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Experimental Investigation of Free Convection from Foam Heat Sinks in an Inclined Rectangular Channel

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Abstract. In this present study, free convection from an in-line 5x1 array of aluminum foam heat sinks which were placed on the bottom wall of the inclined rectangular channel was investigated. Aluminum foam heat sinks were made of Al-6101 alloy. All heat sinks mounted on a bottom wall of the channel was discrete form and exposed to uniform heat flux. Aluminum foam materials with three different pore densities (10, 20 and 40 PPI) were used in experiments. Air was used as coolant fluid (Pr =0.7). The experiments were made for Rayleigh Number range from 2.03 x10^7 to 1.33 x10^8 and the channel inclination angles were varied from 0° to 90°. Moreover, the heat transfer results from the aluminum foam heat sinks were also compared to the results obtained for foam-free (without foam) surfaces. The results showed that both the channel inclination angle and the usage of foam heat sinks are significant influences on heat transfer results.

***Keywords*:** Free convection, foam heat sinks, electronic cooling, inclined channel.

Eğimli Dikdörtgen Bir Kanalda Köpük Isı Alıcılardan Doğal Taşınımın Deneysel Olarak İncelenmesi

Özet. Bu çalışmada**,** eğimli dikdörtgen bir kanalda, 5x1diziliminde alüminyumdan yapılmış köpük ısı alıcılarından doğal taşınımla ısı transferi incelenmiştir. Alüminyum köpük ısı alıcılar Al-6101 alaşımında yapılmıştır. Kanalın alt duvarına monte edilen bütün ısı alıcılar ayrık formda yerleştirilmiş ve sabit ısı akısına maruz bırakılmıştır. Deneylerde farklı gözenek yoğunluğuna sahip (10, 20 ve 40 PPI) metal köpük malzemeler kullanılmıştır. Soğutma akışkanı olarak hava (Pr =0.7) kullanılmıştır. Deneyler Rayleigh sayısının 2.03 x10^7 ila 1.33 x10^8 arasındaki değerleri ve kanal eğim açısı 0° ila 90° arasında değişen değerleri için yapılmıştır. Ayrıca, alüminyum köpük ısı alıcılardan elde edilen ısı transferi sonuçları düz yüzeyler (köpük olmayan) için elde edilen sonuçlarla karşılaştırılmıştır. Elde edilen sonuçlar, hem kanal eğim açısının ve hem de köpük ısı alıcıların kullanılmasının, ısı transfer sonuçları üzerinde önemli etkileri olduğunu göstermiştir.

***Anahtar Kelimeler*:** Serbest taşınım, köpük ısı alıcı, elektronik soğutma, eğimli kanal.

1. INTRODUCTION

Computer technology is among the fastest growing technologies in the world. CPU's are the most important part of computers that act as managers. Processors are not only the hardware which found in computers but also found in all electronic systems besides computers. For example, automatic washing machine, automatic dishwasher, automatic devices situated in factories, televisions and telephones have processors. Since the processors are electronic circuit elements which are heavily operated, their temperatures are very high. A higher temperature than a certain temperature value causes damage to the processor. For this reason, cooling of the processor is a very important in such systems

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The cooling operation consists of two steps; the first is to reduce the heat of the processor by absorbing the heat on the processor. The second is to remove the absorbed heat from the processor. Even if the system automatically turns itself off when the processors become very hot, this technology may not always work, and the processor may burn out. Choosing an improper cooler assembly or improper mounting of the cooler assembly can sometimes result in minor temperature increases, causing system crashes, lockouts or even processor burns. The processor coolers (heat sinks) are usually manufactured from aluminum and copper materials. In addition to the aluminum coolers, there is also aluminum fin coolers mounted on a copper base on the processor. Conventionally CPU coolers are made up of aluminum fins. Today, a wide variety of cooling methods and different fin types are used to cool the electronic systems. In recent years, open celled metal foam materials have been investigated that have superior heat transfer properties instead of fins. Because of low weight, high heat transfer area and high cooling capacity make the metal foams more attractive for use in various industrial areas. Because of the internal structures are like network, having a large surface area to volume ratio, and provide a better heat transfer by mixing the fluid flow, metal foams are also seen as a preferred heat transfer enhancement method.

Convection heat transfer over electronic elements placed on circuit boards has been the subject of many articles for the last 15 years. To summarize briefly the studies on this subject are;

Xu et al. [1] conducted experimentally free convection in horizontally positioned copper metallic foams with open cells. They have investigated how the total thermal resistance is affected by porosity and pore density. It has been seen from the results the porous surface can augment the natural convection and reduce the thermal resistance approximately 20% in comparison with a non-foam case.

Natural and mixed convection from discrete heat sources mounted near the Bottom on a Printed Circuit Board (PCB) were studied by Hotta and Venkateshan [2]. Experiments have been conducted for 5 protruding discrete heat sources having differently sized placed on a PCB and placed on a vertical channel. The results were obtained for different Grashof and different Reynolds numbers in steady state conditions. The effects of Reynolds number and Richardson number on surface temperature and Nusselt number have been discussed.

Habib et al. [3] studied experimentally the natural convection over an array of discrete surfaces placed on a flat vertical board. The study covered the case where the discrete surfaces/substrate unit forms a wall of a closed cavity. It was also examined that how the Rayleigh number affects the velocity and flow fields inside the cavity. The uniform and non-uniform heat source distributions were considered as well.

Durgam et al. [4]. investigated both free and forced convection effects by positioning a substrate plate horizontally and vertically. Experiments were conducted for three different heat flux values of 1500, 2000 and 2500 W/m^2. The experiments were conducted for these heat fluxes at air velocity values of 0.27 m/s and 0.45 m/s.

The natural convection heat transfer in horizontal channels partially filled with aluminum foam heated from below was investigated by Buonomo et al. [5] experimentally and numerically. A horizontal channel with and without foam case conditions was compared. Air was used as working fluid. The effects of different types of metal foam materials on heat transfer were investigated. One from “M-pore”, with 10 and 30 PPI, and the other one from “ERG”, with 10, 20 and 40 PPI. The numerical model was investigated by a simplified two-dimensional simulation.

Experimental study of air free convection on the metal foam-sintered plate at different tilt angles was investigated by Qu et al. [6]. The experimental study was made for 7 copper foam materials with 10–40 PPI, porosities of 0.90 and 0.95 and different aspect ratios. The tilt angles were varied from 15^0 to 90^0 with increments of 15^0.

Deng et al. [7] simulated the steady natural convection induced by multiple discrete heat sources in two-dimensional horizontal enclosures. They used a general combined temperature scale method to define the DHSs of both external and internal types. Four different calculation cases were analyzed and the effects of the Rayleigh number, the thermal strength, and the separation distance on the interaction between DHSs on heat transfer were investigated.

Awasarmol and Pise [8] analyzed experimentally the free convection from solid and permeable fins and the influence of tilt angle of fins on free convection heat transfer as well. This study was made for to compare the heat transfer rate with solid and permeable fins and the influence of the angle of fins inclination. The solid and permeable fin blocks were placed in an isolated chamber in order to ensure natural convection conditions. They compared free convection heat transfer results obtained for the base and the tip in steady state temperature conditions.

The natural convection in an open-ended inclined channel-partially filled with porous media was numerically investigated by Kiwan ve Khodier [9]. Numerical simulation was made under steady-state, laminar, two-dimensional flow condition. The Darcy-Brinkman-Forchheimer model together with Boussinesq approximation was used to define the fluid flow in the porous domain. The Darcy-Brinkman-Forchheimer model was used in conjunction with the Boussinesq approach to describe the fluid flow in the porous domain. The Navier-Stokes equation was used with the Boussinesq approach in order to define the flow region outside the porous region.

Oztop [10] numerically simulated natural convective flow an inclined rectangular enclosure filled with porous medium. The numerical study was made for finite difference method using the SIMPLE algorithm. The Darcy-Rayleigh number was varied from 10 to1000, the center location of the heater was varied from 0.1 to 0.9, inclination angle $0° \leqslant \phi \leqslant 90°$ and length of cooler $0.25 \leqslant w \leqslant 0.75$. The obtained results showed that the inclination angle is the very important parameter on heat transfer rate and fluid flow and also the aspect ratio.

The study of natural convection from an array of 3x3 discrete heaters mounted in a vertical wall of a 3-dimensional rectangular cavity was examined numerically by Saravanan et al [11]. The finite volume method was used for the numerical solutions. Flow and heat transfer properties were analyzed according to Rayleigh number (Ra), aspect ratio (A) and Prandtl number (Pr).

Dogan et al. [12] experimentally conducted natural convection from aluminum foam blocks placed in a volume. The aluminum foams having different pore densities (10, 20 and 40 PPI) were used in experiments. The 3x3 array of foam blocks subjected to uniform heat flux were placed horizontally at the bottom of the volume. The experimental study was made for modified Rayleigh number $8.33\ x10^5 \leq Ra^* \leq 5.24x\ 10^6$ and the foam height varied from 10 to 30 mm. The effects of aluminum foam block height and pore density on Nusselt number were determined.

De Schampheleire et al. [13] were investigated experimentally natural convection flow in open-cell aluminum foam heat sinks. The effects of the pore density, the foam height and the bonding method were examined for Rayleigh numbers ranging between 4000 and 6500. The effects of the pore density, the foam height, and the bonding method were examined for Rayleigh numbers ranging between 4000 and 6500. The length-to-width ratio of the substrate was kept on 10. The foams having 10 and 20 PPI pore density and 93% porosity were examined. The height of 10 PPI foam ranged from 6 to 40 mm, while that of the 20 PPI foam ranged from 6 to 18 mm.

In the extensive literature survey mentioned above, heat transfer studies with metal foam heat sinks placed in the inclined channel and in discrete form were not found. In this study, free convection from open-celled metal foam made of Al-6101 alloy material heat sinks having different pore densities mounted in a discrete form in an inclined rectangular channel was investigated experimentally. Experiments were conducted for both foamed and non-foamed conditions and the obtained heat transfer results were compared.

2. MATERIAL AND METHOD

In this study, the free convection heat transfer in an inclined channel with heat-dissipating elements, which represent electronic components and whose surfaces are extended with aluminum foam heat sinks, have been experimentally investigated. Air was used as a refrigerant. The experimental setup for examining the natural convection heat transfer in the inclined channel is given in detail in Figure 1. The whole circumference of the rectangular channel is completely insulated. The upper wall, side walls, lower entrance wall and exit area of the channel are made of plexiglass material with a thickness of 5mm. The whole channel size is 60x60x485mm.

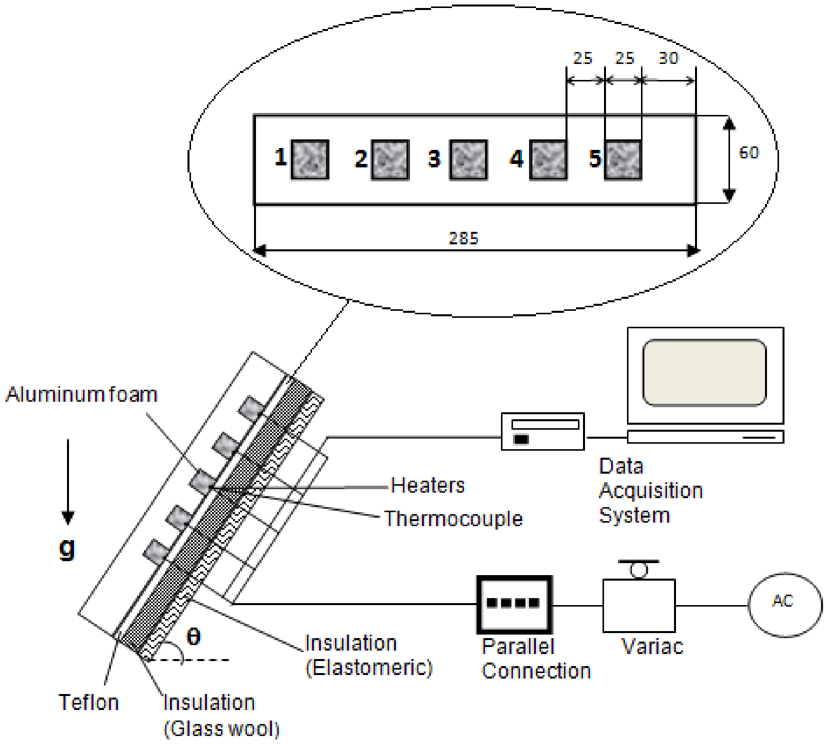


Figure 1. Schematicdiagram of the experimental set-up, heat source layout and assembly (measures are in mm).

The bottom wall of the channel is made of 5 mm thick Teflon substrate (k = 0.25 W/mK). The copper plates which represent the electronic chips used as heat sources are embedded in teflon substrate. The dimensions of copper plates were 25x25x5 mm. The square shape holes in which copper plates were placed in Teflon material were cut with an industrial laser. The photograph of with and without foam case in test section is given in Figure 2.

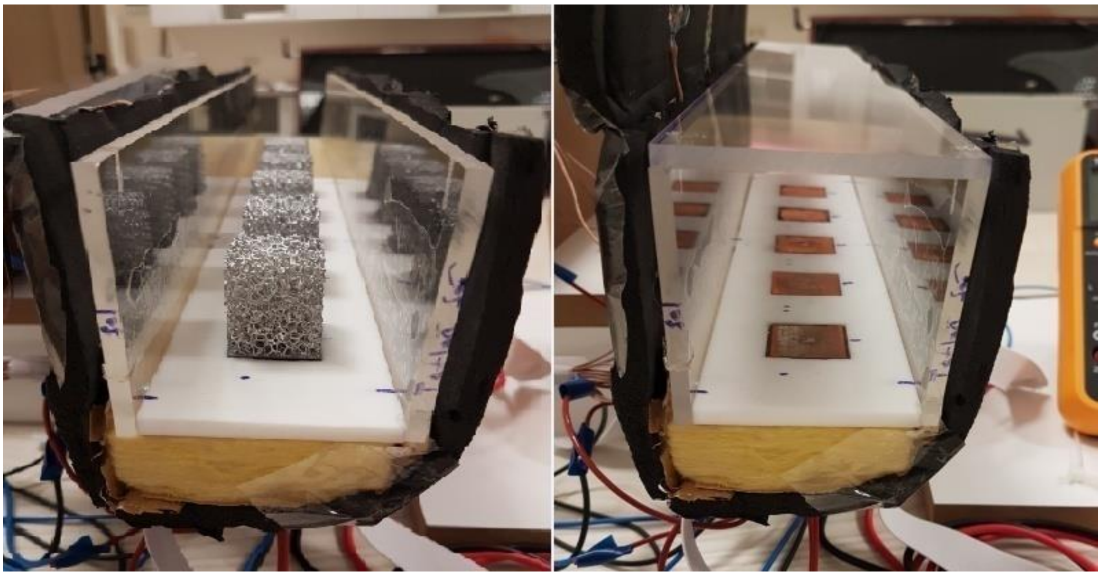


Figure 2. The view of the test section with and without foam case (only copper plate).

The electrical resistances, copper plates and aluminum foam heat sinks have the same base area. Thermal paste with high thermal conductivity (k= 8.5 W/mK) is applied between copper plates and foam blocks to minimize thermal resistance. The 30-gauge T-type thermocouples soldered to the exact center of the copper plates and connected to the data acquisition system to measure the heater temperatures. The bottom of the rectangular channel in which the heaters located was insulated with 20-mm Glasswool (k=0.040 W/mK). The entire circumference of the rectangular channel is insulated with elastomeric rubber material with a thickness of 10 mm. (k=0.034 W/mK). The temperature values were measured using thermocouples from total of 22 points. Five of these were soldered to copper which plates used as heat source. The rest were used at the inlet, outlet, ambient air and several locations on the top and side walls of the insulation applied. The parallel connection circuit via a variac was used to supply the electric current to the heaters. Electrical power was determined by measuring the voltage drop and resistance across the heater plate. A Goldstar multimeter with a voltage accuracy of ± 0.5% and a resistance accuracy of ± 0.2% was used. During the experiments, it was seen that experimental conditions achieve a steady-state condition after about 3–5 h. After reaching the steady condition, differences in temperatures between two intervals became negligible (ΔT< 0.1 0C) and all measured temperature values were recorded. The test procedure and measurements are similar to the experimental method that Dogan et al. [12].

Experiments were carried out in 2 different stages:

1.Without foam heat sinks for 4 different channel angles (0°, 30°, 60°, 90°),

2.With foam heat sinks for 4 different channel angles (0°, 30°, 60°, 90°),

Photographs of Al-6101 foam heat sinks having different pore densities used in the experiments are shown in Figure 3. Table 1 gives the physical properties of foam heat sinks.

The porosity and PPI values of the aluminum foam heat sinks were determined by the manufacturer (Duocel, ERG Materials and Aerospace Corporation, Duocel, Oakland, CA). The permeability values of aluminum foam heat sinks are calculated from Darcy's law [14].

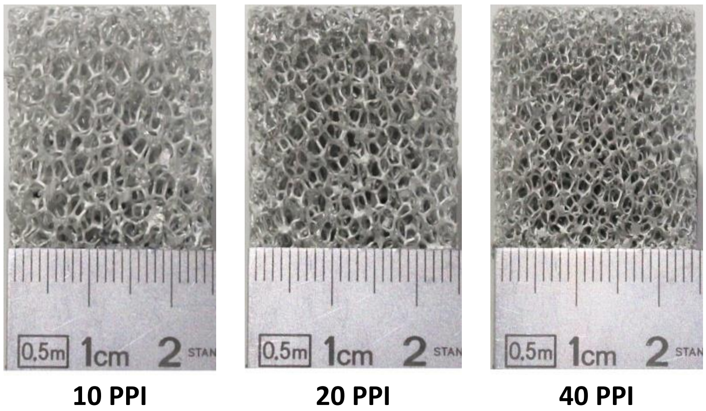


Figure 3. Pictures of aluminum foam heat sinks with various PPI values.

Table 1. Particulars of the Aluminum-foam heat sinks**.**

Specimen	Porosity (ε)	Pore density (PPI)	Permeability(m^2)	Height (m)
1(Al-6101)	0.912	10	$7.73x10^{-8}$	0.03
2(Al-6101)	0.906	20	$4.93x10^{-8}$	0.03
3(Al-6101)	0.914	40	$2.40x10^{-8}$	0.03

3. CALCULATING PROCEDURE OF EXPERIMENTAL DATA

The measurements obtained from the experimental set up to examine the heat transfer from 5 aluminum heat sinks arranged in a single row in an inclined channel are temperature, voltage and electric current.

Using these values which obtained, the Nusselt number for each heat sink and Average Nusselt number was calculated as following equations:

$$Nu = \frac{Q_{conv}\, D_h}{A_s(T_s - T_0)k_{air}} \qquad (1)$$

Nu: Nusselt number

Q_{Conv}: Convection heat transfer (W)

D_h: Hydraulic diameter (m)

A_s: Surface area (m^2)

T_s: Surface temperature (^{0}C)

T_0: Ambient temperature (^{0}C)

k_{air} :Thermal conductivity (W/m K)

$$Nu_{avg} = \frac{Nu_{tot}}{5} \qquad (2)$$

Nu_{avg} = Average Nusselt Number of the heaters

Nu_{tot} = Total Nusset Number of the heaters

D_h is the channel hydraulic diameter calculated as below;

$$D_h = \frac{4A}{P} \qquad (3)$$

A: Channel cross-area (m^2)

P: Channel perimeter (m)

Q_{conv} is the amount of heat transferred by convection from foam heat sinks to air and calculated as follows:

$$Q_{Conv} = Q_{Elect} - Q_{Cond.tot} \qquad (4)$$

$Q_{Cond,tot}$: Total conduction heat transfer (W)

Q_{Elect}: Power dissipation of each heater (W)

$Q_{Cond,tot}$ is the total heat loss through the conduction from the test section surfaces. Q_{Elect} is the electric power delivered to each heater and is calculated as follows:

$$Q_{Elect} = \frac{V^2}{R} \qquad (5)$$

V: Voltage (V)

R: Electrical resistance (Ω)

V is the voltage drop across each heater and R is the resistance of each heater. The accuracies of V and R are ±1.3and ±2 respectively. Conduction loses from each test section surface were calculated from the following equation by using the temperature measurements from thermocouples which were symmetrically located on the inner and outer surface of the insulation materials.

$$Q_{Cond} = -k_{ins} A_{ins} \frac{\Delta T_{ins}}{L_{ins}} \quad (6)$$

Q_{Cond}: Conduction heat transfer (W)

k_{ins} :Thermal conductivity of the insulation (W/ m K)

A_{ins}: Insulation area (m^2)

ΔT_{ins}: Temperature difference (^0C)

L_{ins}: Thickness of the insulation (m)

Heat transferred by radiation is negligible due to the low temperatures in this study. The conduction losses in insulation materials are calculated and found to be approximately16% of the total heat transfer rate.

The dimensionless number affecting the heat transfer is Rayleigh number and can be calculated as shown below:

$$\mathrm{Ra} = \mathrm{Gr}^*.\mathrm{Pr} \quad (7)$$

Ra: Rayleigh number

Pr: Prandtl number

Gr: Modified Grashof number

Modified Grashof number can be calculated as shown below:

$$\mathrm{Gr}^* = \frac{g.\beta.q_c.D_h^4}{k_{air}.v^2} \quad (8)$$

g: Gravitational acceleration (m s^{-2})

β: Thermal expansion coefficient (K^{-1})

q_c: Average convection heat flux (W/m^2)

ν: Kinematic viscosity (m^2 s^{-1})

The uncertainty analysis was conducted to determine the reliability of the experimental results.

The calculations were accomplished in the standard procedure according to the literature (Holman J. P. 1994 [15]).

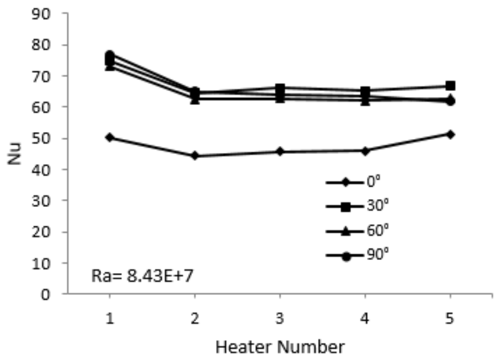
From the calculations, the uncertainty in heat addition from the electrical resistances is ±6.3%, for Nusselt number it is approximately ±4.9% and for Rayleigh number it is around ±4.1%, which is primarily due to uncertainties in the convective heat flux values.

4. RESULTS AND DISCUSSION

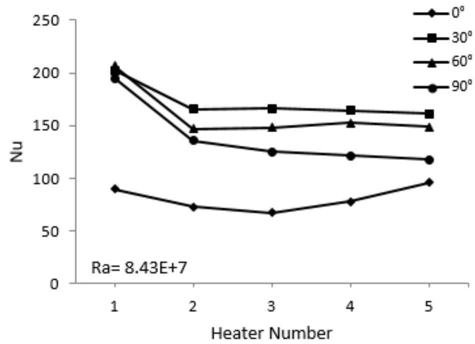
The free convection heat transfer from an in-line five foam heat sinks with pore densities of 10, 20 and 40 PPI has been investigated experimentally inside an inclined rectangular channel. The experimental study was made for inclination angles varying from 0^0 to 90^0, while the Rayleigh number ranged from $2.03x10^7$ to $1.33x10^8$. The results obtained without aluminum foam case were compared that the results with foam case.

In Figure 4, the effect of channel inclination angle on Nu number for Ra= $8.43x10^7$ is presented for without foam and with foam heat sinks having different pore densities of 10, 20 and 40 PPI. As can be seen from the Figure 4(a) under the horizontal orientation (θ=0) Nu number exhibited symmetrical distribution and minimum values were obtained at the middle heaters in the channel. This behavior can be explained that the convection heat transfer in the horizontal position is a Rayleigh–Bénard convection qualified by a fluid that moves up through the center of the Bénard cell. This resulting cell causes a symmetric Nu distribution. Maximum values of Nu numbers were reached at 1stand 5th heaters. In the case of increasing inclination angle, 1st heater took the maximum value of Nu number. Between 2nd and 5th heaters, Nu number approached approximately constant values because of the boundary layer development. Overall, when compared to the channel inclination angle of 0^0, it was observed that Nu number increased for channel inclination of 30^0, 60^0 and 90^0.

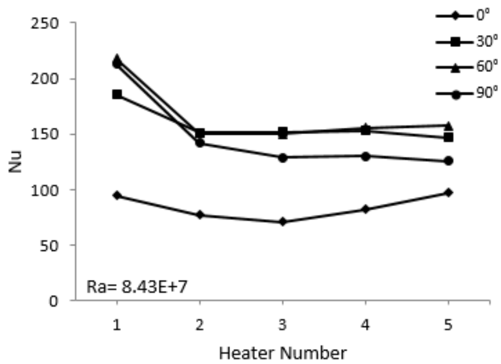
When metal foam heat sinks are placed on the heaters, the heat transfer rate under the same conditions is considerably increased (Figure 4(b)-(d). From the figures, Nu values obtained for with foam heat sinks are higher than that of the without foam heat sink case for all inclination angles. The Nu number has the minimum value in the center foam heat sink because of symmetrical distribution for $\theta=0^0$. The highest Nu number for 10 PPI was obtained at the angle of 30^0. However, as the channel inclination angle further increases, the heated air moving from downward to upward under the influence of buoyancy effect did not supply enough cooling effect on the foam heat sinks. As a result, minimum heat transfer rate is obtained at 90^0 for 10, 20 and 40 PPI. For the angle of 30^0 and 60^0, Nu number values of 20 and 40 PPI foam heat sinks were seen to approach each other along to 2nd, 3rd, 4th and 5th heaters.



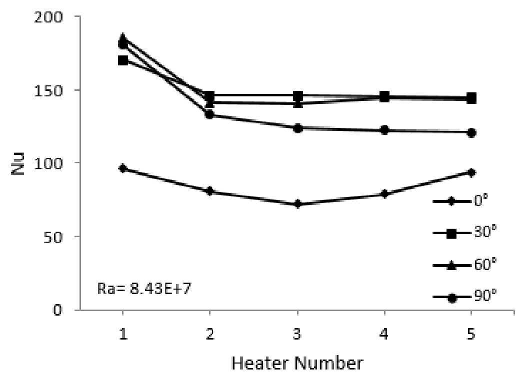
a) Without foam



(b) 10 PPI



(c) 20 PPI



(d) 40 PPI

Figure 4. Effect of channel inclination angle on Nusselt number for Ra= $8.43x10^7$(a) without foam, (b)10 PPI,(c)20 PPI,(d)40 PPI

In Figure 5, the effect of channel inclination angle on the average Nu number for Ra= $8.43x10^7$ is shown for with and without foam heat sink. When the heat transfer results for foam heat sink case compared to without foam heat sink, a significant increase in heat transfer was observed. The average Nu number is increased approximately 2.3 times in case of using foam heat sink. The maximum value of the Nu number is obtained for a pore density of 10 PPI and inclination angle of 30^0.

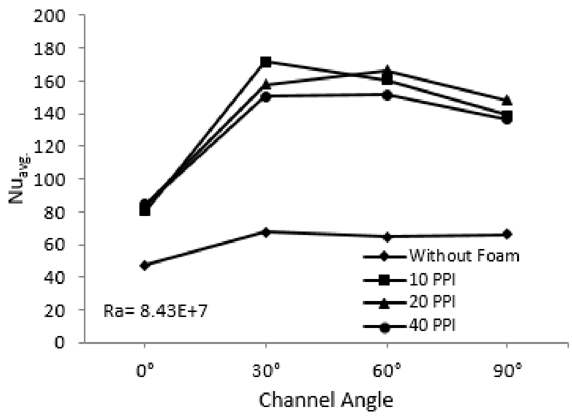


Figure 5. Effect of channel inclination angle on the average Nusselt number with and without foam heat sink for Ra= $8.43x10^7$.

The variation of average Nu numbers according to different Ra numbers in case of channel inclination angle of 30° is also shown in Figure 6. As is clear from the figures, the average Nu number increases with the Ra number increases. At high Ra numbers, the effects of free convection flow on Nu number becoming more important. When the obtained Nu numbers are evaluated together, it was seen that due to its large surface area to volume ratio and strong mixing of fluid flow, the use of foam heat sinks can an important role in increasing heat transfer rate.

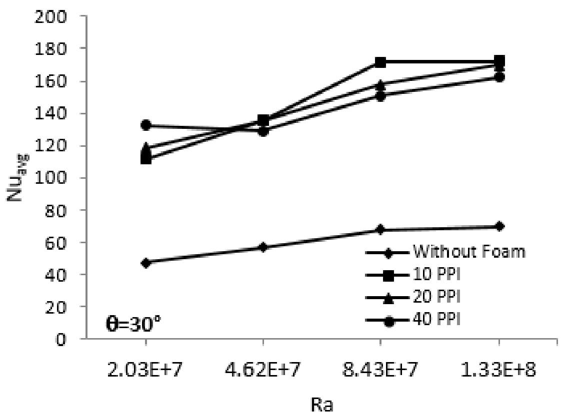


Figure 6. Variation of mean Nusselt numbers according to different Rayleigh numbers in case of channel inclination angle of 30°

5. CONCLUSION

The free convection from discrete heat sources extended their surfaces by Al-6101 heat sinks with pore densities of 10, 20, and 40 PPI has been investigated experimentally in an inclined rectangular cross-sectional channel. Experiments made for 5x1 array of aluminum foam heat sinks subjected to uniform heat flux placed on the bottom wall of the channel. The results showed that heat transfer increases with increasing channel inclination angle. It was also found that the heat transfer results obtained by using the foam heat sinks were 2.3 times better than the results obtained without foam heat sink case. When the results assessed according to the averaged Nusselt numbers ($Nu_{avg.}$), the highest heat transfer was obtained for 10 PPI, when the inclination angle was 30^0. Consequently, the use of metal foam can be seen as a good heat transfer enhancement method in electronic cooling systems.

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