Infrared visualisation of flow within inclined lo uvered fins

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Abstract

Modern compact heat exchangers used in e.g. air conditioning have complex fin types to increase the heat transfer rate. Two commonly corrugated designs are the offset strip fin and the louvered fin. In this study a hybrid design, the inclined louvered fin, is studied. It is aimed at increased performance at low Reynolds numbers. To determine the local heat transfer coefficients, a scaled fin model is tested in an open wind tunnel and the local surface temperature is measured using thermocouples. To determine the air side temperature profile, a novel measurement method using a paper sheet and IR-thermography was proposed.

1. Introduction

The main thermal resistance in a heat exchanger transferring heat to air is located on the air side. To reduce the heat transfer resistance, fins are added to the airside. These fins increase the available exchanger surface and break up the boundary layers. Common designs include louvered fins, offset strip fins… These corrugated fin types increase the heat transfer rate when compared to plate fins, but also cause a large increase in pressure drop. Air conditioning manufacturers seek out new fin types to further enhance the performance of their evaporators/condensers and thus increase overall unit efficiency. The 'inclined louvered fin' (figure 1) is a hybrid design between a standard louvered fins and an offset strip fin (the flow through both these fin types are shown in figure 2) t. It is aimed at improved performance (less pressure drop for a given heat transfer rate) at low Reynolds numbers.

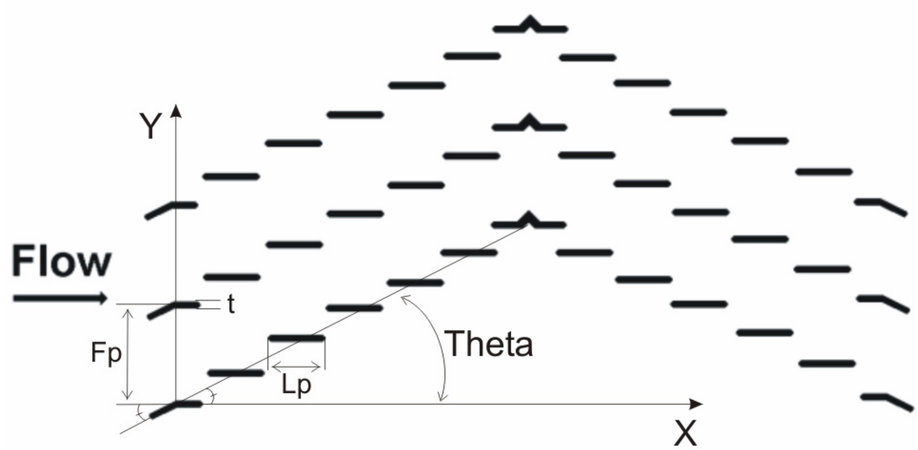

Fig. 1. *Inclined louvered fin: geometric parameters.*

In standard louvered fins the deflection is caused by the louvers: these are set at an angle against the main flow; for inclined louvered fins the louvers are parallel to the flow, but these are set in a staggered layout, creating a deflecting channel. Standard louvered fins show two different flow profiles, as can be read in Cowell et al. [1]. For low Reynolds numbers the thick boundary layers block the passage between the louvers, forcing the flow between the different fins. This is referred to as 'duct oriented flow'. As the Reynolds number increases the flow passages open up and 'louvered oriented flow' is created. The flow is deflected at high Reynolds numbers; extending its flow path throughout the fin array (the top side of figure 2 gives an impression of the modified flow path). This increases the heat transfer rate. But as the flow path is extended so is the frictional pressure drop.

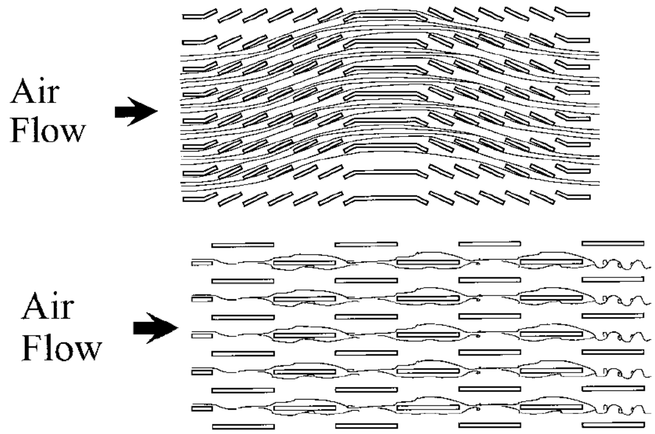

Fig. 2. *Flow through the standard louvered fin (top) and offset strip fin (bottom).*

Though no such study has been done on inclined louvered fins, it can be expected from the geometry that a similar phenomenon will occur. But the deflection would now be present at low Reynolds numbers, and the blocked louver passages would force the flow to wind up and down. This should be an improvement when compared to duct oriented flow in plain louvered passages. For applications such as air conditioning, which operate in the lower velocity range, this could have significant impact. At high Reynolds numbers the behavior of inclined louvered fins should be comparable to offset strip fins, which would result in a reduced pressure drop when compared to standard louvered fins.

Another important aspect of the corrugated fin designs is that these fin types promote unsteady flow. Starting from a certain Reynolds number, the flow throughout the fin will become unsteady, enhancing the mixing of the bulk stream and thus enhancing the heat transfer rate. By using flow visualization DeJong and Jacobi [2-3] were able to show that both in louvered fins and offset strip fins the transition to unsteady flow occurs initially downstream of the exit louver and then gradually moves upstream into the fin array, as the Reynolds number is increased. The recorded mass transfer data showed that vortex shedding results in an increased local transfer coefficient. This transitional behaviour can be explained if the various louvers are considered as individual roughness elements interacting with a fluid stream, as shown by Zhang et al. [5]. Each louver perturbs the flow and as a fluid element passes over the fin, the cumulative effect of these perturbations causes the flow to develop instabilities.

To determine the thermo-hydraulic characteristics of inclined louvered fins a measurement setup was constructed at the Ghent University. In order to construct a correlation, the geometric parameters must be varied over certain ranges. The inclined louvered fin design consists of 6 geometric parameters: fin thickness, fin angle, louver angle (set to 0°), fin pitch, louver length and the number of louvers. This can be seen in figure. 1. Two more parameters appear when considering a heat exchanger in operation: the heat flux and the velocity of the flow. The Reynolds number is determined using the louver pitch (Eq. (1)). All substance properties are determined at the average temperature:

$$\mathrm{Re} = \frac{\rho \cdot U \cdot L_p}{\mu} \qquad (1)$$

So in total an initial set of 7 variables has to be studied. Traditionally one would vary a single parameter while all others remain constant to study the impact of this parameter. This method would require a very large number of experiments and can provide misleading information as shown by Schmidt and Launsby [5]. A numerical screening experiment was set up. The aim of this study was to determine which parameters have a strong impact on the thermo-hydraulic behaviour of this fin type. These will then be studied in more detail. It was found that the fin pitch, the fin angle and louver thickness had the largest impact (T'Joen et al. [6]).

2. Experimental setup

Two approaches exist for determining the thermo-hydraulic characteristics of a new fin design. The most used method is the Wilson-plot method as described by Rose [7]. This requires a complete heat exchanger to be manufactured with the new fin design. It provides accurate data on actual heat exchanger performance. This method however is expensive considering the complex fin shapes that have to be machined in aluminum when doing a parameter study for a fin design. This method also provides insufficient information for validation of CFD simulations. Therefore another approach was chosen. A scale model of the fin design is used. Similar studies have been performed on both slit fins (Dejong and Jacobi [2]) and louvered fins (Dejong and Jacobi [3] - Lyman et al. [8]).

The test rig is an open wind tunnel. The section measures 0.4 m by 0.2 m. The complete test rig (figure 3) consists of a large fan (3), driven by a frequency controlled drive (2+1). The flow then passes through a settling chamber filled with honeycomb (4) (1.10 m x 1.10 m). To generate a uniform flow in the test section (7) the settling chamber has two sinusoidal shaped walls (5). The actual test section is followed by a settling channel (8) of 5 m. This prevents any impact upstream of the mass flow rate measurement using a calibrated orifice plate (ISO 5167). The orifice plate has a diameter ratio of 0.62 and is set in a tube with an inner diameter of 18.29 mm. The pressure taps are set according to the standard. The Reynolds number in the current research ranges from 800 to 2000 (when expressed using the louver pitch as length scale). The lower boundary was imposed due to the nozzle dimensions.

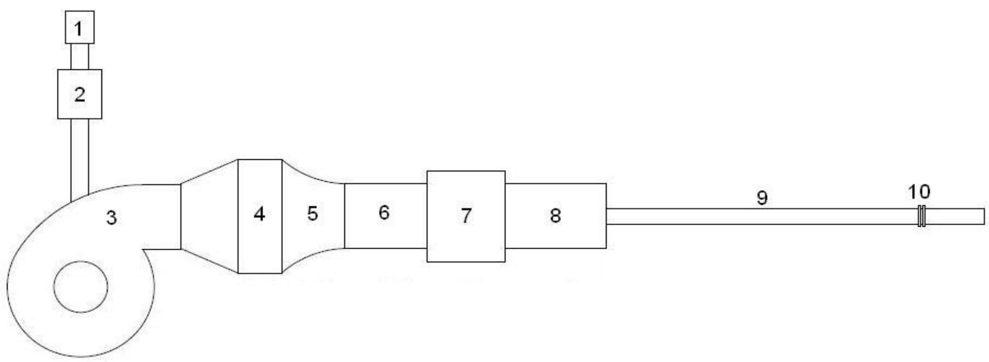

Fig. 3. *Test rig.*

The test section is made up out of a scaled model of the fin array. The fins are scaled up 20 times, resulting in a louver pitch length of 2 cm. This allows for sufficient spatial resolution when taking measurements. To maintain flow similitude, the air velocity must be scaled down by 20 as well. The walls of the test section are made out of balsa wood. The actual fin parts are made out of print board material coated with a thin copper layer on both sides. This material can be obtained at a required thickness (2 mm – maintaining flow similitude) and can be easily cut down to a given size. In the first series of experiments the fin thickness is kept constant to one tenth of the louver pitch.

To provide heating (a uniform heat flux) a current is sent through the copper layer. The current is controlled using an external supply. The maximum current sent through the louvers is 10 A. This restricts the heat flux that can be imposed in experiments to 187.5 W/m². The different louvers are connected to each other electrically using wires, thus ensuring the same heat flux is dissipated throughout the fin array. The top and bottom of each louver are connected using soldered copper contacts. The test section and the settling channel is insulated using 5 cm thick PUR foam in order to reduce the heat loss to the environment.

The air inlet temperature was measured using a thermocouple. To determine the heat transfer coefficient the local surface temperatures must be measured (Eq. (2)). A measurement louver was made using balsa wood. Nineteen K-type thermocouples where inserted in the centre along the flow direction in order to measure the surface temperature change along the flow path. The detailed construction of the measurement louver can be seen in figure 4. The tips of the thermocouples were covered in thermal conductive paste to ensure good thermal contact. The balsa wood was covered with a thin sheet of paper and then a copper foil was placed on top to allow for heating of the louver. As the resistance of the copper foil on the measurement louver and the copper layer on top of the louvers differed, a control circuit was used to ensure that both circuits dissipated the same heat flux.

$$h = \frac{q}{T_w - T_{ref}} \qquad (2)$$

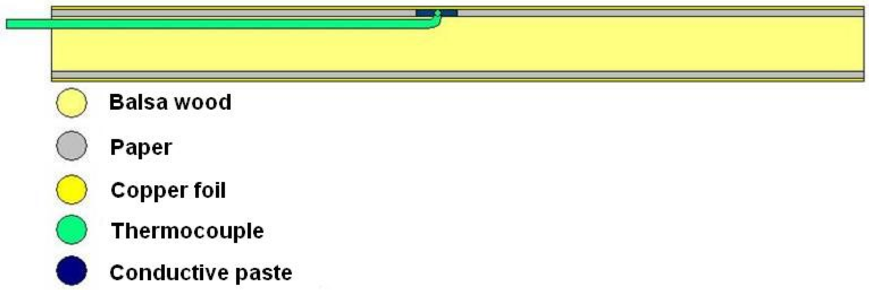

Fig. 4. *Measurement louver construction detail*.

Previous studies on louvered fins by Zhang and Tafti [9] and Lyman [8] showed that each fin row is affected by the row above and below it. Fluid may be heated up by the nearby rows and then be deflected towards the row where the measurements are done. Therefore, to ensure accurate measurements, both the studied fin row and the two adjacent rows above and below the studied fin row are heated. When performing measurements on a scale model the impact of nearby walls can disturb the measurements. This has lead in the past to strong differences between measured values for identical configurations, as described by DeJong and Jacobi [10]. To prevent this, a 2D CFD simulation was performed to determine how many rows of fins were required to ensure that a zone in the centre of the array presented undisturbed flow. It was found that at least 10-15 louver rows were needed. Similar results were found by Springer and Thole [11] when performing a study of louvered fins.

To verify the uniform heat flux assumption a single louver was painted with high emissivity black paint and an infrared image was taken during heating. The resulting temperature image was found to be uniform (indicating a uniform heat flux) except around the wire contacts. As these zones lie outside of the channel and at 0.1 m of the thermocouples it has no effect on the measurements. The heat balance for a measurement was checked by measuring the local exit air temperature over the section (along the central axis in the vertical sense) combined with local velocity measurements at the same points. A single temperature measurement proved to be inaccurate due to strong thermal stratification (only a small zone receives heat). By integrating the measured local velocity and temperature profile a difference of less than 5% was found on the heat balance. The measured heat input (using the current) was used in the calculations. To increase the accuracy of the temperature measurements each thermocouple was calibrated separately. Radiation is assumed to have a negligible impact on the measurements, as was shown by Lyman [8].

All measurements were taken at steady state conditions. To verify that start-up or transient phenomena had ceased, the local surface temperature was monitored during the measurements. It was found that on average 10-15 minutes was required before steady state was reached.

3. Local heat transfer coefficients

In the previous paragraph the experimental setup was described in great detail. If one considers Eq. (2), it is clear that three variables must be know in order to determine the local heat transfer coefficient h: the heat flux q (imposed in the experiment), the surface temperature T_w (measured using the thermocouple) and the reference temperature T_{ref}. For free stream problems such as the flow over an airfoil, the choice of the reference temperature is simple; it is the temperature of the undisturbed fluid at a large distance T_∞. But for the flow through a fin array this value cannot be used. As the fluid passes through the fin array it heats up. The local surface temperature of the louvers increases as one moves further downstream. If the inlet temperature is used as a reference temperature then the temperature difference in the denominator of Eq. (2) increases as one is further downstream into the fin array. This will result in artificially lower heat transfer coefficients downstream in the fin array. This was clearly shown by e.g. Lyman et al. [8].

An alternative reference temperature would be the mean bulk temperature T_b (Eq. (3)) just upstream of every louver. This value can be determined by averaging out the total heat *Q* added by the upstream louvers over the fluid stream. When using the T_b as a reference temperature, the convective heat transfer coefficient becomes a strong function of the local fluid temperature. The impact of heated wakes from upstream louvers thus appears in the convective heat transfer coefficient data, as shown by Lyman [12].

$$T_b = T_{in} + \frac{Q}{\dot{m} \cdot c_p} \qquad (3)$$

4. Thermographical measurements

In order to have a clearer view of the local air temperature throughout the fin array a thermographical experiment was set up. IR-thermography was selected as a measurement method because it provides temperature data with a large spatial resolution (compared to point measurements using thermocouples) and it does not disturb the air flow (thermocouples would). To study the flow within the fin array a thin paper sheet (emissivity 0.94) was placed at the center of the wind tunnel (Fig. 5), and several IR windows (using mica foil, 80% IR-transparent) were made on one side of the channel. In order to ensure the correct placement of the louvers as well as maintain the stiffness of the side wall, only small windows could be used over the channel width. IR-images have been recorded for various geometric configurations (with different fin pitch and fin angle), at various Reynolds numbers (ranging from 800 to 2000) and for a range of heat fluxes.

4.1 Calibration IR-camera

A Midas infrared camera was used for the experiments. It is a long wave (LW) camera which has a spectral response from 8 µm to 14 µm. In order to do accurate measurements the emissivity ε of the paper and the transmittance τ the mica foil window must be known. The emissivity of paper vas found in open literature to be 0.94. To calculate the transmittance of the mica foil, the fin is heated until it reaches steady state. Two pictures are taken, one with the mica foil attached and one without the foil. The factor with which the temperature decreases is a measure for the transmittance of the mica foil. For the pictures with the mica foil, the transmittance value is adapted, until its temperature equals the temperature of the correspondent infrared picture without mica foil. Several tests gave a transmittance of 0.82

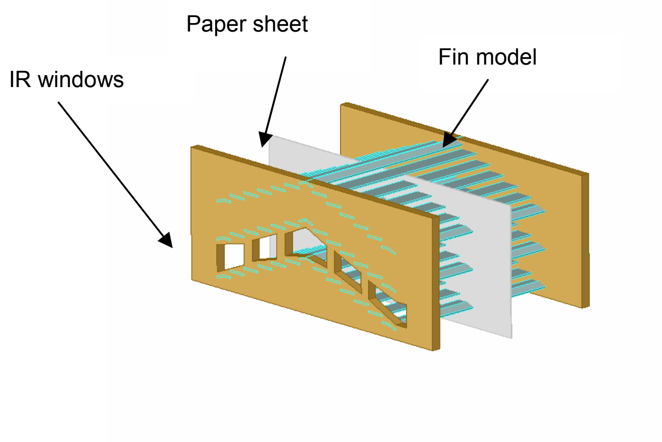

Fig. 5. *Thermographical setup.*

4.2 Results

IR-images have been recorded for various geometric configurations (with different fin pitch and fin angle), at various Reynolds numbers (ranging from 800 to 2000) and for a range of heat fluxes. These images can be compared to numerical simulations (CFD) of the flow through the fin array. These simulations were performed using Fluent©. The settings used to perform these simulations are shown in Table 1. The results for 4 measurements on a single fin design will be presented below. The Reynolds number and heat flux were varied between 800-2000 and 80-180 W/m² respectively. The numerical simulations for Reynolds = 800 converged stationary, while the higher Reynolds number of 2000 required unsteady simulations to converge. A timestep of 0.001s was used for the unsteady simulations.

Table 1. *Fluent settings*

Solver	Segregated
Air properties	Constant (density: 1.225 kg/m³, viscosity: 1.789 10^{-5} Pas)
Pressure-velocity coupling	SIMPLE
Discretization	Second order
Residuals (x and y velocity – continuity)	10^{-7}
Boundary conditions	Periodic top – bottom / velocity inlet / pressure outlet
Energy equation	disabled

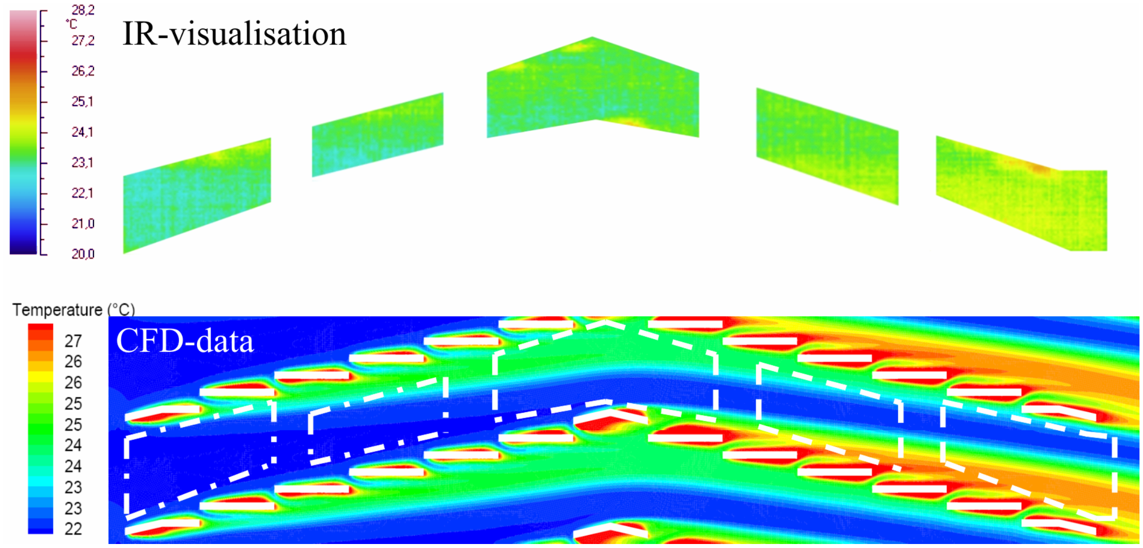

Fig. 6. *Comparison of the IR-measurement (top) and CFD prediction (bottom) of the air temperature – Re: 800, q: 80 W/m².*

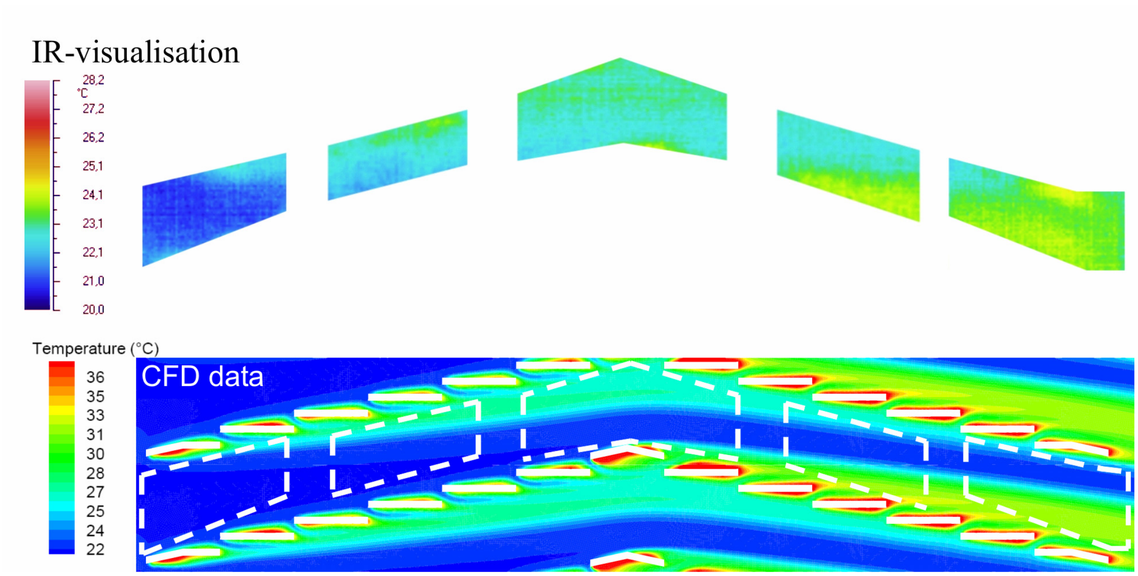

Fig. 7. *Comparison of the IR-measurement (top) and CFD prediction (bottom) of the air temperature – Re: 800, q: 180 W/m².*

When comparing the IR-data to the CFD data in figures 6-7-8-9, it is clear that good agreement can be seen for the laminar cases (figures 6-7). Do note that the temperature scale of the IR-image and the CFD data is not the same. The temperature scale of the CFD data had to be set larger in order to see the complete flow image clearly. Otherwise large blank spaces would distort the view. The temperature scale of the IR-image was set to provide the highest image contrast. The long cold wake which flows through the array is clearly visible in the IR data on figure 7. Local hot wakes behind louvers can be seen clearly in figure 6. The IR-images of figure 8 and 9 show a much more 'spotted' temperature profile. As the flow has become unsteady (CFD simulation) vortices are being shed by the louvers. By using the IR-camera one records a snapshot of the flow (a single moment in time). The local hot spots on the IR-images thus represent hot pockets of air, vortex cores which are transported downstream. If one considers the temperature difference of the IR data in figure 8 it is clear that the temperature resolution is insufficient for a clear view of the flow profile.

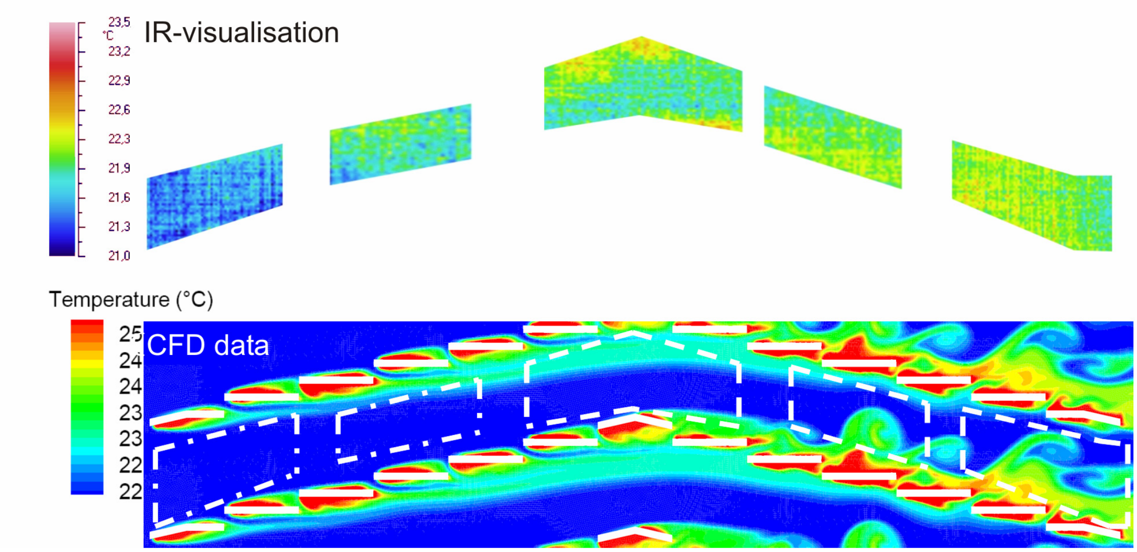

Fig. 8. *Comparison of the IR-measurement (top) and CFD prediction (bottom) of the air temperature – Re: 2000, q: 80 W/m².*

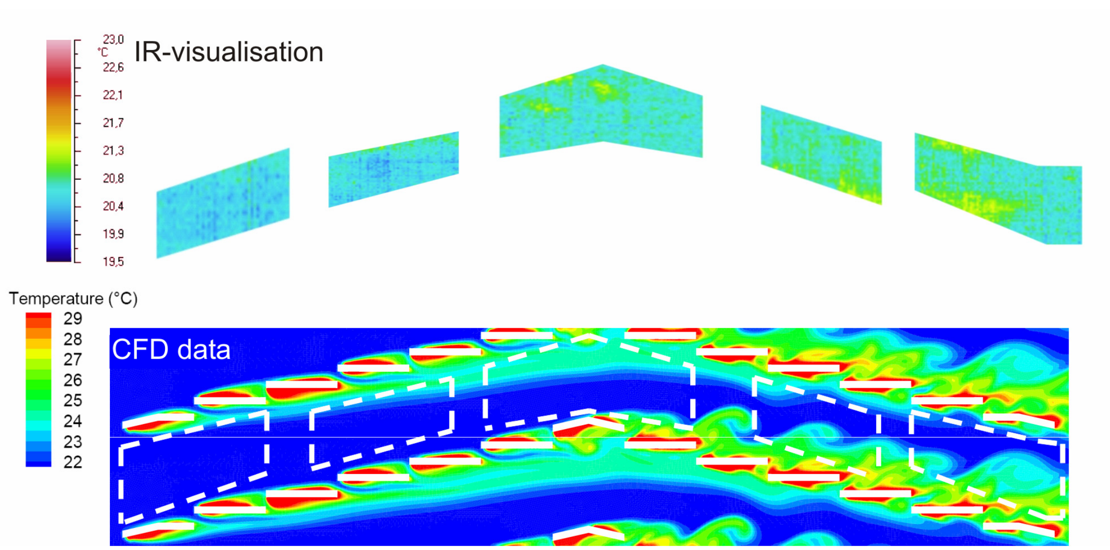

Fig. 9. *Comparison of the IR-measurement (top) and CFD prediction (bottom) of the air temperature – Re: 2000, q: 180 W/m².*

5. Conclusions

The proposed measurement technique for the air temperature throughout a complex fin array has been compared to CFD simulations. It is found that this technique provides a qualitative image of the flow pattern through the fin array and shows several distinct flow features: local hot wakes just downstream of louver, hot recirculation zones behind the turnaround louver… The images can also be used to distinguish the type of flow: the IR-images of figure 6 and 7 clearly show a steady laminar flow, while figure 8 and 9 present more erratic patterns. This is due to the unsteady nature of the flow: hot vortex cores are transported downstream. In order to have a good resolution, there must be sufficient temperature difference between the inlet and outlet.

6. Nomenclature

c_p	specific heat capacity, J/kgK
h	convective heat transfer coefficient, W/m²K
q	heat flux, W/m²
L_p	louver pitch, m

$\dot{m}$ mass flow rate, kg/s
Q heat transfer rate, W
Re Reynolds number, -
T temperature, K
U mean inlet velocity, m/s

Greek symbols
ε emissivity of an object
μ dynamic viscosity, Pas
ρ density, kg/m³

subscripts
b bulk fluid property
in value at the inlet of the fin array
ref reference value
w value at the wall

7. Acknowledgements

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