Experimental Thermal and Fluid Science 88 (2017) 234–245

Contents lists available at ScienceDirect

Experimental Thermal and Fluid Science

journal homepage: www.elsevier.com/locate/etfs

Temperature and velocity measurements in a buoyant flow induced by a heat source array on a vertical plate

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ARTICLE INFO

Article history:
Received 28 November 2016
Received in revised form 23 May 2017
Accepted 3 June 2017
Available online 08 June 2017

Keywords:
Heat source array
Conjugate natural convection
Velocity measurements
Particle image velocimetry
Micro-thermocouple
Proper orthogonal decomposition

ABSTRACT

Heat source arrays are common in engineering applications. Natural convection is a reliable and silent cooling strategy. Therefore, an array of flush-mounted heat sources has been studied under conjugate conduction and natural convection condition. This studies was performed for a system with relatively large dimensions, typical for power electronics, and a modified Rayleigh number up to $2 \cdot 10^{10}$. A modular set of heaters was designed to vary the distribution of heat sources on the plate and investigate the influence of the spacing. Velocity and temperature were measured in the convective flow with particle image velocimetry and micro-thermocouple. The velocity field was analyzed with proper orthogonal decomposition. The first instabilities of the convective flows were described. The results gave a better understanding of the heat transfers in these configurations and are valuable for model validation.

1. Introduction

The phenomenon of natural convection is present as soon as a temperature gradient appears in a fluid. It is therefore of importance in many engineering applications [1]. Arrays of heat sources are common configurations in systems such as electronics. Electrical losses in components give rise to cooling issues, for instance with power electronics for renewable energies [2].

Interests for conjugate natural convection grew with the development electronic. Miniaturization in microelectronics and the consequent increase in heat flux density made it necessary to optimize cooling. In 1995, an array of identical discrete heat sources was studied numerically and experimentally with conjugate conduction and natural convection [3]. The importance of 3D-effects was highlighted and a numerical model was validated for laminar steady conditions. Based on these results, a parametric study was presented for the fluid/substrate thermal conductivity ratio and for the modified Rayleigh number ($< 10^9$) [4].

The thermal coupling between the sources in a array under natural convection is nonlinear and complex. A first optimization study was performed for discrete heating with heaters of an infinite length and to solve the problem with a two-dimensional equations [5]. The convective flow driven by linear heat sources was studied experimentally in [6]. Similar problems with forced convection or with protruding sources have also been studied, like for instance in [7,8].

An array of flush mounted sources with diverse sizes was studied experimentally and numerically in [9]. The purpose was to test different configurations in order to optimize the heat transfer. The influence of the spacing between identical heat sources on the overall thermal resistance has been studied for optimization purposes in [10].

A 3×3 heater array in an enclosure with liquid cooling has been studied numerically for low Rayleigh numbers ($< 10^8$) in [11]. It showed that, the local Nusselt number presented important variations in space and the influence of the enclosure aspect ratio was studied. In [12], the orientation of the cavity was studied on the convective flow due to an array of heat source.

In [13] a problem with multiple non-identical sources was optimized at low Rayleigh number. The temperature was measured at the source and a non-dimensional distance parameter was introduced to implement a first order optimization strategy.

Artificial neural network was used together with genetic algorithm in [14] for an array of heaters with different sizes. The temperature excess was also experimentally studied at the source. The distribution of protruding heat sources in a vertical duct was optimized under forced convection with an ANN-GA approach in [15].

This type of problems has been the subject of several reviews [16,17]. Most of the published work was conducted in the context

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http://dx.doi.org/10.1016/j.expthermflusci.2017.06.002
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Nomenclature

ρ	density [kg/m^3]
λ_f	fluid thermal conductivity [W/m K]
λ_s	solid thermal conductivity [W/m K]
α	thermal diffusivity [m^2/s]
δT	thermal boundary layer thickness [m]
δV	velocity boundary layer thickness [m]
ν	kinematic viscosity [m^2/s]
θ	dimensionless temperature [–]
L	heat source width [m]
T	temperature [K]
T_∞	ambient temperature [K]
q''	heat flux density [W/m^2]
H	heat source height [m]
Nu	Nusselt number [–]
Ra	Rayleigh number [–]
Y^*	dimensionless distance to the wall ($= y/H$) [–]
Z^*	dimensionless vertical coordinate ($= z/H$) [–]
$\tilde{Z}^*$	dimensionless conductive/convective length [–]
ΔX	horizontal spacing [m]
ΔZ	vertical spacing [m]
BL	Boundary Layer
PCB	Printed Copper Board
PIV	Particle Image Velocimetry
POD	Proper Orthogonal Decomposition
RMS	Root Mean Square

of a sealed cavity. Steady flow and temperature has been a commonly used assumption. Despite all the available literature, a lack of flow measurements is mentioned in [6]. Experimental setups have been limited to temperature measurements. Besides, most of the work has been performed for low Rayleigh number and laminar regime.

This present study provides experimental results of the combined temperature, heat transfer and velocity measurements for a system with discrete heat sources with large dimensions (e.g. typical for power electronics components) and open boundaries. The dimensions result essentially in higher Rayleigh number which determine the flow regime. Considering the unstable character of natural convection [18], the fluctuations of the flows were also studied.

2. Method

2.1. Experimental setup

Fig. 1 presents the experimental setup. All the tests were performed in a big glass chamber located inside an air conditioned room with a constant temperature. Nine rectangular heat sources were flush-mounted on one side of a vertical metal plate. The heat sources consisted of 3 rows and 3 columns of sources. An array of modular heat sources was designed in order to study the effect of the spacing between the heat sources (see Fig. 2). The other side of metal plate is directly in contact with air and is cooled essentially by natural convection. The plate was chosen to have a low emissivity (approximately 0.1) in order to limit radiative effects.

The heat sources consisted of printed copper board (PCB) with an electrical resistance of around 0.5 Ω. As shown in Fig. 2 (right), a thermal sheet was placed between the PCB and the plate. It was used to insulate electrically while enhancing the thermal contact. Its thermal conductivity was 7 W/m K. A piece of polyurethane foam was mounted in the back of the PCB with a conductivity of 0.028 W/m K. A PVC plate was added on the back of the polyurethane foam in order to apply a uniform mechanical pressure. Pressure was applied with two screws with approximately 0.4 N m on each (see Table 1).

The rest of the backside of the plate was covered by a set of modular pieces of thermally insulating materials. Extruded polystyrene foam, with a thermal conductivity of 0.031 W/m K, was used to obtain adiabatic conditions while still being able to change easily the position of the heat sources mounted on extruded aluminum bars along which they could be moved. This insulating material was chosen for its good mechanical properties.

All the heat sources were controlled by one single DC-supply unit. The power dissipated by each heat source was balanced by tuning the resistances and cables prior to the tests. A thermocouple of type T was installed at the centre of the heaters, to monitor the

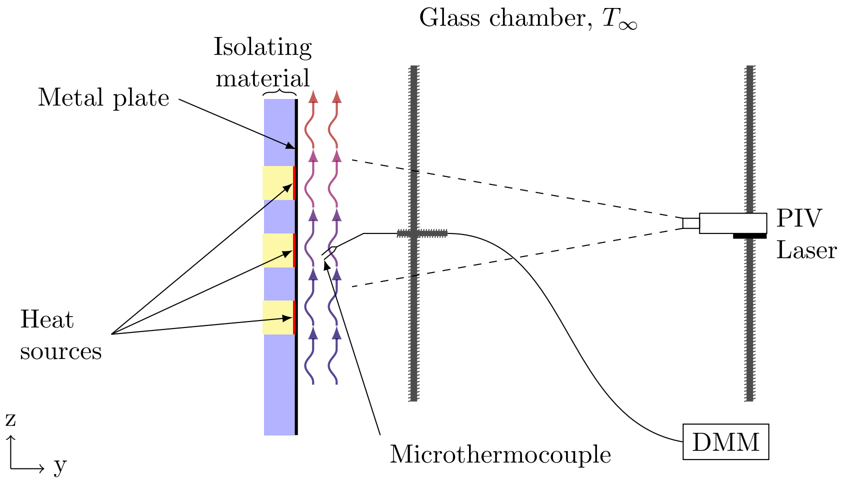

Fig. 1. Schematic view of a middle plane of the setup.