CZECH TECHNICAL UNIVERSITY IN PRAGUE

FACULTY OF MECHANICAL ENGINEERING

DOCTORAL THESIS STATEMENT

CZECH TECHNICAL UNIVERSITY IN PRAGUE
FACULTY OF MECHANICAL ENGINEERING
DEPARTMENT OF PROCESS ENGINEERING

DOCTORAL THESIS STATEMENT

Contactless heat transfer measurement methods in processing units

Ing. Stanislav Solnař

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Supervisor: prof. Tomáš Jirout, Ph.D.

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Disertant: Ing. Stanislav Solnař
Department of Process Engineering, FME CTU in Prague
Technická 4, 160 00 Praha 6

Supervisor: prof. Tomáš Jirout, Ph.D.
Department of Process Engineering, FME CTU in Prague
Technická 4, 160 00 Praha 6

Supervisor: Ing. Martin Dostál, Ph.D.
Department of Process Engineering, FME CTU in Prague
Technická 4, 160 00 Praha 6

Oponenti:

Theses were sent out:

The defense of the dissertation will take place on .... at . . . . o'clock in the meeting room No. 17 (on the ground floor) of the Faculty of Mechanical Engineering of the Czech Technical University in Prague, Technická 4, Prague 6 before the commission. It is possible to get acquainted with the dissertation at the Department of Science and Research of the Faculty of Mechanical Engineering of the Czech Technical University in Prague, Technická 4, Prague 6.

prof. Tomáš Jirout, Ph.D.
chairman of the board Konstrukční a procesní inženýrství
Faculty of Mechanical Engineering CTU in Prague

1 Introduction

For a good design of the apparatus where heat exchange takes place, it is necessary to know very well the map of the intensity distribution of heat transfer in such apparatus. For the common engineering task of heat exchanger design we usually settle for the average value of heat transfer coefficient, but to improve design, reduce apparatus or save money, it is necessary to display the intensity of heat transfer in 2D and study its changes with changing input parameters. The industry also calls for very fast and very accurate measurements of such quantities.

Despite the huge development of numerical simulations and thousands of scientific articles in the area of determining the values of heat transfer coefficient, experimental measurement still has its place also due to the validation of these numerical models. Experimental technique is constantly evolving, thus improving experimentally measured results and also increasing the speed of measurement and decreasing financial demands.

2 Basics of heat transfer

Heat transfer in general can be understood as energy that is in motion between two points that show different temperatures. We can say that there is a heat transfer between any two points with different temperatures. From the transmission point of view, three principles of heat transfer are known, namely conduction in solids or stationary fluids, convection between a solid wall and a flowing fluid, and radiation between two solid walls (Incropera et al. (2007)).

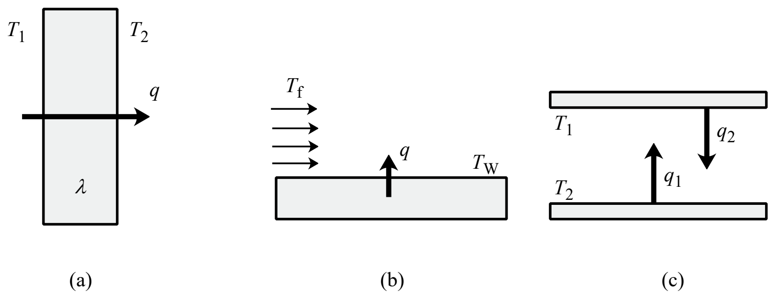

Figure 1: Three types of heat transfer (a) conduction through a solid wall (or not moving fluid), (b) convection from surface to a fluid and (c) radiation between two walls.

Conduction is associated with the mechanical energy of the particles at the microscopic scale and the energy is transmitted mechanically, by the movement of the particles. Conductive heat transfer can also occur in stationary fluids, but it is very often difficult to keep fluids stationary. Conductive heat flux can be expressed as the product of the material property and the temperature gradient, known as the Fourier law

$$q = -k\nabla T \tag{1}$$

where k represents the thermal conductivity of a material. This equation is named after J. B. J. Fourier, who first expressed this equation at the beginning of the 19th century. A negative sign in the equation is that the heat flows in the direction of the decreasing temperature(Incropera et al. (2007), VDI Heat Atlas (2010)).

Convective heat transfer is associated with macroscopic movement of flowing fluid. This type of transfer is actually a superposition of energy transfer due to the macroscopic movement of the fluid and the conductive heat transfer in the fluid. For this reason, convective heat transfer is dependent not only on the material properties, but also on the process properties such as fluid velocity, mixing intensity, shape of process equipment, etc.

The knowledge of convective heat transfer has a great influence on the design of new process equipment. A velocity and temperature profiles are created in the flowing fluid which has a direct impact on the magnitude of the heat transfer coefficient. Fluid behavior in the near wall region where large velocity and temperature gradients are known as boundary layers. This concept of boundary layers was first formulated by L. Prandtl at the beginning of the 20th century and is still in use.

The heat flux is directed towards a lower temperature so that in convection heat transfer both directions are possible (depending on the wall and fluid temperature, the wall can be heated or cooled) and its direction is considered normal to the wall surface. The amount of heat transferred depends on the velocity and temperature profile in the fluid and thus this task can become very complex e.g. for turbulent flow. We use the fairly simple relation to calculate the heat flux

$$q = h\,(T_\mathrm{W} - T_\mathrm{f}), \tag{2}$$

which is known as Newton's law of cooling. Symbol h represents the convective heat transfer coefficient that carries information about the material properties of the fluid, the geometric properties of the task, the surface roughness of the wall, etc.

Some typical values of heat transfer coefficient for various situations are shown in the Table 1 (Incropera et al. (2007), VDI Heat Atlas (2010)).

Table 1: Typical values of heat transfer coefficient VDI Heat Atlas (2010).

System	$h\,(\mathrm{W}/(\mathrm{m}^2\,\mathrm{K}))$
Free convection in gases	2 – 25
Free convection in liquids	10 – 1 000
Forced convection in gases	25 – 250
Forced convection in liquids	50 – 20 000
Boiling and condensing fluids	2 500 – 100 000

Radiation is the transfer of heat by means of electromagnetic waves, and so this transfer is also possible in vacuum, unlike the others. Any body emits radiation that is associated with the body's temperature. The basis for heat flux calculations is the so-called black body, which in practice is usually created by a body with an internal gap, which is heated to a certain temperature and has a small hole, which emits radiation energy out. The radiation heat flux of such a body can be described by relation

$$q = \sigma T^4, \tag{3}$$

where σ represents the Stefan-Boltzmann constant ($\sigma = 5.67 \times 10^{-8}\,\mathrm{W}/(\mathrm{m}^2\,\mathrm{K}^4)$).

3 Heat transfer measurement methods

Methods of measuring heat transfer coefficient can be divided into many groups as stationary and dynamic, contact and non-contact, direct and indirect, etc. of which the most known are about three groups, namely stationary, dynamic and comparative methods.

Stationary methods are closely related to Newton's cooling law from whose definition these methods are based

$$\dot{Q} = hS\Delta T. \tag{4}$$

The heat exchange surface S is generally known or can be measured relatively simply and then to determine the heat transfer coefficient h there are two possibilities: to set the temperature difference ΔT and read the supplied heat rate $\dot{Q}$ or to set the heat rate $\dot{Q}$ and read the temperature difference ΔT.

These methods are widely used in the science world for their simplicity and accuracy, but these methods are very time consuming and also conditions such as perfect thermal insulation etc. must be ensured.

Dynamic methods work with system response to supplied thermal information (lonely or periodically repeating). These methods are very fast but demanding on experimental components (usually very fast reading) and these methods are generally less accurate than stationary methods, but they can offer some positives such as complete contactlessness of the method, which may be useful in industrial measurement.

Comparative methods work on the principle of similarity between heat transfer and mass transfer. On this basis, mass transfer is measured in a geometrically identical or similar system, which is then converted into heat transfer. These methods are very interesting with different accuracy and speed.

In the literature we can also find some special methods such as optical, but they are very rarely used.

The first experimental work of measuring the heat transfer coefficient can be found during the 1920s and mostly in Europe. Rummel, Nusselt, or Hausen's work was primarily focused on describing and calculating the thermal characteristics of regenerators or air heaters, which was the main target of heat transfer at that time. In the pre-war period it is possible to find several works, in the vast majority of German. During World War II, it was not published very much and further experimental work can be found in the early 1950s, but mostly from the 70s onwards.

4 Aims of the work

Application of TOIRT method to complex flow geometry
Temperature oscillation method seems to be very suitable for measuring local and average heat transfer coefficient values in process units and apparatuses. Since it does not require any contact with the processing unit or the liquid in which the heat transfer takes place, it makes it possible to measure the heat transfer coefficient in containers and vessels with dangerous or even toxic liquids. Measurement results can be used to predict flow mode in devices or to detect changes in the vessel with regular measurements. However, this method has been tested on simpler geometries and I want to test its applicability to complex flow geometries such as vessels equipped with agitators.

Numerical and technical research of the oscillation method
The method has been thoroughly theoretically studied in the past and has also been experimentally verified on simpler geometries. I have prepared a numerical simulation study that allows to predict very quickly the surface temperatures in the measured wall. From the results of numerical simulations, I determined, among other things, the minimum number of heat waves for experimental measurements or the transient effect that occurs in the wall. Investigations of experimental equipment helped to improve the application of the method to individual geometries, mainly by the methodology of synchronization of used

heat flux sources and data processing.

Heat flux jump method

Heat flux jump method is a logical continuation of the oscillation method. The transient effect of the oscillation method would make it impossible to use all the measured data, or it is necessary to measure a large amount of data to minimize the transient error. Both approaches are not very suitable and so I have analytically derived the wall temperature change for these experiments and its results can serve to remove the transient phenomenon. In addition, the derivation serves as a stand alone measurement method, which is very suitable for low values of the heat transfer coefficient or as a measurement method for determining the incident heat flux on a wall.

5 Temperature oscillation method

The temperature oscillation method is a relatively new and not very often used method for measuring the local values of the heat transfer coefficient. The basis of the method consists in measuring the surface temperature of the measured wall at individual points by IR camera. The wall temperature is influenced by several factors, two of which are the strongest, namely the modulated heat flux and the heat transfer coefficient. If we influence the wall with the sinusoidal periodically repeated heat flux and measure the surface temperature, we find that the surface temperature is also modulated by the sine function, but is delayed behind the heat flux. It is the delay of the surface temperature behind the heat flux that is directly related to the heat transfer coefficient and thus can be determined. A schematic drawing of the method can be seen in the Fig. 2.

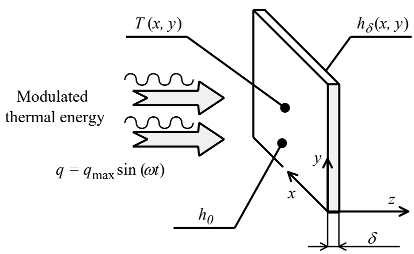

Figure 2: Schematic drawing of temperature oscillation method. The measured wall is shown in a Cartesian coordinate system where the wall in x and y has infinite length and in the z direction has a thickness δ.

The phase difference between the signals (see Fig. 3) is directly related to the value of the heat transfer coefficient on the other (unlit) side of the board. It can be said with exaggeration that this method allows to see behind the opaque wall.

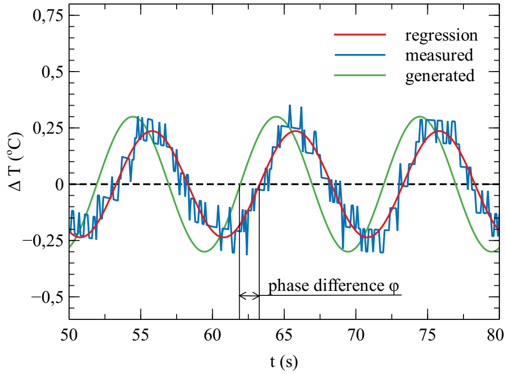

Figure 3: Example of comparison of generated and measured signal (obtained from measured data by regression).

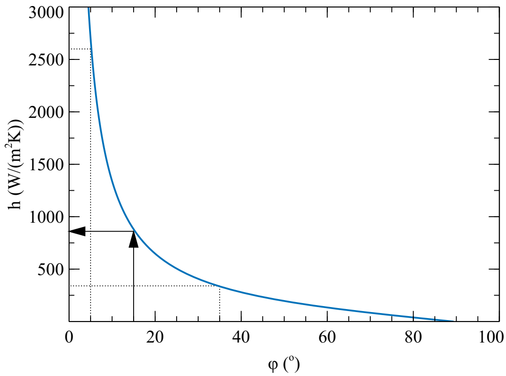

Figure 4: Analytical solution according to Wandelt and Roetzel (1997) presented as dependence of phase delay φ and heat transfer coefficient h for one experimental setup. The results are then interpolated from this dependency.

If I use several contactless thermometers or IR camera to measure the surface temperature during the experiment, it is possible to obtain a map of the heat transfer coefficient intensity.

Vessels equipped with agitators

Vessels equipped with agitators are probably the most frequently seen equipment in the process industry after pumps and heat exchangers. These apparatuses are primarily intended to prepare homogeneous mixtures or heterogeneous with the same concentration in the batch, but can also be used as heat exchangers. Particularly in the duplicator or in the tubular baffles inside the agitated vessels it is possible to exchange heat with another medium. For these reasons too, values (mostly only overall) of heat transfer coefficients in such apparatuses were very often measured in the past. There are a myriad of papers on this subject in the literature that also carry a huge number of different correlations and different geometries that have been experimentally measured.

However, the fluid flow in the vessel with impeller is very complex. For two basic operations (mixing and dispersing) two basic types of agitators, axial and radial, are used. However, neither of those perfectly creates an axial or radial flow, and there are always other streams present in the fluid. In addition, a tangential flow (mostly limited by baffles, but not perfectly) is also produced during the agitator operation, thus creating a 3D flow in the vessel.

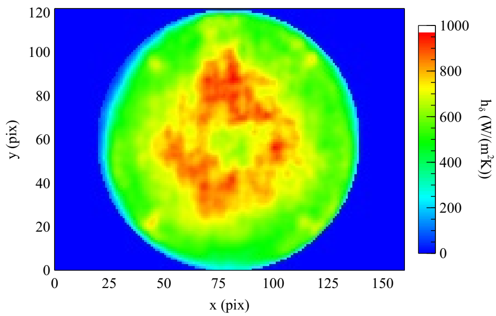

Figure 5: Distribution of heat transfer coefficient h_δ at the bottom of the agitated vessel with four baffles. 6PBT45 axial impeller with $d = 131$ mm ($d/D = 1/3$), impeller distance from bottom $h/d = 3/4$, Re $= 2.9 \times 10^4$. The bottom diameter of the container corresponds to approximately 120 measured pixels.

Other TOIRT method experimental measurements

■ **1D water flow in the pipe - validation experiment**
No. of measured series: 17
$\mathrm{Re} = 10000 - 35000$
variable parameters: material, tube dimensions, no. of waves, frequency of waves, generated heat flux, position of measurement
difference to main reference: (Gnielinski correlation) lower Re about 30%, higher Re about 5%

■ **2D water flow in a limited impinging jet**
No. of measured series: 9
$\mathrm{Re} = 8000 - 28000$
variable parameters: z/d ratio, generated heat flux, no. of waves, frequency of waves
difference to main reference: (Lytle and Webb, Persoons et al.) overall Nu about 10%, local values of Nu about 10%

■ **3D flow - bottom of vessel equipped with agitators**
No. of measured series: 8
$\mathrm{Re} = 5000 - 110000$
variable parameters: H_2/d ratio, generated heat flux, no. of waves, frequency of waves, agitator type, draft tube
difference to main reference: (various) overall Nu about 7%

■ **3D flow - wall of vessel equipped with agitators**
No. of measured series: 8
$\mathrm{Re} = 5000 - 110000$
variable parameters: H_2/d ratio, generated heat flux, no. of waves, frequency of waves, agitator type, draft tube
difference to main reference: (Karcz) overall Nu 50-70%, different tendency of Reynolds and Nusselt numbers dependence

■ **2D water flow - impinging jet with gravitational force**
No. of measured series: 1
$\mathrm{Re} = 4000 - 22000$
variable parameters: z/d ratio

6 Heat flux jump method

My idea of a new method and the derivation of gradual heating of the measured wall is shown in the Fig. 6. The radiative heat flux is modulated by the Heaviside function and is delivered evenly to the measured wall by halogen flood lights where a temperature profile is created by changing the heat flux. The thermophysical properties as well as the heat transfer coefficient h_0 are known (or calculateable) and I assume that the change in wall temperature will not be so great as to fundamentally alter the thermophysical properties.

By the analytical solution of the development of the temperature profile of the surface temperature of the measured wall and its comparison with the experimentally measured surface temperatures (IR camera, TLC color layer ...) it is possible to determine the heat transfer coefficient h_δ on the other side of the wall. In the latter case, the modulated sinusoidal heat flux is replaced by a constant heat flux, and for the known heat transfer coefficient h_δ, the heat flux that best matches the measured experimental data is found.

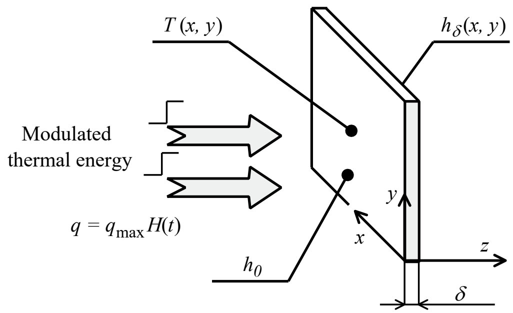

Figure 6: Schematic drawing of a new, heat flux jump method.

The method expects the thickness of the measured wall to be in the order of units or tenths of a millimeter and of a relatively well thermally conductive material (stainless steel, aluminum, copper ...). The temperature will be measured by a contactless IR camera, thanks to which we will obtain a large amount of data depending on the camera resolution.

To solve this problem, I have introduced several simplification and assumptions:

- Wall is homogeneous
- Incident heat flux is delivered evenly

- No lateral conduction in the wall
- Thermophysical properties of the wall are constant
- Heat transfer coefficient on the left-hand side h_0 is constant
- No heat losses to the environment
- Infinitely rapid propagation of heat wave in the wall

which lead to simplification of the task and its ability to analytically solve it.

Since the Heaviside and Dirac delta functions are interconnected by their own derivative, this derivation (with minimal modifications) can also be used for the impulse function. However, from an experimental point of view, such a solution is unsuitable because, due to the thermal capacities of the system, it would be necessary to use very strong heat flux sources in order to have a measurable change in the surface temperature.

I simplified the problem to solve a nonstationary Fourier equation in 1D space and only the temperature change from the initial value is solved

$$\frac{\partial (T - T_0)}{\partial t} = a \frac{\partial^2 (T - T_0)}{\partial z^2}. \tag{5}$$

with two boundary conditions of the third kind (Newton's)

$$-k \frac{\partial (T - T_0)}{\partial z}\bigg|_{z=\delta} = h_\delta \, \left(T - T_f\right)\big|_{z=\delta}, \tag{6}$$

and

$$-k \frac{\partial (T - T_0)}{\partial z}\bigg|_{z=0} = qH(t) - h_0 \, \left(T - T_f\right)\big|_{z=0}. \tag{7}$$

and also one initial condition

$$T_0 = T_f = 0 \tag{8}$$

A general solution using Laplace transformation and dimensionless variables t^* and z^* leads to a known equation of a sum of hyperbolic functions

$$\overline{T}(z^*, s) = C_1 \sinh(\sqrt{s}z^*) + C_2 \cosh(\sqrt{s}z^*), \tag{9}$$

where C_1 and C_2 are constants that has to be solved by boundary conditions.

Application of boundary conditions as well as initial conditions leads to the solution of two equations of two unknowns C_1 and C_2. This system of two equations can be written into matrix and solved, for example, with Cramer's rule for solving the system of equations. Determinant of the matrix A is

$$\det A = \sqrt{s}\left[\sqrt{s}\sinh(\sqrt{s}) + \mathrm{Bi}_\delta \cosh(\sqrt{s})\right] - \mathrm{Bi}_0\left[\sqrt{s}\cosh(\sqrt{s}) + \mathrm{Bi}_\delta \sinh(\sqrt{s})\right] \tag{10}$$

and the constants C_1 and C_2 can be calculated from the subdeterminants (using the Cramer's rule)

$$C_1 \det A = -\frac{1}{s}\frac{\hat{q}\,\delta}{k}\left[\sqrt{s}\sinh(\sqrt{s}) + \mathrm{Bi}_\delta \cosh(\sqrt{s})\right] \tag{11}$$

and

$$C_2 \det A = \frac{1}{s}\frac{\hat{q}\,\delta}{k}\left[\sqrt{s}\cosh(\sqrt{s}) + \mathrm{Bi}_\delta \sinh(\sqrt{s})\right]. \tag{12}$$

Therefore, finding a non-stationary temperature profile in the wall, as a particular solution to equations (5 – 8), can be expressed in Laplace's space as

$$\overline{T}(z^*, s) = \frac{C_1 \det A}{\det A}\sinh(\sqrt{s}\,z^*) + \frac{C_2 \det A}{\det A}\cosh(\sqrt{s}\,z^*) \tag{13}$$

The inverse Laplace transformation of such a complex image of function $\overline{T}(z^*, s)$ by direct translation or decomposition into partial fractions is too complicated, so I decided to perform the inverse transformation numerically. Abate and Whitt (2006) introduced a numerical solver for inverse Laplace transformation.

One of the limit cases of this solution can occur if both heat transfer coefficients are zero (modeled using very low coefficients $h_0 = h_\delta = 1\times10^{-16}$). In this case, the temperature change can be calculated analytically based on the calorimetric equation for the unit dimensions in the x and y directions ($x = y = 1\,\mathrm{m}$). The equation for calculating temperature change is

$$\Delta T = \frac{q\,t}{\delta\,\rho\,c_\mathrm{p}} \tag{14}$$

and is dependent on the thermophysical properties of the wall (ρ, c_p) as well as on the incident heat flux q and duration time t. Comparisons of the analytically calculated temperature changes and those calculated from our derivation are shown in Fig. 7 for various heat flux values q.

For the second limit case, where both heat transfer coefficients are infinitely large, the wall temperature should not change in any way. This prerequisite is also met.

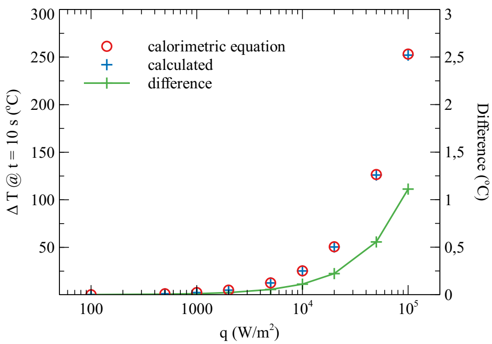

Figure 7: Calculated temperature difference ΔT at time $t = 10\,\mathrm{s}$ for various values of incident heat flux $\dot{q}$ in the limit case of zero heat transfer coefficients. The thermophysical properties of the wall are the same.

I compared analytically calculated temperature differences with numerical simulation in MATLAB to confirm the correctness of the method derivation. The numerical simulation was performed using the function `pdepe`, which is a solver of partial differential equations. Fourier equation in Cartesian coordinates was supplemented with the same boundary and initial conditions and numerically solved for 5 dimensional points (the study shows that 3 dimensional, which is minimum for calculation, points are sufficient). The analytical and numerical values are compared in the Fig. 8.

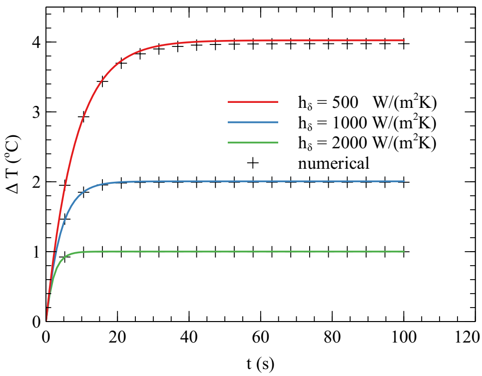

Figure 8: Comparison of analytically and numerically calculated values of temperature differences for different values of heat transfer coefficient h_δ.

The comparison of the numerically and analytically calculated values shows that the temperature differences also coincide for different values of the heat transfer coefficient h_δ. The maximum difference recorded is 0.06 °C ($\approx 1.4\%$), which is related to the numerical calculation.

Data reduction

All measured temperature fields from the IR camera are converted into a 3D matrix for further processing in MATLAB. From the surface temperature record at each point (see Fig. 9), the value of the time constant τ is then obtained by non-linear regression based on the model function

$$\Delta T = A\,(1 - \mathrm{e}^{-t/\tau}) + B. \tag{15}$$

From nonlinear regression results, individual time constants τ are then stored in the matrix $\tau(x,y)$.

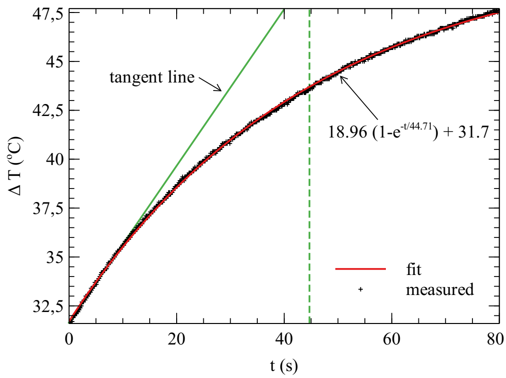

Figure 9: Recording of measured surface temperatures for one measuring point with fitted model function. This point has a time constant of about 44 s, which corresponds to a heat transfer coefficient h_δ of about 105 $\mathrm{W/(m^2\,K)}$ for a given system.

The resulting heat transfer coefficient h_δ is obtained by interpolation from the function $h_\delta = f(\tau)$, which is obtained by analytical solution. We can also see the relative insensitivity of the method, which reaches very good values for low values of the heat transfer coefficient.

Gnielinski with Hausen's correction

As the validation experimental measurement, I chose probably the most well-known experiment, namely heat transfer during flow in a tube. Due to the fact that the new method is more suitable for measuring lower values of the heat

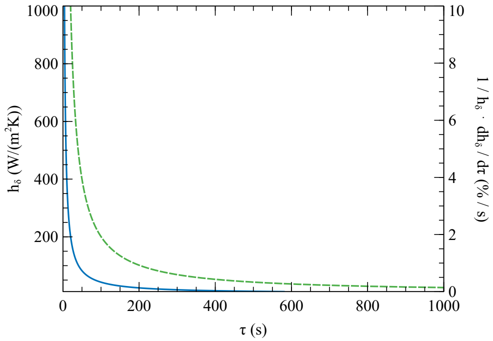

Figure 10: Dependence of heat transfer coefficient h_δ on time constant τ of measured wall temperature response. The dependence is supplemented by relative sensitivity $1/h_\delta \cdot \mathrm{d}h_\delta/\mathrm{d}\tau$

transfer coefficient, I chose an air as a fluid, where I expect a lower intensity of heat transfer.

For the experiment of air flow in a tube, the Gnielinski correlation with Hausen's correction for short tubes still works (and is also most often cited), and the area of Prandtl numbers also covers Prandtl number of air (about 0.7). For this case, I will also try to compare the local values of the Nusselt number from the beginning of the measuring section, where the temperature profile begins to form. For this case, there is a Gnielinski correlation for local Nusselt number values $\mathrm{Nu_x}$

A comparison of the selection of my experimental values of the overarll Nusselt number and the Gnielinski correlation with the Hausen correction for short tubes can be seen in the Fig. 11 as well as with the Dittus and Boelter correlation.

From the comparison we can see that the selected data (the one with the largest deviation) can be described by Gnielinski correlation with a maximum error of 17 %, the average error is around 8 %. The tendency of the dependence of the Nusselt number on the Reynolds number is very similar to that in the literature.

In terms of the average values of the Nusselt number, the method seems to be functional, it may be more interesting to compare the local values that the method allows to measure. In this case, the local values were averaged along only one coordinate (shorter side of the measured bar, about 10 measuring points) and compared with the Gnielinski correlation to calculate the local values of the Nusselt number $\mathrm{Nu_x}$.

In the literature, it is usual to plot local values depending on the dimensionless distance from the beginning of the measuring zone, which is related to

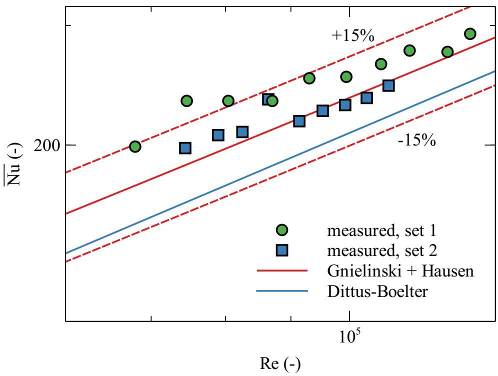

Figure 11: Comparison of a selection of experimentally measured overall Nusselt numbers (averaged over the whole measuring section) with the Gnielisnki correlation with Hausen correction for short tubes and the Dittus and Boelter correlation.

the inner diameter of the pipe x/d for various Reynolds numbers. Alternatively, the presentation of the results can be seen in the literature as the ratio of the local Nusselt number to its value in the very far region from the beginning.

A comparison of the local values of Nusselt numbers along the x coordinate from the beginning of the measuring section with the Gnielinski correlation for various Reynolds numbers can be seen in the Fig 12.

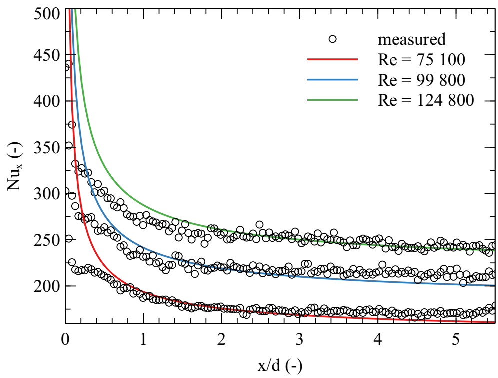

Figure 12: Comparison of experimental local values of Nusselt number with Gnielinski correlation for various values of Reynolds number.

From the comparison we can see that the values are very similar to those in the literature. A small deviation can be seen in the region of dimensionless coordinate $x/d < 1$, however, this is a deviation of up to 10%, which I consider to be a very good agreement. From the coordinates $x/d > 1$, the data are practically identical to the literature with minimal error.

Possibility of measuring heat flux density

When measuring using these methods, I recommend placing the heat flux sources that irradiate the measured wall evenly to avoid temperature gradients in the wall. By slightly modifying the equations, we can also get a tool to measure the incident heat flux. If I insulate the right hand side of the task (see Fig. 6, $z = \delta$, $z' = 1$), I will simulate an environment where the heat transfer coefficient $h_\delta = 0$. In this case, the only variable (if I calculate the value h_0 based on the correlations) is the heat flux q, which becomes measurable. In addition, it is possible to determine the distribution of heat flux $q(x,y)$ over the entire measured wall. Such a process leads to an improvement in the distribution of the incident heat flux on the wall.

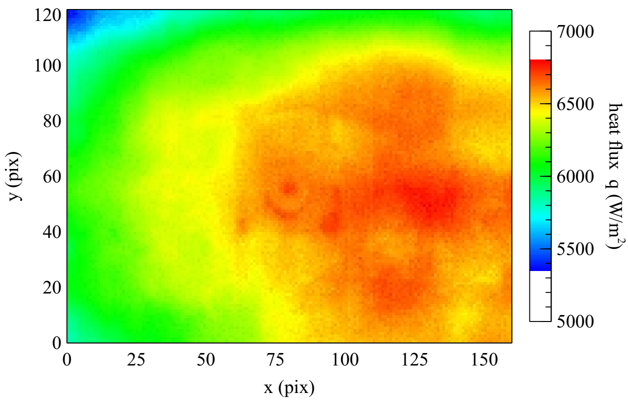

Figure 13: Heat flux distribution on the measured wall $q(x,y)$. The position of the heat flux sources has not been adjusted in any way to achieve a more even distribution.

Further modifications of the method

Since the Dirac and Heaviside functions are interconnected by their own derivation, it is possible to derive the method for Dirac's impulse function with minimal modification of the equations (the solution is even simplified and some members fall out of the equation). Unfortunately, from a practical point of view, such a method would be very difficult to implement (a very strong source of heat flow would be needed to create the impulse) and so this method remains theoretical for the time being.

Other HFJ method experimental measurements

1D gas flow in the pipe

No. of measured series: 6
$\mathrm{Re} = 70000 - 110000$
variable parameters: tube dimensions, length of signal, generated heat flux, position of measurement
difference to main reference: (Gnielinski correlation) overall Nu about 15%, local values of Nu 1 - 20%

2D gas flow - impinging jet

No. of measured series: 5
$\mathrm{Re} = 10000 - 25000$
variable parameters: z/d ration, generated heat flux, length of signal
difference to main reference: (Lytle and Webb) overall Nu about 10%, stagnant point Nu about 15%

2D gas flow - wind tunnel

No. of measured series: 3
$\mathrm{Re} = 10000 - 25000$
variable parameters: d of cylinder, generated heat flux, IR vs. TLC temperature measurement

7 Conclusion

In this work, local values of heat transfer coefficient for various geometric systems were experimentally measured. Two different methods were used to measure these local values, namely the temperature oscillation method (TOIRT) and the heat flux jump method (HFJ). Methods use a different exciting function, but some aspects are the same for them. Both methods are:

- very simple and quick to evaluate,
- fully contactless,
- without having to measure the temperature of the fluid,
- and with selectable spatial resolution

The thermal oscillation method has proven to be applicable to measurements on very complex flows, such as a vessel with a stirrer. Excellent results were obtained when measuring at the bottom of the vessel with the stirrer placed, while measurements on the wall showed other phenomena that are likely to disrupt the flow in this area. Sensitivity analysis of the thermal oscillation method shows that this method is not suitable for low values of the heat transfer coefficient, so I decided to derive my own method that would target this area, especially gas flows.

The heat flux jump method is derived and analytically, numerically and experimentally confirmed to be functional. With a minimum deviation, it is possible to measure 1D and 2D flows, which are very often found in process engineering.

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Total citations: 3 (excluded self citations)
h-index = 1

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