Heat transfer measurement using a thermochromic liquid crystal (TLC)

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Group 2

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1. Introduction

The goal of this experiment was to get a feel for different modes of heat transfer mechanisms and their quantification. In particular the convective heat transfer coefficient was quantified using a thermochromic liquid crystal for temperature measurement.
In order to operate turbomachinery at high temperature cooling inside the blades is required to keep temperatures below the melting point of the material. It is crucial to understand the heat transfer mechanisms as well as the flow conditions in order to design efficient cooling for turbomachinery. Because the inside of a blade under operation is experimentally not easily accessible a ducting channel as experimental model was set up with different geometries but comparable non-dimensional characteristic numbers [1]. The reason to do such an experiment is to gain insight on real life system and offer a setup to test CFD calculations.
The estimation of the heat transfer coefficient was carried out by measuring the temperature over time after the start of cooling of the ducting channel and the numerical evaluation of a mathematical model described in more detail in the theoretical section.

2. Experimental

The experimental dimensions were set up in a way to resemble a real example of turbomachinery cooling. The schematic procedure is depicted in Fig. 1. The dimensionless number matched in this experiment was the Reynolds number with a value of $Re = 1.4 \cdot 10^5$ [1].

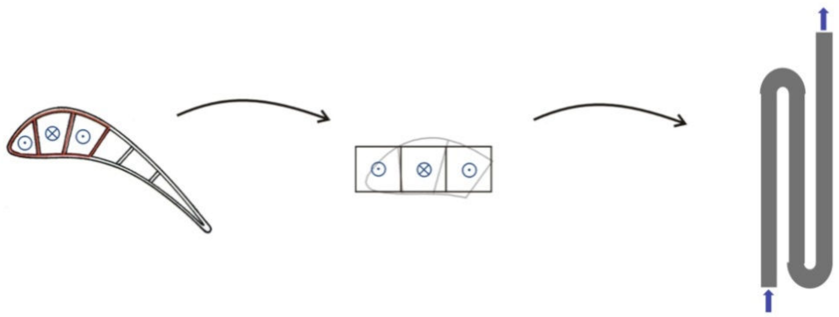

Fig. 1: From the real fin to the lab test section [1].

The experimental setup consisted of a warm water reservoir at $T_i = 39\,°C$. From this reservoir the warm water was continuously pumped through the test section depicted in Fig. 1. This was done for a sufficient amount of time in order to guarantee that all of the test section was heated up to T_i. Connected with a valve to the water loop was a cold water reservoir at $T_s = 27\,°C$. The acquisition was started as soon as the valve was switched and the cold water now circulating in the loop was cooling down the test section.

The acquisition was carried out by a CCD camera, directed towards the ducting channel, controlled by a PC. A total of 100 images were recorded by the camera over a time period of 4.5 seconds.

The test section was a ducting channel made out of aluminum covered by a Plexiglas for visual access of the TLC mounted on the bottom of the channel. The observation section recorded with the camera was located in the middle of the ducting channel and had additional turbulence promoters mounted before and after the observation area. There were no turbulence promoters in the rest of the ducting channel. Fig. 2 shows a shape of the turbulence promoters employed. Two vortices are expected to be created by each structure.

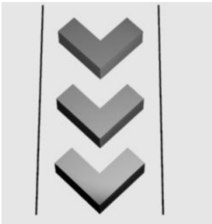

Fig. 2: Turbulence promoter employed in the experiment [1].

3. Theory

Heat transfer is the transfer of thermal energy over a temperature gradient from the hot item to the cool item or to the surrounding. It can be divided into three basic types: heat transfer by conduction, heat transfer by convection and heat transfer by radiation [1].
Heat transfer by radiation becomes dominant at higher temperatures and can be neglected at lower temperatures because it scales with the fourth power of the temperature as written in the Stefan-Boltzmann equation for the power output of a hot surface:

$$P = A \cdot \varepsilon \cdot \sigma \cdot T^4 \quad (1)$$

With temperatures of 25 and 39 °C it can be completely neglected in this experiment.

Conduction is the transfer of heat among all kind of matter without any mass transfer. It originates solely from a temperature gradient within the media and contributes to the equilibration of temperature within and among different media.

$$\dot{q} = -k \cdot A \cdot \frac{dT}{dx} \qquad [k] = \frac{W}{m \cdot K},\ [A] = m^2,\ [T] = K,\ [x] = m \quad (2)$$

k in formula (2) stands for the conductive heat transfer coefficient, a material property.

The main mechanism of heat transfer under investigation in this experiment is heat transfer by convection. It can only take place in fluids and can be divided into free and forced convection. Free convection takes place without an input of external work. The gradient of density in a fluid from the lower density close to a hot surface towards the bulk of the fluid at a lower temperature leads to natural circulation within the fluid and provides a mechanism for the transportation of thermal energy away from the hot surface. Forced convection works by forced circulation within the fluid to carry away heat. It increases heat transport compared to free convection. Typical examples of forced convection are a fan on a hot summer day or the air cooling inside rotor blades of turbomachinery. Heat transfer by convection is given by formula 3.

$$\dot{q} = -\alpha \cdot A \cdot dT \quad [\alpha] = \frac{W}{m^2 \cdot K} \quad (3)$$

Where α stands for the convective heat transfer coefficient. It depends on the fluid properties and on the flow regime. A turbulent flow regime results in higher convective heat transfer and therefore a higher α. Therefore turbulence promoters are introduced into convective cooling systems such as in the inside of turbine fins. The goal of this experiment was to measure and quantify the convective heat transfer coefficient α.

To describe the heat transfer mechanism of a system the Nusselt number is used.

$$Nu = \frac{\alpha L}{k} \tag{4}$$

Where L stands for the characteristic length relevant for the flow. It gives the relationship of convective over conductive heat transfer. A higher Nusselt number means a more convective and therefore more efficient heat transfer. The Nusselt number can be calculated with the Dittus-Boelter equation which is valid at high Re numbers:

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \tag{5}$$

Since Pr and Re were known in our system we could then calculate the Nu number according to (5) to be $Nu_0 = 178.9$. We then divided the measured Nu number from (4) by the calculated Nu number from (5) to quantify the effect of the turbulence promoters shown in Fig. 2.

It is impossible to measure actual heat fluxes in a system. The only measurable changing quantity of heat transfer is temperature. There are several systems available for temperature measurements. Classical thermometers as well as thermocouples deliver reliable but not spatially resolved temperature measurements. To obtain a spatially resolved temperature measurement an infrared camera with temperature evaluation according to formula (1) or a thermochromic liquid crystal (TLC) can be used.

The liquid crystal consists of long chain organic molecules aligned in a preferential direction. The temperature determines the reflected wavelengths of the molecules and the TLC's color delivers an optical indication of temperature. The calibration was done beforehand and the data was reused in this experiment to link the recorded color of the TLC to an absolute temperature value. The temperature range of the TLC was 35 to 37 °C.

The goal of the experiment was to determine α. In order to be able to calculate α the one dimensional heat equation has to be considered [2]:

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} \tag{6}$$

A semi-infinite assumption was then made in order to solve the partial differential heat equation. The semi-infinite assumption states that at an infinite x the temperature will be equal to the initial temperature T_i before convective heat transfer. This is of course a simplification of reality and in our case the temperature at the back of our solid will start to change rather quickly and therefore violating the semi-infinite assumption. However it is valid for a short amount of time and will deliver a reasonable solution for α. The maximum measurement time, meaning the time interval in which the semi-infinite assumption holds true is given by [2]:

$$t_{max} = \frac{L^2}{a} \; with \; a = \frac{\lambda}{\rho \cdot c_p} \tag{7}$$

α is called the thermal diffusivity and L is the baseplate thickness. In our particular setup with an aluminum baseplate $t_{max} \cong 1.5\ s$. The boundary and initial conditions of the heat equation are given by:

$$T(x, t = 0) = T_i \quad (8)$$

$$T(x \to \infty, t) = T_i\ (semi - infinite\ assumption) \quad (9)$$

$$T(x = 0, t) = T_{surface} \quad (10)$$

The heat equation with initial conditions (7) and (8) has a solution which can be numerically solved for α if T and T_i is known.

The data of the CCD sensor was converted in MATLAB from RGB to HSI color space and then matched with the calibration data provided. The numerical solution was also calculated in MATLAB and the resulting heat transfer coefficients were plotted as a function of time.

4. Results and Discussion

During the time period of the experiment the visual observation clearly showed a faster change of color in the observation area than for the rest of the cross section. This is due to the turbulence promoters before and after the observation area increasing α.

Fig. 3 depicts the raw data obtained by the camera during the first 50 frames. The damages on the TLC are clearly visible as the dark spots in the series of frame 20 to 30 (row three in Fig. 3). They will affect the overall result since the calibration was done when the TLC was still intact.

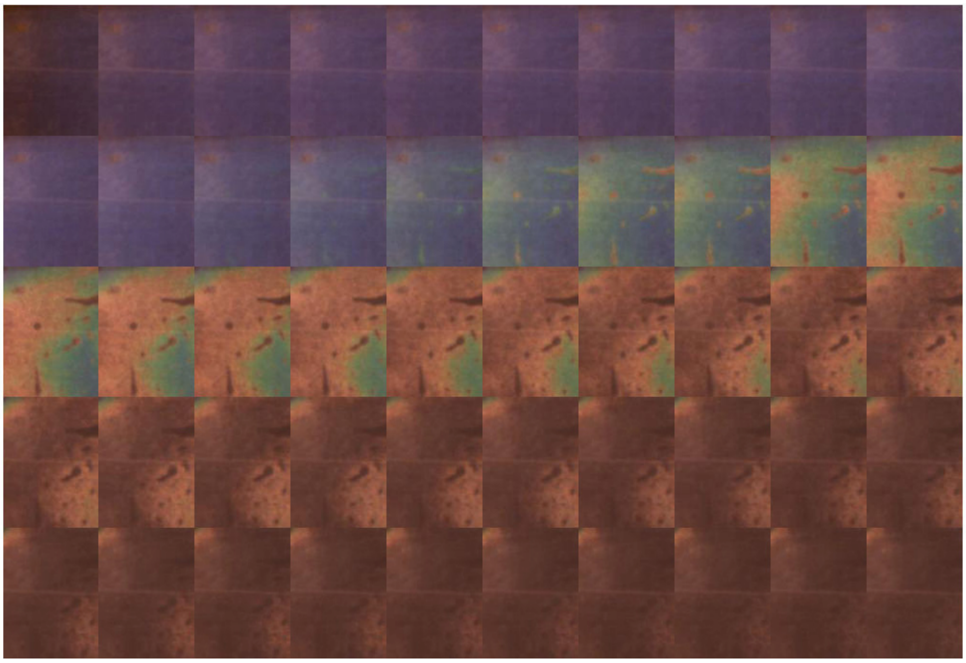

Fig. 3: TLC during frame 1 to 50 in the experiment.

The surface temperatures from frame 1 to frame 100 according to the calibration values are shown in Fig. 5. Clearly there must be something wrong in the matching of the TLC color and temperature. A smooth transition from blue to red is expected from the calibration pictures for the cooling of the TLC. Fig. 3 already shows red spots and a red color tendency associated with lower temperatures in the first frames before cooling. This is not in accordance with the calibration pictures shown in Fig. 4.

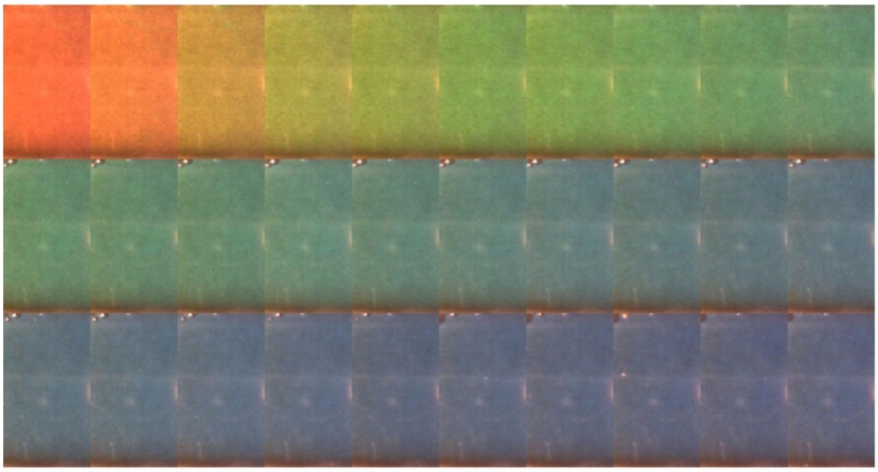

Fig. 4: Calibration pictures with known temperatures used for evaluation. Pictures are from cold to hot.

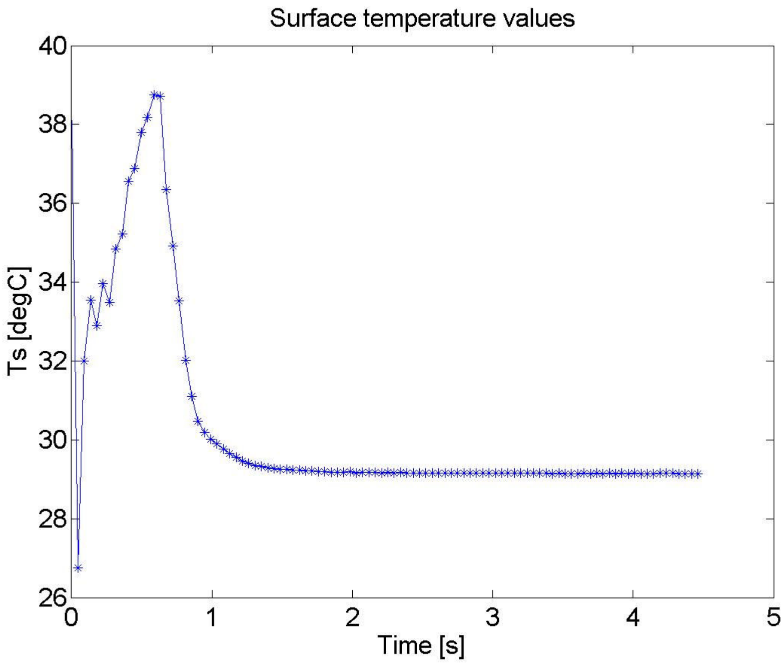

Fig. 5: Surface temperature of TLC according to the calibration values.

The pictures after frame 50 can be cut because they are out of calibration range (temperature too low) and because the take place after t_{max} from (6). Cutting also frame 1 to 15 one obtains a nicer and more realistic distribution as shown in Fig. 6.

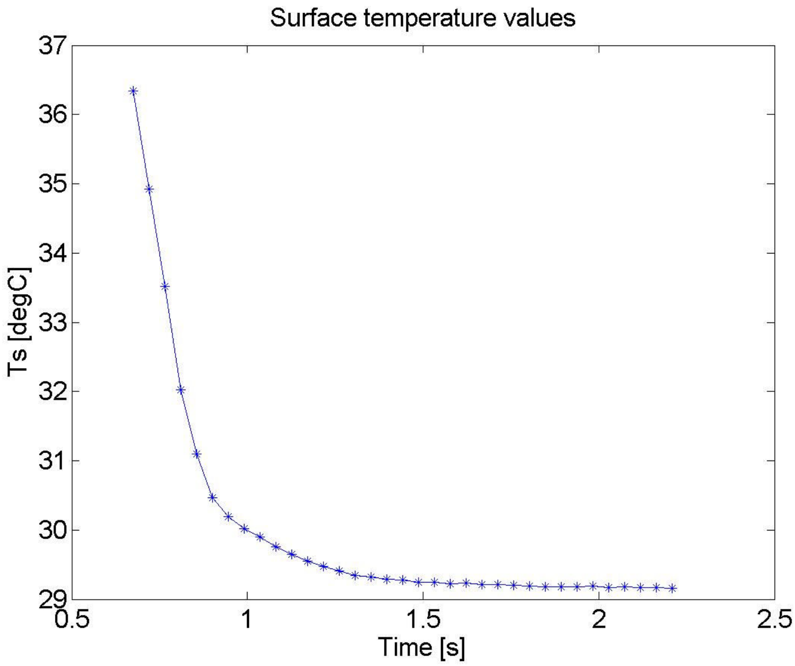

Fig. 6: Surface temperature for frame 16 to 50.

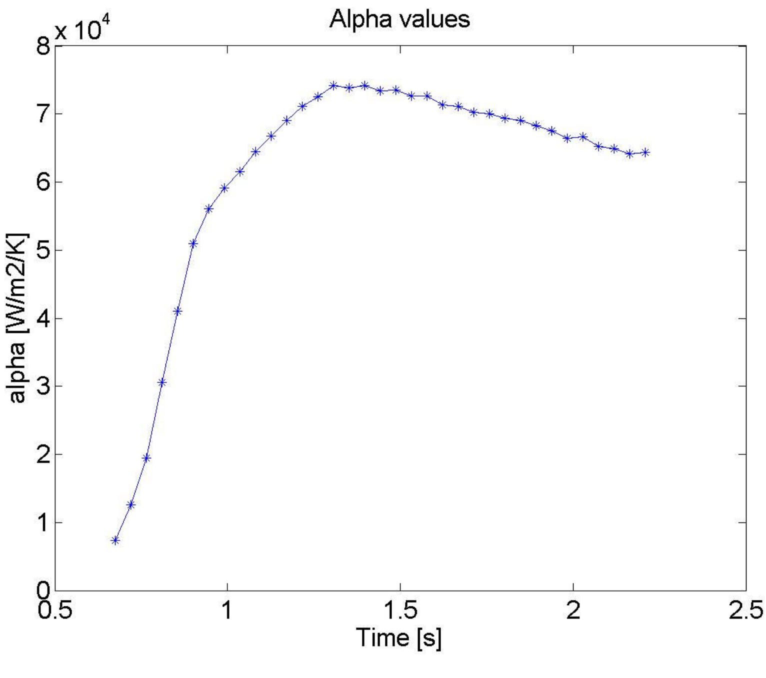

Fig. 7: Convective heat transfer coefficient as a function of time.

The time dependence of the heat transfer coefficient shown in Fig. 7 comes as a surprise. The expected result would have been a constant value since the flow and the turbulence inside the observation area remain the same over the course of the experiment. α also should not depend on the temperature. A possible source of error is the initial condition (9) used to solve for α. It assumes a steep temperature step from $T_i = 39\,°C$ before the experiment to $T_s = 27.6\,°C$ as soon as the valve is switched. In other cases it does not deliver a mathematical reasonable solution. The reason for a gradual rather than an abrupt temperature change are the mixing of the cold water with the hot water already in the loop as soon as the valve is switched. From the valve to the ducting cross section is a tube of around

50 cm in which the water mixes freely and a fluid with a temperature gradient rather than an abrupt temperature change arrives at the observation area. A possible solution to overcome this problem would be to shorten the tube or to introduce a secondary water loop altogether for the cold water injection into the test section.

The time variation of the Nu number depicted in Fig. 8 comes as no surprise because it is proportional to α according to formula (4).

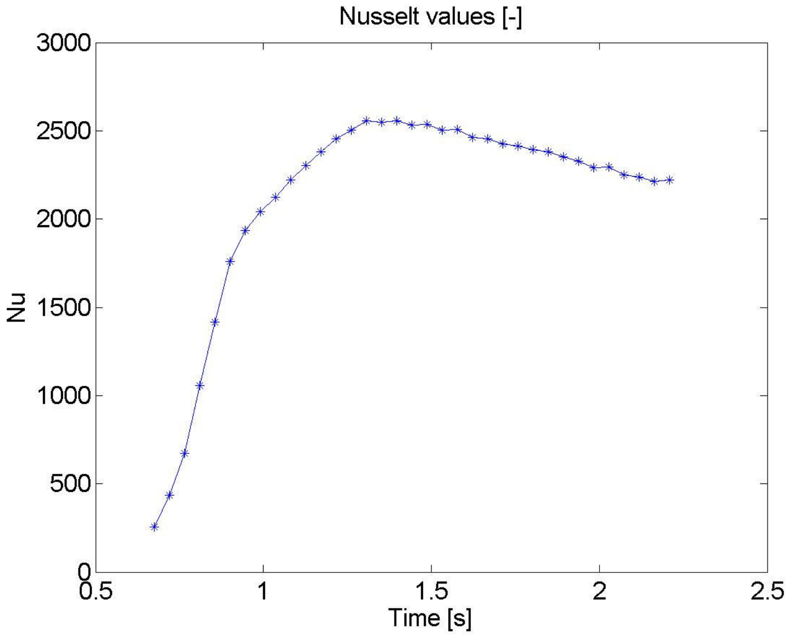

Fig. 8: Nu number over time for frame 16 to 50.

The Nu ratio in Fig. 9 is the ratio of the calculated Nu number over the Nu number expected by the Dittus-Boelter equation for the given Re and Pr numbers. The high ratio is due to the local turbulence promoters before and after the observation area.

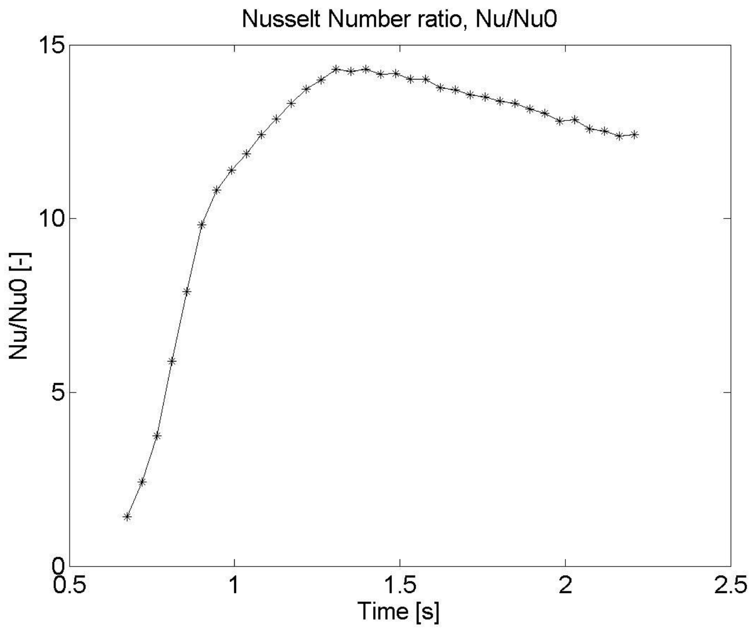

Fig. 9: Nu ratio over time from frame 16 to 50.

The experiment has some systematic problems. One is the synchronization of the CCD acquisition system and the valve which is operated manually. Such a poor synchronization poses problems because it leads to a very low repeatability quality. The semi infinite assumption in the given configuration falls apart already after $t_{max} = 1.5\ s$. This short period of time creates problems in our experiments as well because we chose an experimental time of 4.5 s. A possible solution would be to install a thermocouple in the back of the channel on the aluminum plate and directly measure T_{∞}. Another approach is to heat or insulate the back to reduce natural convection and ensure the semi-infinite assumption.

Another inherent problem of this experiment is the low amount of data. With just one experiment and the problems of exact repeatability statistical error analysis is not possible and it is therefore not possible to define error margins on the results either.

5. Conclusion

In conclusion the TLC measurement of temperature could deliver more accurate results if it was not damaged or if the calibration would be redone. The measurement of the heat transfer coefficient should be taken with care due to all the systematic problems noted above and due to faults in the TLC. Nonetheless it provides a valuable result in the sense that it delivers the order of magnitude of the heat transfer coefficient in the observation area.

The experiment has led to valuable insight into heat transfer problems and exemplified the need for a very careful experimental setup in order to obtain sensible results.

6. References

[1] Lecture on ‘Experimental Methods for Engineers’, Heat Transfer, http://www.ifd.mavt.ethz.ch/education/Lectures/exp_methods/Handouts/Handouts2010/VL_HTC_2010, Dec. 2010.

[2] Lab Notes, Heat Transfer, http://www.ifd.mavt.ethz.ch/education/Lectures/exp_methods/Handouts/Handouts2010/Lab_HTC_2010, Dec. 2010.