Experiments on Natural Convective Air Cooling of a PCBs Array in a Closed Casing with Inclination

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ABSTRACT. This paper presents an experimental study of natural convective air cooling of a printed-circuit-boards (PCBs) array in a closed casing with inclination. The results reveal that in both cases (without and with inclination) an unstable thermal boundary layer exists on the top wall of the casing. This unstable boundary layer, which interacts with the air flow rising up from the channels between the PCBs, produces vortex flow and temperature oscillation. In the inclined casing warm air gathers together in the upper corner of the casing and cold air falls down to the bottom corner. Therefore, the cooling effect on the PCBs in the region near the bottom corner of the casing increases and that in the region near the upper corner decreases. It should be noted that the temperature of the PCBs located near the upper corner can exceed the allowed maximum working temperature designed for levelly adjusted casing and cause an overheating of electronic elements. An empirical correlation deduced from the experimental results is presented in the form of $Nu = f(Ra, \alpha)$ with the modified channel Rayleigh number ranging from 10^2 to 10^5 and with an angle of inclination from 0° to 30°.

1. Introduction

In electronic equipment the temperature of the chips must not exceed a permitted value to protect them from thermal damage. Natural convective air cooling is one important thermal control method and often preferred to maintain the temperature of electronic equipment with low power and packaging density below an acceptable level. Especially in such cases where the casing of the electronic equipment must be closed for dust- and water-proof, natural convection cooling scheme may be the only feasible option. In practice electronic equipment with a printed-circuit-boards (PCBs) array in a closed casing are often working in a position with a varying angle of inclination. Under these circumstances, the inclination will affect the magnitude of the driving force (mostly gravitation) causing natural convection, and has therefore an effect on the temperature distribution and the heat transfer in the casing. In order to design an effective cooling system in a closed electronic equipment and get optimum cooling results the behaviour of natural convection in such complex structures should be investigated in detail.

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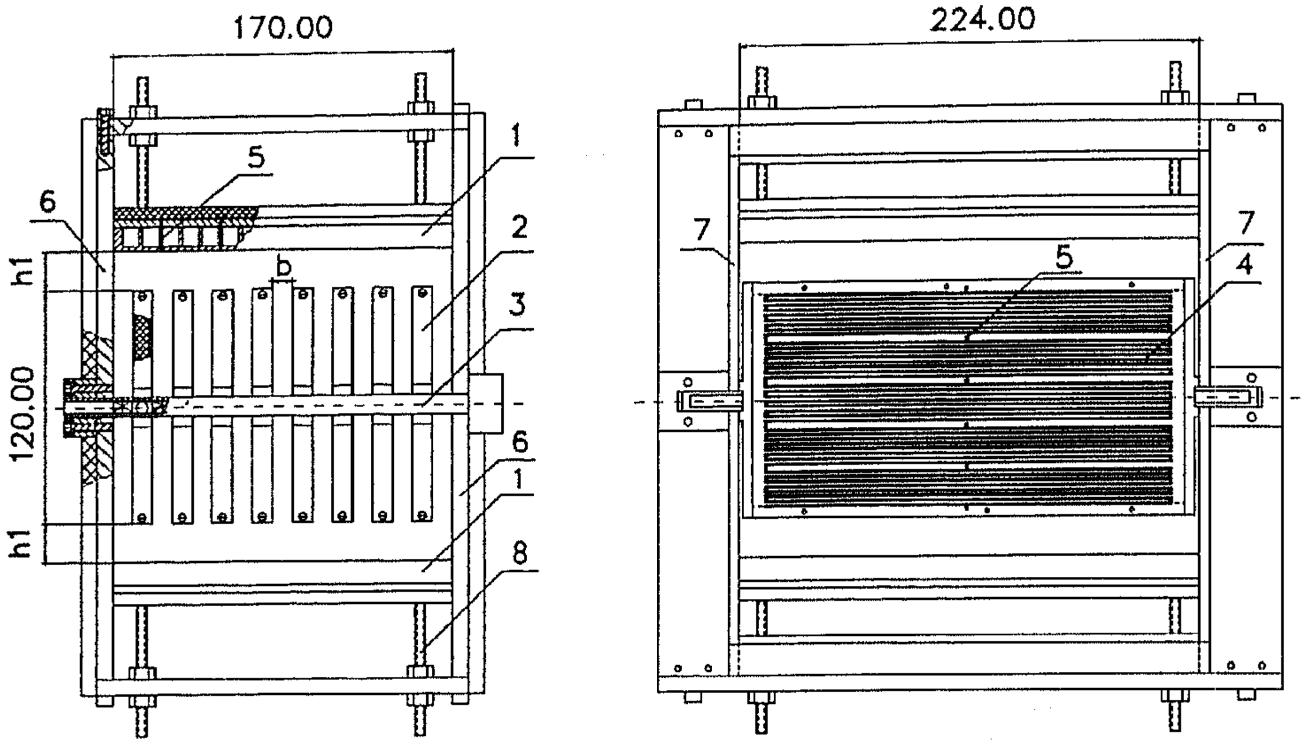

Figure 1: Test section

Much work has been done on the natural convection between two parallel plates under various boundary conditions (Elenbaas [1942], Aung [1972], [1973], Sparrow [1980], Wirtz & Stutzman [1982], Bar-Cohen & Rohsenow [1984] etc.), which laid the foundations of studying natural convection cooling in electronic equipment. Some literature deals with natural convection in closed electronic equipment, e.g. Guglielmini [1985, 1988a, 1988b], Cadre [1988] and Liu [1988].
The present work is concerned with experimental investigation of natural convective air cooling of the PCBs array in a closed casing with an inclination from 0° to 30°. Heat transfer and temperature field in such a PCBs array were measured with a holographic interferometer. From the experimental results empirical correlations were deduced. The effects of the PCBs spacing, the distance between the PCBs array and top or bottom wall of the casing, the inclination and the instability were analyzed.

2. Experiments

2.1. TEST SECTION

The edge and side view of the test section are shown in Figure 1. The simulated PCBs were 220.0 mm long, 120.0 mm high and 10.0 mm thick and constructed of a copper foil laminate similar to that used in electronic industry. The heater elements were designed in a

serpentine fashion and formed on the laminates with etch method. One laminate is mounted on each of the two sides of a frame to simulate a PCB. Voltage and current measurements were made for each heater element giving an estimated accuracy of $\pm 3\%$.
The top and bottom wall of the casing were made of aluminium plates. S-form slots were cut in the plates. Cold water from a thermostat was used to cool the top and bottom wall at the constant temperature of $20°C$. The side walls were made of plexiglass plates ($k \approx 0.184 W/°Km$). The front and the back wall were glass windows suitable for the holographic interferometer. The spacings between the PCBs were equal and could be changed. The distances between the PCBs array and the top or bottom wall were also adjustable with a screw mechanism. The casing was placed on a movable small table and its inclination can

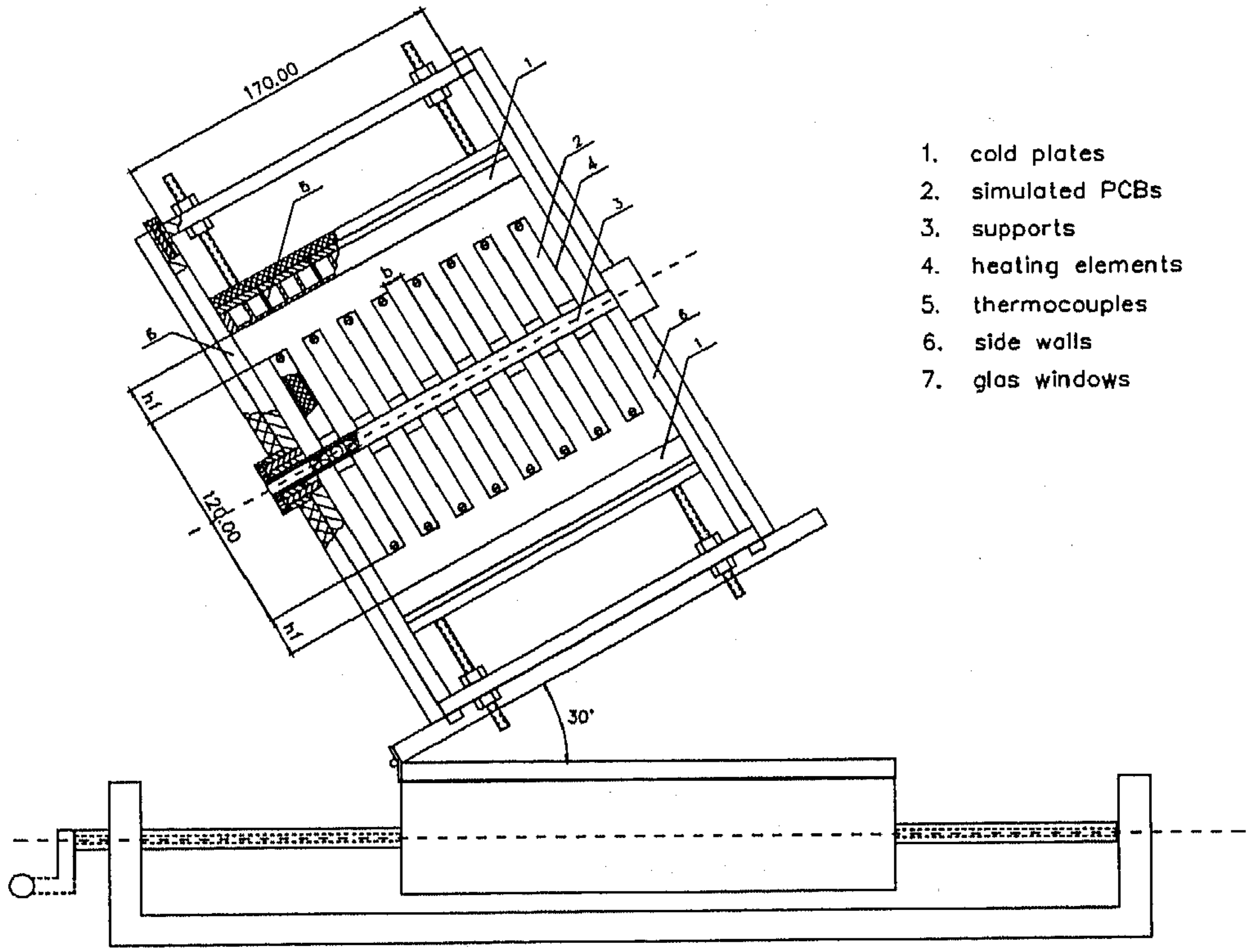

Figure 2: Table with inclined casing

be set with an angle adjuster. Figure 2 shows the table with the inclined casing.

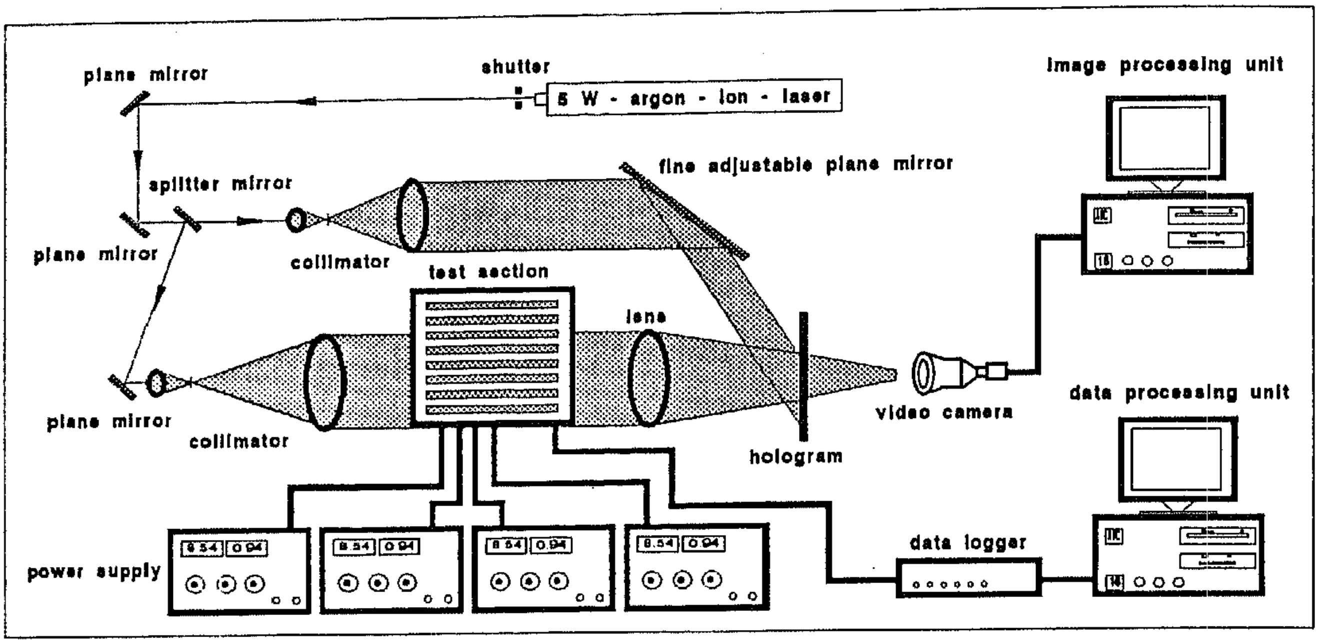

Figure 3: Holographic interferometer

2.2. MESURING TECHNIQUE

In order to get a complete picture of temperature distribution in the casing an optical method was used. The temperature fields were measured with a ϕ 150mm holographic interferometer, which is shown schematically in Figure 3. In order to trace the unstable thermal boundary layer occurring in the casing, real time holographic interferometry was applied to visualize the temperature oscillation. The principle of real time holographic interferometry in heat transfer can be found in the literature (e.g. F. Mayinger [1974]) and will be omitted here.

The wall temperature was also measured with 0.2 mm chromel/alumel thermocouples. Six thermocouples were spaced at the same interval along each side of the PCB and located in the 2-mm wide gaps between the heater strips.

When the temperature in the casing and on the PCBs was measured the heat transfer can be analyzed by the local Nusselt number and the global Nusselt number which can be respectively expressed as follows:

$$Nu_x = \frac{b\dot{q}}{k(T_p - T_w)} \tag{1}$$

where Nu_x is the local Nusselt number, b the PCBs spacing, T_p the temperature on the PCB, T_w the temperature of the top wall, k the thermal conductivity and $\dot{q}$ the uniform heat flux on the PCBs.

$$Nu_p = \frac{b\dot{q}}{k(T_{p_{max}} - T_w)} = f(Ra, \alpha) \tag{2}$$

where Nu_p is the global Nusselt number, $T_{p_{max}}$ maximum temperature on the PCBs, Ra the modified channel Rayleigh number and α the angle of inclination. Ra is defined as:

$$Ra = \frac{g\beta\dot{q}b^5}{k\nu^2 H} Pr \tag{3}$$

where H is the height of the PCB.

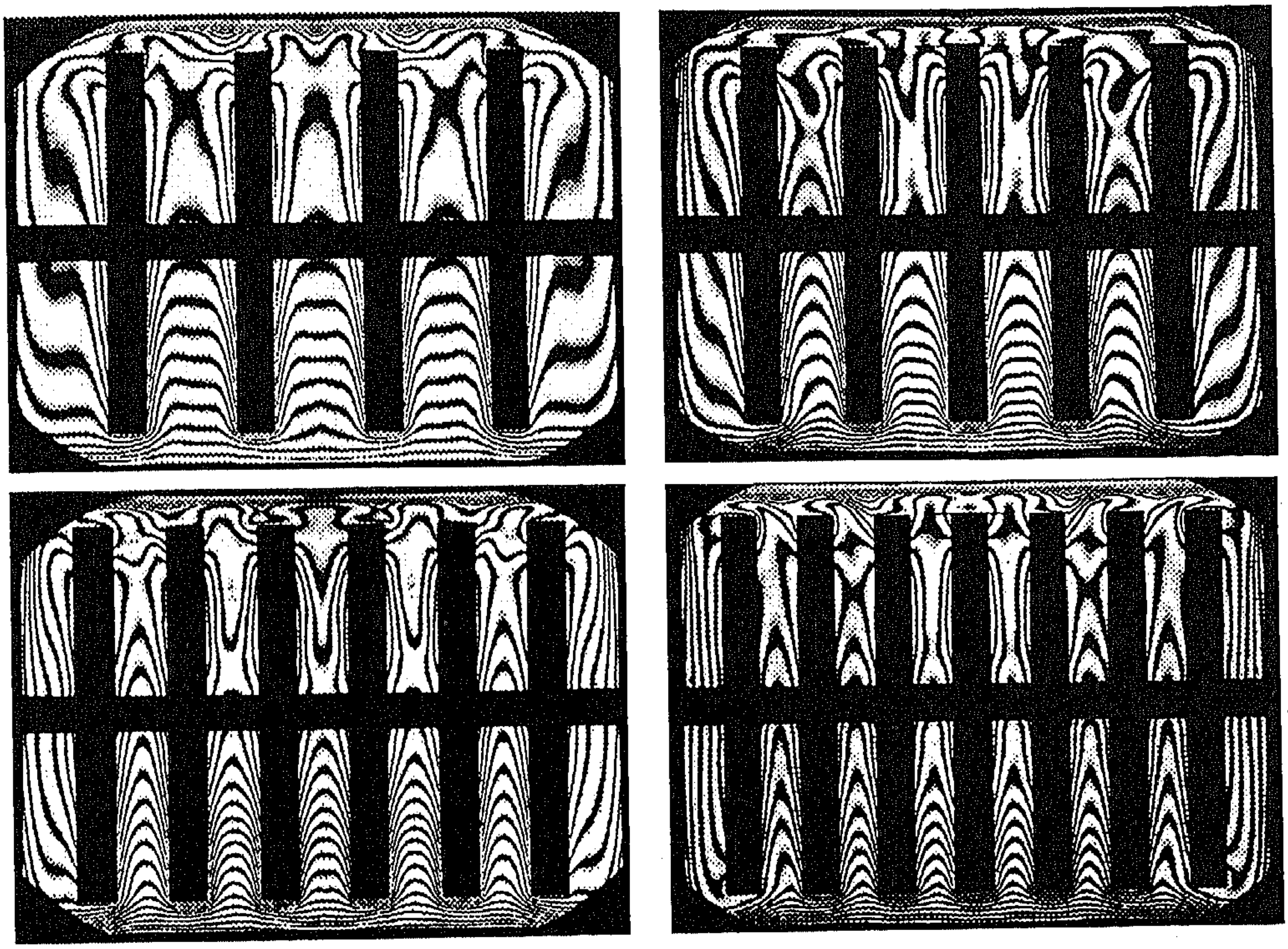

Figure 4: Holographic interferograms in 4-, 5-, 6- and 8-PCBs arrays

3. Results

3.1. TEMPERATURE FIELDS

The holographic interferograms of the temperature fields in a 4-, 5-, 6- and 8-PCBs arrays in a casing without inclination are, for instance, shown in Figure 4. (For the sake of clarity the right halves of the above holograms are presented with the mirror images of the left halves).

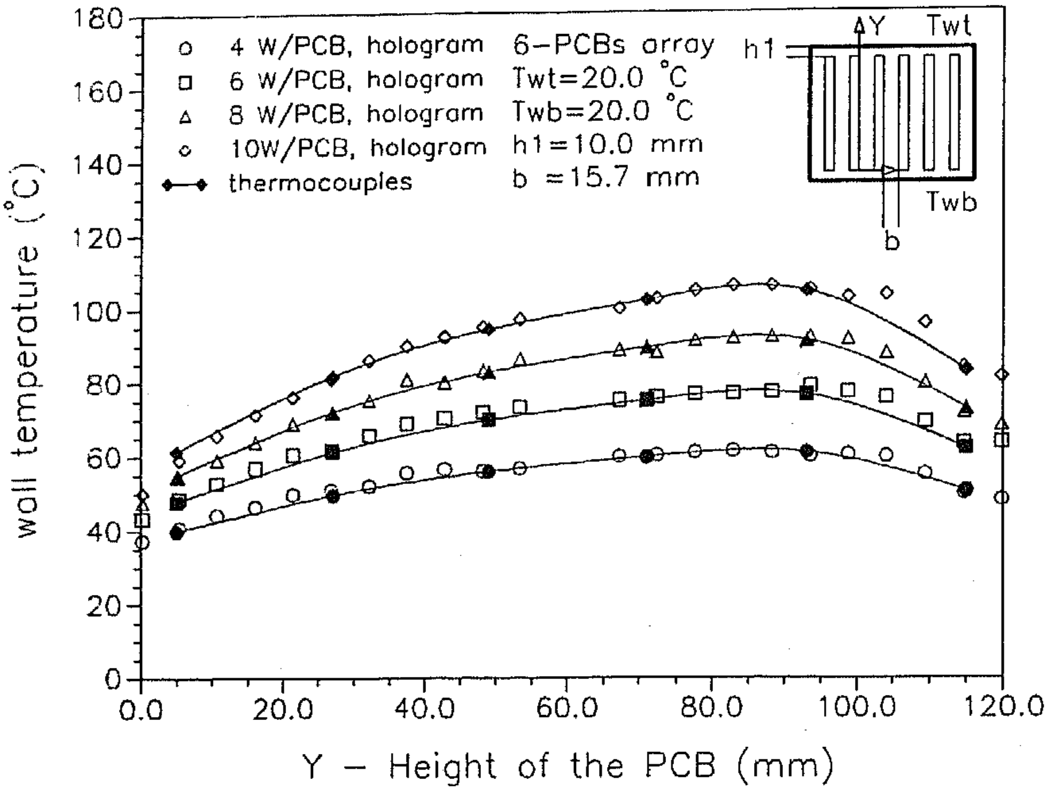

Figure 5: Temperature on the PCB

From Figure 4 it can be observed how the thermal boundary layers develop at the surfaces of the PCBs because of the natural convection. When the PCBs spacing is relatively large (e.g. in a 4-PCBs array), thermal boundary layers are in evidence along each surface. In the lower half part of the PCBs channels, the thermal boundary layers develop from bottom to top and become thicker, as expected. However in the exit region of the channels, the development of the thermal boundary layer is hindered by the vortex flow which is produced by the thermal instability in the reverse temperature layer on the top wall of the casing. The thickness of the thermal boundary layer is reduced by cold air which falls randomly down from the top wall. The thermal boundary layers in the PCBs array have therefore a bulging form. When the PCBs spacing is small enough (e.g. in a 8-PCBs array) the developing boundary layers in the channels join together behind a short developing region. It is obvious that the ability of heat removal in a single channel is lower for a small PCBs spacing than that for a large PCBs spacing. However, in the case of a small spacing the power load which needs to be removed from each PCB will be reduced because more PCBs could be available in the casing. Therefore, one optimum spacing may be found for the optimum cooling effect, which will be discussed later.

The temperature distributions on the second PCB in a 6-PCBs array are shown in Figure 5. The maximum of the surface temperature presents at about 3/4 height of the PCB with the power load varying between 4 and 10 W/PCB.

At the top wall of the casing the thermal boundary layer has a wavelike form. The cold air at the top wall sinks down from the wave crest into the PCBs array and interacts

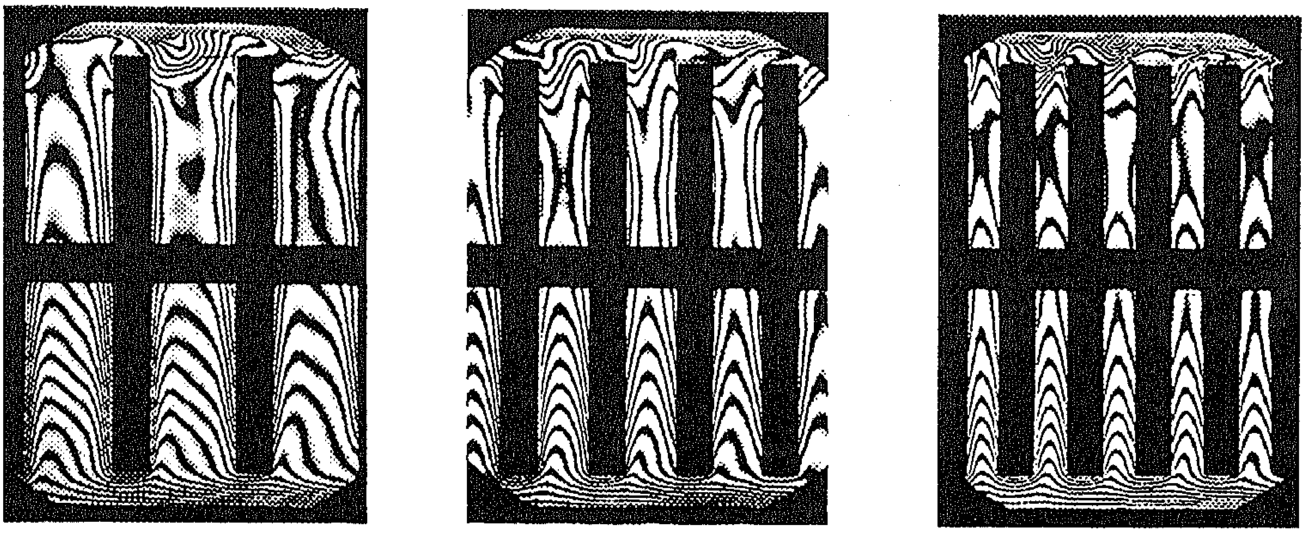

Figure 6: Interferograms in 4-, 6- and 8-PCBs arrays in an 30° inclined casing

with the warm air rising up from the channels to produce vortex flow in the exit region of the PCBs array. Because of this time-varying falling of cold air the temperature in this region is unsteady. With increasing Rayleigh number the temperature oscillation changes from periodic to chaotic. These temperature oscillations attenuate from the centre of the channels to the surfaces of the PCBs. On the surface of the PCBs the temperature oscillates only in the limit of $0.4°C$.
The holographic interferograms of temperature fields in 4- and 6-PCBs arrays in a casing with inclination are, for instance, shown in Figure 6. The casing is inclined to 30°. Holograms in the middle of the casing are presented. From the holograms it can be seen that an unstable thermal boundary layer presents at the top wall of the casing. The temperature distribution in the two half parts of the casing is no longer symmetric. Warm air rises up to the upper corner of the casing and is cooled by the top wall. The cold air partly flows down into the channel between the PCBs array and the upper side wall and partially circulates along the top wall to the lower side wall. Because of this type of air circulation the temperature in the upper half part of the casing is higher than that in the lower half part. The temperature distribution on the PCBs can be clearly seen from Figure 7.

3.2. HEAT TRANSFER

By means of an image processing system, the interference fringes can be measured quantitatively. The heat transfer coefficient can be calculated with the equations in section 2.2. For the thermal design of electronic equipment the relation between total heat load and temperature difference is very important. Determining the total heat load according to the given permitted temperature rise and/or finding the maximum temperature rise on the basis of the given total heat load is one of the fundamental tasks of thermal designers. From the experimental results a correlation between Nusselt number and Rayleigh number

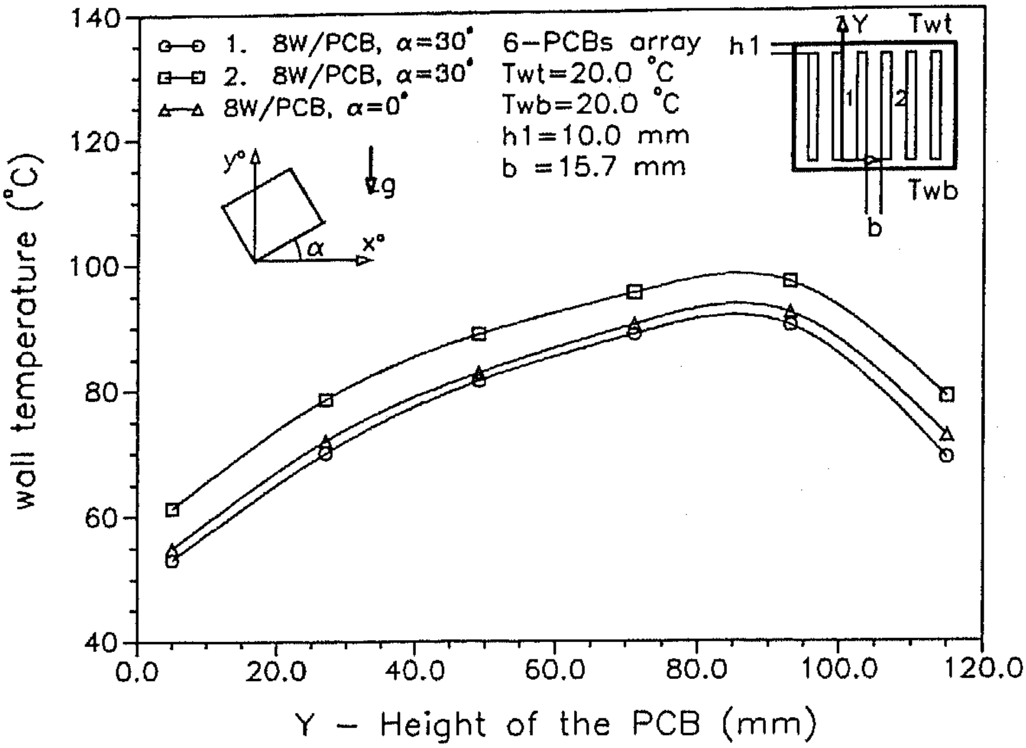

Figure 7: Temperature on the PCBs in an inclined casing

for a PCBs array in a casing without inclination can be expressed as follows:

$$Nu_p = 0.1163 Ra^{0.287} \tag{4}$$

This is shown in Figure 8.
When the distance between the PCBs array and the top wall is increased, the flow resistance in the PCBs array will be reduced. Because of the limitation of the side channel the air circulation can only be accelerated to a certain degree. On the other hand the direct cooling effect of the top wall on the PCBs array will decrease as the distance between the PCBs array and the top wall increases. Because of these two opposite factors the cooling effect of the PCBs array will be insensitive to the distance between the PCBs array and the top wall. That has been confirmed by experiments. Therefore relation (4) can also be used for other arrangements with $h1 \leq 30$ mm.
When the casing is inclined the driving force of the air natural convection in channel direction will be reduced. In vector form it can be expressed as follows:

$$g_x = g \sin\alpha \tag{5}$$

$$g_y = g \cos\alpha \tag{6}$$

If the angle of inclination is small, the main direction of the natural convection is in channel direction. From experimental results it has been found that the correlation of heat transfer

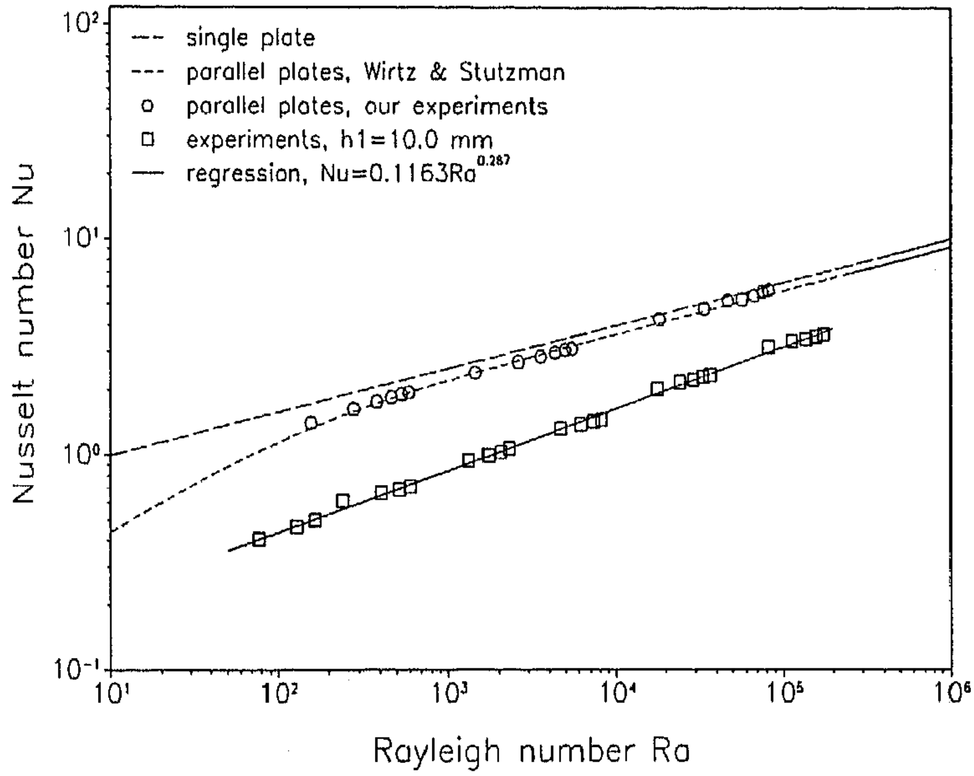

Figure 8: Relation $Nu_p = f(Ra, 0°)$

has the same form as equation (4). If we define the related modified channel Rayleigh number in the case of small inclination as:

$$Ra = \frac{g_x \beta \dot{q} b^5}{k \nu^2 H} Pr = \frac{g \cos\alpha \beta \dot{q} b^5}{k \nu^2 H} Pr \tag{7}$$

the heat transfer relation for the PCBs in a closed casing with an inclination from 0° to 30° can be uniformly expressed by relation (4). Figure 9 shows the experimental results.

3.3. VELOCITY PROFILE IN PCBs ARRAY

Air circulation plays an important role in the cooling of the PCBs array in a closed casing. The air goes up from the PCBs array when it is warmed up by the dissipation heat of chips on the PCBs. Because of the existence of the top wall the air flow has to take a turn to the two side walls of the casing. If the side walls of the casing are cold, the air flow will fall down into the two side channels and rise again into the PCBs array after a turn on the bottom wall. As a whole, the overall natural convection flow can be divided into several circulation layers. If the situation is symmetrically the natural convective air flow produced in the central channel will circulate in the most external layer and the air stream from the channels next to the central one will flow around the interior layers in sequence.

In order to confirm the existence of the recirculation in the PCBs array the velocity profile in the central channel between two PCBs was measured with a LDV. Figure 10 presents

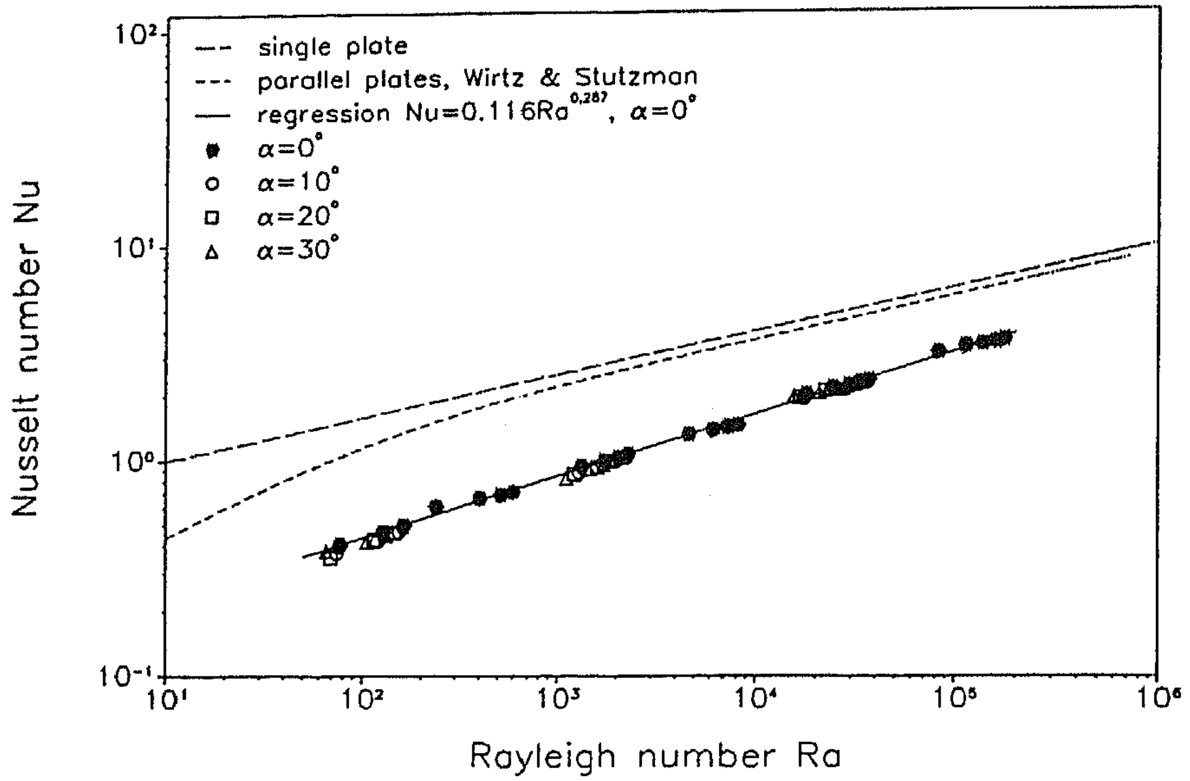

Figure 9: Relation $Nu_p = f(Ra, \alpha)$

these measuring results. From Figure 10 it can be seen that the recirculation exists in the central part of the channel. The penetration of the recirculation is up to the half height of the PCBs. In the exit region the velocity is also unsteady because of the time-varying recirculation. It can be clearly seen that the thickness of the boundary layer is the largest at the entrance because of the 90° turn of the air flow from the side channels. It increases in the channel, as expected, and decreases at the exit region because of the recirculation.

3.4. OPTIMAL SPACING OF PCBs

As mentioned above there may exist an optimum arrangement of PCBs under the same maximal temperature difference between the PCBs array and the cold wall due to the ability of heat removal depending on the PCBs spacing as well as on the number of PCBs. The correlation derived in a previous section can be used to optimize the spacing of the PCBs. With definition (3) and correlation (4) the Nusselt number can be expressed as:

$$Nu_p = ARa^B = \frac{\dot{Q}_p b}{2Sk\Delta T} \tag{8}$$

where A and B are constants, $\dot{Q}_p$ is the heat dissipation of one PCB, b is the spacing of PCBs, S is one side area of a PCB. Under the same temperature difference $\dot{Q}_p$ is a function

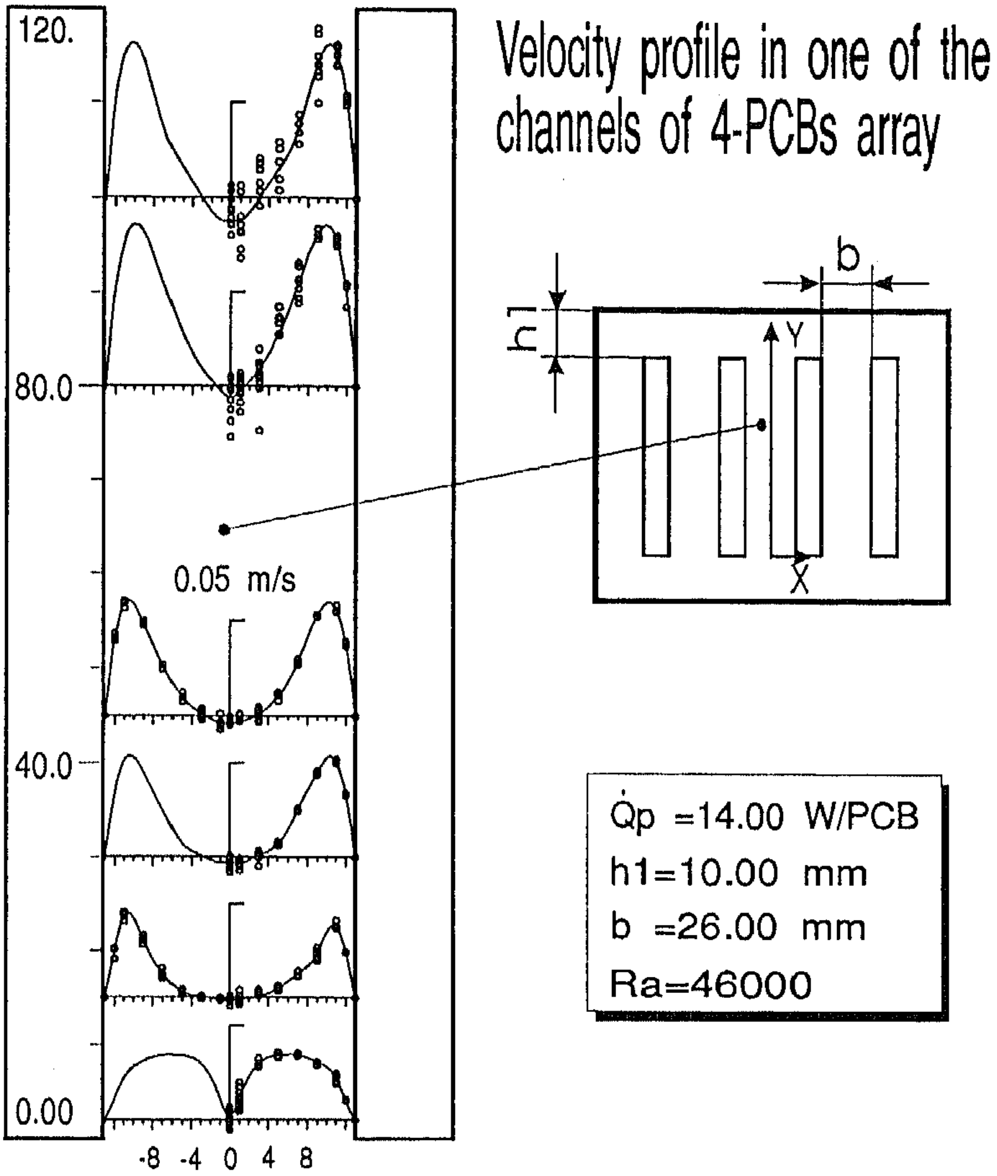

Figure 10: Velocity profile

of the spacing of the PCBs, which can be derived from the equation above:

$$\dot{Q}_p = Cb^{\frac{5B-1}{1-B}} \tag{9}$$

where C is the following expression:

$$C = A\left(\frac{g\beta}{2SH\nu\alpha k}\right)^{\frac{B}{1-B}}(2Sk\Delta T)^{\frac{1}{1-B}} \tag{10}$$

The total heat dissipation can be calculated as:

$$\dot{Q} = N \cdot \dot{Q}_p = C\frac{l-b}{d+b}b^{\frac{5B-1}{1-B}} \tag{11}$$

where N is the number of the PCBs in the casing, which equals (l-b)/(d+b), d is the thickness of the PCB, l is the length of the casing. Differentiating equation (11) with respect to b and setting the derivative to zero leads to:

$$f(b) = D(l-b) - b\frac{d+l}{d+b} = 0 \tag{12}$$

with the constant D as

$$D = \frac{5B-1}{1-B} \tag{13}$$

Solving above equation yields the optimum spacing of the PCBs in the form:

$$b_{opt} = \frac{\sqrt{E^2 + 4D^2dl} - E}{2D} \tag{14}$$

with

$$E = d + l + D(d-l) \tag{15}$$

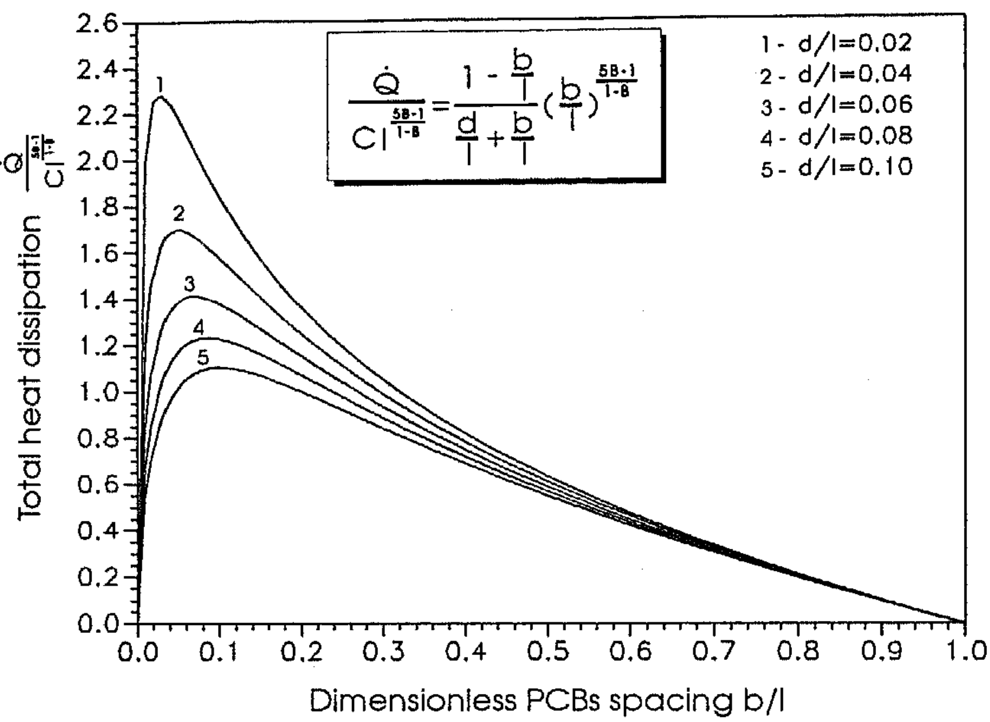

Figure 11: Optimum spacing of the PCBs

Equation (10) is graphically presented in Figure 11. When the values of d=10.0 mm, l=170.0 mm and B=0.287 for the experiments are substituted in the above relation the optimum spacing of the PCBs is found to be 11.59 mm.

4. Conclusions

The experimental results show that the unstable thermal boundary layer at the top wall of the casing plays an important role in natural convective air cooling of the PCBs array in a closed casing with an inclination from 0° to 30°. Because of the thermal instability vortex flow and temperature oscillation exist in the casing. Time-varying recirculation presents in the PCBs array. The heat transfer in such a PCBs array is greatly influenced by the distance between the PCBs and relatively insensitive to the distance between the PCBs array and the top or bottom wall. Uniform empirical correlations was deduced from the experimental results for the heat transfer in a casing with inclination from 0° to 30° and the optimum spacing of the PCBs and presented in the paper.

Nomenclature

A	Constant
B	Constant
b	PCBs spacing
b_{opt}	Optimum spacing of the PCBs
C	Coefficient
D	Constant
d	Thickness of the PCB
E	Constant
g	Gravitational acceleration
H	Height of the PCB
k	Thermal conductivity
l	Length of the casing
N	Number of the PCBs in the casing
Nu_p	Global Nusselt number
Nu_x	Local Nusselt number
Pr	Prandtl number
$\dot{Q}_P$	Heat dissipation of one PCB
$\dot{q}$	Uniform heat flux on the PCBs
Ra	Modified channel Rayleigh number
S	One side area of a PCB
T_P	Temperature on the PCB
$T_{P_{max}}$	Maximum Temperature on the PCBs
T_w	Temperature of the top wall
α	Angle of inclination
β	Thermal expantion coefficient
ν	Kinematic viscosity

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