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Convective Heat Transfer in Open-Cell Metal Foams

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Convective heat transfer in aluminum metal foam sandwich panels is investigated with potential applications to actively cooled thermal protection systems in hypersonic and re-entry vehicles. The size effects of the metal foam core are experimentally investigated and the effects of foam thickness on convective transfer are established. Four metal foam specimens are utilized with a relative density of 0.08 and pore density of 20 ppi in a range of thickness from 6.4 mm to 25.4 mm in increments of approximately 6 mm. An exact-shape-function finite element model is developed that envisions the foam as randomly oriented cylinders in cross flow with an axially varying coolant temperature field. Our experimental results indicate that larger foam thicknesses produce increased heat transfer levels in metal foams. Initial FE simulations using a fully developed, turbulent velocity profile show the potential of this numerical tool to model convective heat transfer in metal foams.

I. Introduction

Metal foam sandwich panels have been proposed as alternative multi-functional materials for structural thermal protection systems in hypersonic and re-entry vehicles[1].[2] This type of construction offers numerous advantages over other actively cooled concepts because of the unique properties of metal foams. These materials, when brazed between metallic face sheets, are readily suited to allow coolant passage without the addition of alien components that may compromise structural performance. Moreover, the mechanical properties can be varied to suit different structural needs by varying the foam relative density. From a heat transfer point of view, these materials have been shown to be exceptional heat exchangers primarily due to the increased surface area available for heat transfer between the solid and fluid phases.

The thermo-mechanical response of metal foam sandwich panels has been recently studied and characterized.[2] In particular, it has been shown that using air as coolant at sufficiently high velocities, the strain due to buckling of these structures under thermo-mechanical loads can be virtually eliminated. The implementation of these materials in thermal protection systems, however, requires that a proper heat transfer model exists that allows the coupling between the thermo-mechanical and heat transfer problems to be properly analyzed. In other words, it is necessary to understand how different foam properties such as relative density, pore density, and foam thickness will affect the heat loads that this type of structural component can remove.

Heat transfer in metal foams has been a subject of active research in recent years. Lu et al.[3] developed an analytical model envisioning the foam as an array of mutually perpendicular cylinders subjected to cross-flow. In this study, a closed-form expression for the convective coefficient of a foam-filled channel with constant wall temperatures was presented based on foam geometry and material and fluid properties. These authors reported that the simplifying assumptions used in their analysis were likely to lead to an over-prediction of the actual heat transfer level. This model has been partially validated by Bastawros and Evans[4] who performed forced convection experiments on aluminum foams adhered to silicon substrate face sheets. These authors reported that the predictions of Lu et al.[3] regarding the dependence of the convective coefficient on coolant velocity and strut diameter were qualitatively consistent with their observations, but that the foam thickness effects were not adequately modeled. In particular, they reported that the heat dissipation rate

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and convective coefficient decrease with increasing foam thickness. Moreover, they introduced an empirical modification to this model based on a tetrakaidecahedral unit cell and their experimental results. These modifications, however, do not completely solve the qualitative limitations of the model.

Calmidi and Mahajan[5] also carried out experimental studies on forced convection in aluminum metal foams using air as the cooling medium. This study focused on the effects of porosity and pore density. In their numerical analysis, they solved a semi-empirical form of the momentum equation governing the flow of fluid through porous media to obtain the velocity profile in the foam-filled channel. This profile is then used in solving a volume-averaged form of the energy equation which yields models for the foam thermal dispersion conductivity and interstitial heat transfer coefficient. Their results reportedly agree favorably with their experimental data and those published in the open literature.

The literature review performed by the present authors did not yield a definite model that properly includes the effects of foam thickness on the convective heat transfer of metal foams. As a result, the objective of this study is to experimentally investigate the size effects present in convective heat transfer of aluminum metal foams and to numerically model them using a finite element approach. Four different foam thicknesses are investigated to determine the dimensions at which size effects disappear and metal foams can be considered as a continuum for heat transfer. In the analysis, the metal foam is envisioned as an array of cylinders in cross flow but a highly ordered structure is not assumed, as has been previously done. The velocity profile inside the foam-filled duct is calculated using the approach of Mahajan,[5] along with the permeability and inertial coefficient determined experimentally. The governing equation for cylinders in cross-flow is then used to derive a potential that yields the necessary element thermal stiffness matrices and forcing vectors. These results are later used to calculate the temperature distribution on the surfaces of the sandwich panel.

II. Experiments

Four metal foam sandwich panels were used to experimentally study the effect of foam thickness on the convective heat transfer coefficient of aluminum metal foams. The sandwich construction consists of a metal foam core brazed to two aluminum face sheets. The ratio of foam thickness to face sheet thickness was maintained constant at approximately 6 to 1 to ensure that thermal resistance effects were similar in all cases. The face sheets are constructed of aluminum T6061-T6 whereas the metal foam core alloy is T6101-T6. The panels used belong to the Duocel family of metal foams manufactured by ERG Materials and Aerospace Corporation. The set of specimens used consisted of four different foam thicknesses ranging from 6.4 mm to 25.4 mm in increments of 6.4 mm. The length and width of each specimen was kept constant at 203.5 mm and 51.0 mm respectively. In order to properly isolate the thickness effects, the foam relative density and pore density were maintained constant at 0.08 and 20 pores per inch (ppi) respectively (values provided by manufacturer). A schematic of a typical specimen is shown in Figure 1.

Each specimen was placed inside a custom-built Plexiglas case which provided insulation from the ambient conditions as well as a means to prevent airflow bypass. Low conductivity styrofoam ($k = 0.037$ W/mK) was placed on the sides of the case to prevent and measure existing heat losses. The top of each specimen was insulated using insulating melamine foam. A sample insulated specimen is shown in figure 2. The specimens were heated on the top side using four patch heaters (Minco Inc, HK913P), as shown in figure 3. The upper side was heated in order to avoid natural convection and buoyancy effects. In this way, the density of the cooling fluid increases from top to bottom, therefore avoiding the formation of circulating patterns (which would increase the resistance to forced convection). The heat entering each panel was controlled by varying the voltage supplied to the heaters using an adjustable Variac, and measured through Ohm's Law. The other side of the panel was insulated using a Plexiglas sheet. Temperature measurements were taken at three axial locations on both the top and bottom face sheets using type K thermocouples (Omega Engineering, 5SC-TT-K-36-36); these thermocouples were then connected to a data acquisition system for recording.

Forced convection was achieved by running air at different velocities through the metal foam. The mass flow rate was measured using an in-line pneumatic flow meter (Omega Engineering, FL6711A). The average air velocity was then calculated using the cross sectional area and volumetric flow rate. For each specimen a range of velocities between 0 and 10 m/s was covered; in some cases the velocity was increased until 19 m/s. The temperature and pressure of the air entering and exiting the foam core were also measured. Each experiment lasted until all temperatures reached steady state conditions.

The heat losses through the sides were calculated using the temperature reading from two thermocouples embedded in the insulating styrofoam, along with this material's thermal conductivity and thickness. This procedure has been shown to accurately represent the lateral heat losses in this type of experimental arrangement.[6] The average convection coefficient was calculated using the average temperature of the heated surface, the temperature of the incoming air, and the geometry of the foam through

$$h = \frac{Q}{LW\Delta T} \tag{1}$$

The heat entering the sandwich panel and the temperature difference are defined based on the resistance of each heater as well as the losses through the sides. The term k_s represents the thermal conductivity of the styrofoam, A_c the lateral foam area, b the thickness of the styrofoam and ΔT is the difference in the readings of the thermocouples embedded in the insulation. These terms are defined in equation 2.

$$\Delta T = \frac{1}{3}\left(\sum_{i=1}^{3} T_i\right) - T_\infty \tag{2a}$$

$$Q = V^2 \sum_{i=1}^{3} \frac{1}{R_i} - k_s A_c \left(\frac{\Delta T_{loss}}{b}\right) \tag{2b}$$

III. Experimental Results

Figure 4 and Figure 5 show, respectively, a typical set of experimental results for the temperature distribution of the heated and unheated side of the sandwich panel as a function of time. The temperature of both surfaces increases with axial position indicating that the cooling fluid does not reach a constant temperature shortly after entering the foam duct. The effect of thickness on the temperature gradient across the metal foam (difference between top and bottom surfaces) is illustrated in Figure 6. Clearly, as the foam thickness is increased, the difference between both surfaces increases. This trend indicates that the heat transfer rate increases with increasing thickness, as a lower temperature implies that more heat has been carried away by the cooling fluid. This result is also consistent with the convective coefficient trends presented below for the same reasons. It is important to mention that the temperature gradient decreases with increasing air velocity due to the constant heat flux boundary condition. If a constant temperature were prescribed at the heated surface, we would observe an increase in the temperature gradient with increasing air velocity.

The experimental results obtained for the convective heat transfer coefficient are summarized in Figure 7. It is clear that in all thicknesses studied, increasing the average velocity of the cooling fluid increases the convection coefficient. This can be physically explained by an increase in the turbulence level of the fluid in the foam core which enhances the heat transfer between the two phases. This trend is observed until average velocities near 15 m/s where the coefficient seems to reach a plateau condition, possibly due to constant turbulence intensity.

The effect of foam thickness is also evident from the figure: an increased convection coefficient is observed for larger foam thicknesses. This trend, although relatively weak, is expected, because a greater foam thickness implies a larger amount of effective surface area available for heat transfer. The trends observed in these results are inconsistent with those presented by Bastawros[4] where the convection coefficient reached a maximum value at approximately 2 m/s and steadily decreased thereafter, and where the convective coefficient was observed to decrease with increasing foam thickness. The experimental results obtained for the pressure drop across the sandwich panels are shown in Figures 8 through 10.

IV. Numerical Model

In this section, the governing differential equation for a cylinder in cross flow is used to derive a weak form that allows a solution to the convective heat transfer problem using a finite element approach. This description is more suited to metal foams as its constituent struts can be easily envisioned as individual components, that is, the method will not lead to the discretization of a continuous structure, as in the case of a beam or a rod, but will instead embrace the nature of metal foams in their description. Additional advantages of this approach include:

- Easy calculation of properties using cylinders of different orientations as all calculations take place in the local coordinate system
- Networks of cylinders with different geometries (lengths and diameter) can be easily modeled
- Cylinders with different material properties can be employed, thereby describing "composite foams"
- The discretization of the medium allows the introduction of an axially varying temperature field for the coolant phase

A. Derivation of Element Matrices

The convective heat transfer model proposed in this study envisages the metallic foam as an array of randomly oriented cylinders subjected to cross-flow. To develop the temperature distribution of such a cylinder, a standard fin analysis is used regarding the heat flux as a negative source that is proportional to the cylinder's geometry and convective coefficient. For a cylinder with diameter d, thermal conductivity λ, and convective coefficient h subjected to cross flow of a coolant with temperature T_∞, the governing equation has the form:

$$\lambda \frac{d^2T}{dz^2} - \frac{4h}{d}(T - T_\infty) = 0 \tag{3}$$

Equation 3 is used to derive a weak form that will allow the determination of the corresponding element thermal stiffness matrix and forcing vector. To find this weak form, (3) is multiplied by a small temperature variation δT and integrated over the length, l, of one cylinder:

$$\int_0^l \left(\lambda T'' \delta T - \frac{4h}{d} T \delta T \right) dz = - \int_0^l \frac{4h}{d} T_\infty \delta T dz \tag{4}$$

Using integration by parts on the first term of the left hand side, the following is obtained:

$$-\frac{1}{2} \int_0^l \left[\lambda \left(\frac{dT}{dz} \right)^2 - \frac{4h}{d} T^2 \right] dz + \frac{1}{2} \int_0^l \frac{4h}{d} T_\infty T dz = constant \tag{5}$$

The temperature of the cylinder is now described using two interpolating shape functions N_1 and N_2. Defining $\mathbf{T}$ as the vector of temperature degrees of freedom and $\mathbf{N}$ as the shape function vector, (5) takes the form:

$$-\frac{1}{2} \int_0^l \left(\lambda \mathbf{T^T N'^T N' T} + \frac{4h}{d} \mathbf{T^T N^T N T} - \frac{4h}{d} T_\infty \mathbf{N T} \right) dz = 0 \tag{6}$$

One clear advantage of using a finite element approach is now introduced. The interpolation functions N_1 and N_2 are defined as the solution to the homogenous form of the governing equation thereby becoming "exact" shape functions. As a result, we have:

$$N_1 = \frac{\exp\left(-\sqrt{K} z\right) - \exp\left(-2\sqrt{K} L\right) \exp\left(\sqrt{K} z\right)}{1 - \exp\left(-2\sqrt{K} L\right)} \tag{7}$$

$$N_2 = \frac{\exp\left(-\sqrt{K} L\right) \exp\left(\sqrt{K} z\right) - \exp\left(-\sqrt{K} L\right) \exp\left(-\sqrt{K} z\right)}{1 - \exp\left(-2\sqrt{K} L\right)} \tag{8}$$

The parameter K is defined for simplicity as

$$K = \frac{4h}{\lambda d} \tag{9}$$

Substitution of (7) and (8) into (6) yields:

$$K_{11} = K_{22} = \frac{\left[\exp\left(2\sqrt{K}L\right) + 1\right]\sqrt{K}}{\exp\left(2\sqrt{K}L\right) - 1} \tag{10a}$$

$$K_{12} = K_{21} = \frac{-2\sqrt{K}\exp\left(\sqrt{K}L\right)}{\exp\left(2\sqrt{K}L\right) - 1} \tag{10b}$$

$$P_{11} = P_{21} = \frac{\sqrt{K}T_\infty\left[\exp\left(\sqrt{K}L\right) - 1\right]}{\exp\left(\sqrt{K}L\right) + 1} \tag{10c}$$

The calculation of the heat removed by each element is easily carried out as:[7]

$$q_e = h\left(T_{av}^e - T_\infty\right)A_s \tag{11}$$

In (11), the term h is the convective coefficient associated with a cylinder in cross-flow, A_s is the surface area through which heat transfer takes place, and the average temperature of each cylinder is defined as:

$$T_{av} = \frac{1}{l}\int_0^l \left(T_1N_1 + T_2N_2\right)dz \tag{12}$$

B. Development of Velocity Profile

The velocity of the cooling fluid plays a critical role in the convective heat transfer coefficient associated with a cylinder in cross flow (see section C), and therefore its accurate representation is essential in the present analysis. In order to determine what the velocity of the fluid is, the momentum equation describing the flow through a porous medium (equation 13) is utilized.

$$\frac{\rho}{\epsilon^2}\nabla\cdot\mathbf{uu} = -\nabla p + \frac{\mu}{\epsilon}\nabla^2\mathbf{u} - \frac{\mu}{K}\mathbf{u} - \frac{\rho f}{\sqrt{K}}\|\mathbf{u}\|\mathbf{u} \tag{13}$$

Equation 13 describes the full three dimensional velocity field based on the properties of the medium such as its permeability (K), inertia coefficient (f), and porosity (ϵ). These foam characteristics are determined experimentally from the pressure drop by fitting the data to a Forchheimer extended Darcy's equation[6] (a summary of these results is presented in table 1). In the present analysis, only the one-dimensional velocity distribution along the thickness of the foam is necessary. This is a consequence of the assumption of a fully developed coolant velocity (parallel flow). Using only the axial component of the velocity and neglecting changes along the axial direction, (13) can be simplified into the following scalar equation:

$$\frac{\mu}{\epsilon}\frac{\partial^2 u}{\partial y^2} - \frac{dp}{dx} - \frac{\mu}{K}u - \frac{\rho f}{\sqrt{K}}u^2 = 0 \tag{14}$$

Following the approach of Vafai and Tien,[8] equation 14 is non-dimensionalized using the foam half-duct height (H) and the centerline velocity u_∞ (see Figure 9 for a description of the geometry used in this process):

$$u^* = \frac{u}{u_\infty} \tag{15a}$$

$$y^* = \frac{y}{H} \tag{15b}$$

In this way, (14) can be expressed in non-dimensional form as follows:

$$\frac{\partial^2 u^*}{\partial y^{*2}} = \frac{1}{Da}\left(u^* - 1\right) + \frac{\Lambda}{\sqrt{Da}}\left(u^{*2} - 1\right) \tag{16}$$

The Darcy number, Da and inertia function Λ are defined as:

$$Da = \frac{K}{H^2 \epsilon} \tag{17a}$$

$$\Lambda = \frac{\sqrt{\epsilon} f H u_\infty}{\nu} \tag{17b}$$

The exact solution to (16) is derived by Vafai[8] and is reproduced below:

$$u^* = 1 - \frac{A+B}{A} sech^2 \left[D \left(y^* + C \right) \right] \tag{18}$$

For simplicity, the following constants are defined:

$$A = \frac{2\Lambda}{3\sqrt{Da}} \tag{19}$$

$$B = \frac{1}{Da} + \frac{4\Lambda}{3\sqrt{Da}} \tag{20}$$

$$D = \frac{\sqrt{A+B}}{2} \tag{21}$$

$$C = -\frac{1}{D} sech^{-1} \left(\sqrt{\frac{A}{A+B}} \right) - 1 \tag{22}$$

It is important to mention that the use of (18) requires the knowledge of the centerline velocity of the fluid in the foam-filled channel. As was mentioned in section II, the experimental results only yield the average velocity in the duct. In order to resolve this, a mass balance calculation is performed in which an initial value for the centerline velocity is specified (usually equal to the average velocity since the centerline value is expected to be larger[9]) and used to integrate a modified version of (18) until the mass flow rate is equal to the one obtained using the average velocity. This process is mathematically stated as:

$$\int_{-H}^{H} u_c u^* dy = 2 u_0 H \tag{23}$$

A sample velocity profile obtained using (18) and (23) is shown in Figure 11. The boundary layer thickness is very small due to the low permeability of the medium. The significance of this result and its impact on the heat transfer rate across the thickness of the foam is clear from the figure: the velocity at the ends is zero thereby enforcing the condition that only conduction occurs at both ends. Neglecting the viscous effects of the flow through the foam, would mean that the heat transfer rate at the foam-facesheet joints is assumed to be equal to that at any other point, which could in turn overpredict the actual heat transfer level.

C. Effect of Cylinder Orientation

The final piece necessary to complete the description and obtain a solution for the problem using the finite element model developed above is to account for the orientation of each element, which is here assumed to be random. This parameter may also be determined from microscopic examination of the metal foam samples. The orientation of the cylinder is introduced through the velocity of the cooling fluid, which is vectorially decomposed into its parallel and perpendicular components with respect to the cylinder coordinate system based on the angle θ that it makes with its local vertical.

Even though the determination of the convective coefficient for a cylinder in cross flow is still an active area of research, several semi-empirical correlations have been proposed[10] and are adopted in the present analysis. Most of these results indicate that the convection coefficient is proportional to the fluid Prandtl and Reynolds numbers as:

$$h = \frac{C \lambda Re_D^m Pr^{\frac{1}{3}}}{d} \tag{24}$$

In (24), Re_D is the Reynolds number based on the cylinder diameter and pore velocity. The constants m and C are empirical coefficients that have been shown to depend on the geometry of the cylinder, in particular, its specific cross-sectional area. In the analysis presented in this paper the following empirical correlation will be used:

$$h = \frac{1.8\lambda Re_D^{0.55} Pr^{\frac{1}{3}}}{d} \tag{25}$$

The only variable in (25) becomes the Reynolds number which will vary along the length of the cylinder according to the solution of the momentum equation through porous media (Eq. 18). Since the velocity of the cooling fluid is only included in this term, the modification mentioned above is included in the Reynolds number, so that the effective h which takes into account the orientation of the cylinder is:

$$h_{eff} = \frac{1.8\lambda Re_D^{0.55} cos(\theta)^{0.55} Pr^{\frac{1}{3}}}{d} \tag{26}$$

One important assumption made in the analysis is clear from (26); heat transfer in the axial cylinder direction is assumed to be zero, that is, heat transfer of the cylinder in longitudinal flow is not included. The implications of this assumption are discussed in section V.

V. Results and Discussion

In order to determine how the numerical predictions compare to the experimental results, simulations were run with the experimental data as input for a range of foam thicknesses and air velocities covering the specimens and conditions studied. In particular, the temperature of the entering and exiting air was specified. The heat flux boundary conditions were enforced by specifying a constant heat flux equal to that used experimentally (accounting for heat losses) on the top side, and insulation on the bottom side. Several other sandwich panel parameters were also specified such as its length, width, cross-sectional area and thermal conductivity. For a summary of the data, see table 2. The finite element model utilized 160 elements along the axial dimension (this number was selected based on the pores per inch and length of the specimens), and 100 elements along the thickness.

Figures 13 through 16 show the results obtained for three different foam thicknesses. It can be seen that the predictions of the FE model become more accurate as the foam thickness is increased. For the 6.4-mm panel we see that the slope of the temperature profile produced numerically is somewhat deviated from the slope of the temperature profile obtained experimentally, especially for the temperature of the insulated side. Because of the small thickness of the metal foam, it is possible that size effects are predominant in this case, in other words, the number of metal foam struts across the thickness of the foam is so small (approximately four in average) that the entire heat transfer process is dominated by boundary layer effects. The hypothesis that the physical process governing the heat transfer in the thin panel is not completely similar to that in the other specimens is also apparent in Figure 7, where the results for the convective coefficient of the 6.4-mm panel lie considerably far from the others.

As the thickness is increased we observe a significant improvement in the agreement between the experimental and numerical results, especially for the insulated side. Figure 13 shows that for the 12.7-mm-thick panel the model predictions are within 0.5 degrees of the experimental results for the insulated side at two axial locations. Finally, Figure 14 shows how at the largest foam thickness tested the results agree very well.

Figures 15 through 18 show the effect of varying the cylinder orientation on the numerical results for the temperature of both surfaces. The cylinder orientation is defined through the angle that it makes with its local vertical. In the tests performed, a random cylinder orientation was compared with several regular configurations, i.e., by selecting a particular angle and assuming the cylinders were arranged in a "zig-zag" manner alternating between positive and negative values of the angle. As expected, increasing the angle that the cylinder makes with its local vertical, produces a decreased heat transfer level (as the axis of the cylinder becomes parallel to the flow) which is evident from the higher temperatures resulting for the 70 degree case in all thicknesses. The figures also show that altering the orientation of the cylinder has a stronger effect on the temperature of the heated surface, as the envelope of temperatures is thinner for the insulated part. Additionally, the variations that occur in cylinder configuration from specimen to specimen are also clear from these results, as different sandwich panels require different angle configurations to closely match the experimental results.

Although varying the orientation of the cylinders in the metal foam produces different results, the magnitude of these differences is in most cases smaller than 15 %, indicating that the exact configuration of the metal foam is not a crucial factor in determining its heat transfer properties. Other foam properties, in particular its thickness and strut size appear to have a much more important influence on the performance of these materials as heat exchangers.

The results indicate that modeling the metal foam as an array of randomly oriented cylinders in cross flow properly captures the influence of the foam thickness on its heat transfer properties. This is evident from the fact that as the foam thickness is increased, the numerical results for the temperature difference between the heated and insulated surfaces increases accordingly.

VI. Concluding Remarks

The effect of foam thickness on the convective heat transfer of metal foam sandwich panels has been experimentally investigated and modeled using a finite element approach. Four different foam thickness ranging from 6.4-mm to 25.4-mm have been tested and results for the convective heat transfer coefficient have been obtained. It has been determined that increasing the foam thickness produces an increased heat transfer level as revealed by a larger convective coefficient and a larger temperature gradient. The heat transfer process in the foam has been modeled using a finite element approach that accomodates metal struts of different orientations. These results have been shown to agree favorably with experimental data. Both the experimental results and the numerical predictions indicate that size effects are present in the heat transfer properties of metal foams as revealed by the substantial differences between the 6.4-mm thick panel and the rest of the specimens. The numerical predictions capture the appropriate trend of increased heat transfer with increasing foam thickness and indicate that the exact orientation of the foam struts is of secondary importance, while the foam thickness and strut size are the dominant properties on convective heat transfer.

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Table 1. Summary of Foam Thermophysical Properties.

Foam Thickness (mm)	Permeability (K, m^2)	Inertia Coefficient (f)
6.4	1.9e-6	0.2084
12.7	6.7e-8	0.0327
25.4	9.1e-8	0.0048

Table 2. Properties of Sandwich Panel and Metal Foam.

Parameter	Description	Value Used
L	Sandwich Panel Length	0.203 m
w	Width	0.051 m
d	Diameter of Foam Strut	0.00044 m
λ	Thermal Conductivity	218 W/m^2K

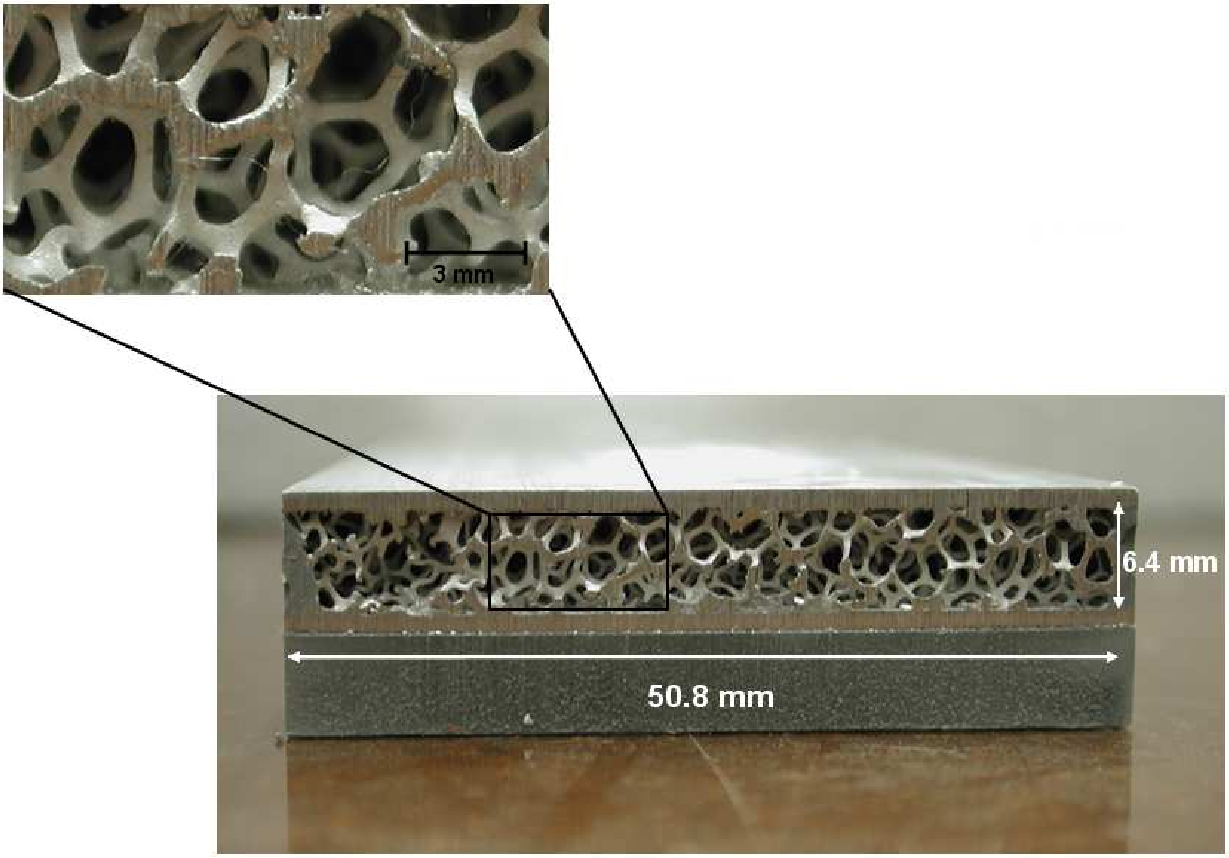

Figure 1. Geometry of a sample metal foam sandwich panel used in the present study. The specimen shown has a foam thickness of 6.4 mm and a face sheet thickness of approximately 1.1 mm.

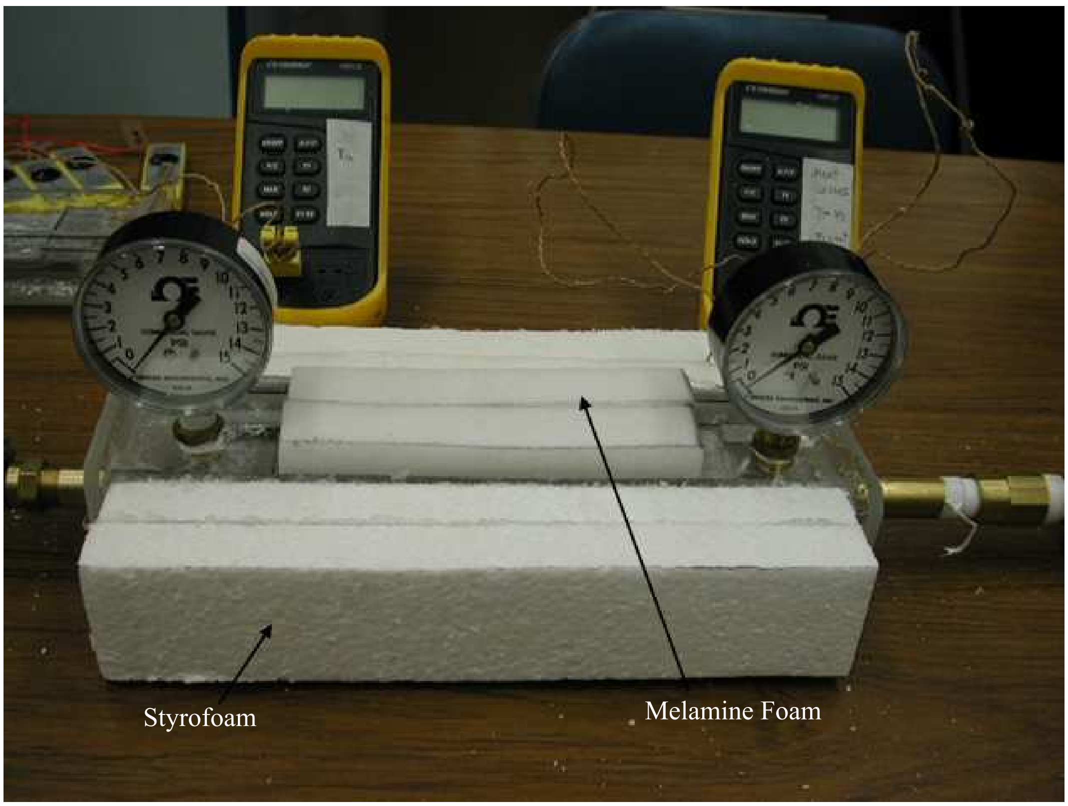

Figure 2. Detail of fully insulated metal foam sandwich panel specimen.

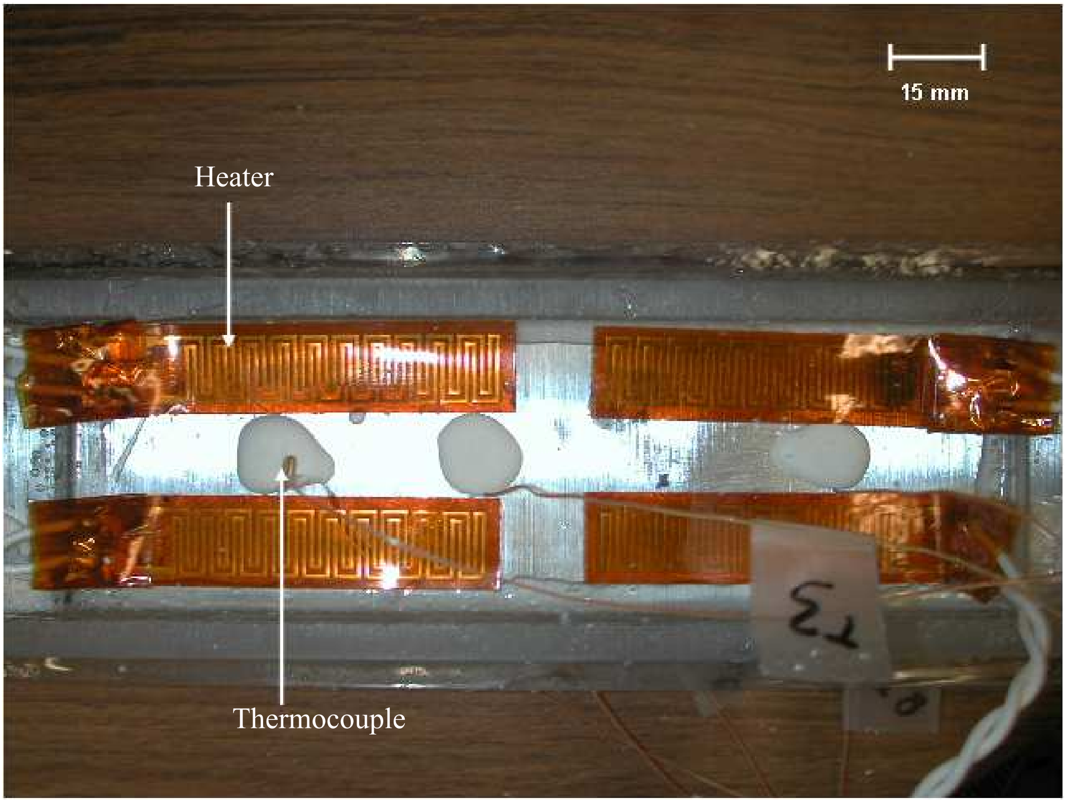

Figure 3. Detail of Heated Surface in metal foam sandwich panel specimen. Each heater has dimensions of 76 mm long by 15 mm wide.

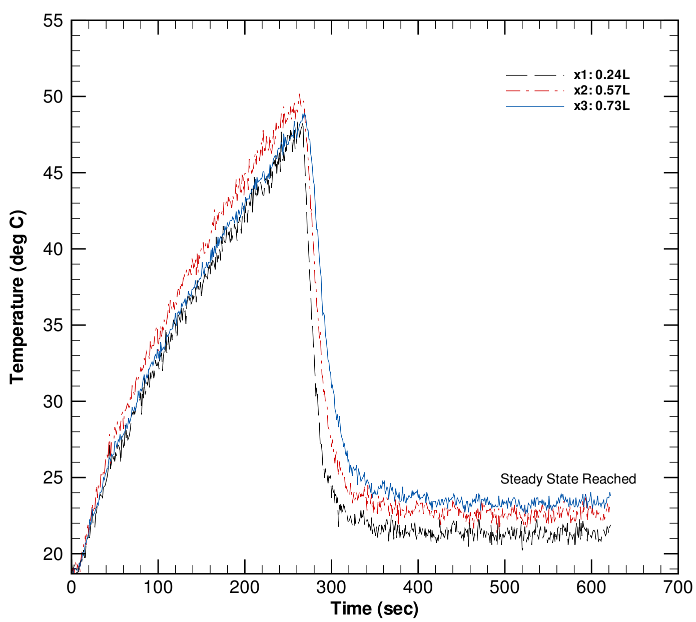

Figure 4. Sample Thermocouple readings for Heated Side.

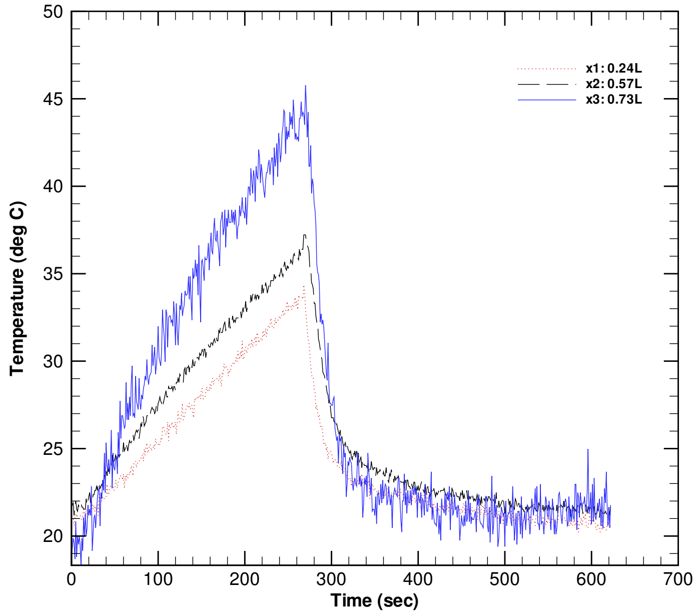

Figure 5. Sample Thermocouple readings for Insulated Side.

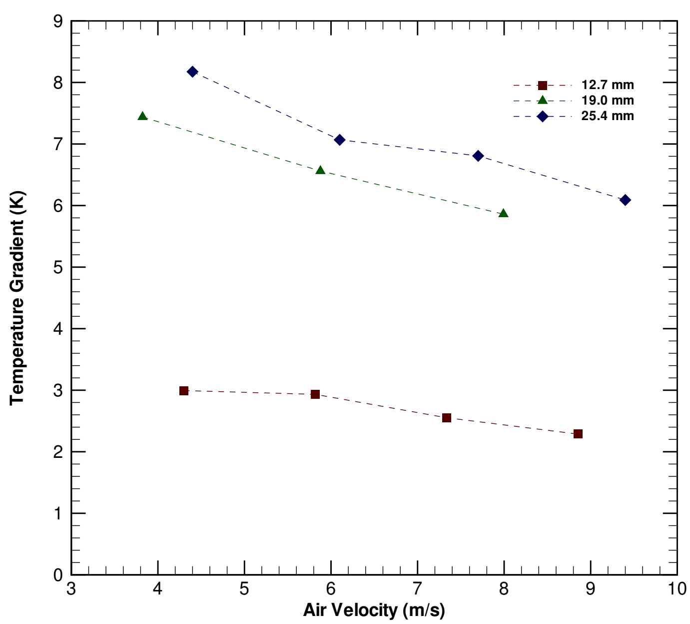

Figure 6. Effect of Thickness on Temperature Gradient across Metal Foam.

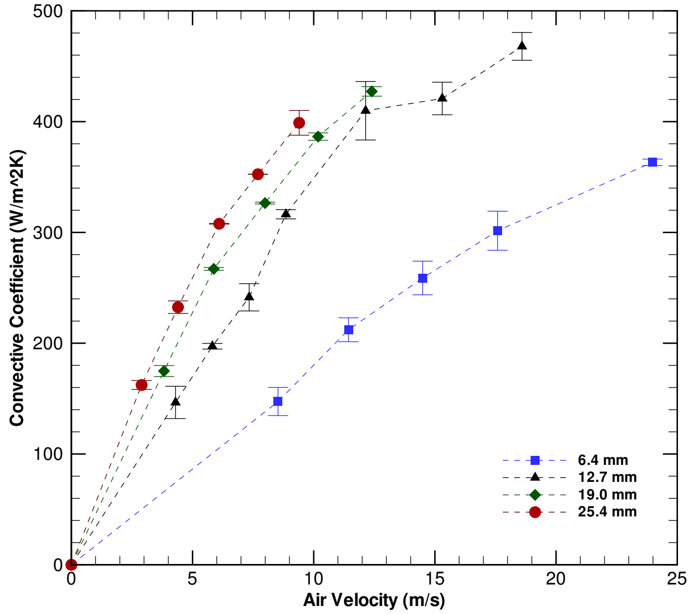

Figure 7. Dependence of Average Convection Coefficient of Metal Foam on Foam Thickness.

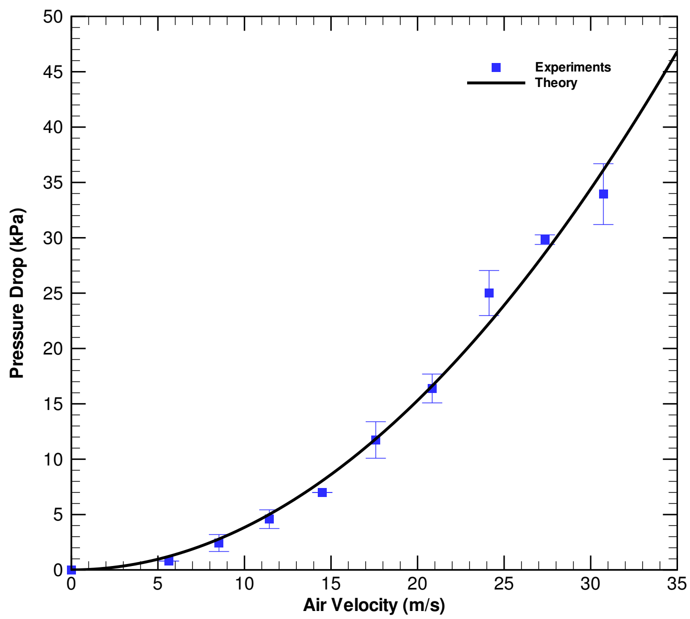

Figure 8. Pressure Drop across 6.4-mm-thick Metal Foam Core.

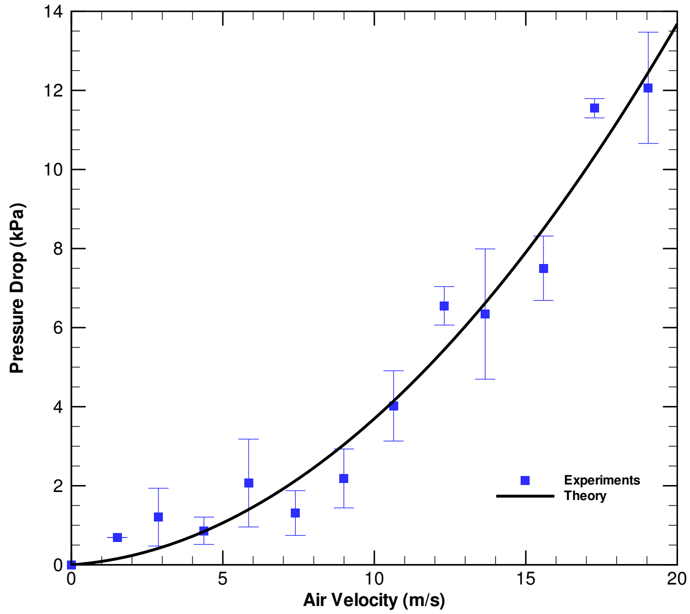

Figure 9. Pressure Drop across 12.7-mm-thick Metal Foam Core.

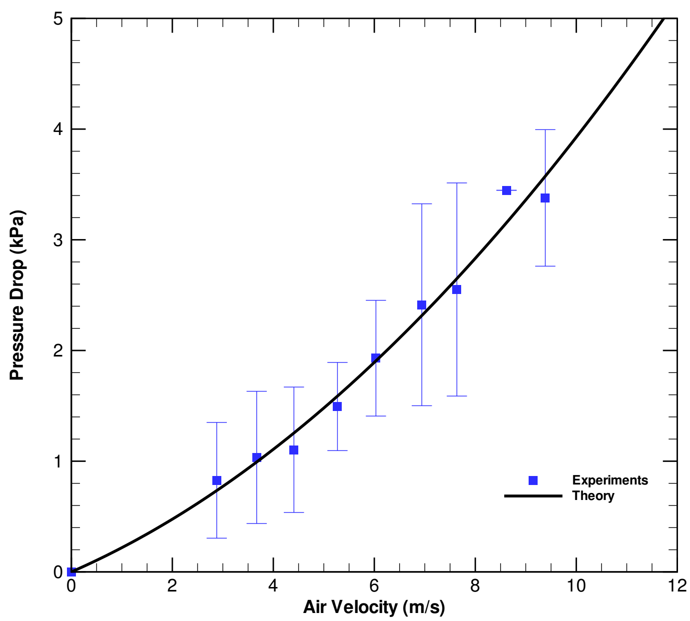

Figure 10. Pressure Drop across 25.4-mm-thick Metal Foam Core.

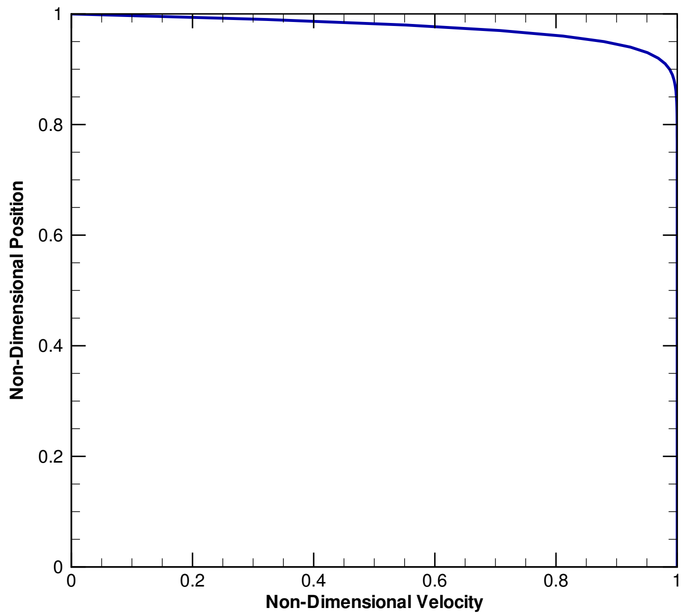

Figure 11. Sample Velocity Profile across Metal Foam.

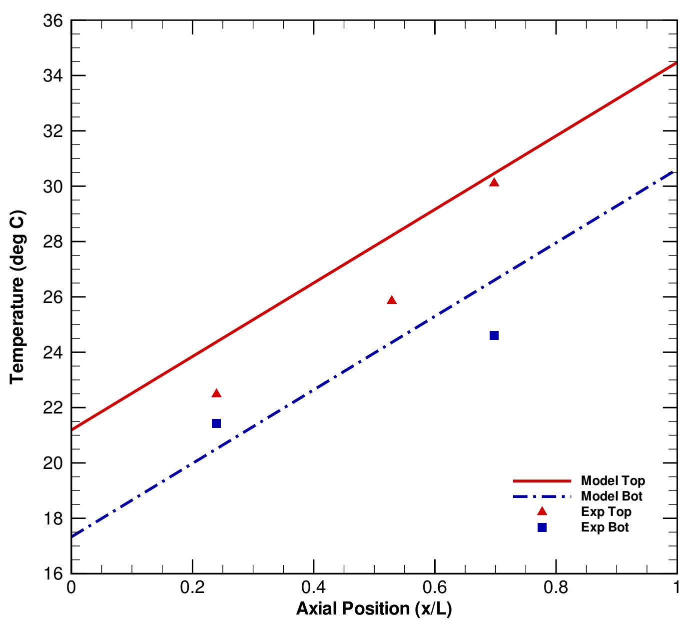

Figure 12. Comparison of Finite Element Model prediction and Experimental Results for Temperature distribution on Sandwich Panel. The foam thickness is 6.4 mm and the air velocity is 17.6 m/s – Random Cylinder Orientation.

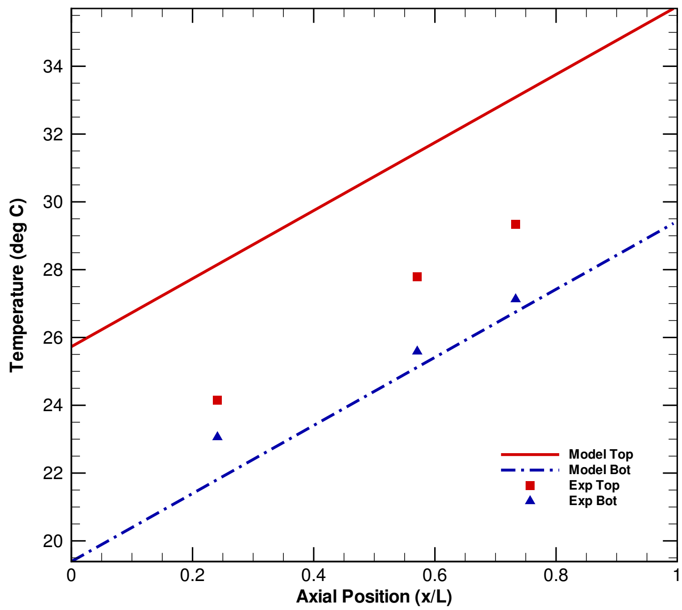

Figure 13. Comparison of Finite Element Model prediction and Experimental Results for Temperature distribution on Sandwich Panel. The foam thickness is 12.7 mm and the air velocity is 5.9 m/s – Random Cylinder Orientation.

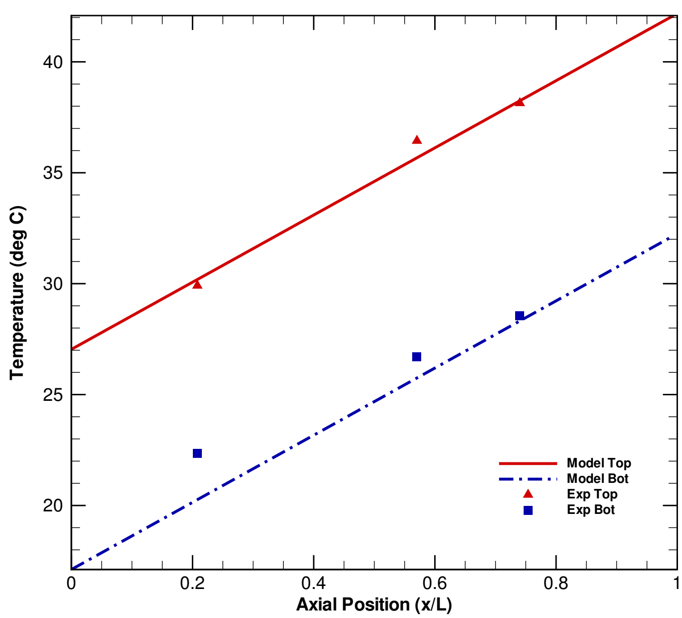

Figure 14. Comparison of Finite Element Model prediction and Experimental Results for Temperature distribution on Sandwich Panel. The foam thickness is 25.4 mm and the air velocity is 2.9 m/s – Random Cylinder Orientation.

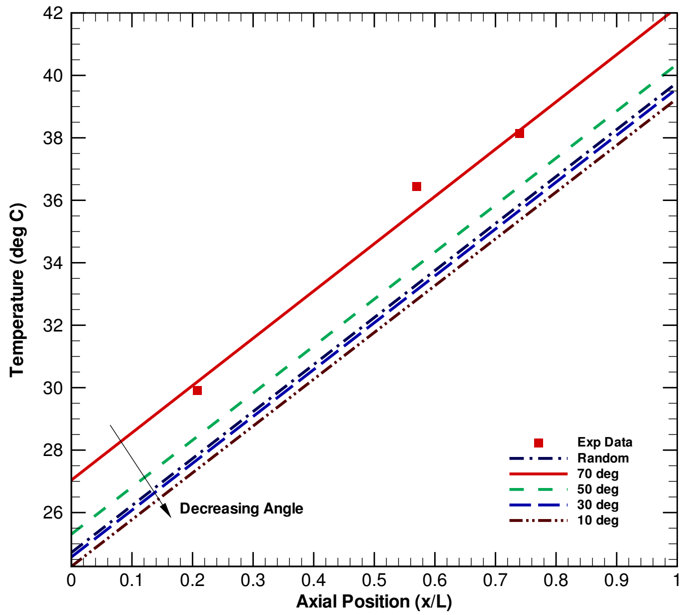

Figure 15. Effect of Varying Cylinder Angle on Model Prediction – Foam thickness is 25.4 mm, Heated Side

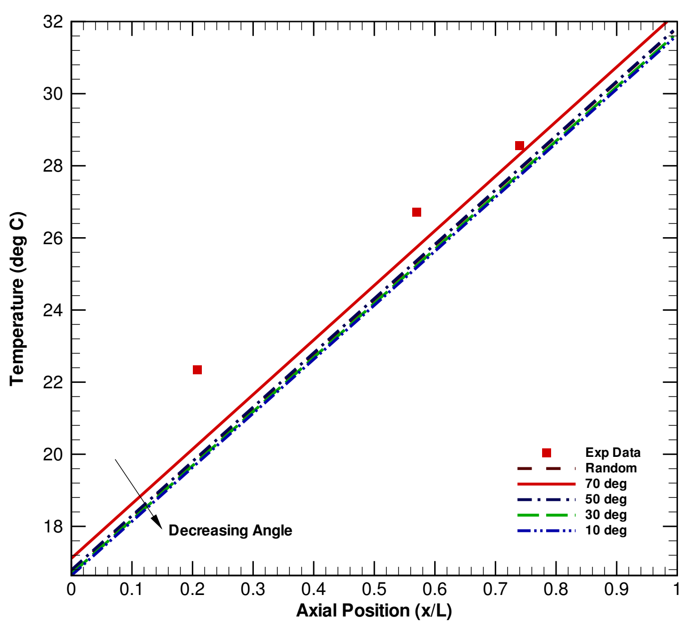

Figure 16. Effect of Varying Cylinder Angle on Model Prediction – Foam thickness is 25.4 mm, Insulated Side

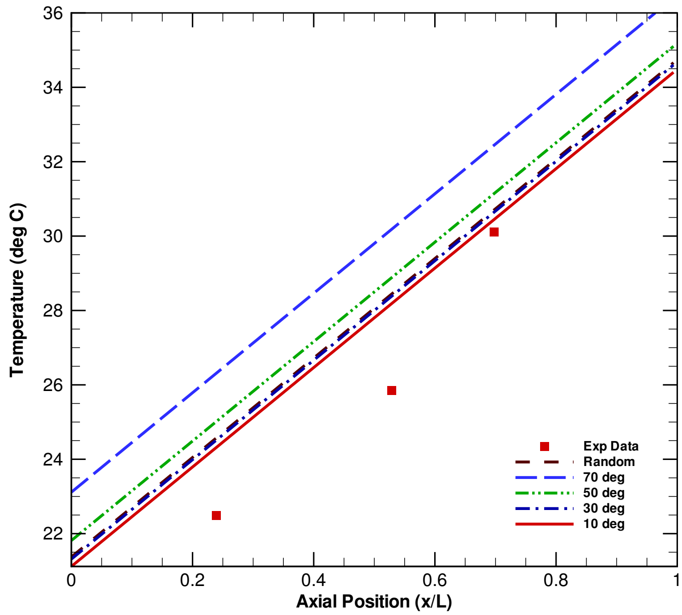

Figure 17. Effect of Varying Cylinder Angle on Model Prediction – Foam thickness is 6.4 mm, Heated Side

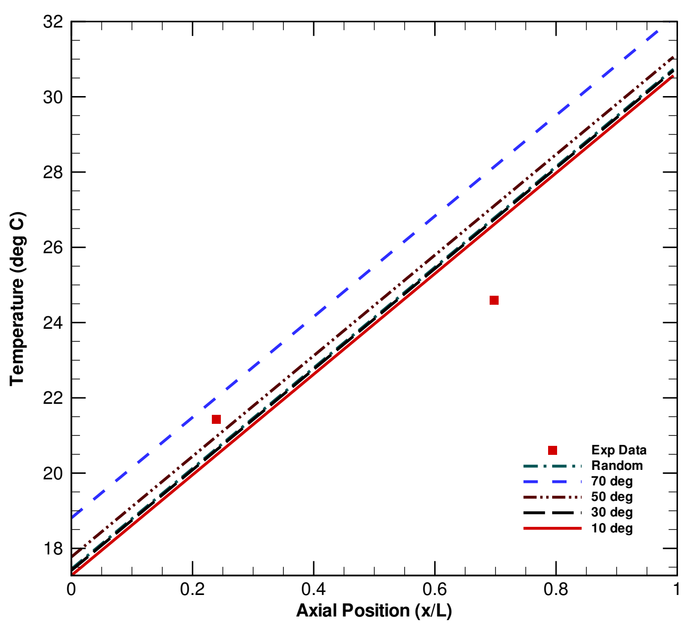

Figure 18. Effect of Varying Cylinder Angle on Model Prediction – Foam thickness is 6.4 mm, Insulated Side