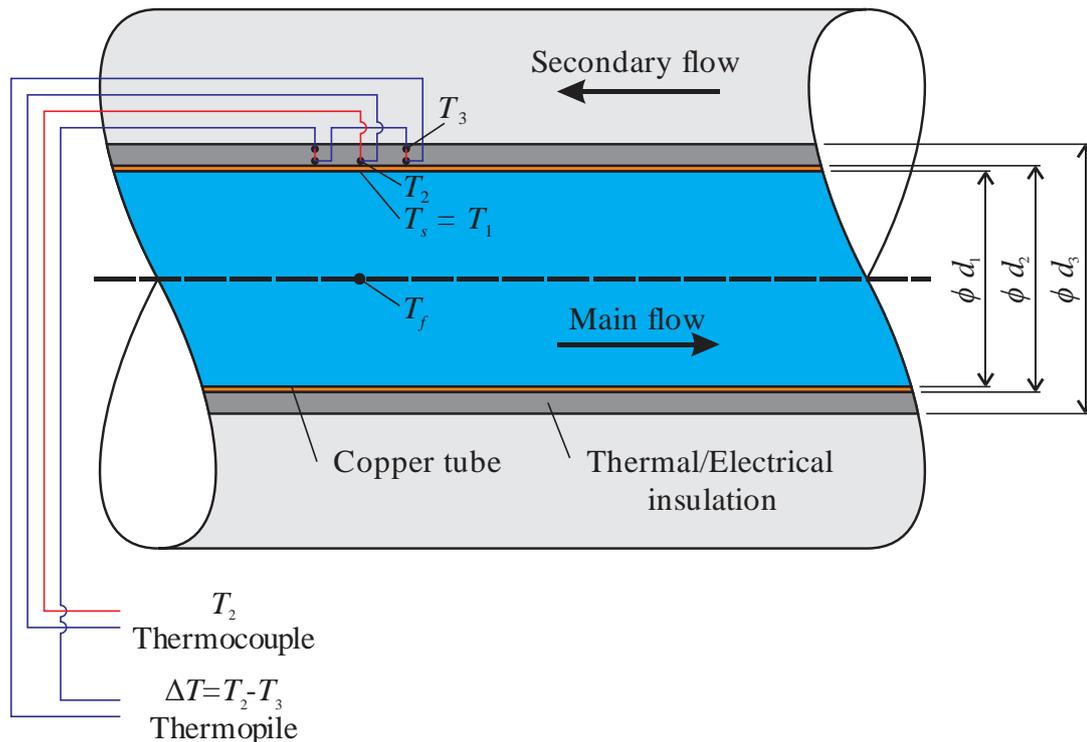


Figure 3. Schematics of the test section.

Therefore, is possible to determine the temperature difference along the resin layer  $\Delta T = T_2 - T_3$  from the thermopile, as described by Doebelin (1990), by measuring the voltage difference between the terminals. The heat flux imposed to the main flow  $q$  can be related with the temperature difference  $\Delta T$  by considering one-dimensional heat flux along the resin with constant properties, as follows:

$$q = \frac{2k_{iso}(T_3 - T_2)}{d_1 \ln\left(\frac{d_3}{d_2}\right)} \quad (2)$$

where  $k_{iso}$  refers to the resin thermal conductivity, and the diameters are represented in Fig. 3. Similarly, the internal wall temperature  $T_s$  can be estimate by solving the energy diffusion equation along the copper wall, as follows:

$$T_s = T_1 = q \frac{d_1 \ln\left(\frac{d_3}{d_2}\right)}{2k_{cu}} + T_2 \quad (3)$$

where  $k_{cu}$  is the copper thermal conductivity and the heat flux  $q$  is determined by Eq. (2). With the estimative of the inner surface temperature of the inner tube,  $T_s$ , it is possible to estimate  $h$  by Eq. (1). Hence, the main idea of the proposed methodology is positioning temperature sensors in the external surface of the tube and the thermal/electrical insulation manufactured around it without perturbing the external flow or drilling the tube wall, in order to measure the local heat flux and wall temperature.

## 2.1 Test facility

The test bench used in the experimental campaign presented in this work was built in the Laboratory of Thermal Sciences at Universidade Federal Fluminense, and is schematically depicted in Fig. 4.

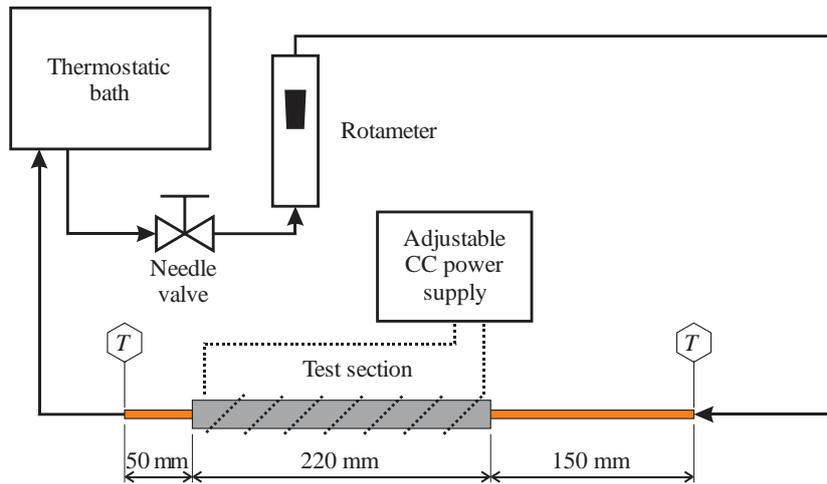


Figure 4. Schematic view of the test bench.

The experimental facility consists in a closed circuit composed by three main components: the thermostatic bath (model Q-214M2) to set the fluid temperature and that counts with a centrifugal pump to propel the working fluid through the circuit; a rotameter to measure the fluid volumetric flow rate with measuring range between 6 and 60 L/h, and corresponds to mean velocity between 0.09 and 0.9 m/s in the test section; and the test section. The conditioning system was used to establish the thermodynamic operation condition at the inlet of the test section, making possible to vary the temperature of the working fluid at atmospheric pressure in a range between 10 and 80°C. Hence, as represented in Fig. 4, water was initially conditioned at the desired temperature and then pumped through the circuit, passing by the rotameter, and then is directed to the test section, where the measurements of the heat transfer coefficient were made.

The test section consists of a plain copper tube with external diameter of 6.35 mm and 200 mm long, covered with a 3 mm cylindrical epoxy layer. A type T thermocouple was positioned on the outer surface of the copper tube in the central section of the test section and a two junction thermopile was positioned in such a way that was possible to measure the radial temperature difference occurring in the epoxy layer, as schematically depict in Fig. 3. In order to impose a heat flux into the working fluid, an electrical heater was tightly wrapped around the test section, and supplied by adjustable power supply. All the setup was thermally insulated, including the piping between the thermostatic bath and the test section, so the actual heat flux supplied at the fluid could be estimated based on the electrical power and system geometry. It is important to mention that a thermopile was installed along the insulation layer to account for the heat gain/loss to the external ambient.

### 3. DATA REDUCTION AND EXPERIMENTAL UNCERTAINTY

The main purpose of this new methodology is to establish a proper way to measure the local heat transfer coefficient of flows inside channels with low uncertainties. However, before proceeding to the heat transfer coefficient calculations, it is important to ensure that the thermal resistance of the epoxy layer used in the calculation of the local heat flux is well-known, because the procedure of manufacturing the test section can imply in great uncertainty in this parameter. Therefore, heat flux was imposed via electrical heaters tightly wrapped around the test section with the entire setup thermally insulated from the external ambient, so the thermal resistance of the epoxy layer could be estimated based on a reference value by energy conservation as follows:

$$\dot{q}_{net} = \dot{q}_{el} + \dot{q}_{amb} \quad (4)$$

where  $\dot{q}_{el}$  is the known electrical power supplied to the system, evaluated based on measurement of electrical current and voltage, and  $\dot{q}_{amb}$  is the heat gain/loss to the ambient estimated by the thermopile positioned at the insulation. Assuming a one-direction heat flux and constant proprieties of all the materials composing the system is possible to relate the net heat supplied to the fluid  $\dot{q}_{net}$  with the thermal resistance  $R_{epo}$  of the epoxy layer:

$$R_{epo} = \frac{(T_3 - T_2)}{\dot{q}_{el} + \dot{q}_{amb}} \quad (5)$$

Fig. 5 depicts the variation of the experimental thermal resistance estimated according to Eq. (5) as function of the net heat flux imposed via the electrical heater to the fluid flowing through the copper tube for inlet temperatures of 15, 24 and 35°C, and for heat flux ranging between 2200 and 3200 W/m<sup>2</sup>. According to this figure, it can be noticed that the thermal resistance is independent of operating temperature and heat flux, agreeing with the hypothesis of constant proprieties used in the development of Eq. (2), with mean value of 2.73 K/W.

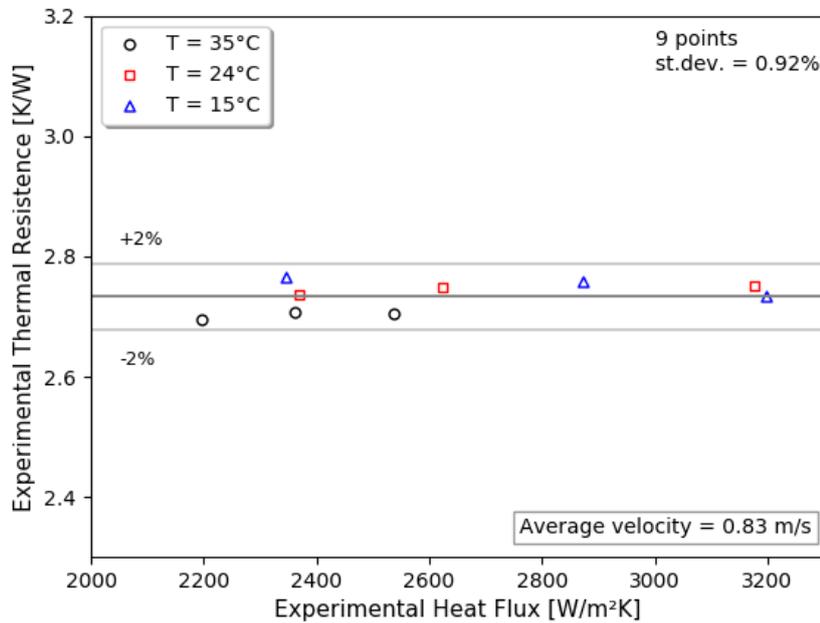


Figure 5. Experimental thermal resistance of the epoxy layer.

Is important to mention that for reduced heat fluxes, the experimental heat flux tends to present higher deviation from the mean value due the high uncertainty of  $\dot{q}_{amb}$ , which has the same order of magnitude of  $\dot{q}_{net}$  in these circumstances.

#### 3.1 Single-phase heat transfer coefficient

Considering that a new experimental methodology is proposed, it is important to ensure the validity of the measurements. Therefore, it is crucial validating the experimental results provided by the proposed method with the main correlations existing in the open literature. The heat transfer coefficient during single-phase flows has well known

correlations that describe the heat transfer phenomena as a function of the Reynolds and Prandtl numbers, for a variety of conditions with reduced uncertainties, under turbulent conditions and as a constant Nusselt number for the laminar region. Hence, the experimental data is compared with the predictions according to Gnielinski (1976) correlation and with a constant Nusselt number of 4.36, in order to evaluate its deviations and to validate the methodology proposed.

Therefore, experiments have been performed for Reynolds numbers ranging from 500 up to approximately 6000, hence all the data points were characterized by a flow relying in the laminar region and in the transition between laminar to turbulent. Based on these aspects, the Gnielinski (1976) correlation is suitable to be used under these circumstances, as its validity relies on Reynolds number higher than 2300. The laminar region is interesting for compare data as well, as the heat transfer phenomena is characterized by a constant Nusselt number of 4.36 for the case of constant surface heat flux.

Fig. 6, thus, presents the comparison between the experimental data and the Gnielinski (1976) correlation, as well for a constant laminar Nusselt number, for Reynolds number ranging between 982 and 4744, inlet temperature of 20°C, and 3155 W/m<sup>2</sup> of imposed heat flux. The figure shows a satisfactory agreement between the experimental and the predicted values. The maximum deviation was about 30%. The mean absolute deviation (MD) of the data set can be defined as follows:

$$MD = \frac{1}{N_p} \sum \frac{|h_{EXP} - h_{CALC}|}{h_{EXP}} \quad (6)$$

where  $N_p$  is the number of datapoints. For the present dataset the MD is approximately 13%, indicating that the procedure is coherent. It must be emphasized that these results were developed by accounting for the heat flux evaluated according to the heat flux sensor.

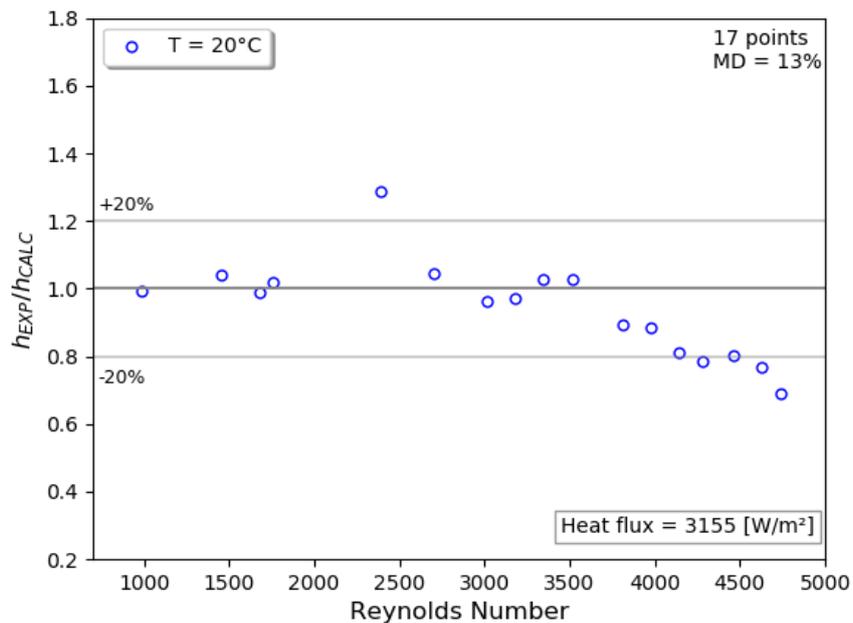


Figure 6. Variation of heat transfer coefficient ratio with Reynolds number. Predictions according to Gnielinski (1976) correlation for turbulent flow, and constant Nusselt number for laminar regime.

### 3.2 Error analysis

The purpose of this section is summarizing the uncertainties of the parameters investigate in this study, and to clarify how the uncertainty calculation of the proposed methodology was evaluated. The physical parameter of interest is the heat transfer coefficient and its uncertainty is estimated based on propagation of uncertainty of Eq. (1). The main concern in this analysis was to properly estimating the uncertainty of the measured thermal resistance of the epoxy layer used in the calculation of the heat flux. Therefore, as described in Abernethy and Thompson (1973), the uncertainty of the thermal resistance  $U_{R_{iso}}$  can be estimated as follows:

$$U_{R_{iso}} = B + tS \quad (7)$$

where  $B$  is the bias error,  $t$  is the Student value and  $S$  is the standard deviation of the measurements presented in Fig. 5. The bias can be expressed as the uncertainty of the reference value used to estimate the parameter of interest, in the present case consists on the combination of devices uncertainties, as expressed in the following equation:

$$B = \sqrt{\left(\frac{U_{\Delta T}}{\dot{q}_{el} + \dot{q}_{amb}}\right)^2 + \left(-\frac{\Delta T}{(\dot{q}_{el} + \dot{q}_{amb})^2} U_{\dot{q}_{el}}\right)^2 + \left(-\frac{\Delta T}{(\dot{q}_{el} + \dot{q}_{amb})^2} U_{\dot{q}_{amb}}\right)^2} \quad (8)$$

where  $U_{\Delta T}$ ,  $U_{\dot{q}_{el}}$ ,  $U_{\dot{q}_{amb}}$  are the uncertainties in the temperature difference measured at the epoxy layer, in the electric power supplied and in the heat gain/loss measured at the insulation. Therefore, Tab. 1 presents a list of uncertainties for the sensor and parameters used in this present work, as well as for the results, and according to this table, the heat transfer coefficient can be determined with uncertainty of  $\pm 9.1\%$ , which is lower than reported uncertainties of the modified Wilson plot method by Khartabil (1987), of the order of  $\pm 9.6\%$  until  $14.1\%$ .

Table 1. Accuracy for sensor and parameters

Temperature	$\pm 0.2^\circ\text{C}$	Heat flux	$\pm 6.7\%$
Average velocity	$\pm 0.015 \text{ m/s}$	Heat transfer coefficient	$\pm 9.1\%$
$R_{iso}$	$\pm 5.0\%$		

#### 4. CONCLUSIONS

This work presents a new experimental approach for heat transfer coefficient determination during flow inside channels when a secondary flow is involved. The following aspects can be concluded based on the new methodology proposed:

- The proposed method was developed for laboratorial investigation of local heat transfer coefficient in test section with secondary flow, based on thermopile for estimation of heat flux.
- The developed system has been evaluated for imposed heat flux in controlled conditions, and negligible variation with temperature and heat flux has been verified.
- An uncertainty up to  $9.1\%$  has been obtained for heat transfer coefficient estimation, which 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.

#### 5. REFERENCES

- Abernethy, R., and J. THOMPSON, JR. "Uncertainty in gas turbine measurements." 9th Propulsion conference. 1973
- Cavallini, A., et al. "Experimental investigation on condensation heat transfer and pressure drop of new HFC refrigerants (R134a, R125, R32, R410A, R236ea) in a horizontal smooth tube." *International Journal of Refrigeration* 24.1 (2001): 73-87.
- Doebelin, E. O., 1990. *Measuring Systems: Application and Design*. McGraw Hill Int., New York
- Gnielinski, Volker. "New equations for heat and mass transfer in turbulent pipe and channel flow." *Int. Chem. Eng.* 16.2 (1976): 359-368.
- Kanizawa, F. T., 2011. *Estudo teórico e experimental sobre padrões de escoamento e perda de pressão durante escoamentos monofásicos e bifásicos no interior de tubos com fitas retorcidas*. Master's Dissertation, Escola de Engenharia de São Carlos, University of São Paulo, São Carlos.
- Khartabil, H., 1987. *A modified Wilson plot technique for determining heat transfer correlations*. Doctoral dissertation, The Ohio State University.
- Mogaji, T. S., 2014. *Theoretical and experimental study on convective boiling inside tubes containing twisted-tape inserts*. Doctoral Thesis, Escola de Engenharia de São Carlos, University of São Paulo, São Carlos.
- Rossato, M., et al. "Heat transfer and pressure drop during condensation of low-GWP refrigerants inside bar-and-plate heat exchangers." *International Journal of Heat and Mass Transfer* 114 (2017): 363-379.
- Silva, J. D., 2017. *Estudo teórico e experimental da transferência de calor durante a condensação e perda de pressão no interior de minicanais para os refrigerantes R1234ze(E) e R32 com reduzido GWP*. Doctoral Thesis, Escola de Engenharia de São Carlos, University of São Paulo, São Carlos.

Tibiriçá, C. B., 2011. *Estudo teórico-experimental da transferência de calor e do fluxo crítico durante a ebulição convectiva no interior de microcanais*. Doctoral Thesis, Escola de Engenharia de São Carlos, University of São Paulo, São Carlos.

Wilson, Eo Eo. "A basis for rational design of heat transfer apparatus." *The J. Am. Soc. Mech. Engrs.* 37 (1915): 546-551.