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ENHANCEMENT OF FORCED CONVECTION COOLING OF HIGHLY LOADED HEAT SOURCES IN A RECTANGULAR CHANNEL BY FLOW DEFLECTION

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ABSTRACT

Investigations have been performed using FC-72 to improve the convective heat transfer from 5 heaters in-line, flush and equidistantly mounted to one wall of a vertically oriented rectangular channel. The Reynolds number, based on the hydraulic diameter, ranges from $1.3 \cdot 10^4$ to $1.5 \cdot 10^5$. The improvement of heat transfer is attained by deflecting a stream of liquid directly to each heat source like a submerged jet by means of inclined deflection plates positioned in the middle of the channel. So the heat transfer is enhanced by breaking up the thermal boundary layer. Additionally the plates function as turbulence promoters. The investigation of flow conditions and heat transport was done experimentally and numerically using the code FLOW3D. Inclination angles of 20°, 30° and 40° were examined and in addition the position of the plates relative to the heaters was variied.

1. INTRODUCTION

Efficient and effective cooling of electronic components is directly related to reliability and lifetime of the device. Cooling of high power chips challenges the thermal engineer because temperature limits become more constraining. Some of the available techniques to improve single-phase convective heat transfer are to roughen of the channel surface with various shaped rib or pin turbulators or periodic interruption of the hydrodynamic and thermal boundary layer by means of jets or obstacles in the flow channel.

The object of this work is to enhance the turbulent heat transfer from discrete heat sources in a rectangular channel with inclined plates positioned along the channel centerline. The periodic positioning of the plates accelerates the flow, enhances turbulence and reduces the thickness of the thermal boundary layer periodically at the heaters. With thin boundary layers higher heat transfer coefficients are associated because thermal resistance is reduced. Investigations of flow and heat transfer in a smooth channel without turbulence promoters serve as a reference.

There have been numerous studies on heat transfer in smooth rectangular channels with flush mounted heaters. Since the flow conditions in a rectangular duct represent a three-dimensional problem and - in contrary to pipe flows - secondary flows exist in planes perpendicular to the mean flow, the heat transfer depends on various geometric parameters: the aspect ratio of the flow channel, the heater geometry and distance to the channel walls and also the position to each other. In a detailled report *James* [1] presented experimental results for an asymmetrically heated duct and found the aspect ratio a important for heat transfer in the case $a < 2.5$. Various experimental or numerical investigations dealt with the turbulent heat transfer from single heat sources (*Craig* [2] and *Maddox* [3]), single-row configurations (*Abhyankar* [4]) or arrays of heaters (*Asako and Faghri* [5],[6]). Despite deducing dimensionless relations for the Nusselt number the results differ partially because there exist various determining characteristic lengths and it is difficult to consider all effects of the specific test configuration. In addition, the question of heat losses through the substrate to the environment and the "adiabatic" regions round the heaters, examined by *Abhyankar*, is not always discussed. The numerical studies investigated specific k-ϵ-models for low and high Reynolds numbers as well as Reynolds stress models.

The enhancement of heat transfer by interrupting the thermal boundary layer has been investigated by several researchers. Amongst studies obtaining such an improvement by manipulating the heater surface or by direct jet cooling only a few researchers investigated the effect of obstacles installed in a rectangular channel at positions away from the heated surface. *Shiina* [7] attained enhanced heat transfer in an asymmetrically heated duct by thin plate-type spacers oriented parallel to the heated wall. In a numerical study *Anand* [8] examined the effect of a series of normally in-line positioned plates. *Shiina* pointed out that the effect of flow acceleration caused by the blockage is not the decisive one for heat transfer improvement but the influence of turbulence enhancement in the wake of the plates. Both related the improved heat transport to the increase of pressure drop.

In this study the authors determined the improvement of heat transfer from single, highly loaded heat sources by inclined flow deflectors of a thin flat plate shape. Compared with parallel oriented spacers a higher increase of heat transport was expected and compared with normally positioned plates a decrease in pressure drop.

2. DESCRIPTION OF EXPERIMENT

The flow loop of the experimental system is shown schematically in Fig.1. It includes a

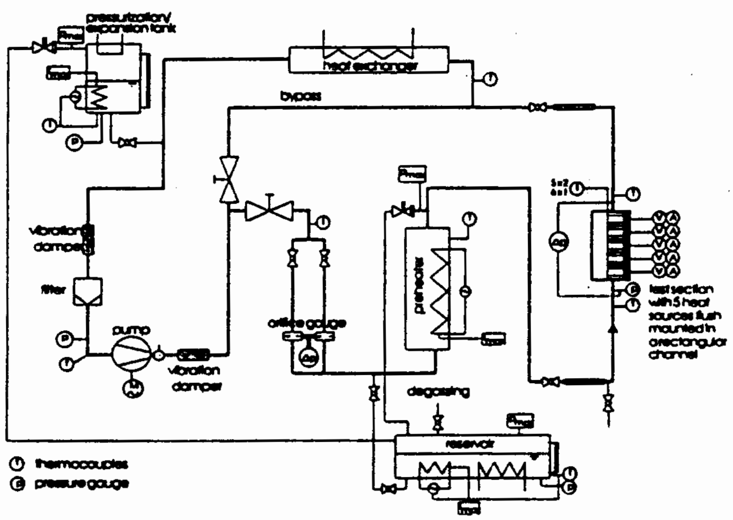

Figure 1: *Schematic diagramm of experimental facility*

magnetically coupled centrifugal pump, a main and a bypass line to provide fluid velocities between 0.5 and 4 m/s in the test section, orifice gauges to determine the mass flow rate and a preheater to adjust various entrance temperatures up to 80°C, furthermore the test section and a heat exchanger. Vibration dampers are installed to make optical measurements possible. The pressure tank maintains a constant system pressure up to 5 bar and functions additionally as expansion vessel. Before filling the loop with the fluorocarbon FC-72 the liquid is degassed by vigorous boiling and depressurization in the reservoir.

The experiments were performed in a rectangular duct of height H=10 mm and width W=20 mm with 5 heaters flush mounted in one narrow side of the vertically oriented channel (Fig.2). The heater length in flow direction is L_h=15 mm, their width H_h=4 mm. They have an equal spacing of S=15 mm to each other. The 5 deflection plates were mounted on the opposite wall on a movable sledge so that the position of the plates P relative to the heaters could be variied. P describes the distance between the end of a plate and the beginning of a heater. Three different sledges with plates positioned at inclination angles of $\alpha = 20°, 30°$ and $40°$ were used. It is important to mention that the plates are designed in such a way that they have the same projected width i.e. blockage effect for any of the three inclination angles. So the distance between the edges of the plates and the short sides of the duct is 5 mm on both sides in any case (see Fig.2). The

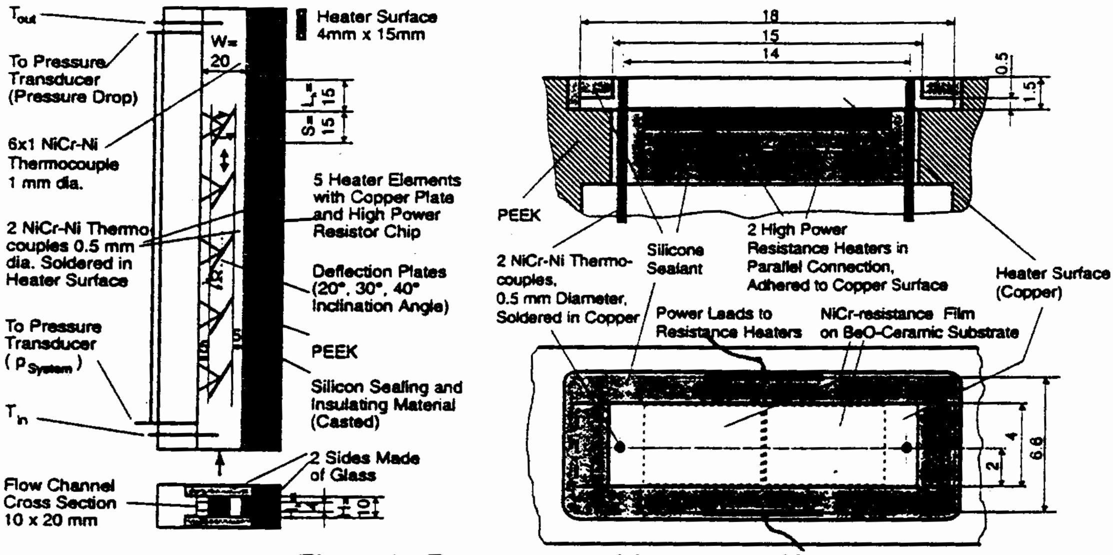

Figure 2: *Test section and heater assmbly*

casing of the flow channel is made of stainless steel while the sides forming the rectangular duct are made of glass or PEEK. The glass sides allow optical measurements by LDV and the shadowgraph method.

The heat source assembly is also shown in Fig.2. Two high power resistance chips are joined to a copper plate by a high temperature epoxy. The surface temperature of the copper block is measured with two thermocouples soft-soldered in the copper at the positions shown in the figure. To seal the whole module in the PEEK substrate a RTV-silicone rubber casting material is used.

All of the experiments were conducted under conditions of equal heat flux from each of the 5 heaters. Therefore, the electrical power to the heaters was in each case determined by measuring the voltage drop at a precision resistor and at the heater itself. To take the individual heat losses into consideration each heater was calibrated. During the experiments each heater was heated with its individual power input to get equal heat flux conditions. To determine the heat input from the heater to the coolant a three-dimensional heat conduction program was developed to simulate the heat losses to the surrounding. They amount to 29% at $Re_D = 1.3 \cdot 10^4$ and decrease with increasing Reynolds number to 11% at $Re_D = 1.3 \cdot 10^5$.

3. RESULTS AND DISCUSSION

3.1 Smooth channel

Figure 3 shows the data for the smooth channel with hydrodynamically fully developed flow. They are correlated in the Reynolds number range $1.5 \cdot 10^4 < Re_L < 1.5 \cdot 10^5$ by the equation

$$\overline{Nu_{L,m}} = 0.158 \cdot Re_L^{0.699} \cdot Pr^{0.33} \quad \text{with} \tag{1}$$

$$\overline{Nu_{L,m}} = \frac{1}{5}\sum_{i=1}^{5}\overline{Nu_{L,i}} \quad \text{and} \quad \overline{Nu_{L,i}} = \frac{\overline{h_i}\cdot L_h}{k} \tag{2}$$

where $\overline{Nu_{L,m}}$ is averaged over the 5 heaters and

$$\overline{h_i} = \frac{\dot{q}w}{(\overline{T_{W,i}} - T_{b,i})} \quad \text{with} \quad T_{b,i} = T_{in} + (i-1)\cdot\frac{\dot{q}w}{c_p} \tag{3}$$

and $\overline{T_{W,i}}$ is the arithmetic mean of the two wall temperatures of each heater. All properties are based on the fluid bulk temperature at the entrance of the flow channel.

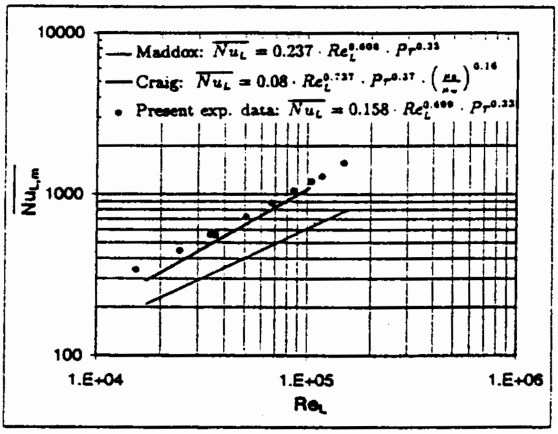

Fig.3: *Comparison of smooth channel data*

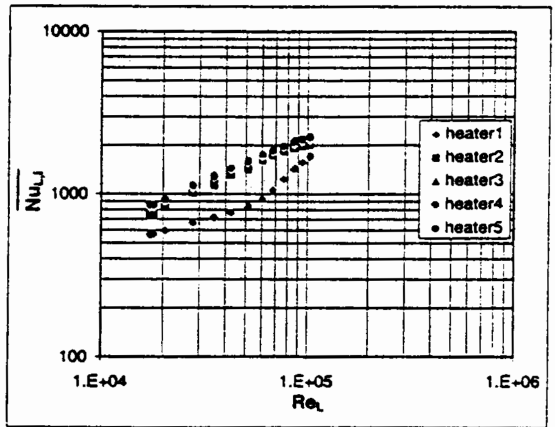

Fig.4: *Effect of heater row for α=20°, P=0mm*

A comparison with the correlations of *Craig* and *Maddox* (Fig.3) shows better agreement with Craigs data but all channel and heater configurations differ in aspect ratio, heater placement and taking the heat losses into accont.

3.2 Channel with deflection plates

Effect of heater row and behaviour of the first heater. In a duct with obstacles of the same distance and shape the flow develops periodically. In this study the flow is periodically developed behind the third plate. That means that the velocity field repeats at corresponding places in successive cycles. At the first plate and in the wake of it the effect of the starting development of the periodic flow causes lower velocities than at the following ones. So the heat transfer certainly improves at the first heater because flow velocities increase due to the blockage effect, but not as much as at the successive ones, as shown in Fig.4. At the last heater the flow changes again because the periodic interruption ends. Due to turbulence enhancement in the wake of the last plate the heat transfer increases at the fifth heater.

Effect of plate position. Figure 5 shows that the behaviour of the first heater depends strongly on the plate position P. In the case of heater 2 to 5 the best heat transfer occurs when the plates end about 5 mm in front of the beginning of the heaters (P=5) for all investigated inclination angles. This is the place of maximal contraction of the deflected flow connected with the highest flow velocities.

Effect of inclination angle. The experimental results presented in Fig.6 show that the 20° and 40° inclined plates enhance the heat transfer more than the 30° configuration. On the other side the pressure losses were also higher for 20° and 40° than for 30° which shows an analogy between momentum and heat transfer. Different inclination causes a different impact of the plates as flow deflectors (jet effect) or turbulence promoters. This is confirmed by measurements of flow velocity and turbulence intensity in the free cross section area at the heater side done by LDV. Considering pressure drop and heat transfer and relating the results in form of the flow area goodness factor, for example, the 30° configuration reaches the highest values.

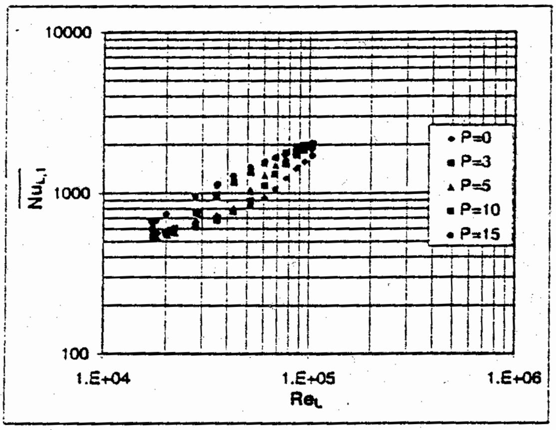

Fig.5: *Effect of plate position P [mm] for heater 1, $\alpha = 20°$*

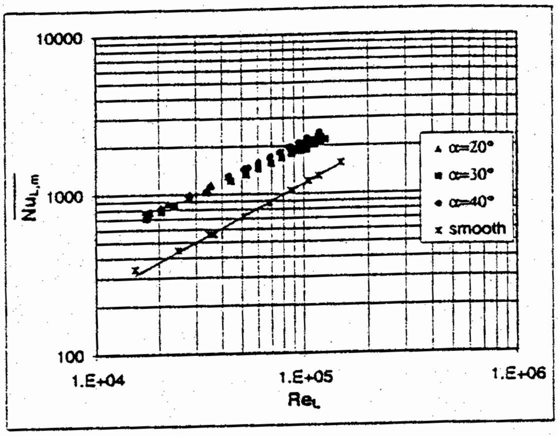

Fig.6: *Effect of inclination angle: Average Nusselt number for all heaters, P=5mm*

Comparison of numerical and experimental results. Fig.7 shows the generated grid and the computed velocitiy field between the 3. and 4. plate using the code FLOW3D.

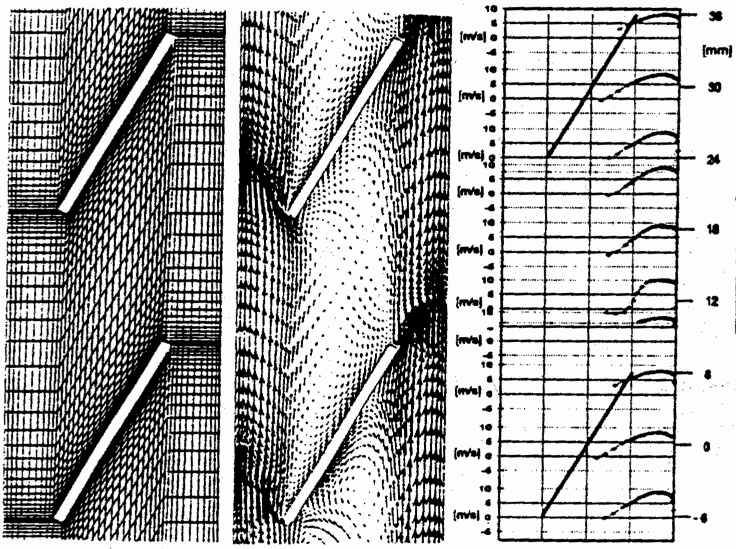

Fig.7: *FLOW3D-grid, velocity field and LDV results between the 3. and 4. plate for $\alpha = 30°$, $U_0 = 3.5 m/s$*

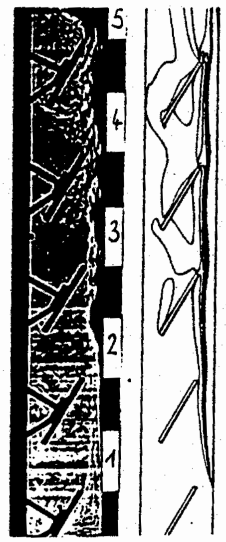

Fig.8: *Shadowgraph picture and lines dT/dx=const. for α=30°, $T_{in} \approx 20°C$, $T_W \approx 70°C$*

Here a $k - \epsilon$ model is applied for the 2D-computation but a Reynolds stress model was used also. Comparisons of single computed and measured velocity profiles as well as profiles of turbulent kinetic energy show good agreement. Postprocessing of the computed heat transfer data in form of lines of constant temperature gradient $dT/dx = const.$ allow an interesting comparison with the shadowgraph pictures. One of these pictures is shown in Fig.8. With the shadowgraph method the thermal boundary layer can be visualized as dark region (shadow) at the wall where the heaters are flush mounted. Additionally it illustrates clearly the turbulent motion.

5. NOMENCLATURE

a	aspect ratio	[-]
c_p	specific heat capacity at constant pressure	$J/(kg\,K)$
D	equivalent hydraulic diameter $= \frac{2HW}{2(H+W)}$	m
h	heat transfer coefficient	$W/(m^2\,K)$
H, H_h	channel hight, heater width	m
k	fluid thermal conductivity	$W/(mK)$
L_h	heater length in flow direction	m
$\overline{Nu_{L,m}}, \overline{Nu_{L,i}}$	average Nu number, over 5 heaters (m) or heater i	[-]
P	plate position relative to the heater	m
Pr	Prandtl number	$[-]$
$\dot{q}_W$	corrected wall heat flux from the heater to the fluid	$\frac{W}{m^2}$
Re_L, Re_D	Reynolds number $= \frac{U_0 \cdot L_h}{\nu}$ or $= \frac{U_0 \cdot D}{\nu}$	[-]
S	spacing between the heaters	m
T_{in}, T_{out}	bulk inlet and outlet temperature	K
T_W	wall temperatures at the heater	K
U_0	average inlet velocity	K
W	channel width	m
α	inclination angle	[-]
μ	fluiddynamic viscosity,	Pas
ν	kinematic viscosity	m^2/s

6. REFERENCES

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