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Enhancement of Heat Transfer in a Passive Cooled Heat Exchanger Using Porous Material

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ABSTRACT

The airside of air-cooled heat exchangers causes poor thermal characteristics resulting in a low heat transfer coefficient (HTC). To overcome this issue, extended surfaces are widely used to improve both thermal performance and HTC. This research presents a numerical investigation of passive cooling by using copper foam porous material as an extended surface to increase the heat transfer area, for laminar flow, on a heat exchanger consisting of a vertical down-faced V-shape porous media attached to a vertical heat source for augmentation of the heat transfer rate in the heat exchanger. The hydraulic and thermal characteristics of passive cooling were studied to evaluate the thermal performance by the overall HTC and Nusselt number. The finite Element Method (FEM) is used to conduct this research by employing the COMSOL software. The study was carried out for different V-shape angles of 30, 60, 90, and 180 degrees and porous insert thicknesses from 0.1-100 mm. The effect of porosities of 50%, 75%, and 95% were investigated. The results show that the hydraulic and thermal characteristics were changed with changing the porous insert height and slightly changed with changing the angles and porous insert thicknesses. The porous insert height was changed from 1-100 mm, and the results indicate that beyond 4 mm height, the performance is almost unchanged, and the effectiveness decreases significantly with increasing the height. Angle 14 degrees gives the best buoyancy and pumping power. Also, porosity 50% gives the best HTC. Accordingly, a fin thickness of 2 mm and a porous insert height of 4 mm were adopted.

KEYWORDS: Passive cooling, Porous materials, Heat exchanger, Effectiveness, Heat transfer coefficient.

تعظيم انتقال الحرارة من مبادل حراري سلبي باستخدام مواد ذات مسامية

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الملخص

يتسبب الجانب الهوائي للمبادلات الحرارية المبردة بالهواء في ضعف الخصائص الحرارية مما يؤدي إلى انخفاض معامل نقل الحرارة بالحمل. للتغلب على هذه المشكلة، يتم استخدام الأسطح الممتدة على نطاق واسع لتحسين الأداء الحراري. يقدم هذا البحث دراسة عددية للتبريد السلبي باستخدام مادة نحاسية مسامية كسطح ممتد لزيادة مساحة نقل الحرارة، على مبادل حراري يتكون من وسط مسامي رأسي على شكل حرف V متصل بمصدر حرارة راسي لزيادة معدل نقل الحرارة في المبادل. تمت دراسة الخصائص الهيدروليكية والحرارية وتتمثل في انتقال الحراره وتعظيم معامل النتقال الحرارة بالحمل الحر. يتم تقييم الأداء الحراري من خلال الرقم الإجمالي لـمعامل انتقال الحراره بالحمل الحر ورقم نسلت. تم استخدام طريقة العناصرالعددية لإجراء هذا البحث من خلال استخدام برنامج كومسيل. تم إجراء الدراسة لزوايا مختلفة على شكل حرف V تبلغ 30، 60، 90، 180 درجة وسماكات مسامية من 0.1-100 ملم. تم دراسة تأثير المسامية 50%، 75% و 95%. أظهرت النتائج أن الخصائص الهيدروليكية والحرارية قد تغيرت مع تغيير ارتفاع النموذج وتغيرت قليلاً مع تغيير الزوايا وسمك المادة النحاسية المسامية. تم تغيير ارتفاع المادة المسامية من 1-100 مم، وتشير النتائج إلى أنه بعد ارتفاع 4 مم، لا يتغير الأداء تقريبًا، وتقل الفعالية بشكل ملحوظ مع زيادة الارتفاع. وبناءً على ذلك، تم اعتماد سمك زعنفة يبلغ 2 مم وارتفاع المادة المسامية يبلغ 4 مم.

الكلمات المفتاحية : التبريد السلبي، المواد المسامية، المبادل الحراري، الفعالية، معامل انتقال الحرارة بالحمل.

1. INTRODUCTION

Heat transfer enhancement techniques, which can be either passive, active, or a combination of both, find widespread application across various engineering fields, including heat exchangers, industrial processes, evaporator systems, thermal power plants, air conditioning units, refrigeration systems, chemical reactors, radiators for spacecraft and automobiles. These techniques are crucial for improving heat transfer rates while simultaneously reducing the size and cost of equipment, especially in heat exchangers. Among these techniques, passive heat transfer methods stand out, as they utilize flow passages to boost heat transfer and offer advantages over active methods due to their compatibility with existing heat exchangers. In the context of compact heat exchanger design, the selection of an appropriate passive insert configuration is vital, considering the specific working conditions of the heat exchanger, encompassing both flow and heat transfer requirements. Air is commonly employed in cooling applications, and the use of extended surfaces, particularly smaller effective fins, has proven effective in reducing air-side thermal resistance and enhancing heat transfer efficiency. Porous fins, characterized by their efficiency and compactness, outperform traditional rectangular cross-section fins, especially in electronic applications. Porous fins exhibit superior heat transfer capabilities compared to solid fins, making them an attractive subject for thermal analysis under natural convection conditions. In recent developments, high porosity metal foams have found applications in aerospace structures, chemical reaction catalysis, and the construction of robust panels, owing to their exceptional surface-area density and efficient fluid mixing properties. Open-cell metal foams have emerged as promising materials for manufacturing compact and efficient heat exchangers. An experimental study was done to show the effect of different parameters like pore density, porous material height, and the number of porous fins, on the thermal performance of the foam heat sinks. As a result of the study, the method of fused bonding causes a reduction in the thermal contact resistance by 19 times, compared to the epoxy-gluing process. The results showed that the Nusselt number is 30% higher by using the fused bonding method compared to using the epoxy-glued method. Additionally, the heat flux from the foam heat sink was twice compared to the normal heat sink [1].

The hydro-thermal performance of forced convective heat transfer in a heat sink combining metal foam and pin fins was numerically analysed. various porosities were tested (ranging from 0.8 to 0.95) under laminar flow conditions. And deionized water was used as a working fluid. The results showed that the hybrid heat sink, benefiting from the synergistic effects of solid pin fins and metal foam, exhibited significantly improved heat transfer performance, the normal pin fin showed an improvement of 267% and the metal foam heat sinks showed an improvement by 36%, at a Reynolds number of 1000 [2].

A numerical study was performed on a circular tube with twisted conical strip inserts fitted on the tube and the laminar flow thermal-hydro performances were tested. The results showed that the twist angle was the most effective parameter on both the friction factor and Nusselt number [3]. [4-8] Studies were carried out on metal foam heat exchangers to enhance their thermal and hydraulic performance, using copper foam and aluminum heat sinks.

The performance of a porous fin made of aluminum alloy 6101 compared to a normal louvered fin and normal louvered fin showed better performance [9].

An experimental and numerical studies on the hydraulic performance corrugated porous heat exchanger. It was concluded that if the dynamic viscosity increased the flow distribution was improved but the pressure drop was increased. A correlation between Reynolds number and the pressure drop was conducted [10].

A numerical study evaluated the hydraulic and thermal performances of tube channels was partially filled with grooved metal foams under laminar steady flow conditions with constant heat flux. Two pitches of helical grooves with a diameter ratio of 0.5 resulted in a reduction in pumping power by approximately 17% [11].

The thermal and hydraulic performances of a heated vertical flat plate heat exchanger with three brass wire mesh materials of different porosities were investigated, showing that increasing porosity led to an enhancement in the rate of heat transfer and decreased the pressure drop [12].

A numerical study examined the thermal and hydraulic performances of three different heat exchangers, including a metal foam baffle heat exchanger, a non-baffle heat exchanger, and a combination of metal foam baffle and fin heat exchanger. The results showed that metal foam baffle heat exchanger achieved up to about a 584% increase in heat transfer rate [13].

An experimental study was performed to evaluate the performance of an annular porous fin rounded in a vertical cylinder by using natural convection. The results indicated that the average Nusselt number was dependent on permeability, the heat transfer enhancement was 8% using one fin and the maximum achievement was 131% with a porous fin layer [14].

In natural convection, the effects of porous material were studied using a nano-fluid inside a triangular chamber. It was concluded that the rate of heat transfer was affected by using a hybrid nano-fluid (water/aluminum and oxide-copper) inside a porous material [15].

In the case of porous flat plate fins placed in a vertical heat source under natural convection, the heat transfer rate from the porous fin could exceed that of a solid fin. Additionally, it was observed that increasing the fin length and effective thermal conductivity improved heat transfer up to a certain limit, with no further enhancement beyond those parameters [16].

A natural convection study around a horizontal solid cylinder wrapped with porous material was conducted to identify optimal conditions for thermal insulation or improving heat transfer rates. It was found that a Darcy number limit needed to be considered, and to achieve this, high thermal conductivity porous media, and high permeability should be used [17].

A hybrid nano-fluid of water/aluminum copper oxide inside a porous enclosure with differential heating on its upright walls was studied in natural convection. Using glass balls as nanoparticles and aluminum foam as the porous material reduces the mean Nusselt numbers and increasing the nanoparticle volume fraction led to a decrease in the heat transfer rates when increasing the nanoparticle volume fraction [18].

An experimental study of forced convection heat transfer was performed on a circular tube filled with a metal foam with nanofluid flow. It was concluded that there was an improvement in the rate of heat transfer along with an increase in the pressure drop. A correlation between Nusselt number and nano-fluid volume fraction was identified [19].

It was numerically shown that adding nanoparticles to a fully developed laminar by using mixed convection including forced and natural convections of water in an annulus-reinforced secondary flow [20].

The study of natural convection between two eccentric cylinders revealed that reducing the Lewis number resulted in fewer nanoparticle collisions and a decrease in the Nusselt number [21].

The utilization of porous media in passive cooling, particularly in the realm of research, remains relatively scarce. To address this knowledge gap, this current research endeavors to contribute to the understanding of passive cooling techniques. The primary aim of this study is to conduct a comprehensive numerical analysis of the hydraulic and thermal performance associated with a compact V-shape porous insert applied to a vertical heat source, with the ultimate goal of enhancing heat transfer rates. The investigation involves the examination of multiple heat exchanger geometries, all incorporating copper foam porous material with varying porosities. The objective is to identify the optimal geometry for the porous material within the compact heat exchanger that delivers superior hydraulic and thermal performance. This research is a step towards filling the existing void in our understanding of passive cooling techniques utilizing porous media.

2. The heat exchanger geometry

The heat exchanger's geometry, as depicted in **Figure 1**, involves the flow of air over a vertical heat source characterized by dimensions (L x W) and a specific height (H). Within this setup, the heat transfer medium is a V-shaped copper foam. The key parameters under investigation for this medium are the effective thickness (t), the V-shape angle (θ), and the porosity (Ɛ). The focus of this research is passive cooling heat transfer utilizing the porous medium, particularly in the context of laminar airflow. It's important to note that this specific shape serves as a modular unit that can be replicated to create the desired overall heat exchanger geometry [21].

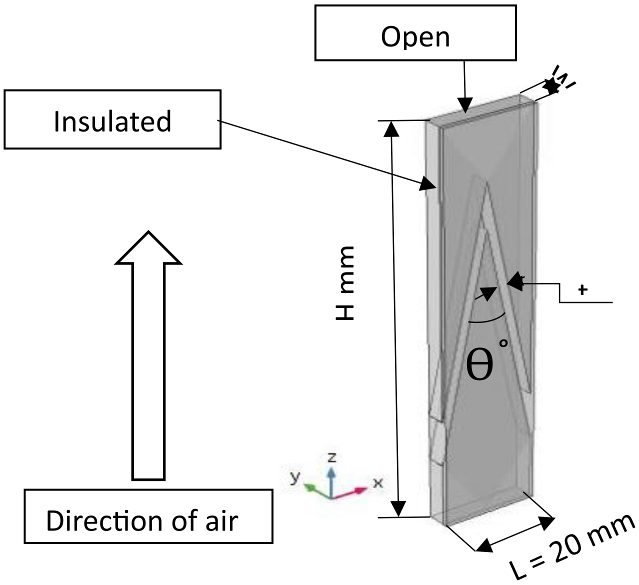

Figure 1. Computational domain.

3. Numerical Model

The domain is divided into two distinct regions, as illustrated in **Figure 1**. The first region is the free airflow domain, comprising an empty channel filled with air, while the second region is the porous matrix domain, where the porous material is located, and air occupies the pores within the matrix. The solve was for the temperature, velocity and pressure fields within the computational domain, the Finite Element Method (FEM) was applied. This numerical analysis was conducted using COMSOL Multi-Physics software [22].

3.1 Assumptions

Some assumptions are made for the present model calculations:

- Porous attachments are assumed to be homogeneous and isotropic and the porosity, thickness, and permeability are uniform .
- Radiation heat transfer is neglected since the temperature change within the porous matrix is less than 2 ^{0}C.
- The copper foam porous permeability is taken as $10 * 10^{-10}$ $(darcy\ or\ m^2)$ [4]

3.2 Mesh dependence test

To choose the best mesh size, coarse (202408 elements), normal (403064 elements) and fine (1062049) sizes were evaluated to find out the effect of mesh size on the bulk temperature (T_{bulk}). The evaluation gave a difference in temperature of only 0.02 °C with a feeble error of 0.0061% between coarse and fine meshes, therefore a coarse mesh size was chosen for all test runs. This saves hours in the time of test runs which are many. **Figure 2** shows the effect of mesh size on the bulk temperature of air flow.

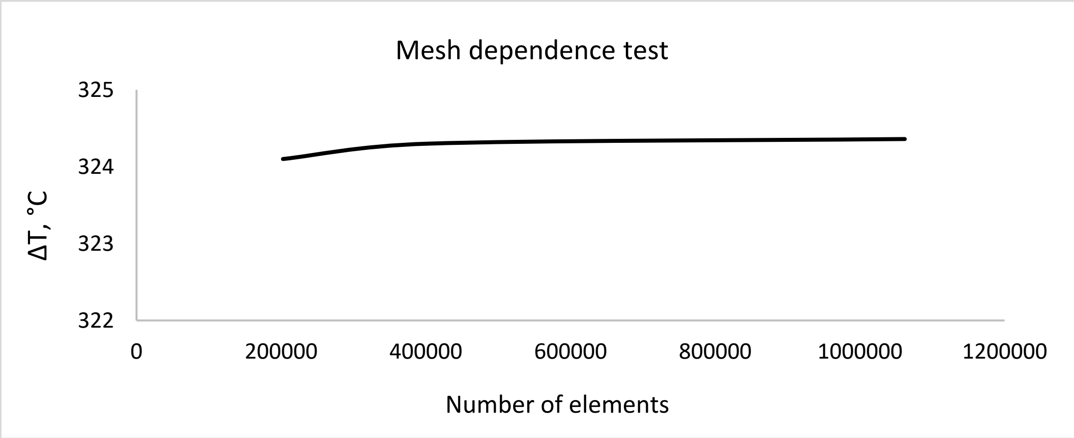

Figure 2. Mesh dependence test

3.3 Boundary Conditions

The boundary conditions such as the velocity and the inlet temperature of the air are listed in Table 1. The symmetry of the geometry results is reduction in the computational time, because of the reduced number of variables.

Table 1 Boundary conditions.

Boundary	Parameter	Value
Inlet	Air velocity V_a	0
Exit	Pressure	0
Wall	Velocity	0
Symmetry	$\nabla v.n = 0$	0
Temperature	Inlet air temperature	293 K
Outflow	n. ∇T	0
Thermal insulation	n. ∇T	0
Heat flux	$\dot{q}$	5000 W/m^2

3.4 Free air flow

3.4.1 Continuity equation

Within the computational domain, a laminar and steady airflow are established. In the free convection model, in free convection the air velocity at inlet is 0 m/s. It's important to note that air density varies as temperature dependent and pressure, and these variables change with position (x, y, z). Consequently, the variations in density have been duly considered, and the continuity equation is employed to account for these variations in the system [15].

$$\nabla.(\rho v) = 0 \tag{1}$$

3.4.2 Momentum equation

In this analysis, the air is treated as a Newtonian fluid, and it's assumed that the air is not subjected to any forces. Additionally, the dynamic viscosity of the air is considered as a function of temperature, and this viscosity value varies with position within the computational domain [15].

$$\rho(v.\nabla)v = \nabla.\left[-p + \mu(\nabla v + (\nabla v)^T) - \frac{2}{3}\mu(\nabla v)I\right] \quad (2)$$

Where μ is the air kinematic viscosity (m^2/s), and p is the air outlet pressure (N/m^2).

3.4.3 Energy equation

The thermal conductivity is considered to be a temperature dependent [15].

$$\rho c_p v.\nabla T = \nabla.(k\nabla T) \quad (3)$$

3.5 Model Validation

The Finite Element Method (FEM) was employed to solve the velocity, pressure, and temperature fields in the computational domain using COMSOL Multi-Physics software, the inputs are heat flux, ambient temperature and pressure, the outputs are bulk temperature and the surface temperature. Nusselt number (Nu) can be calculated from equation (10) and Rayleigh number (Ra) from the following equation

$$Ra = \frac{\rho^2 .g.\beta.\Delta T.Lc^3}{\mu^2} \quad (4)$$

Where g is the gravitiginal acceleration, $\frac{m^2}{s}$, ρ is air density, $\frac{kg}{m^3}$, β is coefficient of thermal expansion = $\frac{1}{T_a}$, where T_a is ambient temperature, K, Lc is characteristic length, m, and μ is dynamic viscosity, $\frac{kg}{m.s}$.

The model validation was confirmed by plotting the relation between Nusselt (Nu) and Rayleigh (Ra) numbers for bare vertical wall without any porous medium inside, and the results are compared with those obtained from equation (5) [23]. which is valid for vertical bare heat source.

$$Nu = \frac{Ra^{1/3}}{[9.742 - 0.1869\ln(Ra)]^{4/3}} \quad (5)$$

The comparative validation results are presented in **Figure 3**. Good agreement is shown between the present model and equation (5)

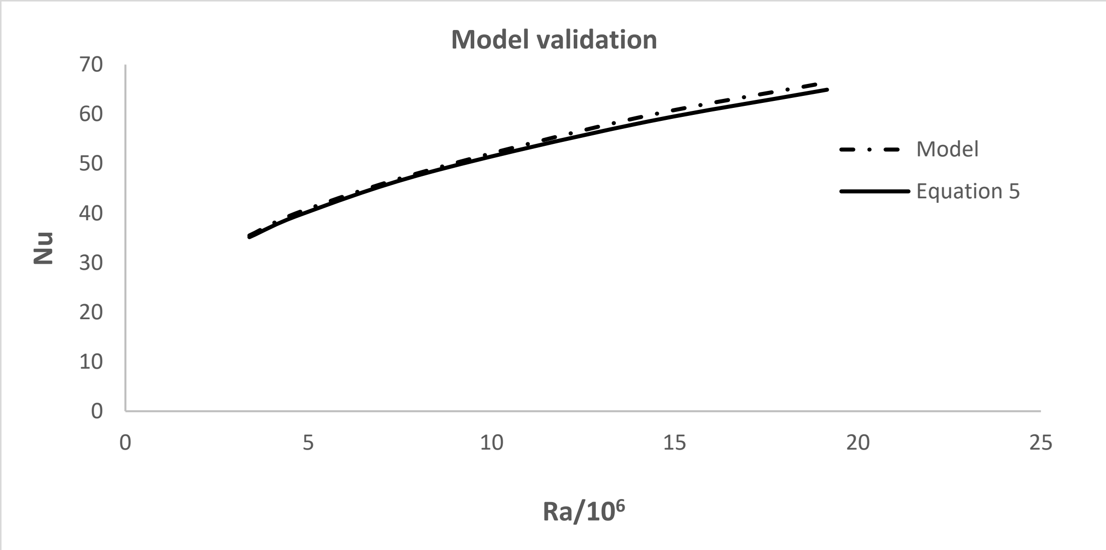

Figure 3. Model validation

3.6 Porous matrix

3.6.1 Continuity equation

The air velocity in the porous matrix is called filtration velocity (v_f). The air density pressure and temperature dependent [15].

$$\nabla.(\rho v_f) = 0 \tag{6}$$

Where the v_f is the flow velocity (m/s).

3.6.2 Momentum equation

Within porous media, the flow behaviour is described by the Brinkman–Forchheimer-extended Darcy equation [15]. The pressure drop, as governed by Darcy's law, is combined with the Forchheimer drag term to formulate the momentum equation, taking into account the porous characteristics of the medium.[15]

$$\frac{\rho}{\varepsilon}(v_f.\nabla)\frac{v_f}{\varepsilon} = \nabla.\left[-p + \frac{\mu}{\varepsilon}(\nabla v_f + (\nabla v_f)^T) - \frac{2}{3}\frac{\mu}{\varepsilon}(\nabla v_f)I\right] - \frac{\mu}{k}v_f - \beta_f {v_f}^2 \tag{7}$$

Where β_f is the Forchheimer coefficient (kg/m^4) and $= \frac{\rho C_f}{\sqrt{k}}$, and $\mathcal{E}$ is the porosity.

3.6.3 Energy equation

Steady flow was considered and no any heat generation in the porous media [15].

$$c_p V_f.\nabla T = \nabla.(K_{eff}\nabla T) \tag{8}$$

Where K_{eff} is the effective thermal conductivity for the fluid and porous material (W/mK).

4. Results and Discussion

This work studied the effect of the heat exchanger geometry on its heat transfer characteristics. The studied parameters were the porous medium thickness and height, the V-angel of the medium, and its porosity. The thermal performance of the heat exchanger is evaluated through the effectiveness, HTC, and buoyancy force.

4.1 Free convection

The efficiency of heat removal in a heat sink, under constant conditions like geometry, air velocity, and heat transfer amount, is reflected by the base temperature of the heat sink at a given inlet air temperature. A smaller difference between base temperature (T_b) and inlet temperature (T_{ai}) results in a higher heat transfer coefficient, and it can be expressed as follows.

$$\mathrm{HTC} = \frac{Q}{A(T_b - T_{a,i})} \tag{9}$$

Then, the Nusselt number is

$$\mathrm{Nu} = \frac{\mathrm{HTC}.\, L_c}{k_a} \tag{10}$$

Where, L_c is the characteristic length of the channel which is the height of the heat source, and k_a is the thermal conductivity of the air at mean film temperature for the natural convection model. The heat transfer rate is denoted by

$$\dot{Q} = \dot{m}.\, c_{p,a}.\, (T_{a,o} - T_{a,i}) \tag{11}$$

The air mass flow rate is

$$\dot{m} = \iint \rho.\, v.\, dA \tag{12}$$

The bulk temperature was generally used to calculate the air outlet temperature and is given by

$$T_{bulk} = \frac{1}{\dot{m}.c_{p,a}} \iint \rho.\, c_{p,a}.\, T.\, v.\, dA \tag{13}$$

In the present study, different geometries were tested to determine the optimum one for the porous heat transfer medium for variable thicknesses of 0.1 to 100 mm. The results reveal that the outlet air bulk temperature is slightly changed with the thickness so as HTC. Varying the angles of the v- shape as 14, 30, 60, and 180^0 (flat plate) did not show any notable difference in the bulk temperature and HTC.

4.2 Effect of porous insert height on the effectiveness and Nusselt number

Figure 4 presents the effect of the porous insert height (H) on the effectiveness for porosities of 50, 75, and 95%. The porosity does not affect the effectiveness, that is because the direction of air flow is perpendicular on the porous insert material so the effective length is the height of the porous material so the height directly affects the effectiveness but not the porosity. since the results coincide for the three porosities. The results show that for porous insert heights from 1- 4 mm, the effectiveness is close to 100%. Increasing the porous insert height above 4 mm decreases the effectiveness, and above 10 mm the effectiveness drops and goes down to 10% at a height of 100 mm, which indicates that above 4 mm the material is a waste. So, a porous height of 4 mm was chosen for economic reasons.

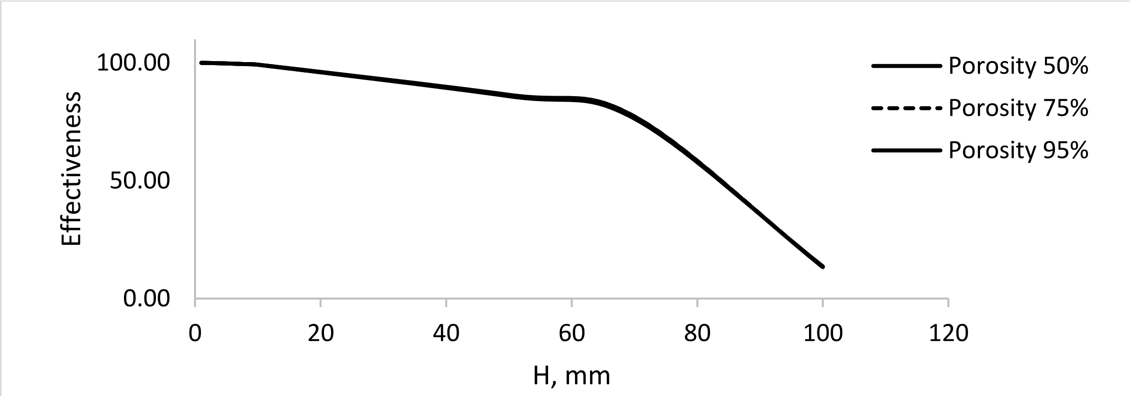

Figure 4. The porous insert height (H) vs effectiveness for different porosities.

4.3 Effect of the porous medium thickness on the effectiveness

Figures 5, 6, and 7 illustrate the effect of the porous insert thickness on the effectiveness of different V angles. V-shape angles below a thickness of 2 mm are extremely difficult to manufacture and give less effectiveness. As seen from the figure, above 2 mm thickness the thermal properties and HTC are not affected. Therefore, 2 mm is considered a suitable thickness, since it is considered material waste in addition to being the minimum machining thickness.

Figure 5. is for a porosity of 95%, and as shown the effectiveness for all angles is very close and almost coincides because of the very little available heat transfer material. Angle 60^0 gives some marginal higher effectiveness. In conclusion, the V angle does not affect the effectiveness.

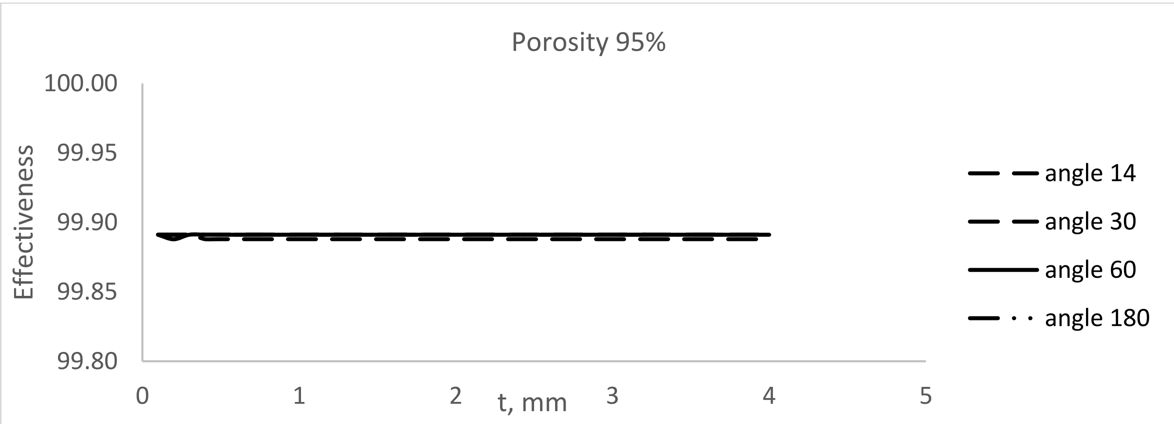

Figure 5. The effect of porous insert thickness on the effectiveness for porosity 95%.

The results in **Figure 6** are for a porosity of 75%. The figure indicates that angle 30 degree results in the lowest effectiveness, whereas angles 60 and 180^0 provide the highest. Although the values for angles 60 and 180 degrees coincide, the flat porous insert could be preferable for being easy to manufacture.

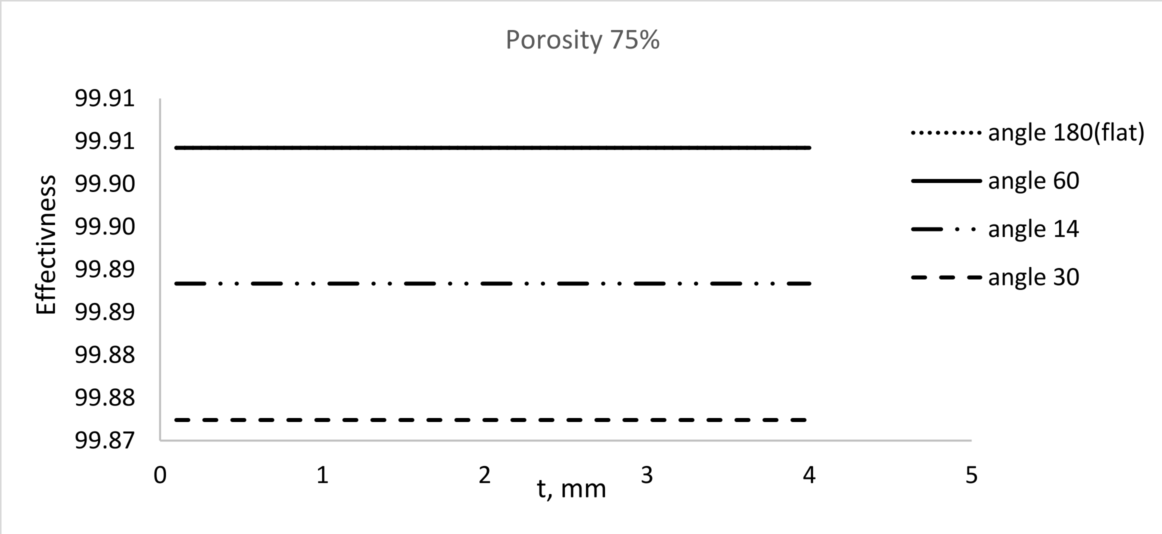

Figure 6. The effect of porous insert thickness on the effectiveness for porosity 75%.

Figure 7 is for porosity 50% and illustrates that angle 30 degrees gives the lowest effectiveness while angle 180^0 produces the best.

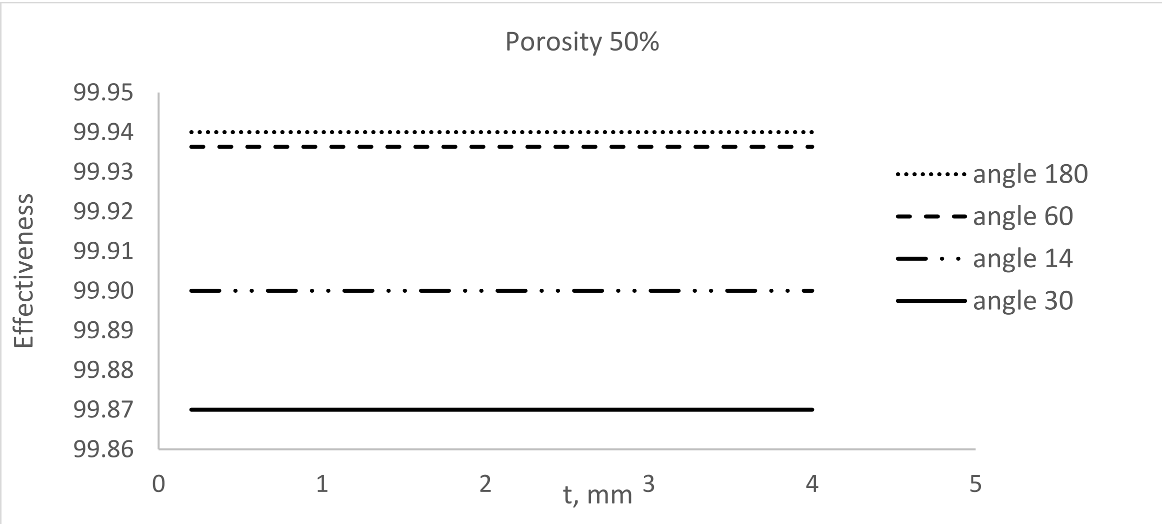

Figure 7. The effect of porous insert thickness on the effectiveness for porosity 50%.

Figure 8 demonstrates the relation between the porous insert height and HTC for different porosities. There is an enhancement in the HTC by increasing the channel height which increasing the mas flow rate of the air. This was done for the chosen porous insert thickness of 2 mm. This finding is opposite to the effectiveness results where the effectiveness decreases as the porous insert height increases. Therefore, one has to choose the optimum porous insert height to give the best effectiveness and HTC.

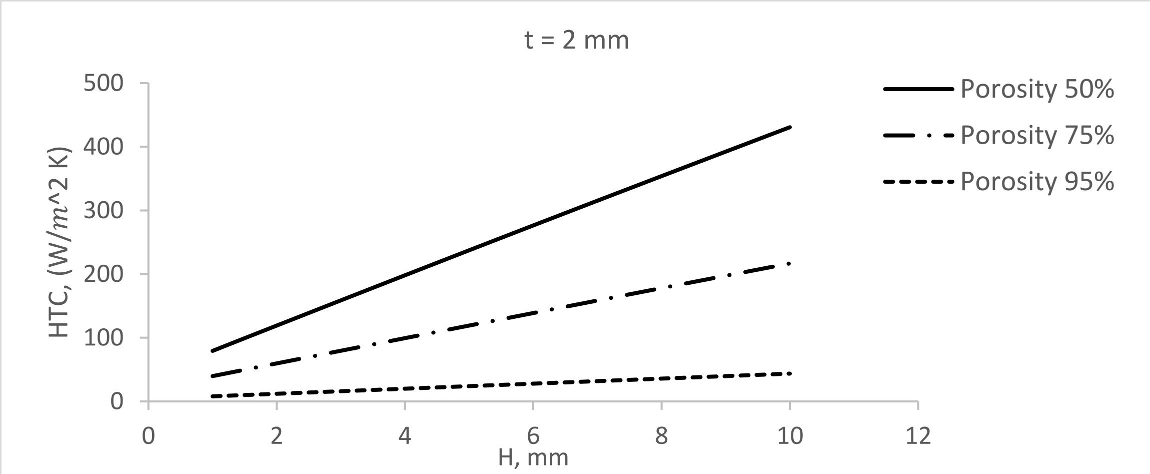

Figure 8. The effect of fin height on HTC for different porosities.

Figure 9 shows the effectiveness versus V angles for different porosities, for a porous insert thickness of 2 mm. It is confirmed that for all porosities angle of 30 degrees gives the worst effectiveness and above that angle, the effectiveness increases to become flat thereafter. The difference is more pronounced between porosities 50 and 95% because the latter contains very less heat transfer material. For porosity of 50%, the conduction heat transfer rate is higher than other porosities which results in higher effectiveness.

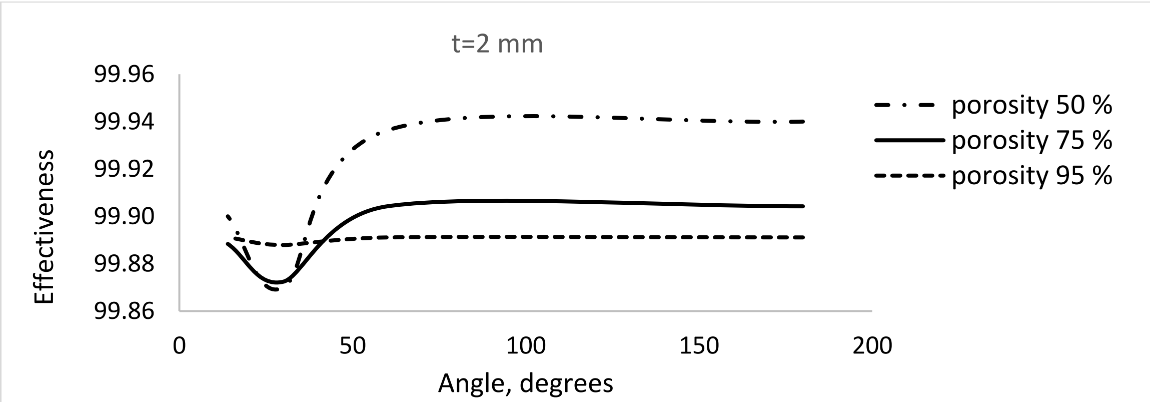

Figure 9. The effect of angle on the effectiveness for different porosities.

4.4 Effect of thickness on the HTC for different angles and porosities

Figures 10, 11 and 12 represent that the porous insert thickness shows no effect on the HTC for all porosities. It is indicated that the V-shape angle affects the HTC. So, porosity 95% with V-shape angle 30^0 gives the highest HTC, while for porosities 75 and 95% angle 180^0 gives the best HTC.

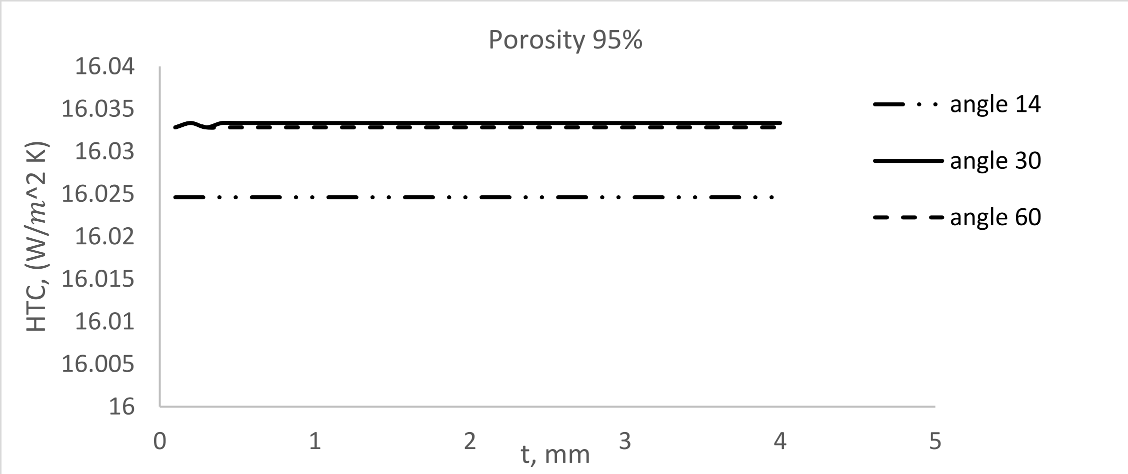

Figure 10. The effect of thickness on the heat transfer coefficient (HTC) for different angles at porosity 95%.

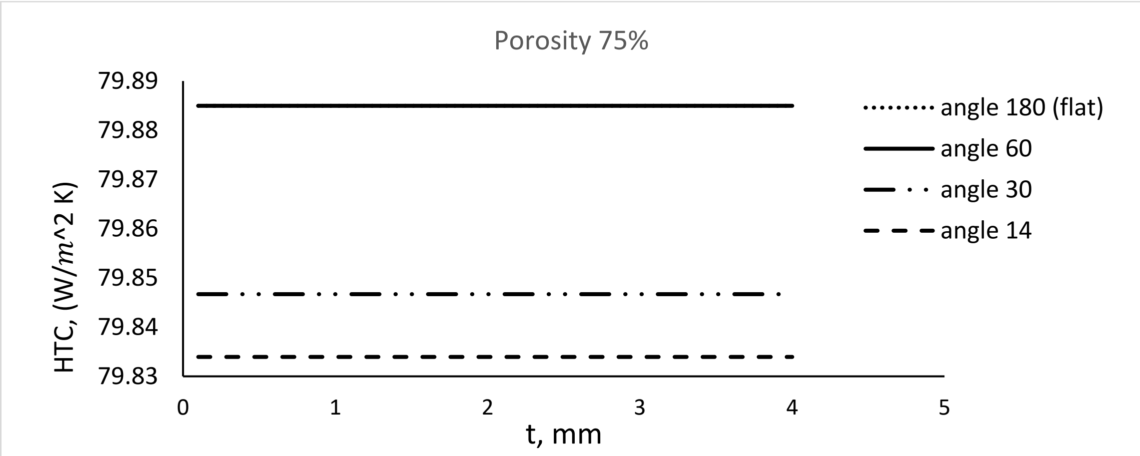

Figure 11. The effect of thickness on the heat transfer coefficient (HTC) for different angles at porosity 75%.

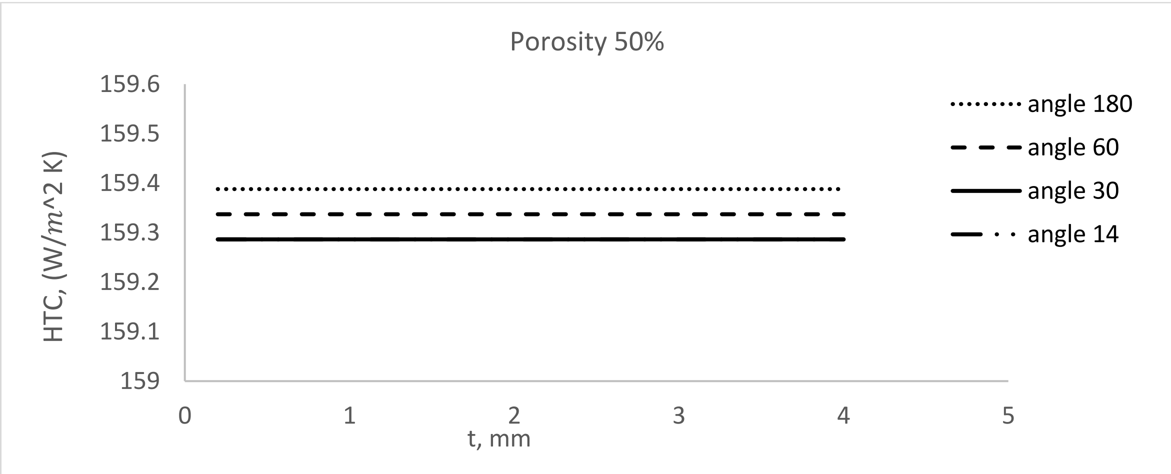

Figure 12. The effect of thickness on the heat transfer coefficient (HTC) for different angles at porosity 50%.

Figure 13 represents the relation between HTC and porous insert angle for different porosities for a porous thickness of 2 mm. It confirms that porosity 50% gives the best HTC that is due to the presence of enough material, hence a large heat transfer area.

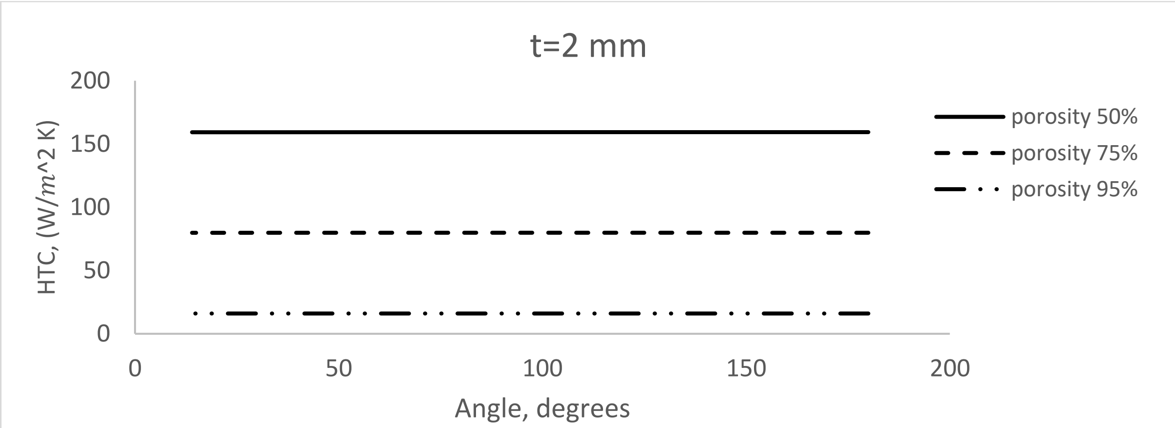

Figure 13. The effect of angle on the heat transfer coefficient (HTC) for different porosities and thickness 2 mm.

4.5 Effect of angle and porosity on and air flow rate represented by buoyancy force

To demonstrate the effect of angle and porosity on the air flow rate, a figure for buoyancy force was developed. **Figure 14** depicts the effect of angle and porosity on the buoyancy force. The figure indicates that a porosity of 95% provides the maximum flow rate of air and a porosity of 50% produces the minimum because of the higher material resistance. However, the results for porosities 75 and 95% almost coincide. Also, increasing the angle decreases the air mass flow rate due to less air passing through the porous material hence less cooling.

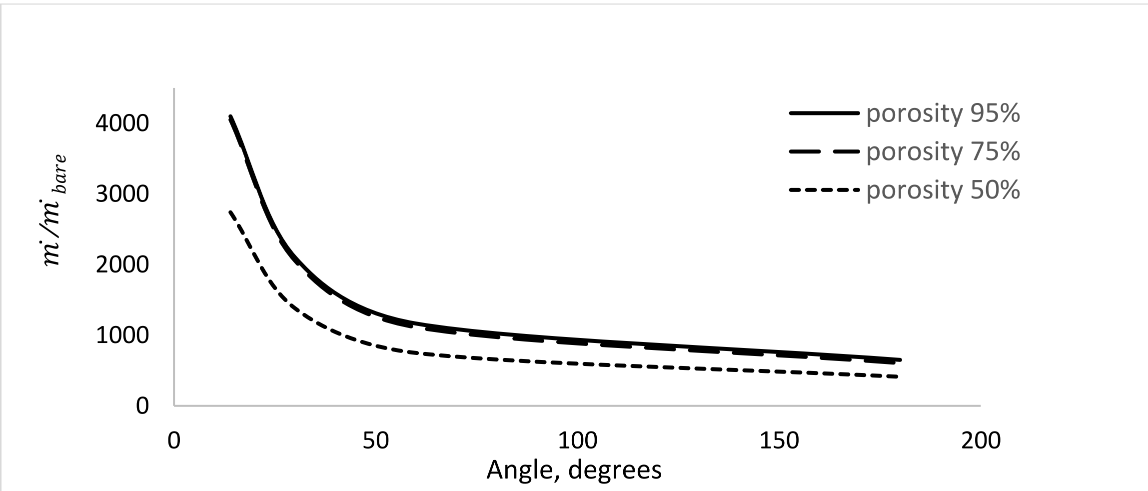

Figure 14. The effect of angle on the buoyancy force with different porosities.

$\dot{m}/\dot{m}_{bare}$ is the buoyancy component of air and represents the pressure drop through the porous material, θ is the V-shape angle, and ε is the porosity of the materialSo, designers can choose the best heat exchanger geometrical parameters according to design requirements, i.e. HTC, effectiveness, buoyancy force or a compromise of all, since there are no optimum fixed parameters for all cases.

4.6 Comparison between Copper, Aluminum and Carbon foam

Figures 15,16 and 17 represents the effect of angle on the HTC for copper, aluminum and carbon foam porous materials at porous thickness of 2 mm. For all angles 14-30-60 and 180 degrees the results show that for all porosities the difference of HTC between copper and aluminum is almost double that is because of higher thermal conductivity of the copper. Copper material shows the best results and carbon foam shows the lowest results increase of HTC. The result for all porous thicknesses ranges from 0.1 mm to 4 mm are showing the same difference, so choosing porous thickness of 2 mm is more reliable.

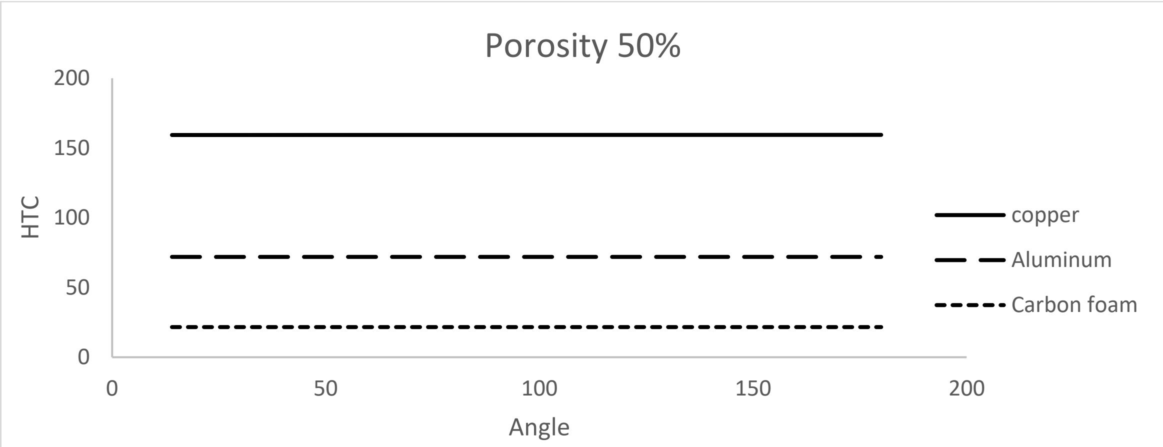

Figure 15. Effect of porous insert angle on HTC.

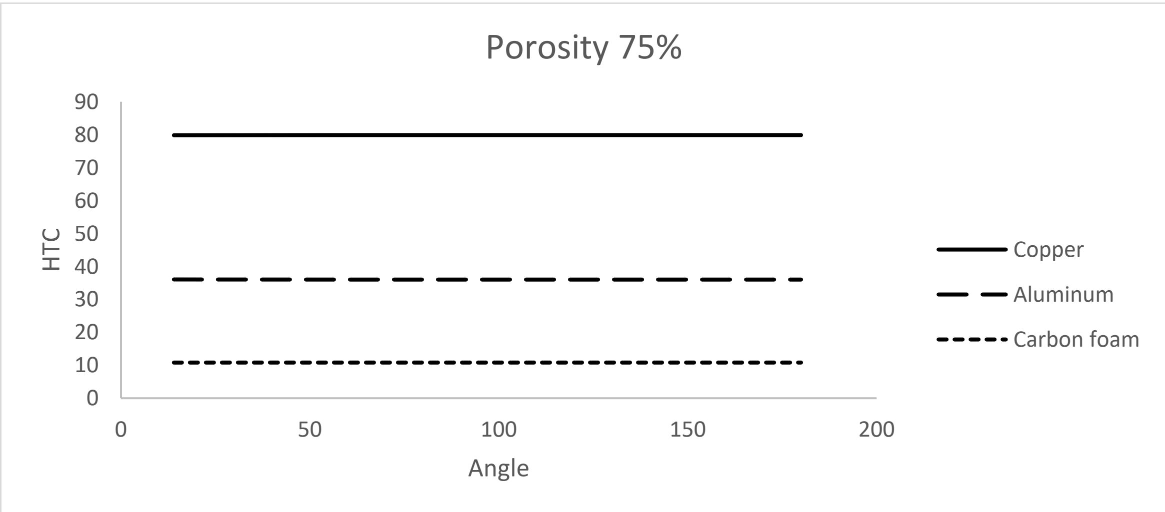

Figure 16. Effect of porous insert angle on HTC.

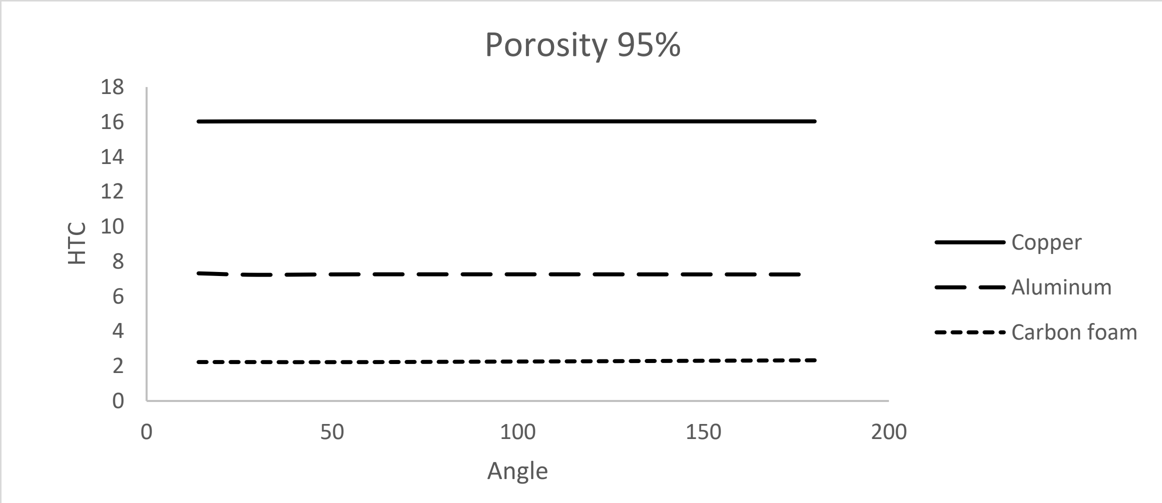

Figure 17. Effect of porous insert angle on HTC.

Conclusions

In the present study, the thermal and hydraulic performance of passive cooling heat exchangers with copper foam porous material was numerically investigated. The heat source is vertical, and variations in the porous medium thickness and its v-angles, and the height of the heat exchanger were simulated using COMSOL MULTI-PHYSICS. Steady-state laminar flow was modeled. The following conclusions are obtained:

-The porous insert height affects the effectiveness and 4 mm is appropriate, since the effectiveness decreases afterward, and above 4 mm is material waste.

-The porous insert thickness does not affect the effectiveness, and 2mm was chosen as the minimum thickness for machining

-For porosity 95%, angle 60^0 gives the highest effectiveness, and for porosity 75%, angles 60 and 180 degrees provide the best effectiveness, where porosity 50%, angle 180^0 results in the best effectiveness.

-Porosity 50% gives the best effectiveness.

-The HTC increases with increasing the porous height, and a porosity of 50% shows the best HTC.

-For porosity 95%, an angle of 30 degrees gives the best HTC, and for porosity 75%, angles of 60 and 180 degrees give the best HTC but for porosity 50%, an angle of 180 degrees gives the best HTC.

-Porosity 95% gives the best buoyancy force.

-angle 14^0 gives the best buoyancy force.

-Three materials were tested, copper, aluminum and carbon foam and copper gives the best results as compared with aluminum and carbon foam.

Nomenclature

HTC	Heat transfer coefficient ($\mathrm{W/m^2K}$)	h_v	Volumetric heat transfer coefficient ($\mathrm{W/m^2\ K}$)
T_b	Base temperature of the heat source (°C)	$\dot{m}$	Air mass flow rate (kg/s)
T_{Bulk}	Air bulk temperature at exit (°C)	k_a	Thermal conductivity ($\mathrm{W/mK}$)
$T_{a,i}$	Air inlet temperature (°C)	ε	Porosity
$T_{a,o}$	Air outlet temperature (°C)	Nu	Nusselt number
V	Air velocity (m/s)	Re	Reynolds number
H	Heat source height (mm)	$\dot{Q}$	Heat transfer rate (W)
W	Heat source width (mm)	K	Permeability ($darcy\ or\ \mathrm{m^2}$)
f	Friction factor	t	Porous insert thickness (mm)
$K_{p,\infty}$	Euler number	V	Gross volume ($\mathrm{m^3}$)
P	Pumping power (W)	PEC	Performance evaluation criteria
ΔP	Pressure drop (pa)		

Subscript

a	Air	o	Outlet
b	Base	v	Volumetric
i	Inlet		

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