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Experimental Investigation of Forced Convection Heat Transfer in Open Cell Copper Fins

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Abstract

Recently, major part of convection heat transfer researches focus on increasing fins efficiency by increasing thermal performance for the same fin volume. Metal foam is a promising way to achieve this aim. Performance analysis has been carried out to investigate the heat transfer characteristics of copper fin foam samples. The samples have been compared with the solid metal fin heat transfer. A forced convection heat transfer had been applied to a four specimens. An electrical heater heats up the fins, which are subjected to a stream of the ambient air driven by a blower fan as heat dissipated. The heat flux had been fixed along the tests with three different air velocity used; the forced heat convection had been simulated. The pores density of copper fin foam is varied in the range of 10, 20 & 40 pores per inches (PPI). Thermal performance of copper fin foam has been evaluated in terms of average Nusselt number and thermal resistance of heat sinks. The increasing in the heat transfer rate and average Nusselt number when used metal foam has been found in the range of 36-133 % compare to solid copper. Furthermore, it has been proven that this increment reaches the maximum value for a given PPI even when raise the air velocity.

Keywords: Fin, pores per inches, Metal foam, heat convection, thermal resistance.

Introduction

Metal foams are a metrical composed of gaps linked to solid material in in structured style. They are a kind of porous mediums [1]. It may be better to use porous mediums in heat exchangers due to the high surface area to volume of them compared to the solid materials. This property is very important in convection heat transfer [2]. Moreover, in forced convection, the air passing through the gaps will be exposed to mixing process due to winding the path to be followed. This will increases the heat transfer rate by convection.

Many engineering fields have been used metal foam in their applications especially in heat transfer purview. In the Heating, Ventilating and Air Conditioning (HVAC) industry, compact equipments are quickly developed. The heat Exchangers, which are the main parts in HVAC systems, have to redesign in new and innovative method. Metal foams with an open-cell structure have physical and mechanical properties promising that might have advantages over conventional fin materials for use in air-cooling heat exchangers. Metal foams are low weight, massive surface area to volume ratio hit 10000 m2/m3, great thermal conductivity and high gas permeability. (for open-cell bodies). Due to these properties, open-cell metal foams are currently regarded as a highly promising material for constructing efficient compact heat exchangers [3].

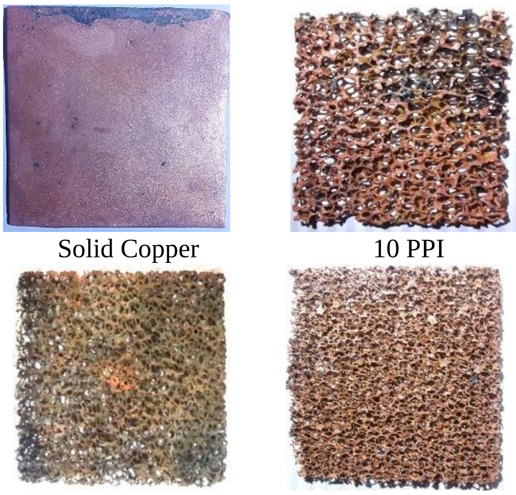
Most of air cooled type of heat exchangers has limited ability due to thermal resistance on the air side [4]. However, this type of heat exchangers is very common in different fields of industry for a long time. Therefore, the airside capabilities have been taken the more attention. Actually, the airside performance can be enhanced if the surface area touching the air stream can be increased or the air stream velocity increased. Nevertheless, there is a unusual way to improve the air side heat transfer, which is using fins of high thermal conductivity and porous material. Porous medium increases the area contacts the passed stream air.

Kobus and Oshio [5] studied how thermal performance of the pin array type of fins effected by thermal radiation. Rao et al.[6] numerically studied the whole heat transfer of N-array type of fin. Thermal radiation is also had been taken into account. Gorla and Bakier [7] Theatrically investigated the thermal performance of porous fins subjected to natural heat convection and thermal radiation. A mathematical model had been derived to simulate the problem case, which consist of non-linear second order ordinary differential equation. They found that radiation and convection heat transfers more than that dissipates heat by convection only. Nawaz et al. [8] experimentally studied forced convection for different pores per inches (PPI) aluminum foams with assorted base metals and various geometry. Esfahani et al. [9] obtained temperature and velocity distributions for heat exchangers of tube type using high porosity metal foam. They show the high improvement in the thermal performance of the heat exchanger when used metal foam inside it. Kundu and Lee [10] theoretically studied the temperature distribution for annular step porous fins (ASPFs) for the continuous movement of fin material. They used double differential transform method to achieve the objective. The analysis was formulated in a generalized form, which is applicable for solid fins also. They found that the porous fins have an ability always to transfer more heat compared to the solid fin for an equal mass of fins at an optimum condition.

In the present study, the thermal performance of copper foams heat sinks for forced air cooling has been investigated. The impact of a pore density in copper foam structure as well as the inlet flow velocity will be studied. Comparison of cooling performance with those of classical heat sink of parallel plate type will be implemented too.

Equipments and experiments

The samples used in this experimental work were Duocel® copper foam from ERG Aerospace Corporation with 10, 20 and 40 PPI pore densities and compare with solid copper fin. All used samples dimensions were 100 mm ×100 mm × 5 mm (H × W × D). Figure (1) shows a pictures of the test samples.



Solid Copper 10 PPI

20 PPI 40 PPI

Figure 1: The different PPI tests samples

The experiments test section is DIDCATA ITALIA® model T110D. As shown in Figure (2), it is 45 cm diameter circular duct with sight glass. The duct attached firmly to air rectifier, which is used to rectify the forced air comes from centrifugal fan with speed regulator. The overall dimensions of the system are 2450 mm long × 480 mm wide × 1400 mm height.

This system can provide the suitable environment to inspect the fin samples. The centrifugal fan gives the forced air in the suitable velocity required to simulate the specific forced heat convection. Using the regulator, the speed of the electrical motor of the centrifugal fan can be changed. The variation in the motor speed gives the produced air its required velocity.

Figure 2: Air duct used in the experimental work

Pitot static tubes have been use to determination static and dynamic pressures for the airflow inside the duct. It fixed in the centerline of the air duct. Anemometer was used to measure air velocity. The samples were heated with 10 cm ×10 cm heaters. Nine type T thermocouples were fitted tightly in the sample in a matrix arrangement as shown in Figure (3). The samples were fixed on an electrical heater and were installed in the air duct to facing the passing air. A voltage regulator controlled the power of the electrical heater to give the suitable base temperature with the feedback of a thermocouple in the base of the fin. All the thermocouples reading are transferring by thermocouples wires to data logger, which stores all the collected data on a memory card. Figure (4) shows a picture of the test apparatus.

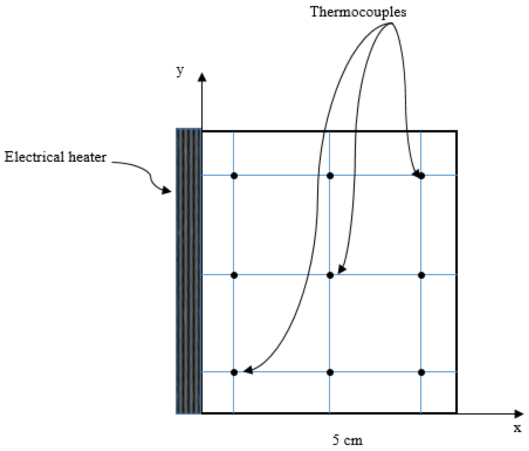


Figure 3: Schematic diagram of the specimen

Figure 4: Test apparatus

A copper foam heat sink is mounted on a heat source that emits a uniform heat flux. Channel wall is thermally insulated, and the blower imposes a uniform airflow at the channel inlet. The schematic diagram of the test rig are depicted in Figure (5)

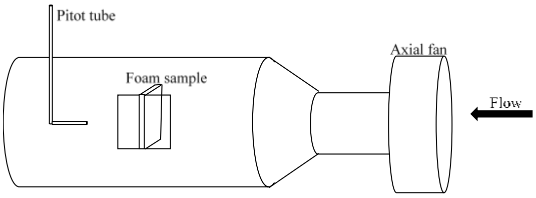


Figure 5: Schematic diagram of the test rig

To calculate the effects of forced air velocity on the thermal performance of the samples, three air velocity values have been tested. The three samples of different PPI with the solid material sample have been examined to investigate the effects of PPI on the thermal efficiency of the fins.

If k_{eff} is the effective thermal conductivity, L is the sample length, then the verage Nusselt is calculated as:

$$\overline{Nu} = \frac{\bar{h}L}{k_{eff}}$$

where $\bar{h}$ is the average heat transfer coefficient which is obtained by:

$$\bar{h} = \frac{q}{A\Delta T_{avg}}$$

where ΔT_{avg} is the average of fin base temperature, which in turn can be calculated by [11]

$$\Delta T_{avg} = \frac{\sum_{i=1}^{3} T_{base}}{3} - T_f$$

where T_f is the temperature of the air at directly front of the metal foam.

Many studies calculate the effective thermal conductivity for porous media,

In porous media field, the effective thermal conductivity draw attentions of several researchers. Furthermore, for estimating the thermal conductivity in metal foams there are a few researchers deal with. Calmidy and Mahajan [12] and Boomsma and Poulikakos [13] used the following formula

$$k_e = (1-\varepsilon)k_s + k_a$$

where k_s is the solid material conductivity and k_a is the thermal conductivity of the air

Results and discussions

In this paper, forced convection heat transfer with air flow across open cell copper fins foam was experimentally studied. The copper foam was with pore densities of 10, 20 and 40 PPI and the last test was with solid copper fin for comparison. Temperatures over the fin surface were measured at different airflow velocity. Temperature variation along x-direction for open cell and solid copper fins are presented in Figure 6.

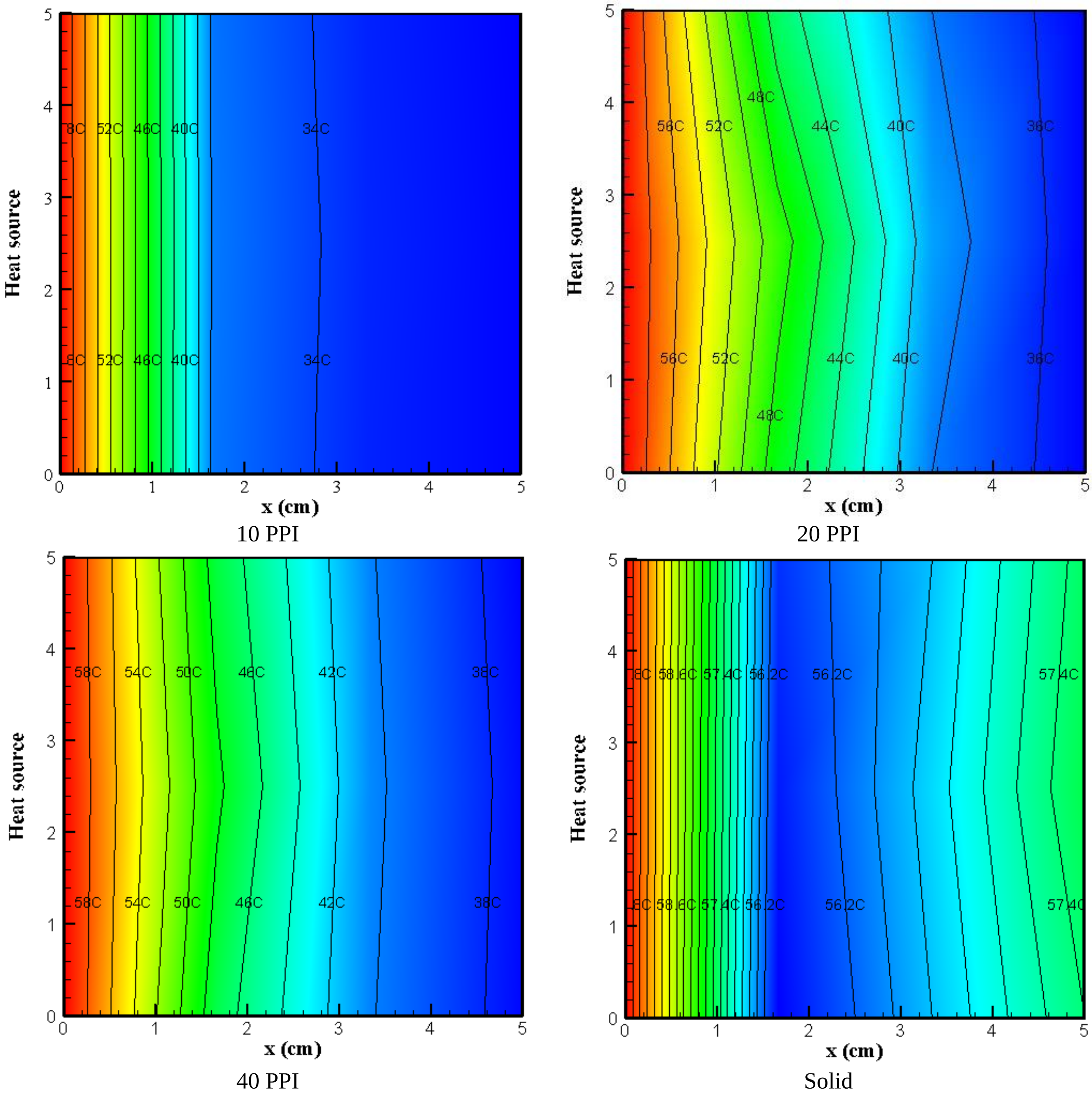


Figure 6: Variation of Temperature along x- axis for 10, 20, 40 PPI copper foam and solid copper fin at a speed of 0.5 m/s.

At air velocity of 0.5 m/s for different foam density. It can be obviously seen that when PPI increases the temperature difference between the fin base and tip decreases, this mean that the increasing of PPI leads to increasing in the rate of heat transfer. Actually, this reflects the fact that PPI increasing leads to increasing in surface area, which will results in increasing in the heat transfer rate.

Same phenomena can be noted in Figures (5) & (6) which also represent the variation of the fin surface temperature for different foam density but at air velocity of 1.0 m/s for Figure (7) and 1.5 m/s for Figure (8). From these figures, it can be found that increasing in air velocity for same PPI causes a decreasing of temperature difference in amount consider small comparing with the effect of increasing of PPI.

Comparisons of the three porous copper fins with solid copper show that the temperature drops are much better for copper foam than that for solid copper fin due to the increasing in the surface area. This leading in turn to increases the interval at which air stays in contact with the hot surfaces and thereby increasing the amount of heat the air takes away.

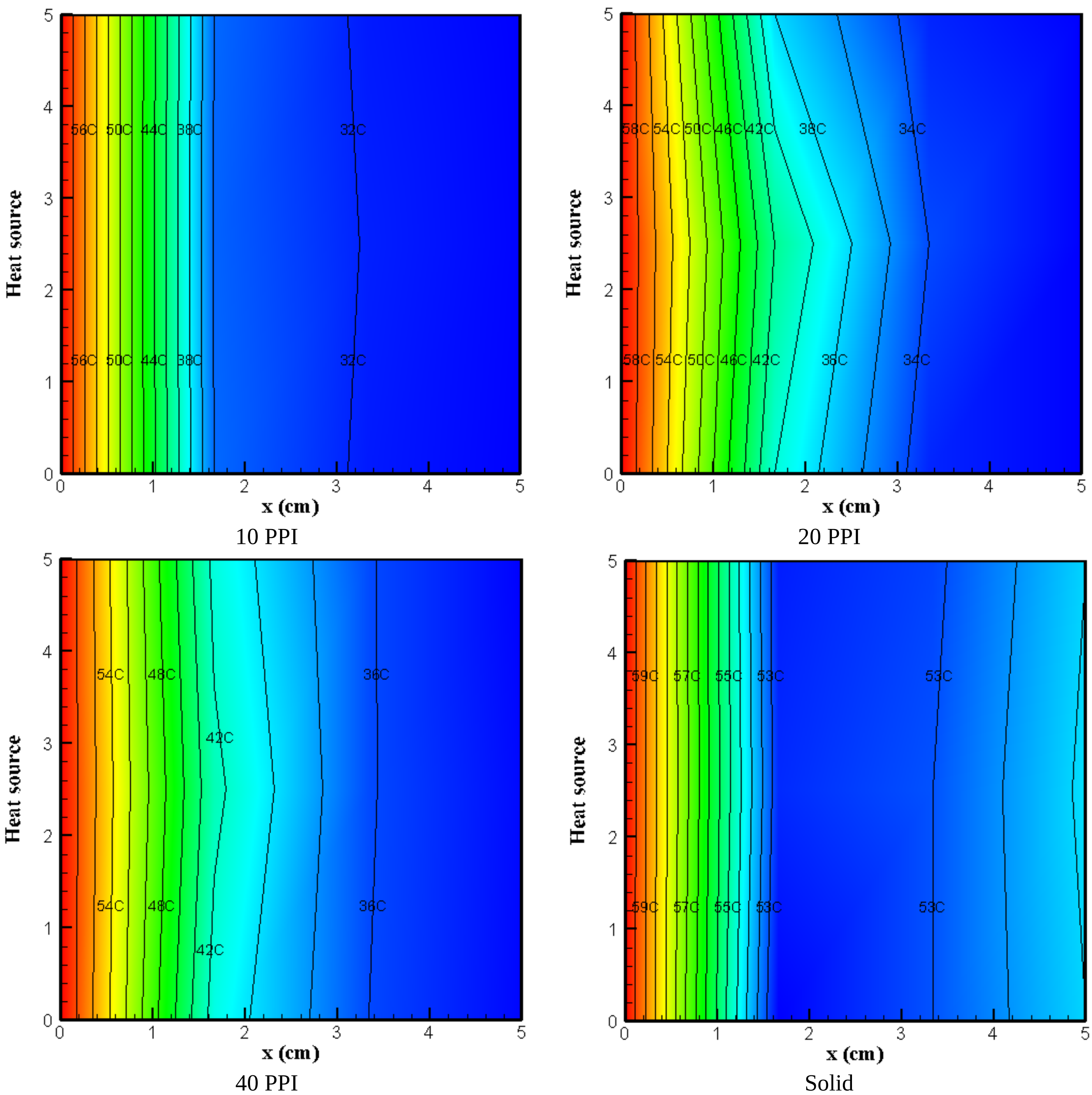


Figure 7: Variation of Temperature along x- axis for 10, 20, 40 PPI copper foam and solid copper fin at a speed of 1 m/s.

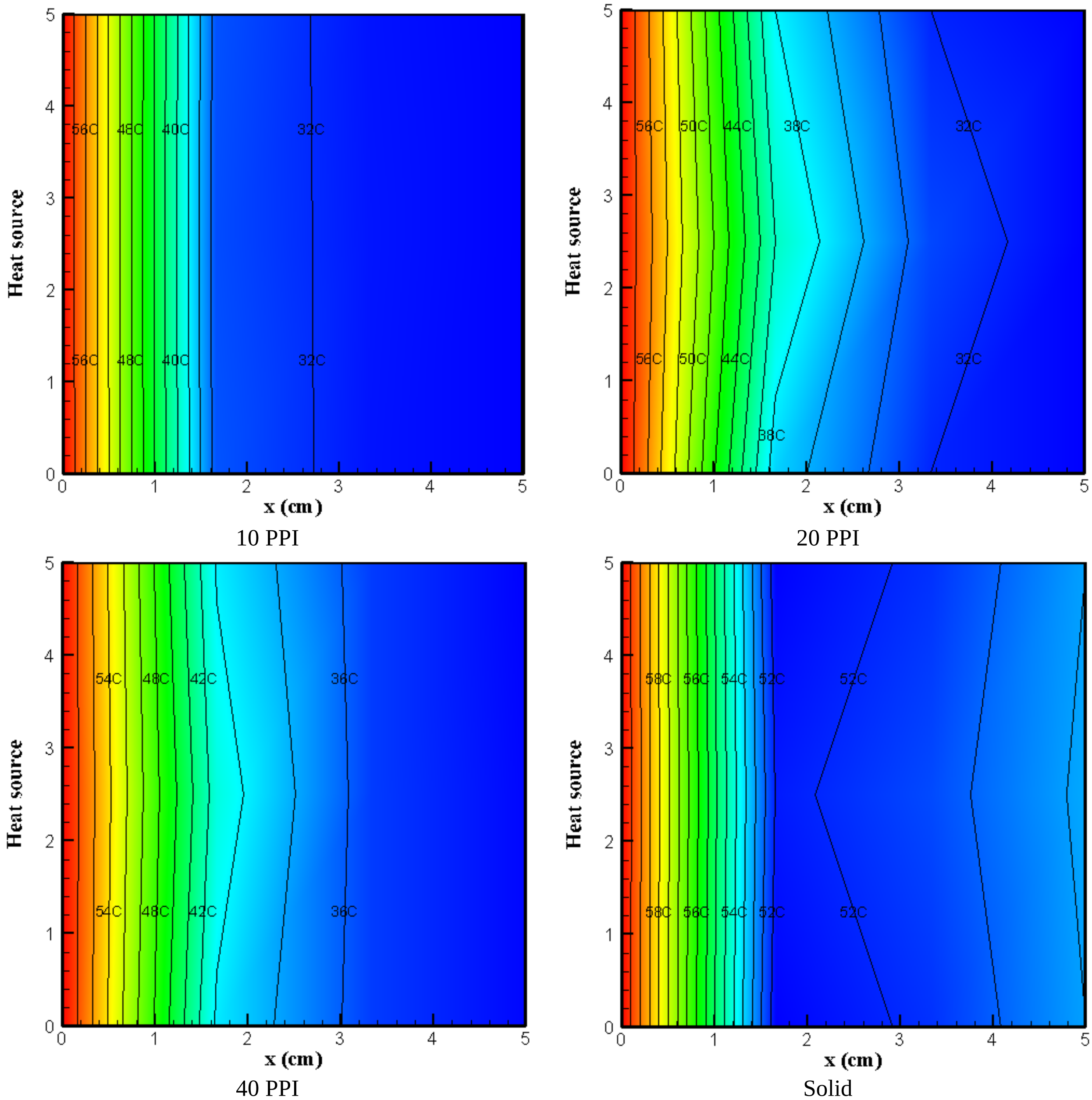


Figure 8: Variation of Temperature along x- axis for 10, 20, 40 PPI copper foam and solid copper fin at a speed of 1.5 m/s.

Figures (9, 10, 11 & 12) show the relation between average Nusselt number and air velocity; it is clearly that whenever velocity increases the average Nusselt number increases due to the increasing of heat transfer rate. From this figure, we can obviously note the difference between the metal foam and the solid metal. Metal foam has a very large heat transfer compare to solid metal because of the high surface area provided by the pores as shown in Figure (13). It display the percentage increasing of the average Nusselt number when increasing the air velocity for different metal foam PPI. For 10 PPI metal foam under air velocity of 0.5 m/s the increasing in average Nusselt number is around 36% while in case of metal foam of 40 PPI with air velocity equal to 1.5 m/s the increment is nearby 133 % compare to solid copper. The figures (7&8) also show that the increasing in the rate of convection heat transfer by metal foam has limitation due to pores sizes. In Figures (9, 10, 11 & 12) for a given PPI, average Nusselt number will reach the maximum value according to air velocity and after that approximately keep at a given value even the air velocity increases much more. Figure (13) shows also that the percentage increasing in average Nusselt number for a given PPI approach to maximum at high air velocity and the curve trend to be horizontal due to reach the average Nusselt number to its maximum value for this PPI.

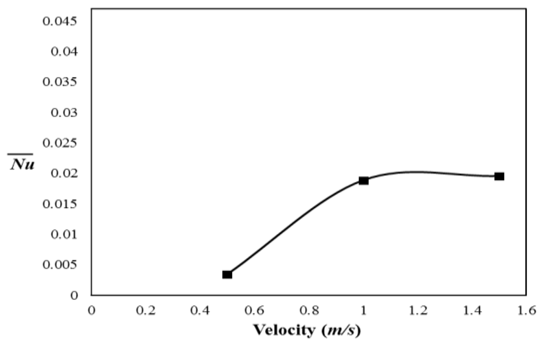


Figure 9: Variation of average Nusselt number with air velocity for solid copper fin.

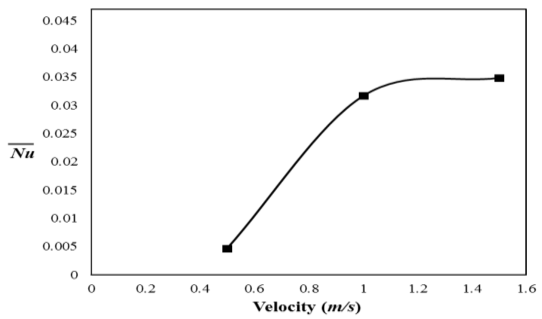


Figure 10: Variation of average Nusselt number with air velocity for 10 PPI copper foam fin.

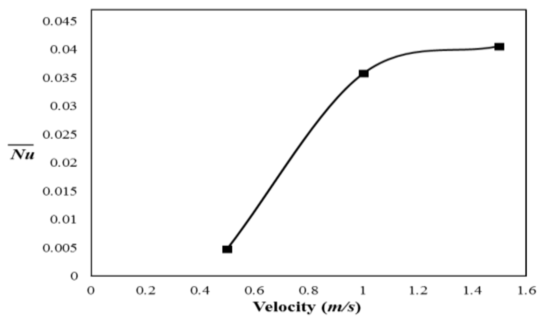


Figure 11: Variation of average Nusselt number with air velocity for 20 PPI copper foam fin.

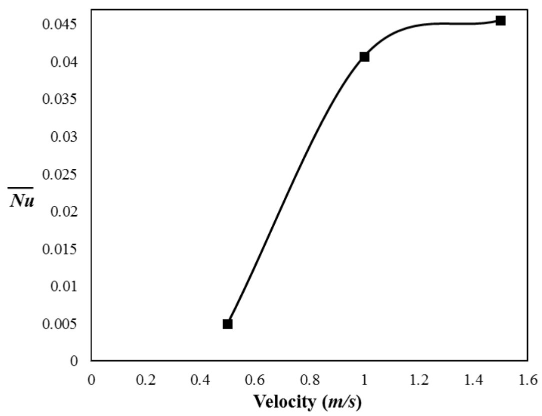


Figure 12: Variation of average Nusselt number with air velocity for 40 PPI copper foam fin.

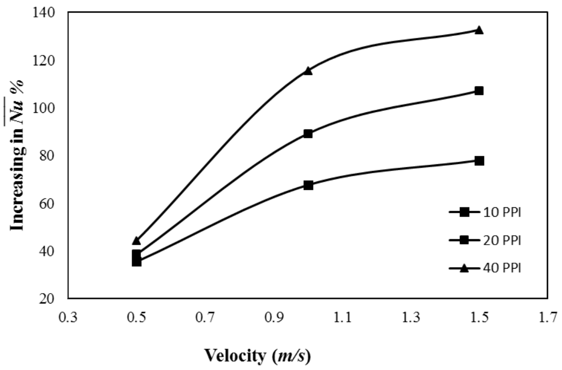


Figure 13: Percentage increasing of average Nusselt number with air velocity for 10, 20, 40 PPI copper foam.

Finally, uncertainty analysis has been done using SPSS software to find the standard error of the experiments. Table (1) shows the obtained results

Table 1: Descriptive statistics

	N	Minimum	Maximum	Mean		Std.
	Statistic	Statistic	Statistic	Statistic	Std. Error	Statistic
Solid metal air speed 0.5 m/s	9	55.80	57.60	56.6222	.2414	.72419
Solid metal air speed 1.0 m/s	9	52.40	54.10	53.2222	.2247	.67412
Solid metal air speed 1.5 m/s	9	51.70	53.10	52.3778	.1839	.55176
40 PPI air speed 0.5 m/s	9	37.20	48.40	41.6778	1.5219	4.56557
40 PPI air speed 1.0 m/s	9	34.20	42.50	37.3556	1.1219	3.36568
40 PPI air speed 1.5 m/s	9	33.60	41.00	36.3111	.9932	2.97970
20 PPI air speed 0.5 m/s	9	35.00	49.00	40.1111	1.8215	5.46453
20 PPI air speed 1.0 m/s	9	32.00	42.00	35.1111	1.2850	3.85501
20 PPI air speed 1.5 m/s	9	31.00	40.00	33.8889	1.1600	3.48010
10 PPI air speed 0.5 m/s	9	32.30	35.60	33.7000	.4676	1.40268
10 PPI air speed 1.0 m/s	9	31.00	34.00	32.2444	.4362	1.30873
10 PPI air speed 1.5 m/s	9	30.80	33.20	31.7444	.3481	1.04416
Valid N (listwise)	9					

Conclusions

Forced convection heat transfer through a copper metal foam fin was carried out experimentally. A solid metal sample with three pieces of metal foam of different PPI all made from copper metal were used to simulate a heat transfer from fin which subjected to constant heat flux. It is found that the temperature decreases as moving from the base of the test sample along the x-axis to the tip at different rates for diverse PPI. Metal foams have been confirmed as the best choice among the materials when used in forced convection heat transfer. The metal foam enhances the Nusselt number in rates directly related to the increasing in pore density. Through that, metal foam has a promising future in the area of convection heat transfer especially for miniature applications such as cooling of electronic chips due to its high rate of heat transfer compared to solid material. This rate of heat transfer mostly depended on the PPI of the metal foam.

When compared between the temperature drops for the 10, 20 and 40 PPI, on x- axes are made, it is found that the drops at air velocity of 1m/s and 1.5m/s are higher than the temperature drops at 0.5m/s.

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بحث تجريبي لحمل الحرارة القسري في زعانف النحاس الاسفنجي مفتوح الخلية

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الخلاصة

في الآونة الأخيرة، جزء كبير من الباحثين ركز اهتمامه على زيادة كفاءة الزعانف عن طريق زيادة الأداء الحراري لنفس حجم الزعنفة الاصلية. المعدن الاسفنجي أحد الطرق الواعدة لإنجاز هذا الهدف. تم اجراء تحليل أداء لبحث الخواص الحرارية لنماذج زعانف من النحاس الاسفنجي. هذه النماذج تم مقارنتها مع انتقال الحرارة في زعانف النحاس الصلد. انتقال حرارة بالحمل القسري تم تسليطه على أربعة نماذج. مسخن كهربائي سخن الزعانف التي كانت تخضع لتيار من الهواء المحيط مدفوع بواسطة مروحة منفاخ والذي يعتبر كمشتت للحرارة. الفيض الحراري تم تثبيته في كل التجارب وباستخدام ثلاث سرع للهواء، حمل الحرارة القسري تم تمثيله. فيض حراري ثابت تم تسليطه خلال التجارب مع سرع هواء معينة كمتغيرات حاكمة. كثافة المسام للنحاس كانت 10، 20، 40 مسامة بمربع البوصة. الأداء الحراري لزعنفة النحاس الاسفنجي تم حسابه من خلال معدل رقم نسيلت والمقاومة الحرارية لمشتت الحرارة. الزيادة في معدل انتقال الحرارة ومعدل رقم نسيلت عند استخدام المعدن الاسفنجي وجدت بحوالي مدى من 36% الى 133 % مقارنة بالمعدن الصلد. علاوة على ذلك، تم اثبات ان هذه الزيادة تصل لقيمة عظمى لكل قيمة من كثافة المسام ولا تتجاوزوها حتى بزيادة سرعة الهواء.