Forced Convection through Discrete Heat Sources and Simple Thermal Model – A Numerical Study

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Abstract: In this paper a forced convection heat transfer through discrete heat source assembly and simple thermal model placed inside the vertical channel is analyzed numerically. The problem domain considered consists of a vertical channel in which a heater plate discrete heat source assembly is placed at the center of the channel. The novelty of the present study is to replace the discrete heat source assembly by a simple thermal model for the uniformly distributed temperature and streamlines. A conjugate heat transfer analysis is carried out since the problem domain consists of aluminum solid strips as well as Bakelite strips in discrete heat source assembly which are replaced by a aluminum solid in case of simple thermal model. Initially the numerical results are compared with experimental results for discrete heat sources for the purpose of validation. The temperature of each heat source in discrete heat source assembly decreases with increase in inlet velocity of the fluid and bottom heat source is able to take higher heat load. The results in terms excess temperature obtained for both discrete heat source and simple thermal model is presented and discussed.

***Keywords*:** Vertical channel, Mixed Convection, simple thermal Model, Discrete Heat Sources

1. Introduction

The rapid growth of electronic industries and reduced size electronic components results in less amount of space for excessive heat dissipation. Hence the cooling of electronic components became essential for the improved life of the components, noteworthy operation and for its safety. In small scale industries components the natural convection is the preferable cooling method using natural air as working fluid whereas in case of higher electronic components the forced convection cooling method more preferred by using air or water or some other fluid as the cooling fluid. Mixed convection is most suitable to overcome this problem of forced convection in some extent and plays an important role in the advantages of both natural and forced convection. Mixed convection in vertical channels, tubes and ducts has been extensively investigated because of its applications to nuclear reactors, heat exchangers, electronic equipment and other areas of particular interests.

There are number of studies carried out for convection heat transfer in horizontal, vertical and inclined channels by many researchers. Chen and Chung [1] analytically studied the linear stability of mixed convection heater transfer in a vertical channel. The side wall of the channel is assigned with heat flux boundary condition. They analysed that the maximum amplifications rates for both gravity assisted and opposed flows was found insensitive for different temperature perturbations. Hadim and Chen [2] carried out a mixed convection analysis in a vertical channel filled with porous medium with discrete heat sources at the left wall. They reported that as Darcy number decreases the flow separation point remains the same but the reattachment point moved still downstream and the average Nusselt number increases with increasing Darcy number.

Premachandran and Balaji [3] numerically investigated heat transfer through protruding heat sources in a channel by keeping the size of channel, thickness of heat sources/substrate and spacing between the heat sources constant. They found that with increase in Reynolds number the maximum temperature and effect of radiation decreases. Hotta et al. [4] conducted steady state experiments with five discrete heat sources under mixed convection cooling and found that the maximum temperature decreases with increase in the value of λ (heuristic non-dimensional geometric distance parameter) and the optimal configuration occurs for the highest value of λ. Gururaja Rao et al. [5] performed a numerical study on mixed convection through a vertical channel in which the flush mounted heat sources are kept in the walls. The study includes the surface radiation emitted by the heat sources and concludes that the maximum temperature decreases with surface emissivity. Chen et al. [6] carried out numerical study of forced convection in horizontal channel partially filled with porous metal foam with discrete heat sources on the bottom wall. They found that the increasing the heat exchange between solid-fluid interface results in decrease in the excess temperature between the solid-fluid phases at a particular Reynolds number. Sankar et al. [7] numerically investigated the natural convection in a vertical channel filled with porous medium saturated by fluid with discrete heat sources. The result reveals that the heat transfer rate increases with increase in Rayleigh and Darcy numbers. It is also presented that the length of the heaters also influences on the maximum temperature and heat transfer in the annular cavity.

Hotta et al. [8] conducted experiments on protruding discrete heat sources to see the effect of surface radiation by natural convection heat transfer. They reported that the highest heat generating heat source should be placed at the bottom and they achieved 12% increase in heat transfer by applying black paint on the sources. Ghorab [9] investigated numerically forced convection heat transfer of passive heat exchanger for non-porous and partial filled porous channels with varying the exit height. The results reports that temperature decreases with decrease in exit height. They also concluded that the overall heat transfer performance decreases with increasing Reynolds number because of increase in pumping power. Kamath et al. [10] carried out experiments on high porosity metal foams filled in a vertical channel. They studied different pore density metal foams with different porosities. The results reveals that the higher pores per inch metal foam gives higher heat transfer with expense of pressure drop. Kotresha and Gnanasekaran [11] numerically studied the forced convection heat transfer through metal foams filled in a discrete heat source assembly placed vertical channel. They considered two different types of metal foam material for the investigation. They concluded that excess temperature decreases for all the heat sources with increase in inlet velocity of the fluid and bottom heat source can take higher heat load. They also reported that the copper metal foam does not show significant improvement in the heat transfer even after having high thermal conductivity. The isothermal condition on all the heat sources is achieved at a particular velocity for both aluminium and copper metal foams. Ahamad and Balaji [12] studied laminar mixed convection conjugate heat transfer through discrete heat sources by applying simple thermal model concept. They revealed that the geometric complexity can be reduced by applying simple thermal model concept instead of discrete heat sources which results in uniformly distributed heat in the domain.

Based on the above literature, it is found that the many experimental and numerical studies are carried out on channels by considering different types of heat transfer enhancement techniques. The numerical study on discrete heat source assembly placed inside the channel is studied very less and the studies on simple thermal model are even adequate in the literature. Hence, this work concentrates on fluid flow and temperature distribution in the channel for both discrete heat sources and simple thermal model. The results in terms of temperature contour, excess temperature obtained are presented.

2. Physical Geometry

The geometry considered consist of a discrete heat source assembly is placed at the centre of the vertical channel. The discrete heat sources assembly is a combination of aluminium and Bakelite strips. The aluminium and Bakelite strips are placed one after the other to make the assembly. A heater is placed in between two aluminium strips and acts as heat source and three aluminium heat sources are kept in the discrete heat source assembly. The size of the aluminium and Bakelire strips is 250 x 20 x 3 (all in mm) and 250 x 22.5 x 7 (all in mm). Figure 1 shows the schematic diagram of the physical geometry considered for the present numerical study.

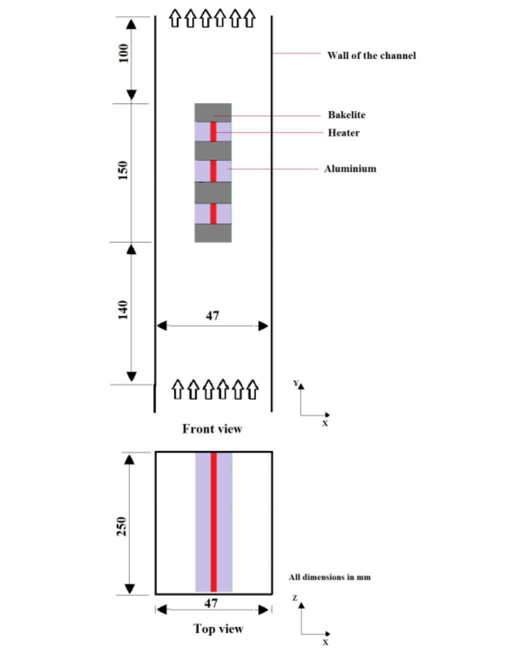

Fig. 1. Physical domain of the vertical channel

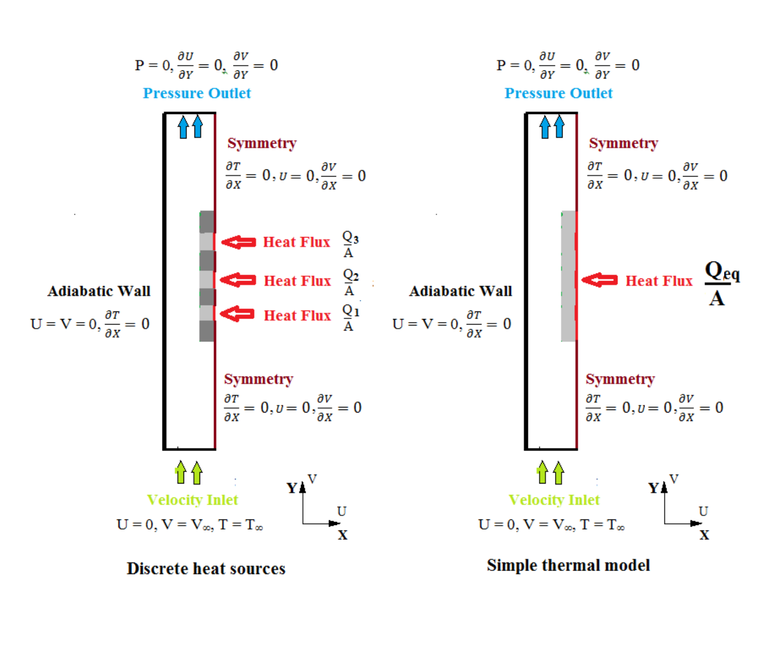

Fig. 2. Computational domain and boundary conditions

3. Computational domain and boundary conditions

As seen from Fig.1, it is clear that the vertical channel is symmetrical about the vertical axis. Hence a two dimensional computational domain is selected for further numerical computation as shown in Fig. 2. A uniform fluid velocity and zero pressure are defined at inlet and outlet of the vertical channel respectively. The side wall of the channel is defined as adiabatic wall and vertical axis is assigned with symmetry boundary condition. A known value of heat input is assigned for discrete heat sources.

4. Numerical details

The numerical computations are performed using commercially available ANSYS FLUENT. The governing equations used in the open region of the channel are similar to flow through pipe. Air is considered as the working fluid with constant properties taken at an inlet temperature of 30^0 C. The inlet velocity of the fluid is varied between 0.42 to 3.5 m/s, hence the Reynolds number based on hydraulic diameter varies from 2000 to 17000. The turbulent characteristic of the flow is captured using k-ω turbulence model. A conjugate heat transfer analysis is carried out for the computational domain since involves both fluid as well as solid domains. The coupled pressure velocity coupling is used with pseudo transient in time. A second order upwind scheme is used for pressure, velocity, energy and for turbulence parameters. The convergence criteria for continuity, momentum is set below $1e^{-5}$, energy is $1e^{-10}$ and turbulence parameters it is $1e^{-3}$.

5. Simple thermal model

The concept of simple thermal model proposed by Ahamad and Balaji [12] is used in the present study by replacing the discrete heat source assembly by a uniform heat generating volume. The thickness of simple thermal model is same as that of discrete heat sources. The net heat input to the heater for simple thermal model is given in Eq. (1)

$$q_{eq} = \frac{\sum_{i=1}^{n} q_i A_i}{A_{eq}} \quad (1)$$

4. Results and Discussion

4.1 Grid independence study:

Grid sensitivity analysis is carried out on the computational domain by selecting three different number grid sizes. Table 1 shows the grid independency results carried out in the present study, based on the results the grid size of 78080 is selected for further numerical computations as it shows less deviation in pressure.

Table. 1. Grid Independence Study

Grid Size	Maximum pressure (Pa)	Maximum temperature (K)	% Deviation	
			Pressure	Temperature
41,160	0.129	370	11.0344	0
78,080	0.14	370	3.448	0
1,11,693	0.145	370	Baseline	

4.2 Validation

The numerical results are compared with the experimental results for the purpose of validating the present numerical methodology. The present numerical result of excess temperature obtained for the bottom heater is compared with experimental results of Kamath et al. [10] and is shown in Fig. 3. The excess temperature results matches fairly well with the experimental results. This confirms the numerical methodology adopted in the present study is correct.

4.3 Thermal Results

Figure 4 and Figure 5 shows the temperature distribution obtained in the vertical channel for both discrete heat source and simple thermal model. The top heater receives higher heat in the case of discrete heat source case since the air moving to top carries heat from the bottom as well as middle heaters. The temperature distribution in the channel is not uniform but whereas in simple thermal model the temperature distribution is uniform through the aluminium strip. The excess temperature in the channel is less in the case of simple thermal model compared to discrete heat source. This confirms that the discrete heat source chips in the electronic components can be replaced by uniform generating chips for the smooth working of the components.

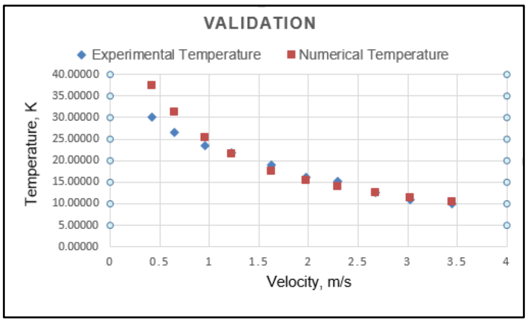

Fig. 3. Variation of excess temperature with inlet velocity

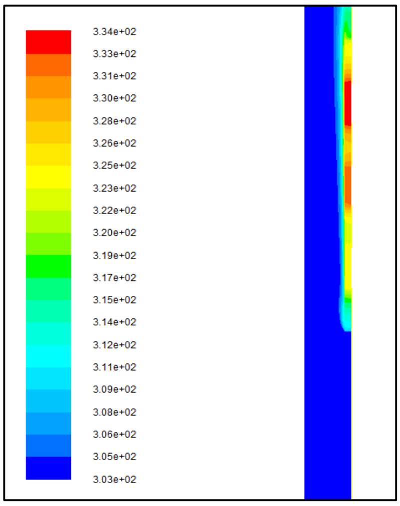

Fig. 4. Static temperature contours for discrete heat sources on vertical channel

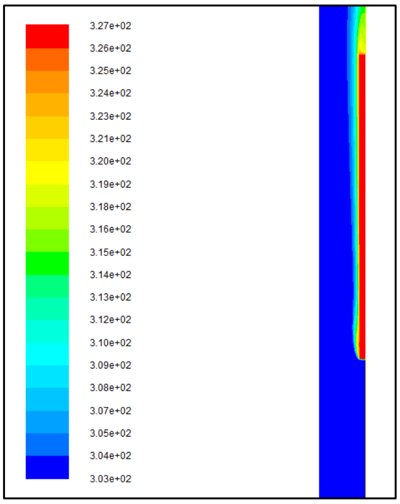

Fig. 5. Static temperature contours for simpler thermal module

The variation of excess temperature with Reynolds number for discrete heat sources and simple thermal model is shown in Fig. 6 and Fig. 7 respectively. The excess temperature shows a general decrease with respect to Reynolds number for both the scenarios. The bottom heater in the discrete heat source assembly can take higher heat load since it shows less excess temperature. The excess temperature obtained in the simple thermal model is almost lies between of excess temperature obtained at the middle heater and bottom heater.

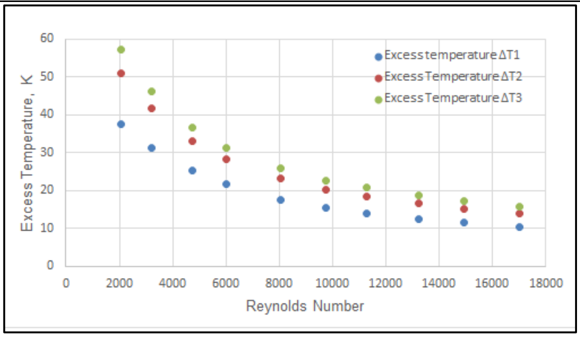

Fig. 6. Excess temperature variation for discrete heat sources

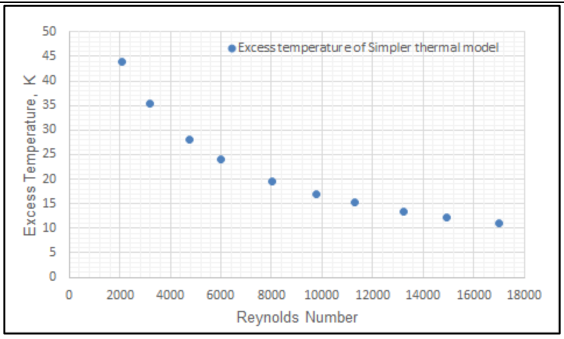

Fig. 7. Excess temperature variation for simple thermal model

Conclusion

A two dimensional numerical simulation is carried out on discrete heat sources and simple thermal model placed at the centre of the vertical channel. The discrete heat source assembly is combination of aluminium and Bakelite strips and only aluminium strip is used in the simple thermal model. The numerical simulations are carried out for a range of Reynolds number with equal heat input to the heaters. The numerical results are compared with experimental results for the purpose of validation of the methodology. The silent conclusions based on the study are

- The excess temperature obtained in the simple thermal model is less compared to discrete heat sources.
- The simple thermal model shows a uniform heat distribution in the channel whereas temperature distribution is not uniform in the case of discrete heat sources.
- The discrete heat sources can be replaced by a simple thermal model which gives less temperature and uniform temperature profiles.

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