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Experimental and Numerical Analysis of one Dimensional Heat Transfer on Open Cell Aluminum Foams

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ABSTRACT

In this study, one dimensional heat transfer of open cell aluminum metal foams is investigated both experimentally and by using numerical methods as well. Open cell aluminum foams with pore densities of 10, 20 and 30 (Number of Pores Per Inch) PPI were shaped into heat exchangers. The foams having sizes of 200 x 100 x 20 mm were insulated on their three faces. Steady heat flux was maintained on the base section of the foam by heating a plate shaped coil electrically. Temperature distributions on the vertical sections and mostly on locations near heaters were measured with the thermocouples located on the aluminum foams. With the help of the recorded temperatures from the tests the graphs of open cell aluminum foams with pore densities of 10, 20 and 30 were plotted. First of all, one dimensional heat transfer equations were derived for the numerical solution of the system. The governing equations obtained were then discretized by using the Central Difference Method and finally solved with the Finite Difference Method. The results obtained were converted into graphs and compared with the empirical results obtained beforehand. The fastest drop in temperature close to the heater was observed at the foam with 10 PPI while the lowest falling rate took place at the foam with 30 PPI pore density. At an interval of three aluminum foams, the temperature difference was found to be higher near the heater and lower away from the heater. It was found that both experimental and numerical results are closely related.

Keywords: *porous media, open-cell aluminum foam, convection-conduction heat transfer*

1. INTRODUCTION

Metal foams have considerable applications in multifunctional heat exchangers, cryogenics, combustion chambers, cladding on buildings, strain isolation, buffer between a stiff structure and a fluctuating temperature field, geothermal operations, petroleum reservoirs, catalytic beds, compact heat exchangers for airborne equipment, air cooled condensers and compact heat sinks for power electronics [1]. They are a relative new class of materials with very promising applications in which their low density and other thermal, mechanical, electrical and acoustical properties make these materials an excellent means of performance improvement. So far, a number of investigations on aluminum foams (both experimental and analytical) have been made. Some are briefly mentioned below. Kim et al. [2] measured the heat transfer coefficient in forced convection of air across aluminum foams. The authors tested six foams: three of those presented 10, 20 and 40 PPI with a constant porosity of 0.92. Bai and Chung carried out a study on analytical and numerical prediction of heat transfer and pressure drop in open-cell metal foams [3]. Zhao et al. developed a model to characterize the radiative transport

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process in highly porous, open-celled metal foams having idealized cellular morphologies in terms of fundamental radiative parameters such as emissivity, reflectivity and configuration factors [4].Writz [5] developed a semi-empirical model for the combined conduction and convection heat transfer in a thin porous wall. The researcher found that, for the same volume of heat exchanger, the porous matrix provided approximately 1.5 times more heat transfer surface than the offset strip fin array. Singh and Kasana studied computational aspects of effective thermal conductivity of highly porous metal foams. They noted that experimental data for the effective thermal conductivity of metal foams with well characterized micro-structures having wide range of porosity are still in short supply [6].

Li et al. conducted experimental and numerical studies on melting phase change heat transfer in open-cell metallic foams filled with paraffin [7]. Bastawros demonstrated the efficiency of metallic foams in forced convection heat removal [8]. Another similar study is the one conducted by Boomsma et al. where open-cell aluminum foams were compressed by various factors and then fashioned into heat exchangers for electronic cooling applications, where large amounts of heat is dissipated [9]. Boyd and Hooman developed a model to study the viability of the use of metal foam for fuel cells interconnect; they found that the air-cooled metal foam heat exchanger is as efficient as a water cooled one with the same pressure drop [10]. Ghosh presented a theoretical analysis in order to compute interstitial heat transfer coefficients and foam-finned surface efficiency. This simple model has been selected because it takes into account both the interstitial heat transfer coefficient and the foam-finned surface area efficiency [11].

Babcsan et al. conducted a study on thermal and electrical conductivity measurements of aluminum foams [12]. In a study by Dukhan et al. one-dimensional heat transfer model for open-cell metal foam is presented. The model combines conduction in the ligaments and convection to the coolant in the pores. Close correlation was found between the analytical and the experimental results [13]. Dukhan and Quinones (2003) carried out analysis on aluminum foams having different areas, relative densities, filament diameters, and with varying numbers of pores per inch. They found that maximum heat transfer for the aluminum foam occurs at a pore with a Reynolds number of 52 [14]. Sertkaya et al. [15] carried out a study on experimental investigation of thermal performance of aluminum finned heat exchangers and open-cell aluminum foam heat exchangers. They found that for both types of heat exchangers, as the velocity of refrigerant increases, the effectiveness drops and pressure loss increases. Kamath et al. reported the results of an experimental study on flow assisted mixed convection in a vertical channel containing aluminum metal foams having different number of pores per inch, with high porosity in the range of 0.9 to 0.95 [16].

In this study one dimensional heat transfer of open cell aluminum foams with 10, 20, 30 PPI pore densities is investigated both numerically and experimentally. Necessary one dimensional heat transfer equations for the system were developed and solved numerically with a help of a computer program. Gaus-Seidel iteration method was used for the numerical solution. Temperature distributions for metal foams with varying porosities were compared to one another. The effect of velocity on temperature distribution was investigated experimentally. The numerical and experimental results were compared.

2. EXPERIMENTAL STUDY

The experimental setup used in this study is shown in Fig.1. The experimental device used consists of the following major parts: test section, canal, a pressure loss measurement section, a digital manometer, a blower, a frequency adjusting device that controls the fan velocity, a velocity measurement section with a digital anemometer, heated section, and a data logger. Canal cross section and length were designed as (200 x 100 mm) and 160 cm respectively. Air was used as a cold fluid in the canal.

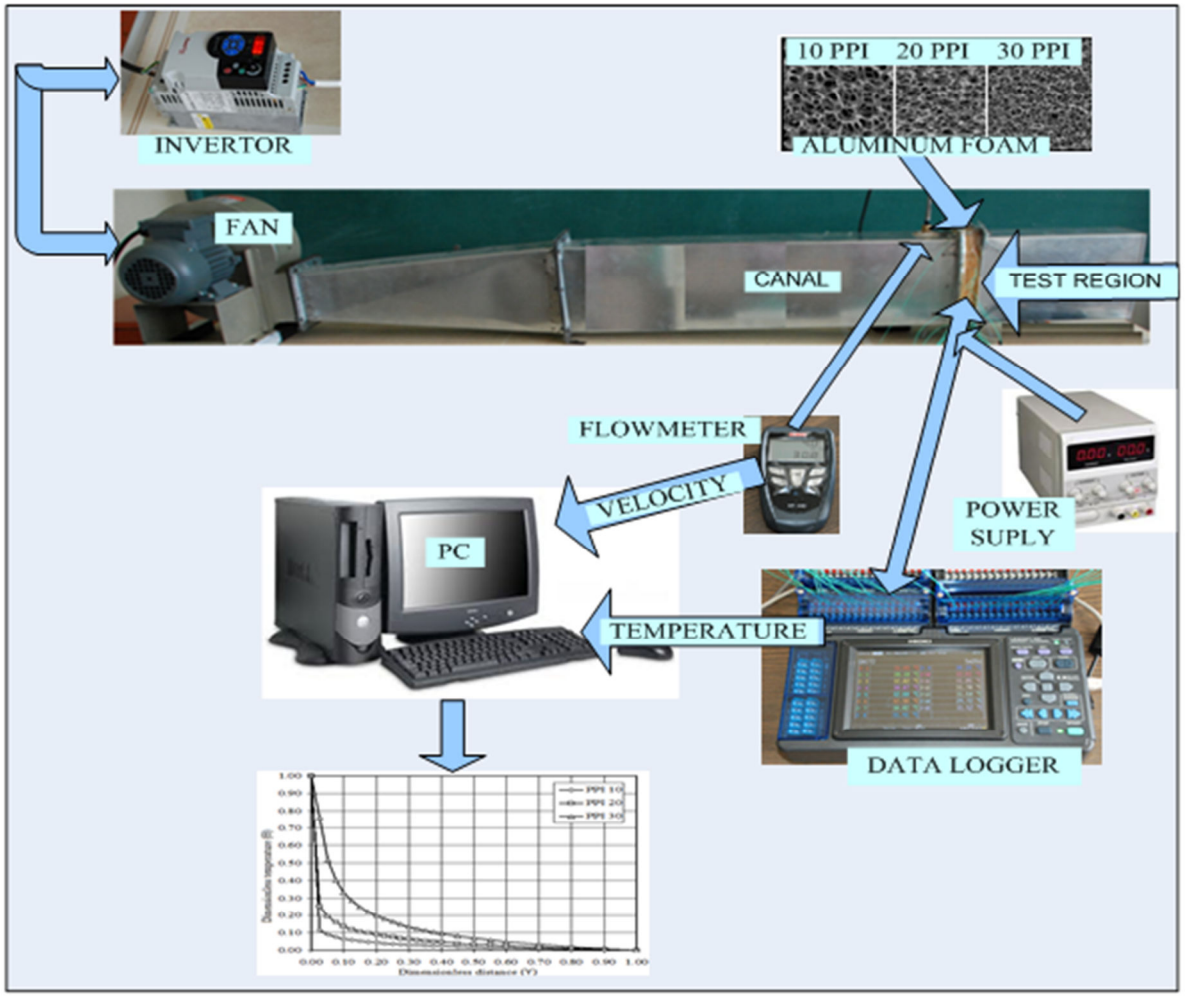

Figure 2.1. Experimental setup

The air flow rate in the canal was adjusted with a frequency adjuster with an accuracy of 1%. In the experiments, input / output temperatures of the cold and hot fluids in the canal, temperature distribution on the aluminum foams and the in-pipe air speed were measured. Temperatures are measured with thermocouples which are having (NiCr-Ni Type, T190-1, 0.5 mm-diameter) properties. Air velocity (u) was raised from 1 m/s to 5 m/s at an increasing interval of 1 m/s. Air velocity measured at the entrance of the test area.

Open-cell aluminum metal foam having dimensions of 200x100x20 mm. In order to maintain homogeneous heat distribution, aluminum plates having dimensions of 200 x 100 x 2 mm were mounted between the flexible heater and the aluminum foams. Air which was used as the cold fluid was blown over the aluminum foams with a blower. Inner channel air velocity was read through a digital anemometer. The characteristic values of open cell aluminum foams used in the study are shown in Table 1 [17]. Locating thermocouples on y direction are shown in Table 2. In order to achieve homogeneous heat transfer to the foams surfaces, a 2 mm thick aluminum plate having the same size as the foams was connected between the flexible heater and the foams. Temperatures of the heating plates were controlled with a 50-150 W ± 0.1 W power source. During the experiments, the base plate temperature varied between 50 ^{o}C and 200 ^{o}C. The system was waited until it reached to a steady state, after which the temperature values for the aluminum foams were transferred into a computer with the help of a data logger. The experimental recorded data were used for calculations and plotting graphs which help determining the system performance.

Table 2.1. Thermo-physical parameters and dimensions for aluminum foams [17]

Pore density (PPI)	**10**	**20**	**30**
Materials	AlSi7Mg	AlSi7Mg	AlSi7Mg
Porosity, ε	0.90	0.90	0.90
Heat conductivity, k(W/mK)	165	165	165
Density, (kg/m^3)	230	230	230
Specific surface, σ (m^2/m^3)	1200	1500	1800
Height, H (mm)	200	200	200
Length, L (mm)	100	100	100
Width, W (mm)	20	20	20

Table 2.2. Locating thermocouples on y direction

Ter. No	**y (mm)**	**Ter. No**	**y (mm)**	**Ter. No**	**y (mm)**
1	0	9	40	17	80
2	5	10	45	18	90
3	10	11	50	19	100
4	15	12	55	20	110
5	20	13	60	21	120
6	25	14	65	22	140
7	30	15	70	23	160
8	35	16	75	24	180
				25	200

3. THEORY

The cellular metal is accepted as a system that transfers heat from a hot surface into a fluid. Heat transfer in open cell aluminum heat exchangers occurs in such a way that convection takes place in between the coils while conduction is involved in transferring the heat between the coils and ambient air. The control volume for numerical solution of heat transfer analysis on the aluminum foam coils is presented in Fig. 2.Aluminum foam samples was insulated in the z direction against outside influences from both sides. These samples are very thin in the x direction. So the temperature decrease is neglected in the direction dx. Therefore, these studies is taken one-dimensional (Y direction) [13].

One dimensional (y-direction) heat transfer is investigated in this study. In addition, bottom surface of the heater is isolated from the external environment. In Fig. 2 for the control element; T_b is base plate temperature, T_∞ is environment (air) temperature, Δx is width, H is height, q_{con} heat transferred by convection in the foam sample, dy is control volume of thickness in the y direction and dz is thickness. The air (cold fluid) was blown onto the aluminum foam on the x direction.

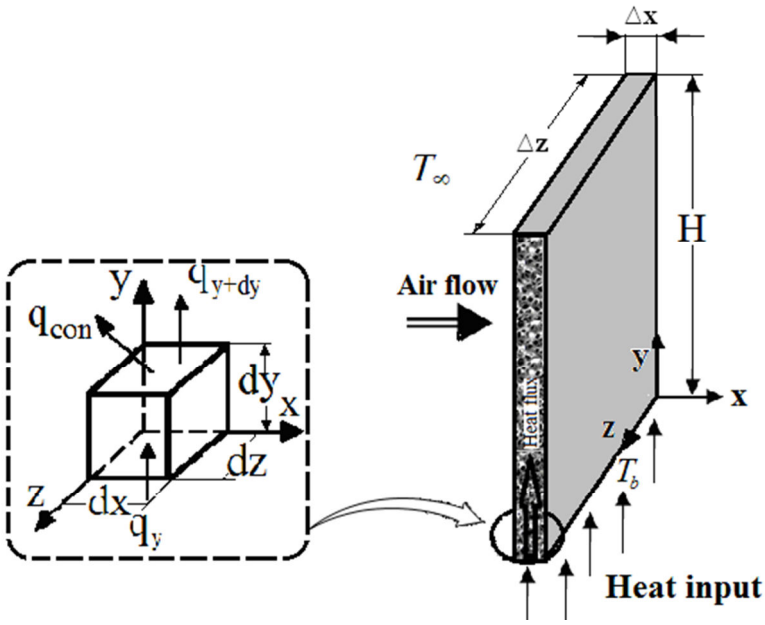

Figure 3.1. Control volume for open cell aluminum foam

The law of conservation of energy for Fig.2 can be written as below:

The Fourier's law of heat conduction and Newton's cooling law of heat convection for the aluminum foam with solid parts and holes are given as follows,

$$q_y = q_{y+dy} + q_{conv} \tag{1}$$

$$q_y = -k_s dxdz(1-\varepsilon)\frac{\partial T_{fm}}{\partial y} - k_f dxdz\varepsilon\frac{\partial T_{fm}}{\partial y} \tag{2}$$

$$q_{y+dy} = -(1-\varepsilon)k_s dxdz\frac{\partial T_{fm}}{\partial y} - (1-\varepsilon)k_s dxdydz\frac{\partial^2 T_{fm}}{\partial y^2} - \varepsilon k_f dxdz\frac{\partial T_{fm}}{\partial y} - \varepsilon k_f dxdydz\frac{\partial^2 T_{fm}}{\partial y^2} \tag{3}$$

and convective heat transfer as follow,

$$q_{conv} = h_{fm}\sigma\, dxdydz\left(T_{fm} - T_\infty\right) \tag{4}$$

where k_s is conductivity coefficient for solid part, ε is porosity, T_{fm} is temperature of aluminum foam, k_f is conductivity coefficient for fluid and h_{fm} convection heat transfer coefficient. The convective surface area is defined using the property of surface density σ, that is the surface area per unit volume. This property is specified by the manufacturer and is expressed as follows.

$$\sigma = \frac{A_{conv}}{dx\, dy\, dz} \tag{5}$$

where A_{conv} is the surface area through which heat transfer by convection takes place.
The heat conduction area in the solid part is given as,

$$A_{cond} = A_c - A_p \tag{6}$$

where A_c is the total cross sectional area and A_p is the area occupied by the pores [19].

Equations 2-5 are substituted into Equation to obtain,

$$-\left(1-\varepsilon\right)k_s\frac{\partial^2 T_{fm}}{\partial y^2}-\varepsilon k_f\frac{\partial^2 T_{fm}}{\partial y^2}+h_{fm}\sigma\left(T_{fm}-T_\infty\right)=0 \tag{7}$$

where s subscript is the solid part, f subscript is the fluid flows, fm aluminum foam and ε is the porosity of the foam. it is a property reported by the manufacturer and ε the porosity of the surface is given as below.

$$\varepsilon = 1 - \left(\frac{V_s}{V_{tot}}\right) \tag{8}$$

where V_s is the volume of solid part and V_{tot} is the volume of fluid section.
Equation 7 can be written as follow,

$$\left(1-\varepsilon\right)k_s\nabla^2 T_{fm}+\varepsilon k_f\nabla^2 T_{fm}-h_{fm}\sigma\left(T_{fm}-T_\infty\right)=0 \tag{9}$$

From here,

$$\nabla^2 T_{fm}=\frac{h_{fm}\sigma\left(T_{fm}-T_\infty\right)}{\left(1-\varepsilon\right)k_s+\varepsilon k_f} \tag{10}$$

Can be written. The terms related to the convective coefficient, surface area per unit volume of the foam, conductivities of the solid and fluid, and the porosity of the metal foam are the foam parameters and they are described by

$$m_{fm}^2=\frac{h_{fm}\sigma}{\left(1-\varepsilon\right)k_s+\varepsilon k_f} \tag{11}$$

where m_{fm} is foam parameters.

$$\nabla^2 T_{fm}=m_{fm}^2\left(T_{fm}-T_\infty\right) \tag{12}$$

Equation 12 can be written as a dimensionless form.

$$\frac{\partial^2\theta_{fm}}{\partial Y^2}-M^2\theta_{fm}=0 \tag{13}$$

where $\theta_{fm}=\dfrac{T_{fm}-T_\infty}{T_b-T_\infty}$ dimensionless temperature $M^2=H^2m_{fm}^2$ is dimensionless the foam parameter, and $Y=\dfrac{y}{H}$ is is dimensionless axial direction along the foam sample.
The boundary conditions for equation (15) are given as follows,

for $Y=0,\quad T_{fm}=T_b,$ and $\theta_{fm}=\dfrac{T_b-T_\infty}{T_b-T_\infty}=1$

for $Y=1,$ and $\dfrac{\partial\theta_{fm}}{\partial Y}=0$

In order to discrete the equation (13), the central difference method is used and written in integral form. If the integral of the expression is written for given boundary conditions, the following equation is obtained.

$$\left[\frac{\theta_{i+2}-\theta_i}{(\delta Y)_{i+1}}-\frac{\theta_i-\theta_{i-2}}{(\delta Y)_{i-1}}\right]-\int_{i-1}^{i+1}M^2\theta\ dY=0 \tag{14}$$

The last term of the equation (18) is,

$$\int_{i-1}^{i+1}M^2\theta_{fm}\,dY=\overline{S}\,\Delta Y \tag{15}$$

where $\overline{S}$ is the linear source term, and can be described as follows [20],

$$\overline{S}=Sc+Sp\,\theta_{fm} \tag{16}$$

Where Sc is a constant term of the linearized source whereas Sp is the temperature coefficient of the linearized source [20]. In this case, equation (18) is given in the following form.

$$\left[\frac{\theta_{i+2}-\theta_i}{(\delta Y)_{i+1}}-\frac{\theta_i-\theta_{i-2}}{(\delta Y)_{i-1}}\right]-(Sc+Sp\theta_i)\Delta Y=0 \tag{17}$$

In order to put the equation given above into a computer form, equation (21) is written in a compact form as below [20],

$$a_P.\theta_P=a_E.\theta_E+a_W.\theta_W+c \tag{18}$$

$$a_P=a_E+a_W+S_P \tag{19}$$

$$a_E=\frac{1}{(\delta Y)_{i+1}}\qquad a_W=\frac{1}{(\delta Y)_{i-1}} \tag{20}$$

Here, $c=Sc=0$ and $Sp=-M^2$ [20],

The boundary conditions are:

for $Y=0$, and $\theta_{fm}=1$

for $Y=1$, $\dfrac{\partial\theta_{fm}}{\partial Y}=0$, $\left(\dfrac{\partial\theta_{fm}}{\partial Y}\right)_e=\dfrac{\theta_{i+2}-\theta_i}{(\delta Y)_e}$, $\theta_{i+2}=\theta_i$

$\left(\dfrac{\partial\theta_{fm}}{\partial Y}\right)_w=\dfrac{\theta_i-\theta_{i-2}}{(\delta Y)_w}$, $\theta_{i-2}=\theta_i$

A program developed by the authors in computer programming language for solution of these expressions was used.
The uncertainty calculations of the experimental results are obtained by the method of Kline and McClintock. The general equation of this method is as follow;

$$w_R=\pm\left[\left(\frac{\partial R}{\partial x_1}w_1\right)^2+\left(\frac{\partial R}{\partial x_2}w_2\right)^2+\ldots+\left(\frac{\partial R}{\partial x_n}w_n\right)^2\right]^{1/2} \tag{21}$$

where, R is a given function of the independent variables $x_1, x_2, \ldots, x_n$. It can be expressed as: $R=R(x_1, x_2, \ldots, x_n)$. Where; $w_1, w_2, \ldots, w_n$ are the uncertainties in the independent variables and w_R is the uncertainty in the result [21]. The

obtained uncertainty ranges using measurement accuracy for U, θ and Y were estimated to be ± 2.51 %, ± 1.20 % and ± 1.41% respectively. The characteristics and uncertainties of the measurement instruments are given in Table 3.

Table 3.1. Characteristics and uncertainties of the measurement instruments

Instrument	Range	Uncertainty
Thermocouple NiCr-Ni T190-1	-25 to +400 [^{o}C]	±0.8 %
Digital anemometer DCFM8901 CFM	125 to 4900 fpm	±2 %

4. RESULTS AND DISCUSSIONS

In this study, modeling of a one dimensional heat transfer model for open cell aluminum foam was made both experimentally and with numerical solution. The materials used for the tests had 10, 20 and 30 PPI features. The porosity for all the three aluminum foams was ε=0.90, specific surface per unit volume for 10 PPI was σ =1200 m^2/m^3, for 20 PPI was σ =1500 m^2/m^3 and for 30 PPI was σ =1800 m^2/m^3. Air was used as a coolant and was blown over the metal foam at varying velocities. One dimensional temperature distribution formulas were developed for the system. The equations were solved by using programming language. Temperature distributions belonging to metal foams with different pore density were compared to each other. Effects of velocity on temperature distribution were studied.

In Fig. 3, experimental variation of θ-Y for aluminum foams with 10, 20 and 30 PPI porous density at 2 m/s is given. It is seen that the temperature drop in specimens with low PPI porous density is faster than that of specimens with high porous densities. The fastest temperature drop occurred at 10 PPI while the slowest was at 30 PPI. An increase in surface area per unit volume led to increasing heat transfer. Variation of θ-Y for numerical solution for the aluminum foams with different specific surface (σ) features from Eq 17 is in given in Fig. 4. As seen from the graph, the temperature drop is mild at higher values of surface area per unit volume while it is rapid at lower surface area values. It is therefore understood from here that, much heat is stored at higher values of surface area per unit volume and that good heat transfer is exhibited here. In both graphs, rapid temperature drop is observed on areas near the heaters while the drop becomes mild away from the heaters. A good correlation was observed when the experimental results were compared with the numerical results.

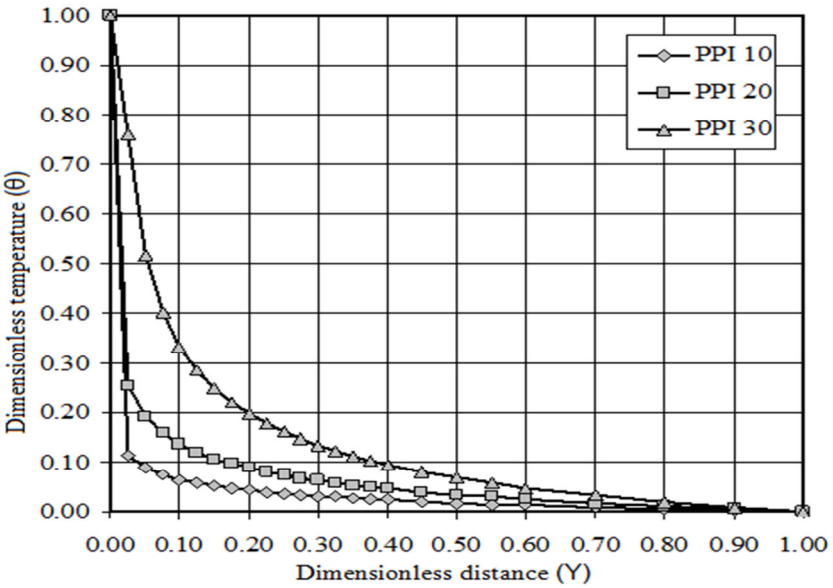

Figure 4.1. Variation of θ-Y for PPI 10, 20 and 30 features foam materials at 2 m/s based on experimental results

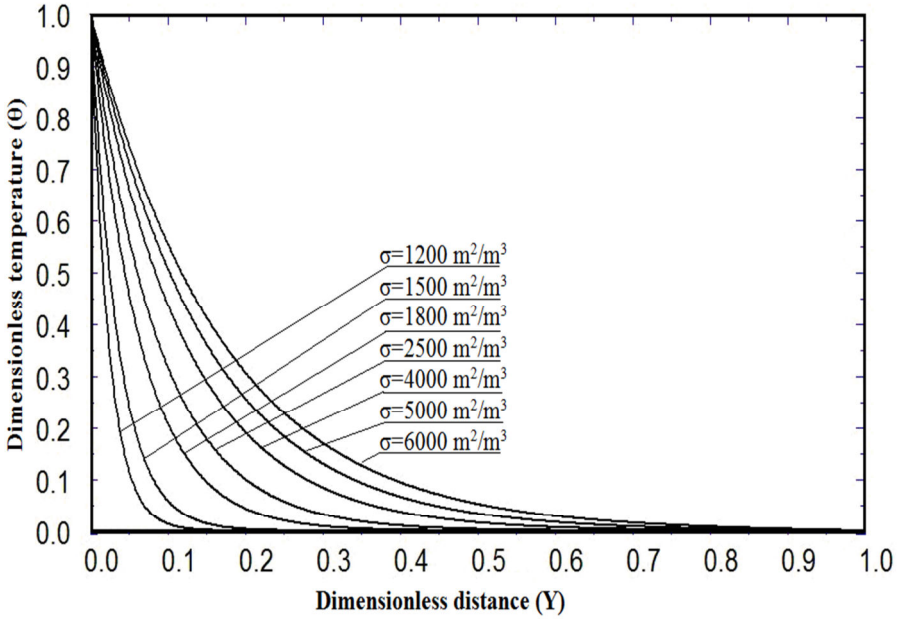

Figure 4.2. Variation of θ-Y based on numerical solution at different σ (m^2/m^3) values (where is ε=0.90, h_{fm}=200 $Wm^{-2}\,{}^oC^{-1}$, k_s=165 $Wm^{-1}\,{}^oC^{-1}$and k_f=0.031 $Wm^{-1}\,{}^oC^{-1}$)

In Fig. 5, variation of θ-Y for 30 PPI obtained from experimental results conducted at various speeds is given. As seen from the figure, at low speeds the temperature drop is slow but as the speed increases, the temperature drop becomes rapid. The slowest temperature drop occurred at the speed of 1 m/s while the fastest temperature drop was observed at 5 m/s. The rapid temperature drop at higher speeds indicates that heat transfer increases with the (air) speed.

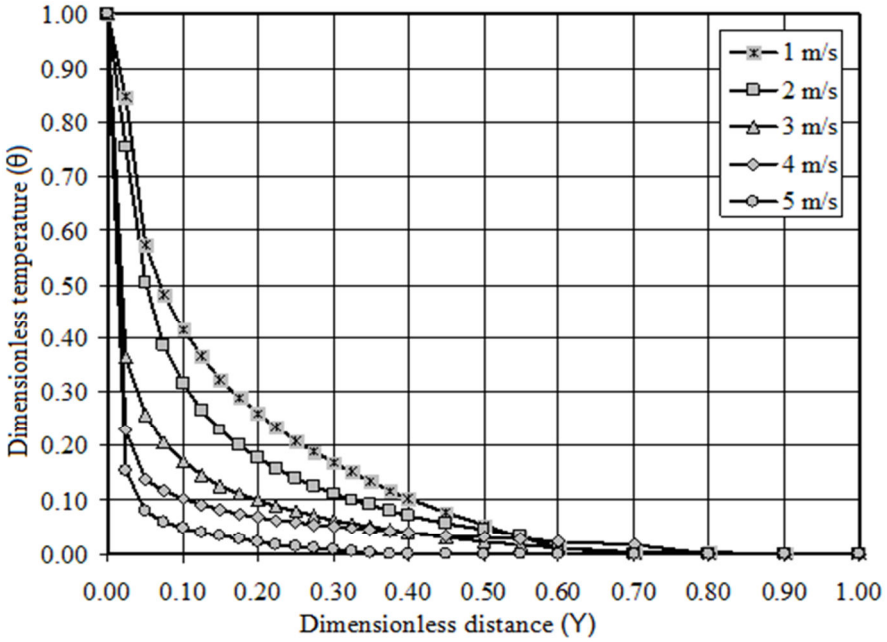

Figure 4.3. Variation of θ-Y for 30 PPI features foam materials at various velocities based on experimental results

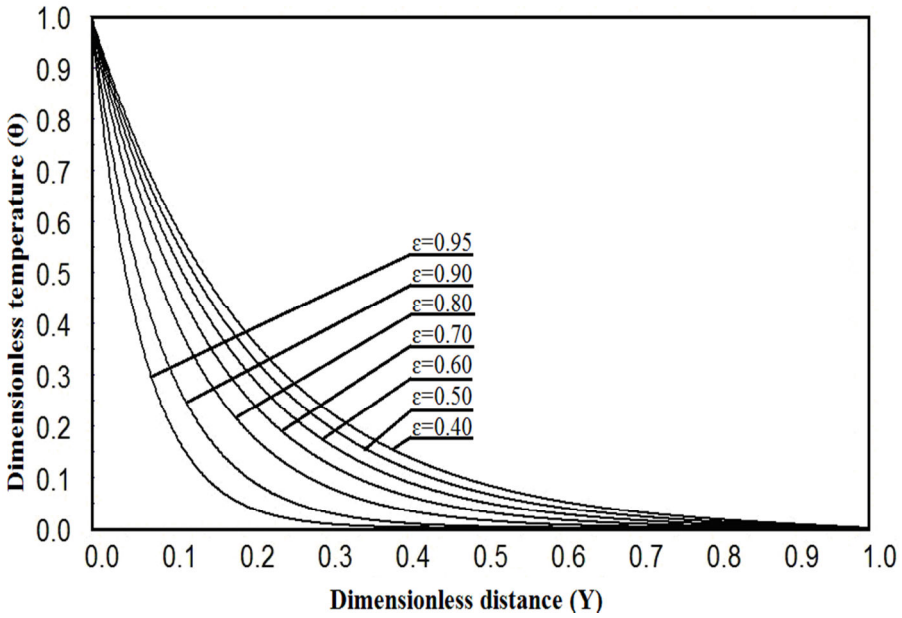

Figure 4.4. Variation of θ-Y for open cell foam materials at various pore density based on numerical solution

In Fig. 6, variation of θ-Y obtained from numerical solution of various pore densities is given. From the figure, it is seen that as the pore density increases, temperature drop becomes rapid. As for the numerical solution obtained from Eq. 20, it was found that the slowest temperature drop is at ε=0.40 pore density and the fastest temperature drop is ε=0.95 pore density at. As the pore density falls the heat transfer by conduction increases too.

5. CONCLUSION AND FUTURE WORKS

In this study, one dimensional heat transfer distribution of open cell aluminum metal foams possessing 10, 20 and 30 PPI features was investigated both experimentally and numerical. The results obtained were summarized as below.

Lower temperature drops are observed as you move along the Y axis. This is situation is faster on sections near the heater and slows down far from the heater. It was found both experimentally and theoretically that the temperature drop is slow on areas with higher surface areas per unit volume (σ) and faster on areas with lower surface areas per unit volume (Figs.3-4). It was experimentally found that temperature drop is slow on lower speeds and becomes faster when the speeds increase. The slowest temperature drop occurred at the speed of 1 m/s and the fastest temperature drop was at 5 m/s (Fig. 5). Temperature drop rate increases with the pore density. In other words; the temperature drop is faster on higher pore densities and slower on lower pore densities. (Fig. 6).

In the θ-Y graph drawn based on the theoretical equation developed, it is seen that heat transfer at low (σ) values is faster than that at high (σ) values. These results are in good correlation with those found on the graphs drawn for PPI 10, 20 and 30. It is hoped that more interesting results will follow in further studies on the topic as follows: Comparison can be made on heat transfer rates of different metal foams. Performance of metal foams as insulators can be studied. Two-dimensional analysis of metal foams temperature distribution can be carried out to study its performance.

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CONFLICT OF INTEREST

No conflict of interest was declared by the authors.

Nomenclature

A	area, m^2
a	general coefficient
c	source term coefficient
dx	height of the control volume, m
dy	depth of the control volume, m
dz	thickness of the control volume, m
h	convection heat transfer coefficient inside the pores, $Wm^{-2}\,{}^oC^{-1}$
k_s	conductivity coefficient for solid part, W m^{-1} $^oC^{-1}$
k_f	conductivity coefficient for fluid, W $m^{-1}\,{}^oC^{-1}$
H	length of foam sample, m
m	foam parameter, m^{-2}
M	dimensionless the foam parameter
PPI	Number of Pores Per Inch
q	heat transferred in the foam sample, W
S	source term
$\overline{S}$	linearized source term
T	temperature, K
V	volume, m^3
x, y, z	axial direction along the foam sample, m
Y	dimensionless axial direction

Greek symbols

ε	porosity
θ	dimensionless temperature
σ	surface area per unit volume of foam, m^2m^{-3}

Subscripts

b	base
c	cross sectional
cond	conduction
conv	convection
f	fluid
fm	foam
p	pore
s	solid
tot	total
∞	environment

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