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Investigation of local heat transfer coefficients in plate heat exchangers with temperature oscillation IR thermography and CFD

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ABSTRACT

A method for the measurement of local convective heat transfer coefficients from the outside of a heat-transferring wall has been developed. This method is contact-free and fluid independent, employing radiant heating by laser or halogen spotlights and an IR camera for surface temperature measurements; it allows for the rapid evaluation of the heat transfer coefficient distribution of sizable heat exchanger areas. The technique relies first on experimental data of the phase lag of the outer surface temperature response to periodic heating, and second on a simplified numerical model of the heat exchanger wall to compute the local heat transfer coefficients from the processed data. The IR temperature data processing includes an algorithm for temperature drift compensation, phase synchronization between the periodic heat flux and the measured temperatures, and Single Frequency Discrete Fourier Transformations. The ill-posed inverse heat conduction problem of deriving a surface map of heat transfer coefficients from the phase-lag data is solved with a complex number finite-difference method applied to the heat exchanger wall. The relation between the local and the mean heat transfer coefficients is illuminated, calculation procedures based on the thermal boundary conditions are given. The results from measurements on a plate heat exchanger are presented, along with measurements conducted on pipe flow for validation. The results show high-resolution surface maps of the heat transfer coefficients for a chevron-type plate for three turbulent Reynolds numbers, including a promising approach of visualizing the flow field of the entire plate. The area-integrated values agree well with literature data. CFD calculations with an SST and an EASM–RSM were carried out on a section of a PHE channel. A comparison with the measured data indicates the shortcomings of even advanced turbulence models for the prediction of heat transfer coefficients but confirms the advantages of EASM–RSM in complex flows.

1. Introduction

As technology progresses and efficiency requirements increase, greater emphasis must be laid upon heat exchanger development to meet the challenge of maximum heat transfer at minimum pressure drop. A precondition for design enhancement with respect to effectiveness is a measurement method that allows quantifying the performance of convective heat transfer. Measuring local heat transfer coefficients allows gaining insight into the flow field, validating correlations and testing the performance of heat transfer equipment. Measurement results from the plate heat exchanger are used for the validation of CFD models for turbulent heat transfer. CFD simulations of plate heat exchangers can be carried out with these models in order to optimize the geometry to maximize performance while minimizing pressure loss. Such improvements, made possible due to detailed convection measurements, can lead to cost and energy savings in future heat exchangers.

Local heat transfer coefficients generally vary over the considered heat transfer area, just as the local temperature and the heat flux vary. For practical purposes, like heat exchanger layout design, not the local heat transfer coefficients but the mean value h_m is significant, as it factors into the overall heat transfer coefficient U. The relation between the local heat transfer coefficient $h(x,y)$ and the mean value h_m is illuminated in the following. By defining h_m, the total heat flow from the wall surface of a heat exchanger into the fluid becomes

$$Q = Ah_m\Delta T_m, \tag{1}$$

with A as the wetted surface area and ΔT_m being the mean of the local temperature differences $\Delta T = T_{surface} - T_{fluid} = q/h$ (neglecting any viscous dissipation and radiation) integrated over the area from 0 to x_A and y_A:

$$\Delta T_m = \frac{1}{A}\int_0^{xA}\int_0^{yA}\frac{q(x,y)}{h(x,y)}\,dy\,dx \tag{2}$$

In convective heat transfer studies and textbooks, typically two basic boundary conditions are distinguished: constant wall temperature and constant heat flux. Both idealized cases are

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Nomenclature

A	surface area (m^2)	α	thermal diffusivity (m^2/s)
a	plate corrugation amplitude (m)	δ	thickness (m)
c	thermal capacity (J/kg K)	ϕ	phase delay (rad)
h	heat transfer coefficient (W/m^2 K)	ν	kinematic viscosity (m^2/s)
k	thermal conductivity (W/m K)	ρ	density (kg/m^3)
$\dot{m}$	mass flow (kg/s)	ξ	dimensionless wall thickness (–)
Q	heat flow (W)	ψ	dimensionless heat transfer coefficient (–)
q	heat flux (W/m^2)	ω	angular frequency (1/s)
r	radial coordinate (m)		
S	volumetric heat source (W/m^3)	*Subscripts*	
T	temperature (K)	b	bulk
t	time (s)	m	mean
V	volume (m^3)	q	constant heat flux
X	sensitivity coefficient (–)	T	constant temperature
x, y, z	spatial coordinates (m)		

hardly reached even for instance in evaporators or condensers (constant temperature for pure fluids) or electric heaters (constant heat flux), as the local heat transfer coefficient varies and the wall's thermal conductivity allows lateral heat transfer. Consequently, real applications fall somewhere in between these two ideal cases. Interestingly, the relevant mean heat transfer coefficient h_m as defined in (1) is different for these two cases when calculated based on the local values $h(x, y)$. Written as an area integral of the local heat transfer coefficient, h_m becomes for constant temperature (T):

$$h_{m,T} = \frac{1}{A}\int_0^{xA}\int_0^{yA} h(x,y)\,dy\,dx, \tag{3}$$

This equation is well known and often appears in literature, e.g. [1], when local convection coefficients are averaged. To be strictly mathematically consistent with (1) and (2) not only the wall temperature, but also the fluid temperature must be constant over the area, so that ΔT becomes a constant. However, when the considered area of constant surface temperature is small and the heat transfer coefficient or the fluid temperature variation is small, the error introduced into the total heat flow (1) by using the product of two mean values rather than an exact integration over q also becomes small. For the constant heat flux case (q), to fulfill (1) and (2), h_m must be different, as the following simple example may point out. Consider a fluid streaming over two surface areas of 1 m^2 each and a constant heat flux of $q = 10\ kW/m^2$ applied to both areas. The heat transfer coefficient on the first area be $h_1 = 10000\ W/m^2\,K$, while the heat transfer coefficient over the second area be $h_2 = 2000\ W/m^2\,K$. Now Eq. (2) yields the correct $\Delta T_m = 3$ K, which becomes clear when assuming a sufficiently high fluid capacitance rate that the bulk temperature stays at 0 °C and the surface temperatures become $T_1 = 1$ °C and $T_2 = 5$ °C, for the first and the second area, respectively. The mean heat transfer coefficient as defined in Eq. (1) becomes $h_{m,q} = q/\Delta T_m = 3333\ W/m^2\,K$. In contrast, using Eq. (3) to evaluate h_m from averaging h_1 and h_2 delivers the erroneous value of 6000 W/m^2 K, which, at the given heat flux q would give an erroneous ΔT_m of only 1.666 K. This example demonstrates that Eq. (3) is strictly invalid for constant q and that a different formula has to be found. Such an equation can be derived by setting $h_{m,q} = q/\Delta T_m$ from Eq. (1) and substituting Eq. (2) for $\Delta T_m : h_{m,q} = q/\frac{1}{A}\int_0^{xA}\int_0^{yA}\frac{q(x,y)}{h(x,y)}\,dy\,dx$.

The constant q cancels out and the total area A can go into the numerator; h_m for the constant heat flux case ($_q$) becomes

$$h_{m,q} = \frac{A}{\int_0^{xA}\int_0^{yA}\frac{1}{h(x,y)}\,dy\,dx}. \tag{4}$$

This distinction of forming the mean heat transfer coefficient depending on the thermal boundary conditions is especially important when reporting the mean of measured local convection coefficients that vary widely over the area. For discrete measurement data, the integrals in Eqs. (3) and (4) degenerate into summations and $h_{m,T}$ becomes the area-weighted arithmetic mean while $h_{m,q}$ is the area-weighted harmonic mean; at equal grid spacing, the weights become unity and A becomes the count of the measured points.

When neither the heat flux q nor the temperature difference ΔT approach constant conditions over the area, the rigorous mathematical solution for the mean heat transfer coefficient also accounts for q as it cannot be cancelled out. Consistent with the definitions (1) and (2), h_m for arbitrary boundary conditions becomes:

$$h_m = \frac{\int_0^{xA}\int_0^{yA} q(x,y)\,dy\,dx}{\int_0^{xA}\int_0^{yA}\frac{q(x,y)}{h(x,y)}\,dy\,dx} \tag{5}$$

This equation, which follows directly when substituting (2) into (1), is rather impractical to use as it requires a priori the distribution of $q(x, y)$. The value of h_m (5) coincides with $h_{m,T}$ for constant ΔT and $h_{m,q}$ for constant q. For other boundary conditions, h_m is above $h_{m,q}$ and below $h_{m,T}$; the deviations of $h_{m,q}$ and $h_{m,T}$ from h_m depend on how closely the respective boundary conditions are approximated and on the variation of $h(x, y)$. The geometric mean of $h_{m,q}$ and $h_{m,T}$ will give a reasonable estimate of h_m for unknown or mixed boundary conditions.

In this study, the local heat transfer coefficients in single-phase flow in a plate heat exchanger have been measured employing a method that relies on temperature oscillations. The concept of temperature oscillation techniques is to supply periodically modulated thermal energy to a system and measure its temperature response. The measured phase angle of the temperature response is then compared to the solution of a mathematical model of the system. Material properties or boundary conditions of the model such as thermal diffusivity or heat transfer coefficients can be derived from the inverse solution. Temperature oscillation techniques have been applied to heat transfer measurements for decades and have proven to be valuable tools. The temperature oscillation IR thermography (TOIRT) is a new method whose beauty lies in the simplicity of its experimental approach: temperature measurements are taken on a heat-transferring wall with an IR camera and the temperature oscillations are generated by radiant heating, thus it is contact-free and fluid independent. The principle of such a technique is described by Wandelt and Roetzel [2], while major devel-