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THERMO-HYDRAULIC CHARACTERISTICS OF INCLINED LOUVERED FINS

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Abstract

Inclined louvered fins are a novel hybrid fin design aimed at improved performance at low Reynolds numbers for fin-and-tube heat exchangers. A numerical screening experiment is performed to determine which parameters have a strong impact on the thermo-hydraulic behaviour. A test rig was built to determine the thermo-hydraulic behaviour of this fin type. Scaled models are tested in a low speed wind tunnel. The impact of the fin pitch, louver angle and Reynolds number are studied in detail.

INTRODUCTION

When exchanging heat with air (e.g. air conditioning, central heating…), the main resistance to the heat transfer is located on the airside (over 85 %). So, in order to improve the performance of the heat exchanger fins are added on that side. These serve several purposes: they increase the available surface for heat transfer and prevent the forming of thick boundary layers throughout the exchanger. The most simple type, the plate fin, has been replaced by more complex geometries. Amongst the most popular are louvered, (super)-slit, wavy… Louvered fins are the most used type for automotive applications.

New fin designs are implemented continuously, however no data is available on their performance (proprietary information). Past experience has shown that even small changes to a fin design can have a strong impact on the performance. Therefore new fin designs require extensive testing before they can be introduced. Numerical simulation (CFD) should be able to predict the heat transfer and pressure drop performance of a given fin design, but still requires validation. The goal of this research is to study a novel and yet unpublished fin type, the inclined louvered fin. The research is done both experimentally and numerically. The research is set up in such a way as to gather data that can be used to validate the simulations.

INCLINED LOUVERED FINS

A basic representation of inclined louvered fins is shown in Fig. 1. A single short horizontal plate is called a louver, similar as in louvered fins. In the figure the characteristic parameters that define the fin type are indicated. These are the fin pitch (F_p), the louver pitch (L_p), the fin angle theta, the fin thickness t and the number of louvers N. Tanaka et al. (1984) studied inclined louvered fins and considered another parameter: the louver angle. This is set at 0° in the subsequent study.

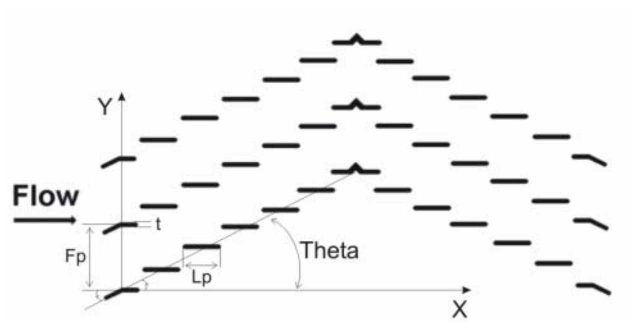

Fig. 1. Inclined louvered fin: parameters.

Inclined louvered fins are a novel fin design. It combines features of both the standard louvered fin (Fig. 2) and the offset strip fin or slit fin (Fig. 3), as described by Shah et al. (2001). Both inclined louvered and standard louvered deflect the main flow. In standard louvered fin 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.

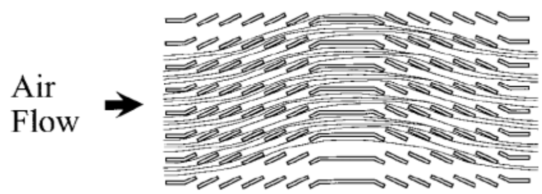

Fig. 2. Flow through standard louvered fins.

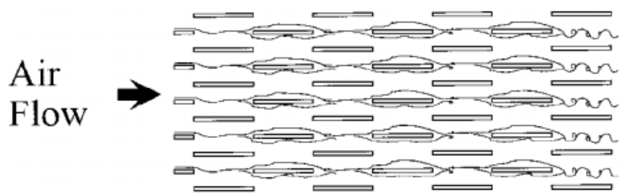

Fig. 3. Flow through offset strip fins - slit fins.

Standard louvered fins show two different flow profiles, as can be read in Cowell et al. (1995). 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. This is indicated in Fig. 4. As is shown, the flow is deflected at high Reynolds numbers, extending its flow path throughout the fin array (Fig. 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. This is usually expressed using the concepts 'flow efficiency' or 'flow angle'. For high Reynolds numbers the flow angle approaches the louver angle.

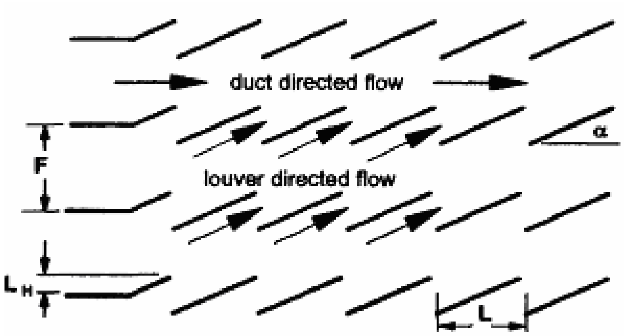

Fig. 4. Duct directed flow versus louver directed flow.

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, as 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 louver passages. For applications such as air conditioning, which operate in the lower velocity range, this could have significant impact. At high Reynolds numbers the behaviour of inclined louvered fins should be comparable to slit fins, which would result in a reduced pressure drop when compared to standard louvered fins.

NUMERICAL SCREENING

Parameter study and range

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 clearly seen in Fig. 1. By choosing a reference length non-dimensional values can be introduced. A common value used to determine Reynolds numbers of standard louvered fins is the louver pitch. Thus two non-dimensional values (indicated by *) can be introduced as parameters:

$$t^* = \frac{t}{L_p} \tag{1}$$

$$F_p^* = \frac{F_p}{L_p} \tag{2}$$

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. (3)). All substance properties are determined at the average temperature:

$$\mathrm{Re} = \frac{\rho \cdot U \cdot L_p}{\mu} \tag{3}$$

So in total an initial set of 7 variables has to be studied. By choosing a reference length and by defining non dimensionless quantities (Eq. (1) - (2)) this number is reduced to 6. 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 (2005). A screening experiment is set up studying six parameters: Reynolds number, heat flux, fin angle, number of louvers, t* and Fp*. The aim of this study is 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.

To limit the number of experiments a L12 Taguchi matrix is used, as described by Schmidt and Launsby (2005). To select an appropriate parameter range published experimental data for standard louvered fins was studied. Limitations imposed by the designed test rig were taken into consideration as well when considering the Reynolds number (fan power) and the heat flux. The resulting parameter range for the screening can be seen in Table 1.

Table 1. Screening parameter range

Parameter	Low Value	High Value
q	50	150
Re	200	2000
θ	12.64	30.96
N	5	7
t^*	0.1	0.15
F_p^*	1.125	2

Numerical simulation

A two level screening experiment is conducted numerically. Heat transfer performance prediction by CFD ('Computational Fluid Dynamics') is of major interest. The ability of a CFD-code to predict the flow pattern and thermal field allows determining the heat transfer characteristics by performing 'numerical experiments'. These still require extensive validation, but as shown by e.g. Perrotin and Clodic (2004) they can give clear qualitative indications. The screening experiment is aimed at providing qualitative information on the impact of different parameters on the thermo-hydraulic characteristics of the inclined louvered fin.

The computations were performed using a commercial code: Fluent©. Two dimensional cases were studied using quadrilateral meshes generated by Gambit©. A single louver row was simulated with an entry region (two fin pitches) and exit region (five fin pitches). The height of the computational domain is one fin pitch. Periodic boundary conditions on the top and bottom of the domain ensure this single element behaves as if placed in a stack of fins. The walls have a constant heat flux boundary condition.

For the low Reynolds cases the flow was found to be laminar. A grid independency study was performed on several cases. The grid size was varied from one-quarter fin thickness to one-sixteenth fin thickness. This resulted in meshes varying from 8000 cells to over one million cells. The average difference between the pressure drop determined when using a 'fine mesh' (cell size of one-tenth fin thickness) and a 'control mesh' (cell size of one-sixteenth fin thickness) was less than 0.5%; for the heat transfer rate this difference was less than 1%. The data used in the screening study was generated using either a 'control mesh' (cases used to study grid independency) or a 'fine mesh'. For the high Reynolds cases a turbulence model was used. The k-ω model was selected for this type of flow. The flow at inlet was assumed to be fully developed. For the high Reynolds number cases the grid was refined near the louvers using a boundary layer mesh.

A double precision solver was used for all simulations. The convergence criteria were set to 1e-8 for continuity and velocity residuals, 1e-6 for energy residual. The pressure drop and heat transfer rate were monitored to evaluate convergence. The pressure velocity coupling was done using the SIMPLE algorithm. Second order discretization schemes were used for all variables. Constant substance properties are used for air, the density was calculated using the ideal gas approximation and the viscosity using the Sutherland equation.

To evaluate the thermo-hydraulic behaviour of a fin type the friction factor and overall heat transfer coefficient are used. The local convective heat transfer coefficient h is defined using the heat flux q, the local wall temperature and the inlet temperature of the flow through Eq. (4). Moffat (1998) showed that the adiabatic wall temperature is a more suited reference value, but due to long computation times this value was not used in the numerical screening it was used however in the experimental study. The local values are then averaged out over a louver (Eq. (5)) and then over the different louvers (Eq. (6)). The friction factor is defined using Eq. (7), as done by Atkinson (1998).

$$h = \frac{q}{T_w - T_{in}} \tag{4}$$

$$\bar{h}_{L_p} = \frac{1}{L_p}\int_0^{L_p} h(x)dx \tag{5}$$

$$\bar{h} = \frac{1}{n}\sum_1^n \bar{h}_{L_p} \tag{6}$$

$$f = \frac{\Delta p}{\frac{\rho \cdot U^2}{2} \cdot \frac{A}{A_c}} \tag{7}$$

The results of the simulations are presented in Table 2. Part of the Taguchi L12 matrix is shown as well using encoded values: 1 stands for a parameter set to a high value, -1 stands for a parameter set to a low value. By averaging out over the high and low values of a parameter Fig. 5 and 6 are found for respectively h (Eq. (6)) and f (Eq. (7)). As expected the Reynolds number has a great impact on both parameters: a strong rise in the heat transfer coefficient and a strong drop in the friction factor. Of the geometric parameters the fin angle has the strongest impact on the heat transfer coefficient. An increase in the fin angle increases the heat transfer area of the inlet, outlet and turnaround louver. As these areas have thin boundary layers this will result in a small rise of the average heat transfer coefficient h. This increased surface will cause a rise in the pressure drop as the flow incurs more form drag. Thick louvers will create more pressure drop as they have a large frontal area, and the larger wake and separation zones will reduce the heat transfer coefficient. A large fin pitch will result in a lower friction factor as the fin structure is more open.

Table 2. L12 matrix and screening results.

Re	F_p^*	t^*	θ	N	q	f	h
-1	-1	-1	-1	-1	-1	0.161	15.42
-1	-1	-1	-1	-1	1	0.167	15.32
-1	-1	1	1	1	-1	0.253	15.26
-1	1	-1	1	1	-1	0.216	19.49
-1	1	1	-1	1	1	0.087	8.393
-1	1	1	1	-1	1	0.252	20.88
1	-1	1	1	-1	-1	0.177	65.12
1	-1	1	-1	1	1	0.063	52.43
1	-1	-1	1	1	1	0.101	62.72
1	1	1	-1	-1	-1	0.062	43.73
1	1	-1	1	-1	1	0.086	74.33
1	1	-1	-1	1	-1	0.067	59.86

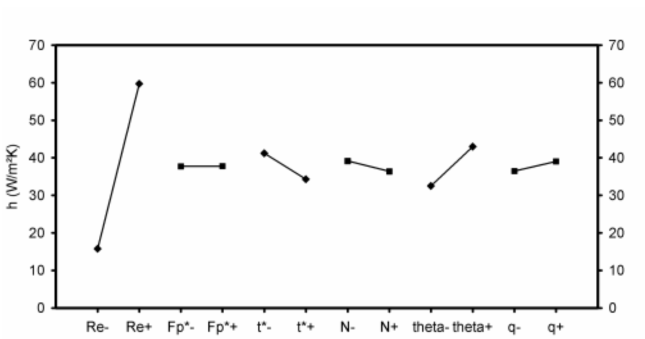

Fig. 5. Average effect plot for the heat transfer coefficient h.

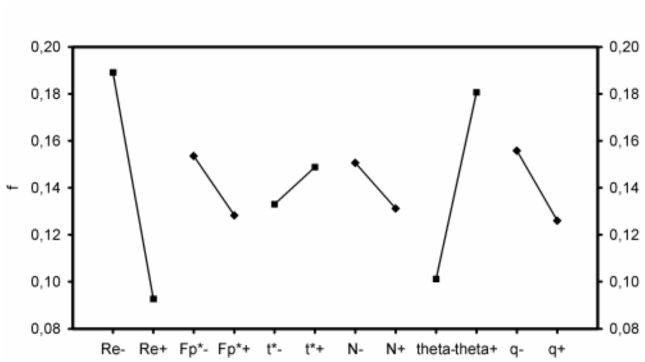

Fig. 6. Average effect plot for the friction factor f.

The numerical screening indicated that the most important parameters to be studied are the Reynolds number, the fin angle, the fin pitch and fin thickness. These are studied in more detail experimentally.

EXPERIMENTAL PROCEDURE

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 (2004). 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 et al. (1997)) and louvered fins (Lyman et al. (2002)). Some researchers use the analogy between heat and mass transfer to determine local heat transfer coefficients, e.g. Dejong et al. (1997).

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.

Test rig

The test rig is an open wind tunnel. The section measures 0.4 m by 0.2 m. The complete test rig (Fig. 7) consists of a large fan (3), driven by a frequency controlled motor (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 1 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.

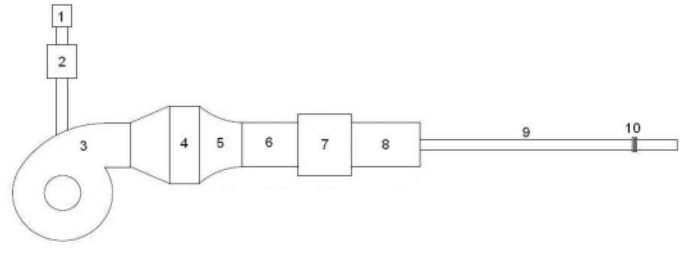

Fig. 7.Test rig.

The test section is made up out of a scaled model of the fin array. The fins are scaled up 20 times. 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 pressure drop over the test section is measured using a differential pressure transducer. These measurements were done without heating. The pressure taps were set along the sides of the channel, as the flow is affected by wall effects on the top and the bottom of the channel. The average value of the pressure drop was used in calculations.

The air inlet temperature was measured using a thermocouple. To determine the heat transfer coefficient the local surface temperatures must be measured. This can be done using an IR camera, provided there is an optical access (IR transparent material). Due to the compactness of the scaled fin model it was found this could not be realized easily. Therefore the surface temperatures are measured using K-type thermocouples. A measurement louver was made using balsa wood. Nineteen thermocouples where inserted in the center 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 Fig. 8. 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.

Previous studies on louvered fins by Zhang and Tafti (2001) and Lyman (2000) 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.

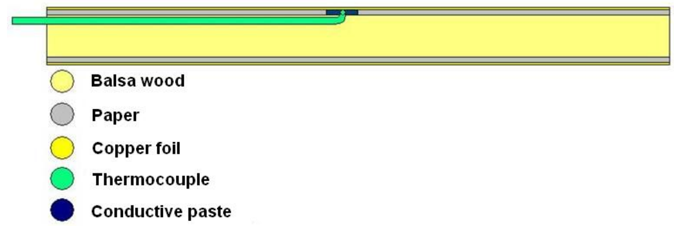

Fig. 8. Measurement louver construction detail.

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 Kim et al. (2003). 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 center of the array presented undisturbed flow. It was found that at least 10-15 louver rows were needed. Similar results by Springer et al. (1998) 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 a 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 neglible impact on the measurements, as was shown by Lyman (2000).

To reduce the number of experiments required to determine a model for inclined louvered fins, the 'Box Behnken' design matrix was used to study 4 parameters (Reynolds number, F_p^*, theta and heat flux) at three different levels (Schmidt and Launsby - 2005). The parameter range is shown in Table 3.

Table 3. Parameter range for the experimental study.

	Low value	Center value	High value
F_p^*	1.125	1.5	2
θ	12.64°	19.78°	30.64°
Re	800	1265	2000
q	80	122.4	187.5

Data reduction

In the experiments the local heat flux is imposed, and the local surface temperature is measured. Thus only a reference temperature is required to determine the local heat transfer coefficient. In the numerical screening experiment the inlet temperature of the fin array was used. However a more accurate reference temperature can be used: the adiabatic wall temperature, as introduced by Moffat (1998). This temperature is measured by performing a second experiment in which the wall temperatures of a louver are measured while all louvers dissipate heat except the measurement louver. The temperature then measured is a better indication of the local driving temperature difference. The convective heat transfer coefficient based on the adiabatic wall temperature, h_{ad} is defined in Eq. (8). Using the adiabatic temperature as a reference temperature removes the effects of the local thermal field resulting from the upstream louvers. This value is then averaged out over a louver and then over the different louvers as in Eq. (5) - (6).

$$h_{ad} = \frac{q}{T_w - T_{ad}} \tag{8}$$

During the experiments it was found that a temperature difference was present between the top and the bottom of the measurement louver, due to the finite conduction of this louver. By measuring the thermal conductivity of the balsa wood (0.045 W/mK) the conductive flux could be determined. This flux through the louver is then added to or substracted from the imposed heat flux boundary when doing the data reduction.

To calculate the friction factor (*f*), the general expression by Kays & London (1984) was used. As the pressure drop measurements were performed without heating the well known equation reduces to Eq. (9).

$$f = \frac{A_c}{A}\left[\frac{2 \cdot \Delta p \cdot \rho_1}{G_c^2}\right] \tag{9}$$

RESULTS AND DISCUSSION

Friction factor

The results for three configurations are shown in Fig 9. An error analysis was performed on these measurements and the resulting error is shown as well. It is hard to compare this data to other relevant fin types as the pressure drop shown here is that only caused by the fin array itself. Published correlations (e.g. for louvered fins by Wang et al. - 1999) use data obtained from full scale heat exchangers. This data combines the pressure drop caused by the tube bundle and the fin array. As is expected the error increases for lower Reynolds numbers due to the stronger impact of the error on the mass flow rate. When comparing the two top curves it is clear that an increase of the fin angle increases the pressure drop. This is consistent with previous studies in inclined louvered fins (Tanaka et al. (1984) found that for a constant louver angle the friction factor increased with the fin angle) and in louvered fins (Dejong and Jabobi - 2003). It is also consistent with the screening experiment (Fig. 6). Figure 9 also shows that a large fin pitch reduces the friction factor, mainly at low Reynolds numbers. The impact of this change is, as shown by the screening experiment, less pronounced than a change in the fin angle.

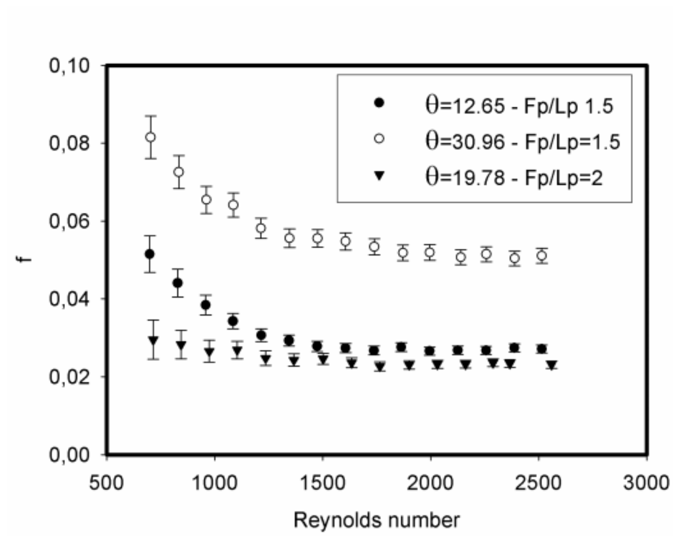

Fig. 9. Measured friction factors.

Heat transfer coefficient

During the data reduction it was found that the heat flux had a neglible impact on the average heat transfer coefficient when determined using the adiabatic wall temperature. When using the inlet temperature or the bulk temperature (as has been used in the past by some researchers) a strong impact was found. Therefore to avoid confounding of the data the results presented below are adiabatic values. Using the adiabatic wall temperature as a reference eliminates the impact of the thermal field on the measurements. The results then only show the impact of the flow field. As the Richardson number is very small (<0.01) forced convection occurs, so the heat flux has no impact on the flow field.

Figure 10 shows the impact of the Reynolds number, the fin pitch and the fin angle on the adiabatic Nusselt number (Eq. (10)) for three different configurations. The Nusselt number increases as the Reynolds number increases. Considering the curves for the same F_p/L_p ratio of 1.5 and different fin angles for varying Reynolds numbers. These three curves have the same slope, this can be expressed by $Nu_{ad} \approx \vartheta^{0.22}$. If the same data out against the Reynolds number (Fig. 11) two curves are found (same fin pitch but different fin angle). These curves have the same slope and can be expressed as $Nu_{ad} \approx \mathrm{Re}^{0.58}$. These findings confirm the results of the numerical screening (Fig. 5). It can be seen that the impact of the Reynolds number is much greater than the impact of the fin angle. The impact of the fin pitch can be seen in Fig. 11 as well. The Nusselt number for the configuration F_p/L_p ratio of 2 and the fin angle of 19.78° nearly coincides with the data for F_p/L_p ratio of 1.5 and the fin angle of 12.64°. So, as predicted by the screening experiment the impact of the fin pitch is neglible.

$$Nu_{ad} = \frac{\overline{h}_{ad} \cdot L_p}{\lambda} \tag{10}$$

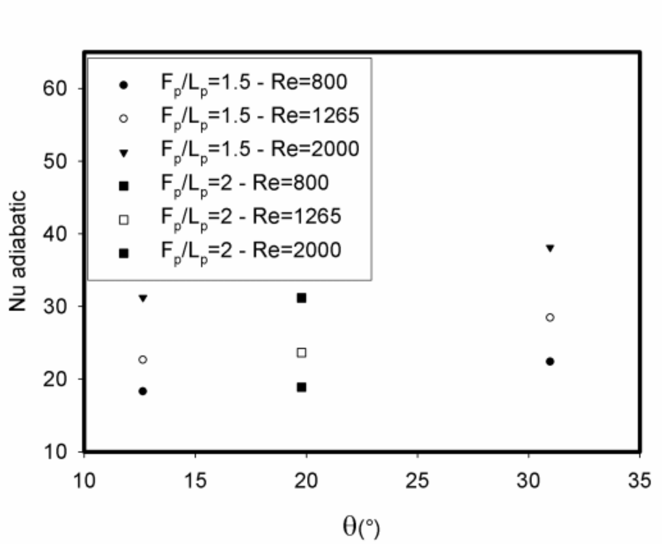

Fig. 10. Nusselt number vs fin angle.

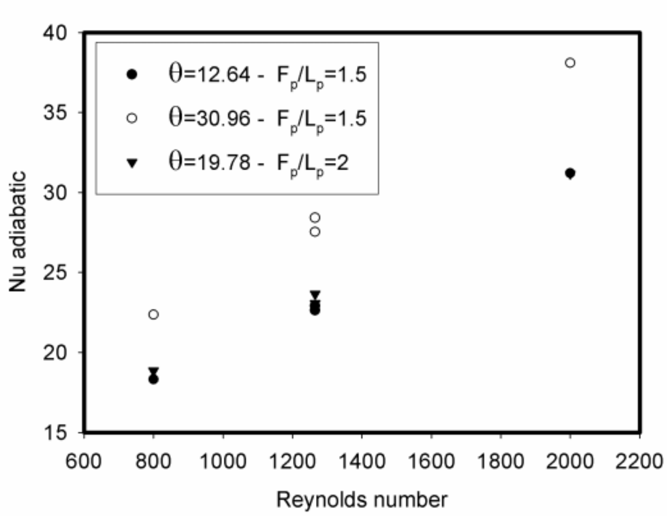

Fig. 11. Nusselt number vs Reynolds number.

The average convective heat transfer coefficients reported in Table 2 are based on the inlet temperature. If one would perform the adiabatic simulations, one would find a higher value. This can be explained as follows. As the flow passes through the fin array and heats up, the (adiabatic) wall temperature rises. This means that the denominator in the heat flux equation (Eq. (4)) is smaller when expressed using the adiabatic temperature. Further along the flow path this effect strengthens. Thus the average heat transfer coefficient based on the inlet temperature is smaller than the value determined on the adiabatic wall temperature. If one compares the measurements to the simulated values, one finds that the simulations overestimate the heat transfer coefficients.

CONCLUSIONS

In this paper heat transfer coefficients and friction characteristic are determined for inclined louvered fins. These results are part of the data to determine a general correlation for heat transfer and friction characteristic of adapted inclined louvered fins. A detailed description of the test rig is given. A numerical screening experiment using a CFD code was used to determine which parameters had the strongest impact on the heat transfer coefficient and the friction factor. These were then further studied experimentally.

Though only a limited set of the required data points for the inclined louvered fin model have been determined, it is already able to determine the trends of varying parameters. These confirm the results of the numerical screening experiment. However there are strong differences in the values obtained through experimentation and through simulation. It is well known that this type of flows (low Reynolds numbers but with unsteadiness triggered due to geometry) is difficult to accurately simulate. More study is needed to estimate the capabilities of CFD for predicting the thermo-hydraulic behavior of fins in this type of flow.

ERROR ANALYSIS

In order to be able to indicate the quality of the measurements a thorough error analysis was made using the procedures found in Taylor (1997).

The air flow rate was measured using a nozzle and a differential pressure transducer with an error of 0.05 Pa. This resulted in a maximum relative error of 1.38% on the air mass flow rate. A calibration measurement was used to determine the error on the thermocouples. All thermocouples were placed in temperature controlled room. The temperature was logged for a long period. The average and standard deviation was determined for each thermocouple. The largest value of standard deviation, 0.1K, was then used in the error analysis.

The average error on the friction factor was 6.36% and the overall error on the louver-array-averaged convective heat transfer coefficients is 1.92%.

NOMENCLATURE

A	=	total surface area [m²]
A_c	=	minimal flow area [m²]
F	=	friction factor [-]
F_p	=	fin pitch [m]
G_c	=	mass flux based on A_c [kg/m²s]
h	=	convective heat transfer coefficient [W/m²K]
L_p	=	louver pitch [m]
N	=	number of louvers [-]
q	=	heat flux [W/m²]
Δp	=	pressure drop [Pa]
Re	=	Reynolds number [-]
t	=	thickness [m]
T	=	temperature [°C]
U	=	velocity [m/s], expressed on the minimal flow section

Greek symbols

θ	=	fin angle [°]
λ	=	heat conductivity [W/mK]
μ	=	dynamic viscosity [Pas]
ρ	=	density [kg/m³]

Subscripts

ad	=	adiabatic
in	=	inlet
w	=	wall

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