Estimating local heat transfer coefficients from thin wall temperature measurements

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Abstract. An approach to experimental estimation of local heat transfer coefficient on a plane wall has been described. The approach is based on measurements of heat-transfer fluid and wall temperatures during some certain time of wall cooling. The wall was a thin plate, a printed circuit board, made of composite epoxy material covered with a copper layer. The temperature field can be considered uniform across the plate thickness when heat transfer is moderate and thermal resistance of the plate in transversal direction is low. This significantly simplifies the heat balance written for the wall sections that is used to estimate the heat transfer coefficient. The copper layer on the plate etched to form a single strip acted as resistance thermometers that measured the local temperature of the wall.

1. Introduction

Estimation of convective heat transfer parameters requires reliable information on the temperatures of heat-transfer fluid and the wall. It is quite challenging to obtain these temperatures as the temperature fields in general are non-uniform in space and vary in time. In practice, these measurements can be performed using the intrusive sensors that are inserted into the flow and/or the wall and can have negative effect on the hydrodynamics and heat transfer.

Different approaches to heat transfer research are employed nowadays. They include the solution to inverse heat conduction problem [1-3]. A temperature field of a solid body (a wall) is studied here. It develops depending, among others, on the boundary conditions of the solid body. Under the boundary conditions of the third kind (heat transfer coefficient $h=\text{const}$), there is a unique relation between the heat transfer coefficient distribution over the body surface and the temperature field within the body of given geometry and thermal properties. Under these conditions, the distribution of heat transfer coefficient along the interface between the solid body and fluid can be estimated using the measurements of their temperatures.

If thin plates (films) are employed for heat transfer research, a two-dimensional field can be considered instead of a three-dimensional one [4-6]. When the temperature is constant across the plate thickness, heat balance is considerably simplified, and so is the estimation of heat transfer coefficient. Thermocouples of different types are traditionally used for wall temperature measurements [6-8]. To obtain more detailed temperature distributions over the wall surfaces, optical methods are employed [5, 9]. However, optical methods on their own only give a qualitative pattern of a body’s thermal state.
To obtain the numerical values of parameters, these methods should be calibrated e.g. using thermocouples.

The integral characteristics of heat transfer are easier to estimate (at least, in terms of the procedure). When the distributions of thermal characteristics are irrelevant or constant across the heat-transfer surface, the total heat flux and surface-average heat transfer coefficients can be estimated from the heat balance written for the relevant channel sections [10].

In general, the research of complex flows requires detailed measurements of their hydrodynamic and thermal parameters [11-13]. In this respect, acquisition of large amount of data is associated with certain technical difficulties. The development of an efficient, reliable, and accurate approach to heat transfer coefficient measurements is therefore a relevant problem of experimental research.

2. The device for measurements of thermal parameters

A printed circuit board was proposed as a main element employed in the study of convective heat transfer and measurements of local wall temperature. The circuit board was made of composite epoxy material. It had the length \( l_h = 450 \) mm, the width of 200 mm and the thickness \( \delta = 1.6 \) mm (figure 1). Small thickness of the heat-transfer wall and its moderate thermal conductivity \( \lambda = 0.3 \) W/(m·K) made it possible to assume that the heat flux due to thermal conductivity along the wall was many times lower than removal of heat by the heat-transfer fluid. Then, allowing for proper external thermal insulation, the heat balance for a wall section contained only a single value of heat flux that was the heat flux from the wall to the heat-transfer fluid. Under these conditions, the measured values (temperature, heat flux) can be considered local and related only to the given section of the channel.

![Figure 1. Printed circuit board (heat-transfer wall) employed in measurements.](image)

Copper layer on the plate was etched to form a strip whose sections acted as 33 resistance thermometers. The area occupied by each strip (thermometer) had a length of 13 mm and a width of 100 mm. So, it was possible to measure the wall temperature local along the length and average across the channel test section with the spacing of 13.5 mm.

Thus, the circuit board took on two roles in the course of experiments. It was involved in heat transfer to the flow and provided the measurements of the wall temperature.

At low Biot numbers (\( \text{Bi} = \frac{h \delta}{\lambda} \ll 1 \)), the heat transfer coefficient, \( h \), on the wall with the area \( F \) and volume \( V \) was estimated from the wall temperature variation rate, \( m \), in the process of cooling or heating [14]:

\[
m = \ln(\Delta T_1/\Delta T_2)/(\tau_2 - \tau_1),
\]

which was derived from the dimensions and thermal properties:

\[
m = \psi h F/(\rho c V).
\]

Here, \( \Delta T_1 \) and \( \Delta T_2 \) are differences between the wall and flow temperatures at the time \( \tau_1 \) and \( \tau_2 \); \( \rho, c \) are the density and heat capacity of the wall; the coefficient of wall temperature field non-uniformity \( \psi = (B^2 + 1.44B + 1) - 0.5 \). Here, the modified Biot number is \( B = h F K / (\lambda V) \), where \( K \) is the body form factor. For a thin plate:
\[ K = \left( \frac{\delta}{\pi} \right)^{2}; \ B = h\delta/(\lambda \pi^{2}); \ m = \psi h/(pc \delta). \]

Parameters for the considered wall:
- at \( h = 20 \text{ W/(m}^2\text{ K)} \): \( B = 0.011; \ Bi = 0.1; \ \psi = 0.99; \)
- at \( h = 100 \text{ W/(m}^2\text{ K)} \): \( B = 0.054; \ Bi = 0.53; \ \psi = 0.96. \)

Thus, for heat transfer coefficients typical of gas flows, even if the temperature field across the wall thickness is assumed to be uniform, the error of the method lies within the range of fractions of a percent to several percent (if deviation of \( \psi \) from 1 is considered).

A circuit diagram was constructed for practical measurements (figure 2). The circuit diagram consisted of two electrical circuits:
- wall temperature measurement circuit;
- heat-transfer fluid temperature measurement circuit.

Each circuit was equipped with a DC voltage source \( E_{w} = 10 \text{ V} \) and \( E_{f} = 1.5 \text{ V} \), respectively. Signals from each resistance thermometer on the board, \( R_{1}, R_{2}, \ldots, R_{n} \), and a fixed resistor, \( R_{0} \), were recorded in the form of voltage, \( U_{1}, U_{2}, \ldots, U_{n} \) and \( U_{0} \), respectively, by two analog-to-digital converter units L-CARD E-14-140-M with 16 differential channels each. To measure the fluid temperature, the resistance thermometer, \( R_{TF} \), connected in a single circuit with the resistor, \( R_{0} \), was used. Voltages across the thermometer, \( U_{TF} \), and the resistor, \( U_{0} \), were also recorded by the analog-to-digital converter. Fixed resistors \( R_{0} = R_{0} = 226 \Omega \) were used to measure the electric current in the circuits of the wall temperature and fluid temperature measurement, respectively.

![Figure 2. Measurement diagram: 1 – measurement board; 2 – analog-to-digital converter; 3 – PC (laptop).](image)

3. Estimation of heat transfer coefficient and test measurements

The measured voltage yielded the resistance of strips of the measurement board:
\[ R_{i}(\tau) = R_{0} U_{i}(\tau)/U_{0}(\tau), \]
where \( i \) is the strip number.

The current wall temperature was determined from the known relation for the electrical resistance of copper:
\[ T_{i}(\tau) = (R_{i}(\tau)/R_{i}^{*}(\tau - 1))/\beta + T^{*}, \]
where \( \beta = 0.004 \text{ 1/K} \) is the temperature dependence coefficient of the electrical resistance of copper; \( R_{i}^{*}(\tau) \) is the resistance of strips in “cold” condition at the temperature \( T^{*} \).
The heat-transfer fluid temperature, \( T_f \), was estimated in the same way. The differences between the wall and fluid temperatures were calculated from these values:

\[
\Delta T(\tau) = T_i(\tau) - T_f.
\]

In the end, cooling rates of the strips, \( m_i \), and corresponding local heat transfer coefficients, \( h_i \), were determined from \( \Delta T(\tau) \).

Test experiments were conducted to check the performance of the submitted approach and dedicated device. The heat-transfer wall in the experimental setup was a part of the wall of the plane channel with the cross section of 115×150 mm\(^2\) and the length of 1.2 m (figure 3). This wall had the length of \( l_h = 450 \) mm and occupied the whole width of the channel bottom wall (150 mm). It was mounted in the middle part of the channel at the distance of \( x = 600 \) mm from the channel entrance. Here, the channel inlet section (where the inlet duct was attached) was taken as \( x = 0 \). To guarantee the turbulent flow in the channel, a wire mesh was mounted at the channel inlet. Its cell size was 5 mm; the diameter of the steel wire was 1.2 mm. The perimeter of the 50-mm long channel section immediately downstream of the mesh was covered with an abrasive with a grit size of 0.6-1.0 mm. To minimize the heat loss to the environment, a 30-mm thick polystyrene was glued to the external surface of the heat-transfer wall.

Test heat transfer experiments were arranged as follows. The suction fan installed downstream of the channel outlet was turned on and sucked the ambient air through the channel. The required flow rate was set by a controlling gate.

At the preliminary stage of experiment, an electric blow drier was installed in front of the channel inlet. The channel walls including the inlet duct were heated by the hot air flow reaching the temperature that exceeded the ambient air temperature by 50-60˚C. The heating time was \( \sim 10 \) min, i.e. until the temperature field of the channel walls became approximately uniform.

Then the blow dryer was turned off and moved away from the experimental setup to prevent its further impact on the air sucked by the fan. Since this moment the wall temperature and ambient air temperature were recorded. Then the heat transfer coefficient distributions along the channel wall were estimated from the temperature dynamics using the technique described above.

![Figure 3. Schematic of experimental setup: 1 – inlet duct; 2 – thermometer; 3 – channel test section; 4 – heat-transfer section.](image)

The studied channel was relatively short. Its length was \( L/d^* = 9.2 \), where the equivalent diameter \( d^* = 130 \) mm. Therefore the obtained experimental data should be considered as heat transfer coefficients of the channel entrance section. Test experiments were conducted for two mean freestream velocities \( U = Q/F = 1.9 \) and 3.1, where \( Q \) is the volumetric flow rate through the channel; \( F \) is the channel cross-section area (figure 4). The considered Reynolds numbers were \( \text{Re} = U d^*/\nu = 1.65 \times 10^4 \) and \( 2.7 \times 10^5 \). For reference, the dash lines show the distributions calculated from the following formula for the turbulent channel flow:

\[
\text{Nu} = 0.021 \text{Re}^{0.8} \text{Pr}^{0.43} \varepsilon_i,
\]
where the Reynolds and Nusselt numbers were calculated using the equivalent diameter $d^*$, the correction for the entrance region $\varepsilon_l$ was determined as a function of $x/d^*$ and Re [15]. The corresponding calculated and experimental data descend streamwise and have close values of $h$. The calculated values are somewhat higher. This is due to the fact that the calculated heat transfer coefficients were averaged over the $x$ length, while their local values were measured in the experiments. This being said, the calculated and experimental values of $h$ tend to converge and become constant further downstream. Thus, the obtained test data agree with the classical concepts of heat transfer in the channel entrance region.

**Figure 4.** Heat transfer coefficient at mean freestream velocity of 1.9 m/s (1) and 3.1 m/s (2): symbols – experiment; lines – calculation.

**4. Conclusions**

The results show that the submitted method of heat transfer coefficient measurement is quite efficient. It can be used to study heat transfer in different channels with a plane wall. Making different (in terms of location and size) strips on the measurement plate, one can obtain the corresponding distributions of heat transfer coefficient on the wall.

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