

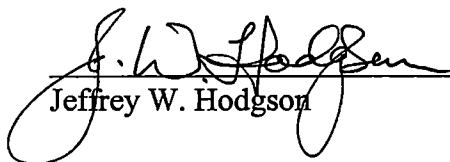
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I am submitting herewith a thesis written by Chin Chien Tee entitled "Modeling of Thermal Conductivity and Pressure Drop, and Thermal/Mechanical Testing of Carbon Foam-Based Materials." I have examined the final copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering

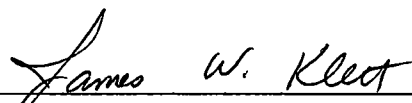


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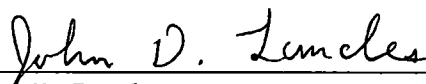
We have read this thesis
and recommend its acceptance:



Jeffrey W. Hodgson



James W. Klett



John D. Landes

Accepted for the Council:



Associate Vice Chancellor
and Dean of The Graduate School

**Modeling of Thermal Conductivity and
Pressure Drop, and Thermal/Mechanical Testing
of Carbon Foam-Based Materials**

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Chin Chien Tee

May, 2000

Dedication

This thesis is dedicated to my parents,

郑顶吉 施玉惠
Teng Kiat Tee and Geok Wooi See

and especially my soul mate,

尤美玉
Bee Gaik Yew

for all their support and encouragement.

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Abstract

The present thesis work can be divided into three parts: (1) modeling of overall thermal conductivity, (2) modeling of pressure drop, and (3) mechanical and thermal testing of the light-weight, high-thermal conductivity carbon foam and carbon foam-based composites that were recently developed at the Oak Ridge National Laboratory. The first model studies the interactions between the overall thermal conductivity of the novel pitch-derived carbon foam and foam porosity, microstructure, and temperature. In a heat dissipation application, the heat exchangers utilizing novel carbon foam may promote high overall heat transfer coefficients in comparison with conventional heat exchangers yet yields relatively high pressure drops. The flow field and the pressure drop across the carbon foam are thus studied in the second model. Furthermore, mechanical testing has been performed on carbon foam infiltrated composite materials to determine their strengths in view of the fact that the novel carbon foam has a potential in replacing carbon fibers in composite materials. Besides, thermal testing on a prototype heat sink and several heat exchanger systems have been carried out in order to determine the overall heat transfer coefficients of the electronic modules in power inverters.

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Nomenclature

A	cross sectional area of the medium
A_c	minimum flow cross-sectional area
A_{foam}	heat transfer surface area of the foam block
A_{fr}	frontal area of the hexagonal surface
A_{min}	minimum free flow area available for the fluid flow
A_n	nominal cross-sectional area of the mechanical test's specimen
A_o	volumetric (specific) surface area of the foam
A_r	cell aspect ratio
A_s	surface area of the unit cell
A_{sj}	cross sectional area of the strut juncture
A_{total}	total surface area of the foam
A^*	equivalent cross sectional area of the tapered strut
a	length of the unit cell
a_c	x-coordinate of the fixed focus point
$a_{\text{hex}} (a_{\text{sq}})$	distance between two parallel hexagonal (square) faces of the cell
B	constant coefficient in the power law relations
b	thickness of the foam block
b_f	width of the specimen for flexural test
C_1	constant term that depends on the microstructure of the cell
C_2	correction factor for a cell shape

c	perpendicular distance from the neutral axis to a point farthest from the neutral axis
c_F	dimensionless form drag coefficient
D	mean pore diameter of the foam and diameter of spherical void
D_h	hydraulic diameter
$D_{\text{hex}} (D_{\text{sq}})$	diameter of circle on hexagonal (square) face of the tetrakaidecahedron cell
d	diameter of circular cylinder strut
d_c	diameter of cylinder
d_f	depth of the rectangular bar for flexural test
dp	differential pressure
dv	differential velocity
f	friction factor
f_{foam}	friction factor for foam in cross-flow
f_{tube}	friction factor for banks of cylinders
f^*	core friction factor
G	flow stream mass velocity
h_c	half of the minimum strut thickness in the middle of the strut
I	moment of inertia
I_1	integral term for the tapered longitudinal strut
I_2	integral term for the tapered gas portion beneath the strut juncture
I_3	dimensionless integral term for the tapered longitudinal strut

K_c	entrance loss coefficients
K_e	exit loss coefficients
K^*	effective thermal conductivity matrix
K^{**}	three-dimensional effective conductivity of misoriented cell
$\overline{K}$	overall effective thermal conductivity of carbon foam
k	thermal conductivity
k_1	Darcian permeability
k_2	non-Darcian permeability
k_g	thermal conductivity of gas
k_K	Carman-Kozeny parameter
k_s	thermal conductivity of the isotropic strut
k_{sj}	thermal conductivity of the cubical strut juncture
k_{sl}	longitudinal thermal conductivity of the strut
k_{st}	transverse thermal conductivity of the strut
k^*	effective thermal conductivity of cubic cell
L	one-half of the length of the unit cell or edge length of each side of the tetrakaidecahedron cell
L_c	general characteristic length of a body
L_f	distance between two support spans
L_p	total length of the foam in the flow direction
l	distance the heat travels
M	resultant internal moment

m	the exponent in the power law
N	number of rows of cylinders in the direction of flow
N_e	effective number for calculating k_k
N_L	number of cylinder rows in the direction of flow for Y- and Z-type cylinders
n_o	probability density function for completely random distribution
$n(\theta, \psi)$	probability density function
P	point load for flexural test
P_{max}	maximum load prior to failure
p_1	inlet pressure
p_e	outlet pressure of cooling fluid
p_o	inlet pressure of cooling fluid
p_w	normal stresses due to the pressure gradient
Q	transformation matrix
q	heat transfer rate
R	thermal resistance
R_{g1}	thermal resistance of the rectangular gas portion
R_{g2}	thermal resistance of the gas beneath the transverse strut
R_{g3}	thermal resistance of the tapered gas beneath the strut juncture
R_{sj}	thermal resistance of the cubical strut juncture
R_{sl}	thermal resistance of the longitudinal strut
R_{st}	thermal resistance of the transverse strut

R^*	effective thermal resistance of the cell
Re	Reynolds number
Re_{max}	maximum Reynolds number
r	taper ratio (ratio of the minimum strut thickness at the middle to the maximum at the junction)
s	coordinate system along the streamline
T	temperature
T_1, T_2	uniform temperature of top and bottom plate, respectively
T_o, T_e	inlet and outlet temperature of cooling fluid, respectively
T_s	heating temperature
$T_{f,in}, T_{f,out}$	inlet and outlet cooling fluid temperature, respectively
t	normalized thickness of strut
U	free stream velocity of the fluid
U_o	overall heat transfer coefficient
$U(s)$	free stream velocity of the fluid along the streamline coordinate
V	volume of the foam block
V_{cell}	unit cell volume
V_{ch}	volume of spherical end caps on hexagonal faces
V_{cs}	volume of spherical end caps on square faces
V_{void}	void volume
V_{avg}	average free stream velocity of the fluid
V_{max}	maximum fluid velocity

v_o	cooling fluid initial velocity
W	mass flow rate
W_p	width of plate
X_1, X_2, X_3	global Cartesian coordinates
x_1, x_2, x_3	local Cartesian coordinates
x, y, z	Cartesian coordinates
y, y_c	dummy variable for cubic equation substitution
Z	rate of straining of the outer layer for 4 point flexural test
α_A	surface area density of the foam
β	angle of the cylinder measured from horizontal
χ	frictional coefficient, for cylinder type X
Δ	thickness and width of the rectangular strut
ΔP	pressure drop across the foam
ΔT	temperature difference
$\Delta T_{in} (\Delta T_{out})$	inlet and outlet temperature difference, respectively
ΔT_{lm}	log mean temperature difference (LMTD)
∇T	temperature gradient
δ	displacement of specimen
ϕ	porosity
γ	exponent in the power law relations
η	height of cell
κ	permeability

μ	average fluid viscosity
μ_f	fluid viscosity
μ_w	fluid viscosity near the wall
ν_f	fluid kinematic viscosity
θ	coordinate rotation about X_3 axis
ρ	foam relative density
ρ_1, ρ_2	fluid inlet and outlet density, respectively
ρ_f	fluid density
ρ_m	fluid mass density
ρ^*	foam density
ρ_s	graphite strut density
σ_{BEND}	maximum bending strength
σ_{UTS}	ultimate tensile strength
τ_w	wall shear stresses due to viscous effects
ψ	coordinate rotation about the X_1 -axis
ζ	constant frictional term

CHAPTER 1

Introduction

1.1 Background

Mesophase pitch-derived open-cell carbon foams with high thermal conductivities have been developed at Oak Ridge National Laboratory (ORNL) by using a relatively simple technique (Klett and Burchell, 1998). The novel technology for fabricating carbon foams has been licensed to Poco Graphite, Inc. for large-scale production (Klett, 1999a).

The novel carbon foam exhibits mostly open-cell microstructure and is derived from either a petroleum or a synthetic mesophase pitch (Klett and Burchell, 1998). Unlike commercially produced vitreous glassy carbon foam, which is mainly used for insulation, the mesophase pitch-derived carbon foam permits high thermal conduction, which is vital in heat exchanger applications.

In comparison with other thermal management materials, the mesophase pitch-derived carbon foam exhibits high isotropic thermal conductivity and low density, see Table 1.1 (Klett *et al.*, 2000a). Although copper has a higher thermal conductivity, its density is much greater than that of the carbon foam. Hence, the specific thermal conductivity, which is the thermal conductivity divided by specific gravity (i.e. ratio of the density of the material to the density of water), of the carbon foam is significantly greater than most of the available thermal management materials. In fact, the specific thermal conductivity of carbon foam is more than seven times greater than that of copper

Table 1 1 Thermal properties of the mesophase pitch-based carbon foams compared with other thermal management materials (Klett *et al.*, 2000a)

Material	Specific Gravity	Thermal Conductivity (W/mK)		Specific Thermal Conductivity (W/mK)	
		In-plane	Out-of-plane	In-plane	Out-of-plane
Mesophase pitch-derived carbon foam	0.56	187	187	334	334
Copper [†]	8.90	400	400	45	45
EWC-300/Cyanate ester resin [†]	1.72	109	1	63	0.58
Aluminum [†]	2.77	150	150	54	54
Aluminum honeycomb [#]	0.19	--	10	--	53
Aluminum foam (Steiner <i>et al.</i> , 1997)	0.50	12	12	24	24

[†] Amoco Product Literature (1997)

[#] Hexcel Product Data Sheet (1997)

and six times greater than that of aluminum, which is one of the popular materials for heat sinks. By impregnating the carbon foam with epoxy resins to form composites or by densifying the carbon foam with chemically vapor deposited carbon, the strength of the material can be improved by 10 times, and it will be comparable with commercial thermal management panels (Klett, 1999b)

1.2 Processing Technique

To produce a carbon foam with high stiffness and high thermal conductivity, a mesophase pitch can be used as a precursor to assure the formation of graphitic struts or ligaments (Hager and Lake, 1992; Sandhu and Hager, 1992). The pitch used to produce carbon foam is in a mesophase state, i.e., an intermediate stage in the formation of graphitizing carbons (Shuba *et al*, 1997). The mesophase pitch used is from either ARA24 naphthalene-based synthetic pitch from Mitsubishi Gas Chemical Corporation or a proprietary mesophase pitch from Conoco Corporation (Klett and Burchell, 1998).

The Conoco mesophase pitch-derived foam exhibits marginally higher densities than the Mitsubishi ARA24 mesophase pitch-derived foam, see Table 1.2. On the other hand, the ARA24 foam has a larger mean pore size than the Conoco foam since the higher melting temperature of the Conoco pitch yields higher viscosities during the processing (Klett *et al*, 1999)

Table 1 2 Properties of carbon foams produced from different precursors (Klett *et al* , 1999)

Carbon Foam	Density (kg/m ³)	Mean Pore Diameter (μm)	Overall Thermal Conductivity (W/mK)
ARA24	550	275	50 – 187
Conoco	600	60	40 – 135

Typically, a foam forming process utilizes a blowing technique or a pressure release to produce the foam of the pitch precursor (Hager and Lake, 1992). As the bubbles in the foam grow, the resulting bi-axial extension orients the mesophase domains parallel to the cell walls, similar to the uni-axial extension during melt extrusion (or spinning) of mesophase pitch-based carbon fibers (Klett *et al.*, 2000a). The foam is then stabilized by heating in air or in oxygen for several hours to cross-link the microstructure and to set the pitch so that the foam would not melt during further heat treatment process (Kearns, 1997). The stabilization process is time consuming and costly. The stabilized pitch is then carbonized in an inert atmosphere to a temperature as high as 1100°C resulting in a structural material suitable for composite reinforcement (Sandhu and Hager, 1992; Kearns, 1997; Hager, 1992). Finally, in order to convert the carbon to graphite, the foam is graphitized at a temperature as high as 3000°C (Hager and Lake, 1992, Sandhu and Hager, 1992).

However, the novel process developed at ORNL does not utilize a thermodynamic flashing or a blowing agent, and most importantly, it eliminates the

requirement for stabilizing the foamed pitch prior to carbonization (Klett and Burchell, 1998) It is believed that the carbon foam fabricated using the novel technique will be less expensive and easier to fabricate than traditional foams (Klett *et al* , 2000a) The proprietary method is fairly versatile in the sense that the pore size and the foam density can be controlled and adjusted easily (Klett and Burchell, 1998)

1.3 Foam Morphology

The carbon foam cell is made up of an interconnecting network of graphitic struts (cell walls), open interconnects, and strut junctures; Figure 1.1 (Klett, 1999b) It is noted that the carbon foam exhibits predominately open, interconnected pores within its cell structure, which resembles the shape of a sphere. Figure 1.1 shows that the graphite layers are oriented parallel to the cell walls and are highly aligned along the direction of the strut or ligament (Klett *et al* , 2000a).

On the other hand, the foam cell geometry of the commercially available glassy carbon foam, such as the reticulated vitreous carbon foam produced by ERG Corporation, is shown in Figure 1.2 (Klett, 1999b). Without the aligned graphite layers, the vitreous carbon foam exhibits porous graphitic microstructure with large openings and is similar to a pentagonal dodecahedron; see Figure 1.3.

Nevertheless, the graphite struts for the novel pitch-derived carbon foam are relatively thicker and the foam has less open pores than the commercially available glassy carbon foam As it is illustrated in Figure 1.4, the struts for the novel pitch-derived carbon foam are relatively thick near their extremities and relatively thin at the

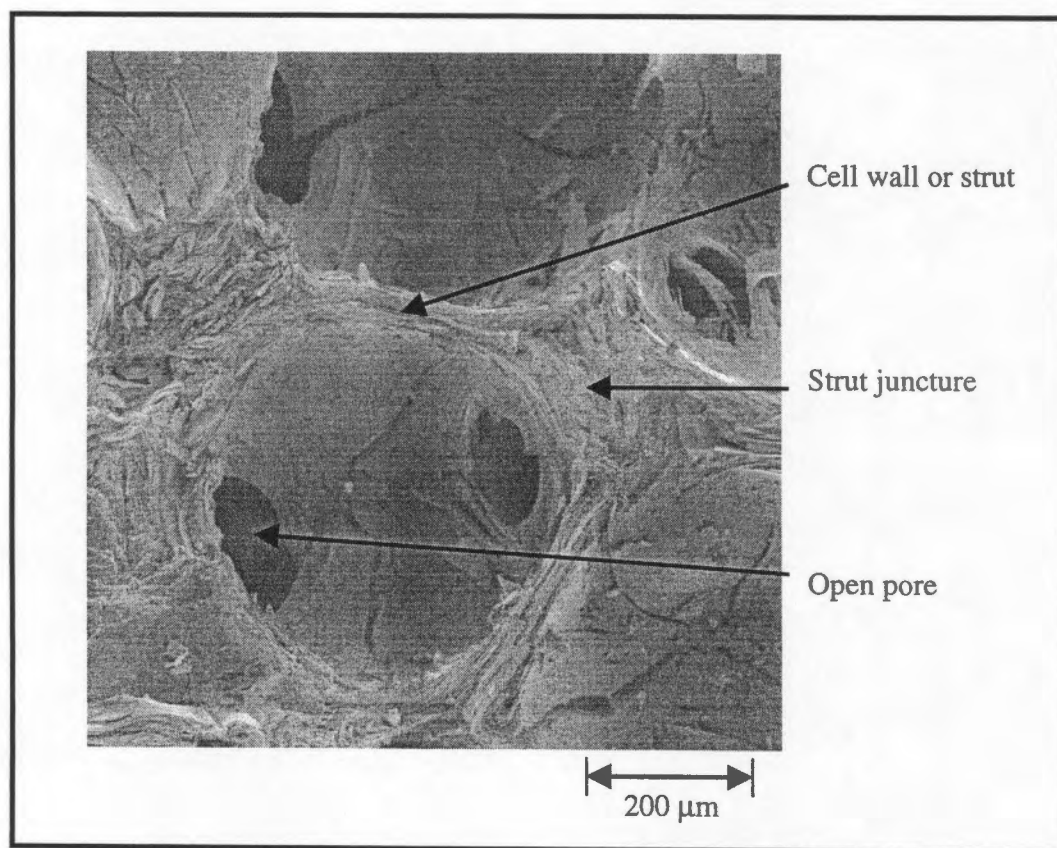


Figure 1.1: Typical microstructure of mesophase pitch-derived carbon foam (Klett, 1999b).

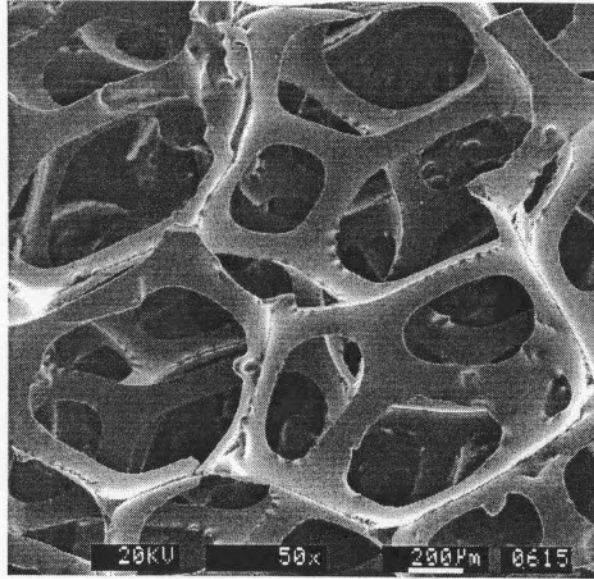


Figure 1.2: Reticulated glassy carbon foam produced by ERG Corporation (Klett, 1999b).

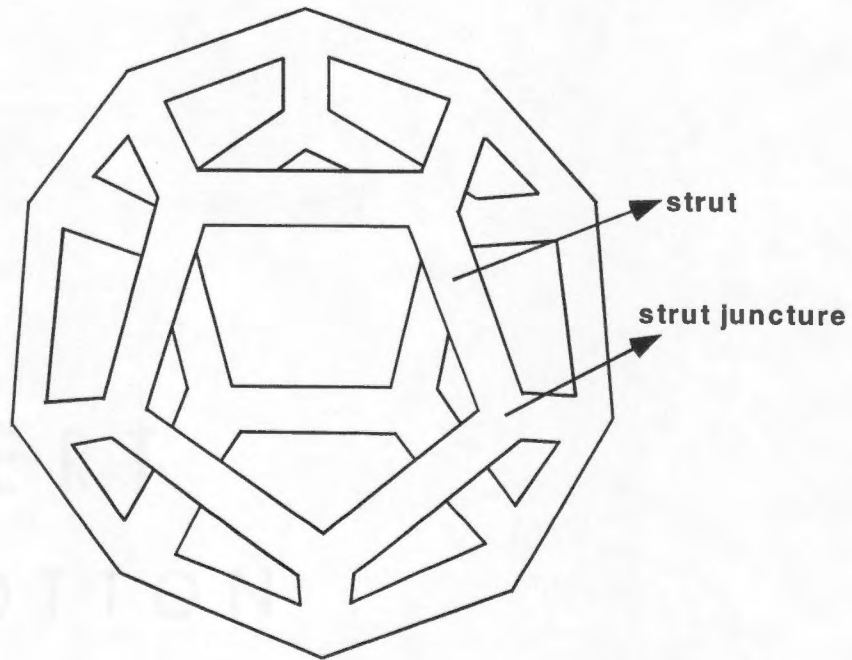


Figure 1.3: Pentagonal dodecahedron model for a glassy carbon foam cell.

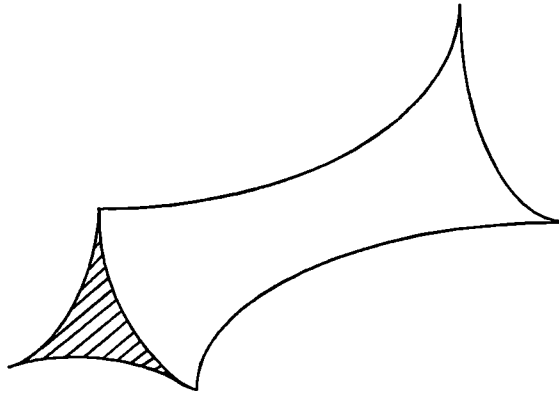


Figure 1 4: Strut with varying thickness.

center. The graphite struts are made of oriented graphitic planes, which is similar to high performance carbon fibers. The cross section of the strut has three concave sides. The strut cross-sectional area is on average two-thirds of the area of the triangle defined by the strut vertices (Doermann and Sacadura, 1994). All struts are almost identical to each other. Furthermore, the orientation distribution of the struts on any plane in the middle of the foam is perfectly random, which renders the carbon foams macroscopically isotropic (Hager, 1992; Klett and Burchell, 1998).

The strut juncture is the intersection of four struts; see Figure 1 5. It can be seen from Figure 1.1, the graphitic structure in the strut juncture is less aligned and has more folded texture, which may arise from the lack of stresses at this location during forming (Klett et al , 2000a). Therefore, the well ordered structure in strut junctures is primarily an artifact of the precursor structure prior to heat treatment and may limit the overall thermal conduction (Klett, 1999b).

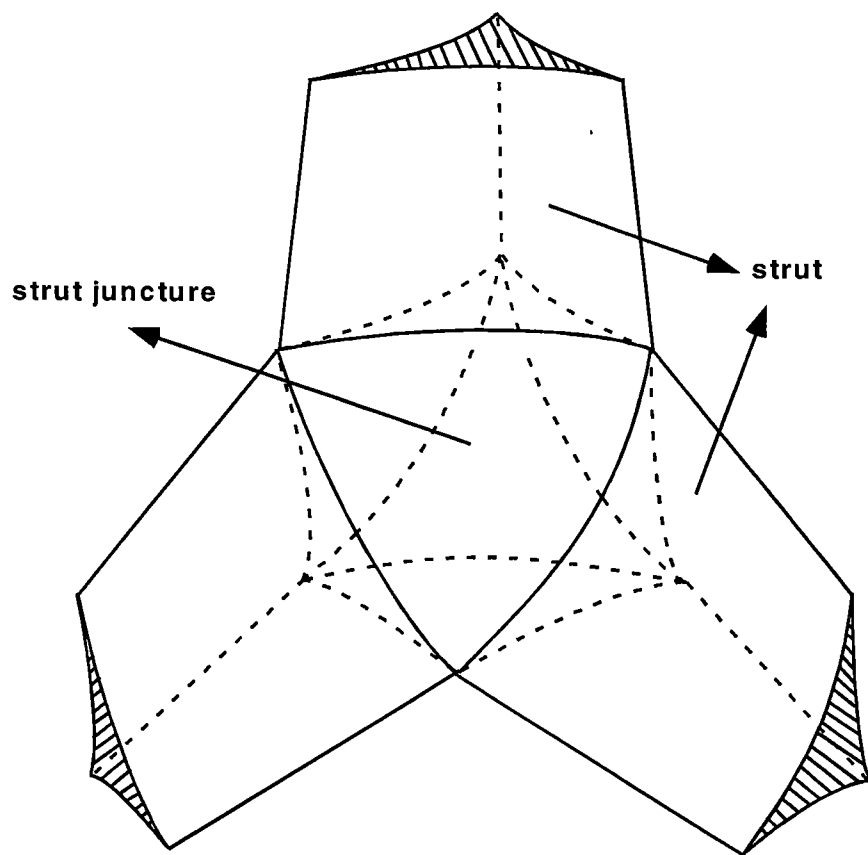


Figure 1.5 Strut junction

1.4 Applications

Recently, foams have received extensive attention in the light of high-efficiency heat exchangers or heat sinks in heat dissipation application. For heat convection processes in industrial heat exchangers, the porous carbon foam with a high thermal conductivity and a large amount of contact surface area between pores and the cooling fluid result in high overall heat transfer coefficients. However, the mechanical pumping power that is necessary to overcome the pressure drop mainly due to cooling fluid friction across the foam is considerably high in comparison with conventional heat exchanger systems at high flow rates (Klett and Burchell, 1998). Therefore, the flow field and pressure drop behavior through the carbon foam is essential to be understood and analyzed in order to design a heat exchanger using carbon foam as the core material.

For automotive industry, the combination of open pores and large specific surface area of the carbon foam allows the improvement in heat exchanger applications used in radiators or intercoolers in vehicles. Specifically, if the air passage within a fin-type heat exchanger is replaced by high-thermal conductivity carbon foams, the surface area of the exchanger will increase significantly. Consequently, the heat will be rapidly transferred from the coolant fluid across the tubes through the foam to the intake air. Such an increase in surface area for heat transfer will also reduce the size of the radiator and therefore, will reduce the weight and improve the efficiency of the automobile. In addition, silicon carbide (SiC) coating layers can be deposited on the surface of the carbon foam to prevent the carbon from oxidation at a working temperature above 500°C (Klett, 1999a).

In addition, the carbon foam may lead to heat sinks for computer processors that do not require electric fans, and thus greatly increase the reliability of the personal computers that are severely limited due to heat generation problems. A finned heat sink made from carbon foam can be used to replace the standard aluminum heat sink in personal computers. The finned heat sink made of carbon foam has a higher efficiency in removing heat because the exposed surface area is larger than that of the standard aluminum heat sink. Moreover, the weight of the finned foam heat sink will be significantly lower than that of the aluminum heat sink (Klett, 1999b).

In addition to its application in heat exchangers, the carbon foam can be potentially used as a reinforcing phase in composite materials. This is because the foam-reinforced composite will have a higher isotropic thermal conductivity than carbon fiber-reinforced composites due to the continuous, isotropic graphitic network. In addition, the mechanical properties of the foam, such as shear strength and fracture toughness, will be enhanced without the interlaminar regions developed in the traditional carbon fiber-reinforced composite laminates (Klett and Burchell, 1998).

Recently, the carbon fiber has evolved from a structural reinforcement to a thermal management material, and has been used in electronic modules, communication satellites, and automotive systems. However, the improved mechanical properties and thermal conductivity of the resultant fiber reinforced composites is limited to the direction of fibers. Therefore, the thermal conductivity in the out-of-plane (through-the-thickness) direction is relatively low because the matrix is the dominant phase for heat transfer in the out-of-plane direction, see Table 1.1 (Klett *et al.*, 2000a). Furthermore,

sandwich structures that consist of a high-thermal conductivity material encapsulating a core structural material, i.e. aluminum honeycomb, is being studied but similarly the overall through-the-thickness thermal conductivity of the aluminum honeycomb based composite is still relatively low (Shih, 1994). Besides, a low-density metallic foams are being explored as a potential core material but the overall thermal conductivity of the metallic foam is still relatively low, 5 – 50 W/mK (Gibson, 1989). Therefore, isotropic carbon foams exhibiting high thermal conductivities in any directions present a unique solution to the problem of low through-the-thickness thermal conductivity. The carbon foam is very versatile, as it can be easily machined, laminated with facesheets, or net shape formed, and made in large samples (Klett, 1999b).

In the automotive industry, a carbon foam-reinforced aluminum piston can be used in place of conventional aluminum alloys within the internal combustion engines (Klett and Burchell, 1998). This foam-reinforced aluminum may have a higher creep resistance and a lighter weight compared with the aluminum alloy and thus, increase the efficiency and reduce the emission of the engine. Moreover, foam-reinforced plastics could be utilized as a piston or an engine block material. Although some plastics can withstand the service temperature of an internal combustion engine (less than 300°C), the low thermal conductivity of the plastic limits the heat removal during the cycles. Consequently, the system will be overheated and the plastic will melt. On the other hand, foam-reinforced plastics have thermal conductivities similar to those of aluminum pistons and therefore, are able to remove heat at the rate that is similar to that of aluminum pistons. A cyanate ester-foam piston saves as much as 40% of the piston

weight. Furthermore, it may be possible that the entire internal combustion engine block be made from a foam-reinforced polymer (Klett and Burchell, 1998)

1.5 Scope of Study

The current research work is focused on the modeling as well as the characterization of the heat transfer and mechanical behavior of the carbon foam and carbon foam-based composites to further delineate the interactions between the microstructure and properties of carbon foam.

Specifically, analytical modeling has been conducted to predict the overall thermal conductivity and the flow resistance behavior of the carbon foam. The effective thermal conductivity of carbon foam is modeled as a function of temperature, constituent properties, and microstructure. Moreover, the effects of misoriented struts (ligament) and the shape of the strut on the overall thermal conductivity of the carbon foam have been investigated. In addition, the overall pressure drop across a carbon foam core under forced convection has been estimated as a function of flow rate and physical property of the fluid, porosity, pore size, and the surface friction characteristic of the carbon foam.

Tensile and flexural tests have been performed to characterize the mechanical properties of the foam and its infiltrated composites, which include carbon foam infiltrated with a carbon, aluminum, or epoxy resin. Furthermore, a prototype heat sink made from carbon foam core has been tested in order to simulate the electronic modules in power inverters and determine the overall heat transfer coefficient. In addition, several different foam density and design heat exchanger tests have been performed in

order to evaluate the performance of the heat exchanger utilizing carbon foam core, to study the effect of different density and geometry of the foam on the heat dissipation system, and to determine the overall heat transfer coefficient as well as pressure drop data

CHAPTER 2

Literature Review

2.1 Heat Conduction

One of the early works on the modeling of the thermal conductivities of porous materials is reported by Russell (1935), where the conductive heat flow through a solid matrix containing in-line cubical bubbles with uniform wall thickness is analyzed. Russell's model regards the solid phase as membranes, rather than struts; therefore, the cell is closed. At low foam densities, Russell's prediction is essentially the same as that by Kerner (1956), who considers an isotropic composite consisting of randomly distributed particles embedded in a continuous phase. By using Kerner's theoretical models, Baxter and Jones (1972) conclude that for foams with a porosity higher than 95%, the thermal conduction of the foam is essentially due to the air and two-thirds of the solid phase present. The portion of the solid phase that is perpendicular to the direction of the heat flow of heat may be ignored (Baxter and Jones, 1972).

Brochhagen and Schmidt (1964), however, point out that the effectiveness of the modeling depends strongly on whether the continuous phase material is simulated as a "cubic area lattice" or a "cubic rod lattice" in view of the fact that foam cells consist of both planar membranes and linear sticks. To this end, Schuetz and Glicksman (1984) propose three models. The first model is the same as Russell (1932), studying conductive heat flows through a solid matrix with cubical bubbles. The second model replaces membranes by struts, while the third model examining the effects of solid path

tortuosity, which is in the form of staggered cubes. It is shown that the first and second model can be extended to bubbles with an arbitrary shape and the cells containing both struts and membranes can be regarded as a linear combination of both models. The difference in the results between the models of in-line and the staggered cubical bubbles is no more than 20% and the limitation of these models is that they apply to foams with densities lower than 80 kg/m^3 (Schuetz and Glicksman, 1984).

Since most of the early foams were developed for insulation, radiative heat transfer in the foams is significant. The model by Schuetz and Glicksman (1984) is further extended by Doermann and Sacadura (1996) to include radiative heat transfer in open-cell glassy carbon foams. Based on microscopic analyses, the foam is modeled as struts with varying thickness along them and the strut junctures.

Dul'nev (1965) develops a cubic strut model for the heat conduction in open-cell foams by using a simple parallel-series resistor diagram. The model is used by Kamiuto (1997) to study the steady-state heat conduction and radiation in a porous psZrO_2 layer bounded by gray, diffusive plates. Numerical solutions are obtained by using a finite difference method. The validity of Dul'nev's model has been verified by comparing the experimental results of several reticulated porous materials, including Ni , Al , Ni-Cr , cordierite and psZrO_2 , with the Dul'nev's analytical model (Kamiuto, 1997).

The first model presented in Helte (1993) involves a dispersion of equisized, disconnected, spherical pores in a solid matrix. The thermomechanical fields inside the porous medium are obtained by considering a single pore surrounded by a concentric shell of the solid phase, which is in turn embedded in an infinite effective medium.

When no radiation is present, the results agree very well with the exact results for a face-centered cubic (fcc) arrangement of spheres (Van Vlack, 1985) and are identical to the approach taken by Hashin (1968) for purely conductive heat transfer in particulate composites. Another model considered in Helte (1993) is a dispersion of aligned, infinitely long, equisized, disconnected, circular cylinders embedded in a solid matrix. The effective transverse conductivity of the fibrous composite is obtained in a similar manner as for the particulate composites in the first model. The third model involves a random distribution of overlapping solid spheres, applicable to consolidated or sintered materials with high porosity. However, few reliable, systematic measurements seem to be available for the material systems discussed in Helte (1993).

Tsao (1961) derives analytical expressions for the thermal conductivities of porous materials, packed beds, and two-phase liquid-liquid systems. In addition to bulk porosity, the expression for the effective thermal conductivity derived involves two experimentally determined parameters - the mean and the standard deviation of porosity.

Chou and Nomura (1981) examine the fiber orientation effects on the effective thermal conductivity of short fiber-reinforced composites. To compute the effective thermal conductivity of randomly distributed short fiber composites, volume averages over a representative volume of various fiber orientations have been used. In order to take into account the interaction among fibers at different orientations, Hatta and Taya (1985) obtain closed-form expression for the effective thermal conductivities of randomly oriented short fiber composites under steady state heat conduction using the equivalent inclusion method (Eshelby, 1957). The effects of volume fraction, fiber

aspect ratio, and distribution function of fiber orientation on the thermal conductivity of short fiber-reinforced composites are discussed

2.2 Forced Convective Hydraulic Resistance

2.2.1 Cylindrical Tube in Cross Flow

Hydraulic resistance is one of the most important characteristics of heat exchangers and may be characterized by the total pressure drop in the flow across banks of cylindrical tubes. King (1914) studies the heat transfer and hydraulic resistance of a single cylindrical tube while Thoma (1921) discusses the effects of the geometry of the tube banks on the heat transfer and flow resistance behavior. Thoma (1921) concludes that the heat transfer per unit area (heat flux) within one tube in a bank of tubes is higher than that of a single tube, and is a function of the arrangement of tubes. Similarity theory, applied to heat convection, may not only suggest the necessary direction of experiments, but also supplies a common basis for the generalization and analysis of the experimental data obtained by different authors (Žukauskas, 1987). Thus, the heat transfer and hydraulic resistance of a bank of cylindrical tubes in a cross flow of gas can be generalized to yield general equations, based on the results of numerous investigations of steam power boilers (Grimison, 1937, Kuznetsov and Lokshin, 1937).

Žukauskas (1987) presents the heat transfer and hydraulic resistance of fluids flowing through a single cylindrical tube and banks of tubes of various arrangements, including the heat transfer between tubes in cross flow. Extensive experimental data, including those associated with Prandtl number ranging from 0.7 to 500 and Reynolds

number ranging from 1 to 2×10^6 are analyzed. It is concluded that the maximum velocity is more acceptable as a reference in computing Prandtl and Reynolds numbers, as it reflects the actual resistance with sufficient accuracy, except for closely spaced banks. Experimental results indicate that the pressure drop across banks of cylinders is proportional to the number of rows and is a function of tube arrangement.

2.2.2 Ceramic Foam

Philipse and Schram (1991) present the experimental results on fluid flow through laboratory-made alumina-silica foams, containing about 40 wt% alumina. The pressure drop as a function of air-flow rates shows that Darcy's law does not hold and that the non-Darcian air-flow through the foams satisfies the following equation:

$$\frac{\Delta P}{L_p} = \frac{\mu_f}{k_1} U + \frac{\rho_m}{k_2} U^2 \quad (2.1)$$

where ΔP is the pressure drop, L_p is total length of the foam in the flow direction, ρ_m is the fluid mass density, μ_f is the fluid viscosity, U is the free stream velocity of the fluid, and the permeabilities k_1 and k_2 are referred as the Darcian and non-Darcian permeability respectively. These permeabilities may be a more sensitive measure for inhomogeneous media than porosity since porous media with approximately the same overall porosity but different microstructure may have different permeabilities. The permeabilities are tabulated for ceramic foam with porosity ranging from 82% to 92%.

Philipse and Schram (1991) show that the pore microstructure strongly affects the flow rate when the flow becomes non-Darcian. Flow paths in the ceramic foam are tortuous, and the magnitude and direction of fluid particle velocities change continuously. Thus, there are inherent inertia effects that are noticeable at flow rates lower than Darcian flow within foam structure at pore size of the millimeter range (Philipse and Schram, 1991). However, even when the inertia effects are appreciable, flow through a porous medium may still be of a stable laminar type (Scheidegger, 1974). Hall and Hiatt (1993) experimentally determine that the intensity of turbulence will increase with an increase in pore size for the fluid flow through a porous partially stabilized zirconia.

2.2.3 Metallic Foam

Antohe *et al.* (1996) develop a microporous heat exchanger for cooling high-power electronics using a mechanically compressed porous aluminum layer to improve the heat transfer and provide relatively uniform cooling. The hydraulic characteristics are obtained experimentally and are used in the numerical simulations of a real microporous heat exchanger.

Lu *et al.* (1998) explore the use of open-cell metallic foams as compact heat exchangers utilizing convective cooling. A simple cubic unit cell consisting of heated slender cylinders is modeled analytically for metallic foams. The model considers the energy flow due to forced convection and conduction through ligaments of the cellular foam. The analogy between flow through the foam and that across a bank of cylinders is

used. The overall heat transfer coefficient and the resulted pressure drop as the fluid flows across the foam are determined and the optimum foam properties for the best heat transfer performance are suggested. Since the present model, which is based on a cross flow across a bank of cylinder, has been idealized to limit the complexity of forced convective flow across open-celled foam, the model may oversimplify the transport phenomenon in metallic foams consisting of non-circular, possibly sharp-edged ligaments. The assumption made in order to analyze the heat transfer in disordered, porous materials likely lead to an overestimate of the actual level of heat transfer but nevertheless capture the approximate functional dependence of the various variables, such as foam density, cell size, and fluid velocity (Tien and Vafai, 1990).

Similarly, Bastawros *et al* (1999) model the thermal properties and hydraulic characteristics of aluminum foams subjected to transverse air-flow. Besides the flow friction over a bank of cylindrical tubes, the hydraulic measurements on aluminum foams are calibrated experimentally and are applied into the Forchheimer (1901) extended Darcy (non-Darcian) model. It is demonstrated that aluminum foams offer improved performance over pin fin arrays together with substantial weight saving and a moderate increase in the pressure drop in the heat sink applications. Nevertheless, the predicted trends of flow friction and heat transfer by the model are in reasonable agreement with available experimental data on Duocel[®] aluminum foams made by ERG Corporation.

2.2.4 Porous Medium

Huang and Vafai (1993) analyze the changes in the flow pattern and heat transfer characteristics in multiple porous block structures. A general flow model that accounts for the effects of the impermeable boundary and inertial effects is used to simulate the flow inside the porous region. Numerical solutions have been obtained using a finite-difference method with the aid of a stream function-vorticity. The effects of Darcy's number, Reynolds number, Prandtl number, the inertia parameter, and other pertinent geometric parameters, on the hydraulic resistance of porous media are thoroughly explored. Two distinct boundary layers are shown to exist for the velocity field, while only one boundary layer is observed for the temperature field. It is concluded that the presence of a porous block array near an impermeable boundary significantly changes the convection characteristics.

Hunt and Tien (1988) examine the non-Darcy flow and heat transfer in high-porosity fibrous media. Several types of fibrous medium, such as carbon, nickels, and aluminum, are tested under forced convection, and several parameters, such as permeabilities, porosities, and thermal conductivities, are experimentally and analytically studied. Traditional analyses based on porous media utilize Darcian or slug flow models (Vafai and Tien, 1981) and neglect the non-Darcian effects, such as the viscous shear force along the solid boundaries, the inertial convective force, and thermal dispersion. However, the fluid in essence moves in tortuous paths throughout the porous medium and recirculates at the back of the solid fibers. This convective or inertial effect increases with an increase in Reynolds number, yielding an increase in pressure drop.

across the medium. If a temperature or a concentration gradient occurs across the fibers, the recirculation or dispersion will mix the fluid particles and increase the net transport. Hunt and Tien (1988) consider these effects with the aid of a volume-averaging process. The momentum equation is obtained by volume averaging the Navier-Stokes equation and relating the drag force from the presence of the solid phase to the Darcian force. However, the averaging method eliminates microscopic phenomena such as local recirculation or dispersion, and thus the governing equations rely on additional empirical relations for closure.

Experimental studies on fluid flow through porous media made up of randomly packed monodisperse fibers have been carried out by Rahli *et al* (1997). The permeability and the Carman-Kozeny parameter, k_K , which is a constant value dependent on porosity and the aspect ratio of the fibrous microstructure, are obtained from experimental results. The permeability as a function of the porosity of the medium are determined using Darcy's law and an experimentally determined relationship. The permeability varies from 20 to 20,000 Darcy when the porosity varies from 0.38 to 0.90. By extending the Carman-Kozeny relation used in semiheuristic porous medium model to randomly packed cylindrical fibers, the evolution of the Carman-Kozeny parameter as a function of the porosity and the fiber aspect ratio is determined. The Carman-Kozeny parameter decreases continuously with an asymptotic tendency toward the value of 3.6 as the porosity approaches 1. The experimental results show that dependence of Carman-Kozeny parameter on the fiber-extremity surface is non-negligible for low fiber aspect ratio. Comparing the existing theoretical models with the experimental results indicate

that there is significant difference in the fluid flow behavior since the theoretical models that were established for regular arrays of fibers do not correctly describe the behavior of randomly packed fibers.

2.2.5 Porous Glass-Ceramics, Woven-Screen and Crossed-Rod Matrices

London and his co-workers have conducted extensive experiments on the heat transfer and flow friction characteristic of crossed-rod matrices, woven-screen matrices, and porous glass-ceramics matrices (Coppage and London, 1956, Tong and London, 1957, London *et al* , 1960, London *et al* , 1970, London and Shah, 1973; Kays and London, 1984) The experiments investigate the conventional steady state isothermal flow, in which the pressure drop is measured for essentially constant density air-flow. All of the experimental results for these matrix materials have been presented graphically in terms of a non-dimensional friction factor, f , as a function of Reynolds number, Re , for the fluid flow.

The porosities of the crossed-rod and woven-screen matrices investigated range from 60% to 83% and the Reynolds number ranges from 5 to 100,000, which covers the laminar flow region as well as the transition region. A correlation of the flow friction data with the *entire* range of Reynolds number is not successfully obtained (Coppage and London, 1956; Tong and London, 1957, London *et al* , 1960). When the porosity of the porous matrices exceeds 70%, the staggered crossed-rod configuration shows the highest friction factor while the in-line crossed-rod configuration has the lowest for a constant Reynolds number. The behavior of woven-screen is similar to that of the random

crossed-rod configuration. In addition, the friction behavior of the inline and staggered configurations at high porosities is consistent with the behavior of comparable tube-bank geometries. When the porosity approaches unity, the friction behavior of the inline and staggered configurations, by extrapolation, agree with the aerodynamic-drag behavior of single cylinders (London *et al* , 1960)

For the porous glass-ceramic matrices with the straight triangular, hexagonal, and circular passage surfaces, the porosities investigated covers the range of 64% - 81% and the Reynolds number are in a narrow range of $60 < Re < 1000$. The passage surfaces have heat transfer area density ratio ranging from 1300 - 2400 ft^2/ft^3 , corresponding to a passage count of 526 - 2215 passages/ in^2 . By comparison, the glass-ceramic with a hexagonal passage has its advantages over the triangular passage geometry, which is commonly used for gas turbine regenerators in vehicles. In one example, the gains were a 30% increase in heat transfer performance, along with a 13% reduction in pressure drop for the same disc envelope geometry tested (London *et al.*, 1970, London and Shah, 1973)

CHAPTER 3

Analytical Modeling on Heat Conduction

3.1 Cubic Strut Model

3.1.1 Approach

In principle, heat may transfer through foams by conduction, convection, and/or radiation. For porous carbon foams, however, the radiative contribution to heat transfer may be neglected since the foams are used at relatively low temperatures. The natural heat convection is also insignificant in comparison with heat conduction.

Dul'nev (1965) develops a cubic strut model for the heat conduction in open-cell foams by using a simple parallel-series resistor diagram. The validity of Dul'nev's model has been verified by comparing the experimental results of several reticulated porous materials with the Dul'nev's analytical model (Kamiuto, 1997). The present model extends Dul'nev's model, which was developed for isotropic struts, to anisotropic struts and strut junctures.

The present cubic strut model considers a porous foam as a collection of cubical unit cells composed of struts. In view of symmetry, only one-eighth of the cubic unit cell is considered; see Figure 3.1. The strut thickness is assumed to be uniform. Each individual strut is considered to be anisotropic with distinct longitudinal, transverse, and juncture thermal conductivities. The surfaces parallel to the y-z plane are assumed to be isothermal, whereas all of the four boundary surfaces that are parallel to the x-y plane and x-z plane are assumed to be adiabatic; see Figure 3.2. Heat is assumed to flow along

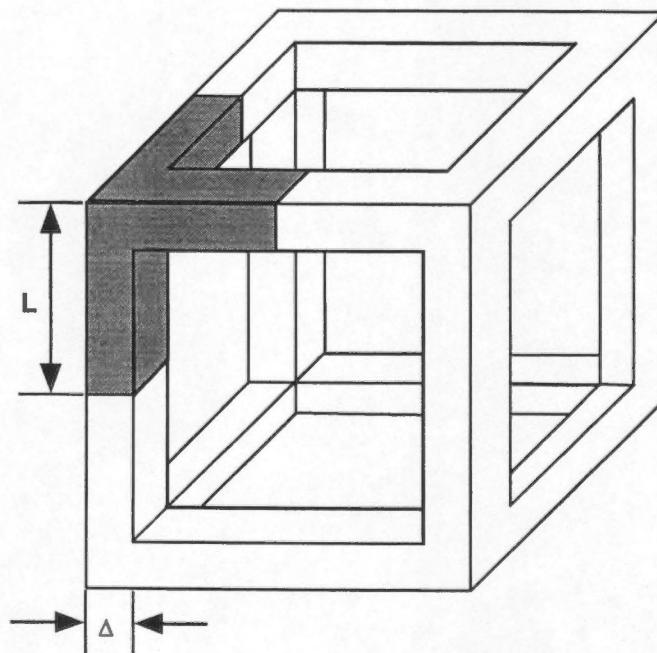


Figure 3.1: Unit cell of cubic strut model.

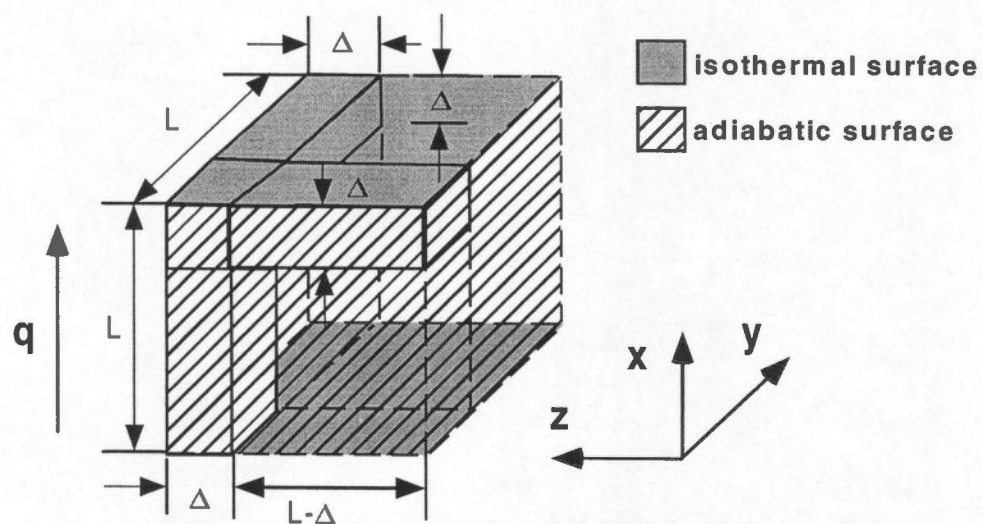


Figure 3.2: One-eighth of unit cell.

the x-direction and perpendicular to the y-z plane.

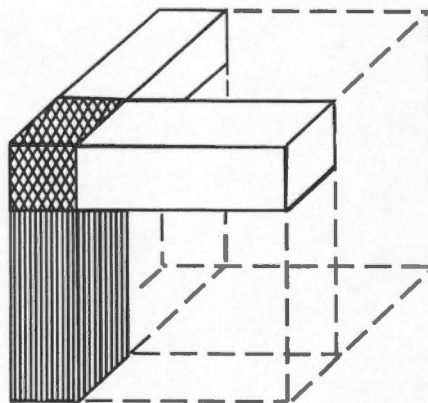
Fourier's law of heat conduction gives

$$q = \frac{kA}{l}(\Delta T) = \frac{1}{R}(\Delta T) \quad (3.1)$$

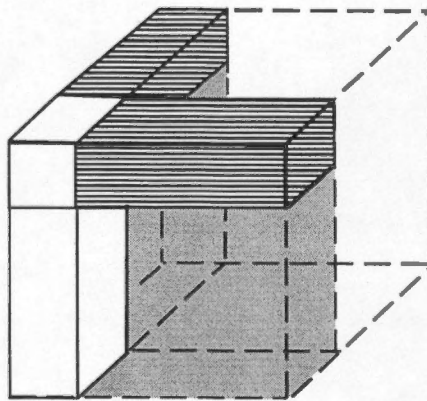
where q is the heat transfer rate, k is the thermal conductivity, A is the cross sectional area of the medium, l is the distance the heat travels, ΔT is the temperature difference, and R is the thermal resistance defined by

$$R = \frac{l}{kA} \quad (3.2)$$

The heat conduction through the unit cell may be simulated by means of four parallel heat resistors. The four heat resistors are composed of. (i) one series combination of the longitudinal strut and the resistor for the small cubical strut juncture, shown in Figure 3.3(a), (ii) two series combinations of the transverse struts and the underlying gas indicated in Figure 3.3(b), and (iii) the rectangular gas portion, see Figure 3.3(c). The resistor diagram of the overall heat resistance is shown in Figure 3.4, where R_{sl} is the resistance of the longitudinal strut, R_{st} is the resistance of the transverse strut, R_{sj} is the resistance of the cubical juncture, R_{g1} is the resistance of the rectangular gas portion, and R_{g2} is the resistance of the gas beneath the transverse strut, which, from Eq. (3.2), the thermal resistances can be given by

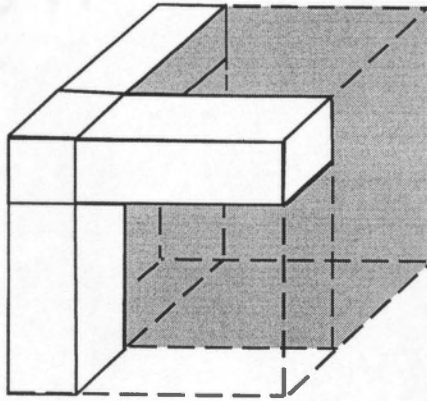


(a)



(b)

Figure 3.3: Heat flow in one-eighth of the cell: (a) Longitudinal strut and cubical strut juncture; (b) Transverse struts and underlying gas; (c) Rectangular gas portion.



(c)

Figure 3.3: Continue.

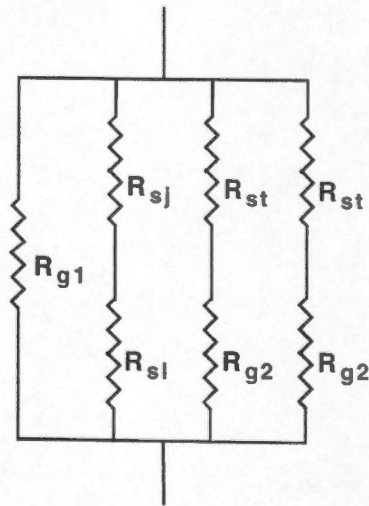


Figure 3.4: Heat resistance circuit diagram.

$$R_{sl} = \frac{(L - \Delta)}{k_{sl} \Delta^2} \quad (3.3)$$

$$R_{st} = \frac{1}{k_{st} (L - \Delta)} \quad (3.4)$$

$$R_{sj} = \frac{1}{k_{sj} \Delta} \quad (3.5)$$

$$R_{g1} = \frac{L}{k_g (L - \Delta)^2} \quad (3.6)$$

$$R_{g2} = \frac{1}{k_g \Delta} \quad (3.7)$$

where Δ is the thickness and width of the rectangular strut and L is the one-half of the length of the unit cell, k_{sl} is the longitudinal thermal conductivity of the strut, k_{st} is the transverse thermal conductivity of the strut, and k_{sj} is the thermal conductivity of the small cubical strut juncture, and k_g is the thermal conductivity of the gas.

Using the rules for parallel and series combinations of resistance, the effective resistance of the cell R^* can be determined by

$$\frac{1}{R^*} = \frac{1}{R_{g1}} + \frac{1}{(R_{s1} + R_{sj})} + \frac{2}{(R_{st} + R_{g2})} \quad (3.8)$$

Consequently, the effective thermal conductivity of the porous foam k^* is given by

$$k^* = k_g (1 - t)^2 + \frac{k_{s1} k_{sj} t^2}{k_{s1} t + k_{sj} (1 - t)} + \frac{2k_{st} k_g t(1 - t)}{k_g t + k_{st} (1 - t)} \quad (3.9)$$

where $t = \Delta / L$ is the normalized thickness. For isotropic struts, $k_{s1} = k_{st} = k_{sj}$, Eq. (3.9) reduces to (Dul'nev, 1965)

$$k^* = k_g (1 - t)^2 + k_s t^2 + \frac{2k_s k_g t(1 - t)}{k_g t + k_s (1 - t)} \quad (3.10)$$

where k_s is the thermal conductivity of the isotropic strut. It is noted that the main difference between the anisotropic and isotropic struts is the second term in the Eqs. (3.9) and (3.10). Furthermore, due to symmetry both the isotropic and anisotropic strut models can be extended to the case where the heat flows in the either y or z direction (Schuetz and Glicksman, 1984)

From the geometry shown in Figure 3.2, the porosity, ϕ , can be expressed by

$$\phi = \frac{[2(L - \Delta)^2 \Delta + (L - \Delta)^2 L]}{L^3} = 2t^3 - 3t^2 + 1 \quad (3.11)$$

To express the normalized thickness, t , in terms of porosity, ϕ , a dummy variable, y , is introduced to remove the t^2 term from Eq (3.11),

$$t = y + \frac{1}{2} \quad (3.12)$$

Thus, Eq (3.11) can be re-written by

$$y^3 - \frac{3}{4}y + \frac{1}{4}(1 - 2\phi) = 0 \quad (3.13)$$

Comparing Eq (3.13) with the following trigonometry identity

$$\cos^3 \theta - \frac{3}{4} \cos \theta - \frac{1}{4} \cos 3\theta = 0 \quad (3.14)$$

one finds

$$y = \cos \left(\frac{1}{3} \cos^{-1} (2\phi - 1) \right) \quad (3.15a)$$

$$y = \cos \left(\frac{1}{3} \cos^{-1} (2\phi - 1) + \frac{2}{3} \pi \right) \quad (3.15b)$$

$$y = \cos \left(\frac{1}{3} \cos^{-1}(2\phi - 1) + \frac{4}{3} \pi \right) \quad (3.15c)$$

In general, Eq. (3.15a) and (3.15b) are no good as they are not the real roots within the positive range. By substituting Eq. (3.15c) into Eq. (3.12), one finds the relationship between t and ϕ .

$$t = \frac{1}{2} + \cos \left(\frac{1}{3} \cos^{-1}(2\phi - 1) + \frac{4}{3} \pi \right) \quad (3.16)$$

3.1.2 Results

In the case of carbon foam, the thermal conductivity of carbon strut, k_{sl} , is given by (Klett, 1999a)

$$k_{sl} = -284.44 \ln(T) + 2344.6 \quad (3.17)$$

where T is the temperature in Kelvin and k_{sl} is expressed in W/mK. The conductivity of air is used for k_g and is determined by (Vargaftik *et al.*, 1994)

$$\begin{aligned} k_g = & [-0.9474 + 11.961 T / 100 - 2.3632 (T / 100)^2 + 0.8406 (T / 100)^3 \\ & - 0.1747 (T / 100)^4 + 0.01904 (T / 100)^5 - 0.001035 (T / 100)^6 \\ & + 0.00002228 (T / 100)^7] / 1000 \end{aligned} \quad (3.18)$$

3 1 2 1 Isotropic Struts

For isotropic struts, the thermal conductivities of the strut are assumed to be the same (k_{sl}) in all directions. The result for isotropic struts is shown in Figure 3.5, where the effective thermal conductivity is expressed as a function of temperature as well as porosity. The effective thermal conductivities of carbon foams with porosity of 70%, 77%, and 80% are compared with the experimental results for $\phi = 77\%$ (Klett, 1999a). The curve for experimental data is smoothened by data regression. The assumption of isotropic carbon struts apparently overestimates the effective thermal conductivity of porous carbon foam.

3 1 2 2 Anisotropic Struts

For the anisotropic case, the struts are assumed to exhibit different thermal conductivities k_{sl} , k_{st} , and k_{sj} , in the longitudinal and transverse direction and at the strut juncture, respectively. Since the struts of the carbon foam are highly aligned with the graphitic in-plane structure in the graphitization process (Klett, 1999b), the longitudinal thermal conductivity of the strut, k_{sl} , is assumed to be the in-plane thermal conductivity of bulk graphite, k_s . Thus, the transverse thermal conductivity k_{st} is considered to be the out-of-plane thermal conductivity of the bulk graphite. The ratio of out-of-plane thermal conductivity to the in-plane thermal conductivity of graphite varies from 0.5 to 0.8 (Mantell, 1968).

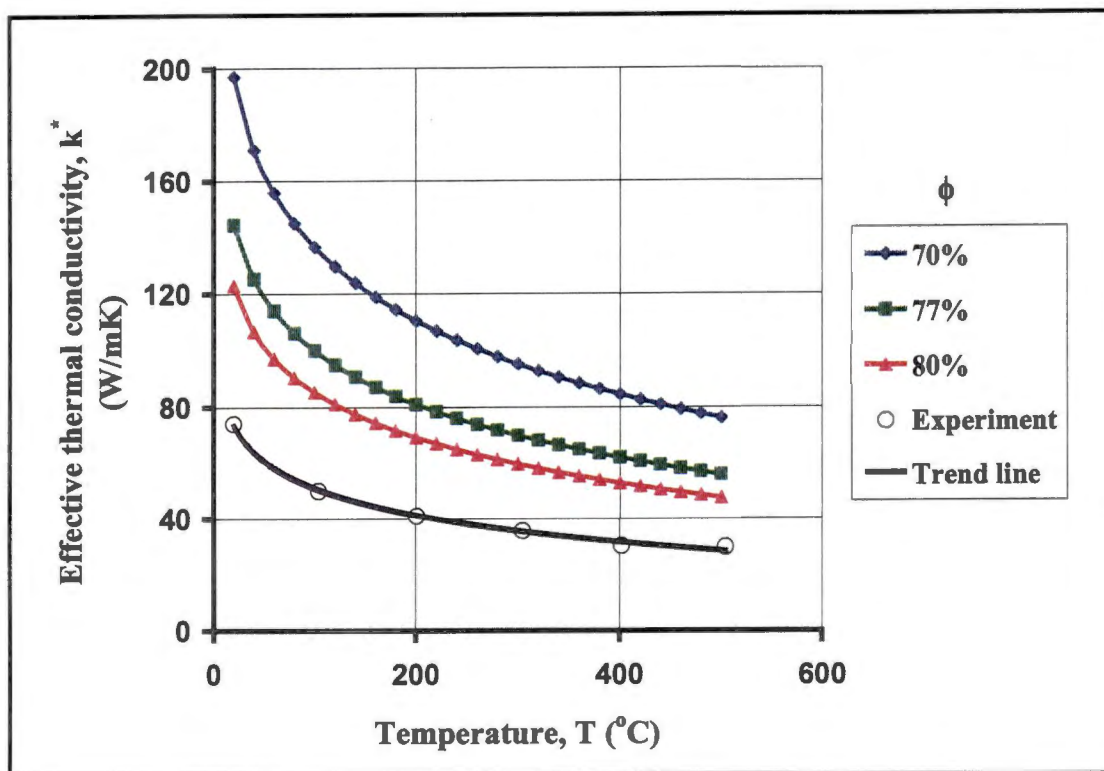


Figure 3.5: Effective thermal conductivity as a function of temperature for $\phi = 70\%$, 77% , and 80% ; $k_{sl} = k_{st} = k_{sj}$.

In Figure 3.6, the effective thermal conductivity of carbon foam consisting of anisotropic struts is expressed as a function of temperature. The porosity of 50% and a constant ratio of $k_{sj} / k_{sl} = 1$ together with $k_{st} / k_{sl} = 10, 1, 0.1$ and 0.01 are used. Similar results are shown in Figure 3.7 for $\phi = 90\%$. From Figure 3.6 and 3.7, one concludes that the effective thermal conductivity of the carbon foam, k^* , is not sensitive to the ratio of k_{st} / k_{sl} . Thus, k_{st} does not affect the overall effective thermal conduction of carbon foam.

On the other hand, the strut juncture is assumed to weaken the foam structure and reduce the overall thermal conductivity because there is more fold-sharpening and crystal misalignment in juncture (Klett, 1999b). Thus, the thermal conductivity of the strut juncture, k_{sj} , is assumed to be much lower than the longitudinal thermal conductivity of the strut k_{sl} . To study the effect of k_{sj} , the carbon foams with porosity of 77% and the ratio $k_{sj} / k_{sl} = 0.25, 0.5$, and 0.75 are considered in Figure 3.8. In view of the fact that the ratio of out-of-plane thermal conductivity to the in-plane thermal conductivity is approximately equal to 0.75 for bulk graphite, the ratio of k_{st} / k_{sj} is set to be equal to 0.75. It shows in Figure 3.8 that the effective thermal conductivity of the foam k^* is significantly affected by the thermal conductivity of the small cubical strut juncture, k_{sj} . The result of the present analytical model also matches with the experiment data (Klett, 1999a) when the thermal conductivity of the strut juncture is assumed to be equal to 25% of the longitudinal thermal conductivity of the carbon strut.

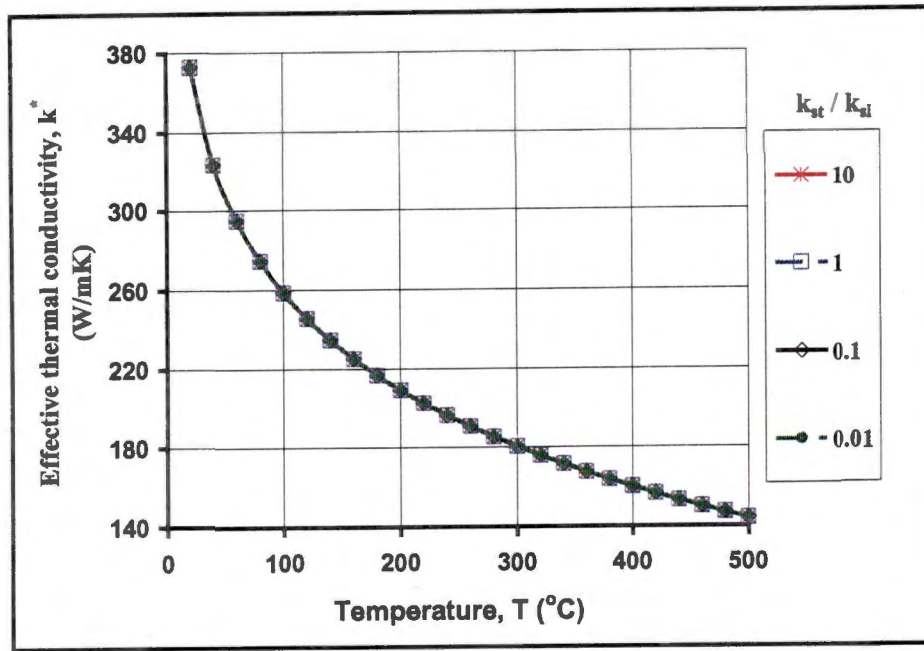


Figure 3.6: Effective thermal conductivity as a function of temperature for $k_{st} / k_{sl} = 10$, 1, 0.1, and 0.01, $k_{sj} / k_{sl} = 1$, $\phi = 50\%$.

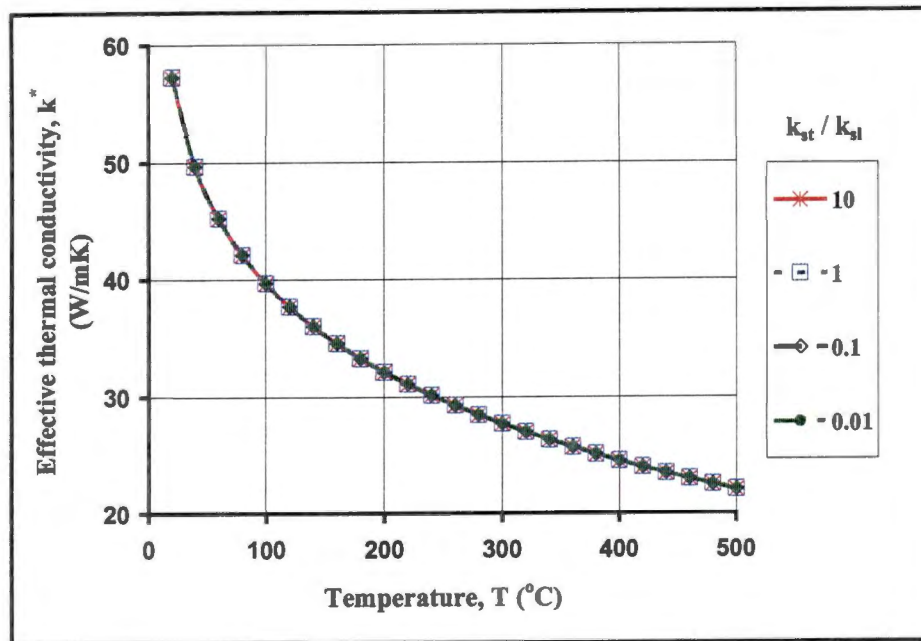


Figure 3.7: Effective thermal conductivity as a function of temperature for $k_{st} / k_{sl} = 10$, 1, 0.1, and 0.01, $k_{sj} / k_{sl} = 1$, $\phi = 90\%$.

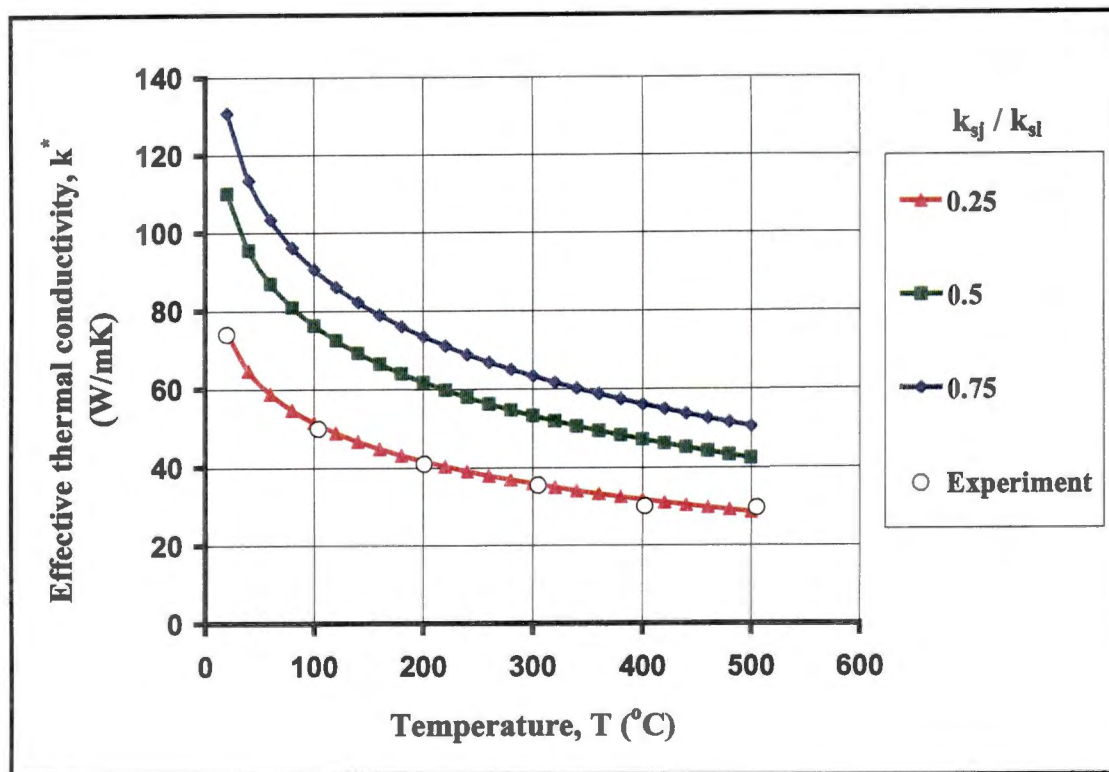


Figure 3.8: Effective thermal conductivity as a function of temperature for $k_{sj} / k_{sl} = 0.25$, 0.50, and 0.75, $k_{st} / k_{sl} = 0.75$, and $\phi = 77\%$.

3.1 3 Discussion

The main difference between the present analytical model and the real carbon foam is the foam geometry. The real foam closely resembles the shape of a pentagonal dodecahedron or tetrakaidecahedron but the present model assumes the foam cell to be in the shape of a cube. However, according to Dul'nev (1965), cubical and spherical shape of the pores will give practically similar values of the effective thermal conductivity of the foam. Thus, the present geometric model may be valid for the modeling of thermal conduction. In addition, the unit cell of the cubical strut model shown in Figure 3.1 involves eight transverse struts and four longitudinal struts, which implies that one-third of the struts are oriented in the direction of heat conduction. From the pentagonal dodecahedron in Figure 1.2, the foam cell has ten fully transverse struts and twenty struts oriented with an angle between the longitudinal and transverse direction. As the twenty intermediate struts are symmetrical and oriented in the same angle, the assumption of ten struts being longitudinal and ten struts being transverse can be made. Thus, the real foam also involves one-third of the struts being oriented in the direction of heat flow.

It is noted that the assumption of isotropic struts, which is used in Dul'nev's (1965) is not a very appropriate approximation of the carbon foam even if the overall carbon foam is isotropic. The present model based on anisotropic struts accurately predicts the effective thermal conductivity of porous carbon foam as functions of temperature and porosity.

3.2 Effects of Misoriented Struts on Overall Heat Conduction in Carbon Foam

3.2.1 Approach

A porous carbon foam may be considered as a collection of randomly distributed cubical cells, as shown in Figure 3.9. The cubical cells are similar in size and shape but differ by orientation.

The thermal conductivity, similar to other transport constants, is a second-rank tensor. For the steady-state heat conduction, the linear relationship between the heat transfer flux, q , and the temperature gradient, ∇T , can be given by Fourier's law of heat conduction in the form of

$$q = k (-\nabla T) \quad (3.19)$$

where k denotes the thermal conductivity and the minus sign is a consequence of the second law of thermodynamics, which requires that heat must flow in the direction from higher to lower temperature; see, for example, Rohsenow (1985). It is assumed that k is a function of position and temperature only. Equation (3.19) can be rewritten in the indicial notation by

$$q_i = k_{ij} \left(-\frac{\partial T}{\partial x_j} \right) \quad (3.20)$$

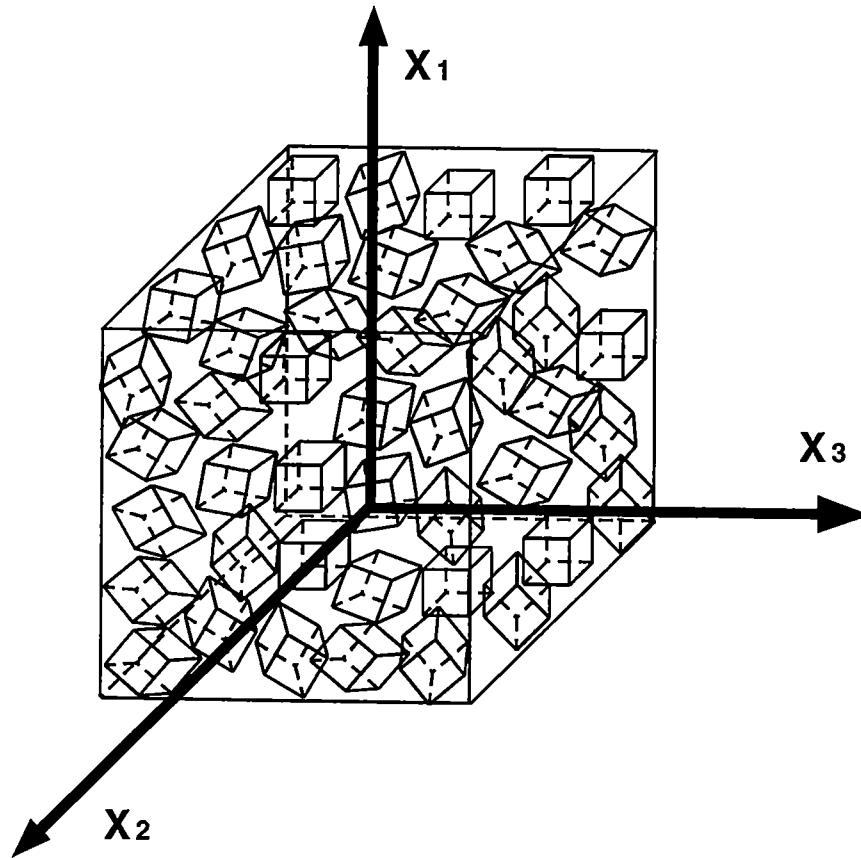


Figure 3 9: Block of carbon foam composed of randomly oriented cubical cells

where i and $j = 1 \sim 3$ and the summation convention is used. That is, a repeated index representing the tensor quantity for the set of three components. Equation (3.20) can also be written in matrix form by

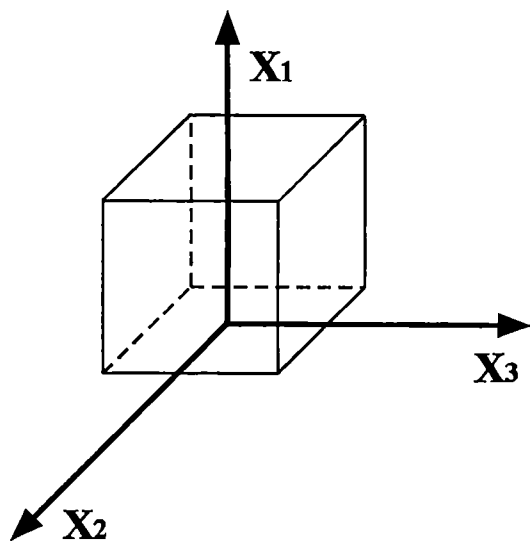
$$\begin{Bmatrix} q_1 \\ q_2 \\ q_3 \end{Bmatrix} = \begin{bmatrix} k_{11} & k_{12} & k_{13} \\ k_{21} & k_{22} & k_{23} \\ k_{31} & k_{32} & k_{33} \end{bmatrix} \begin{Bmatrix} \frac{\partial T}{\partial x_1} \\ \frac{\partial T}{\partial x_2} \\ \frac{\partial T}{\partial x_3} \end{Bmatrix} \quad (3.21)$$

In view of the cubic symmetry of the carbon foam cell shown in Figure 3.1, the estimated effective thermal conductivity of the cell, Eq. (3.9), is assumed to be the same in the X_1 (x), X_2 (y), and X_3 (z), directions. Thus, the conductivity matrix in Eq. (3.21) reduces to

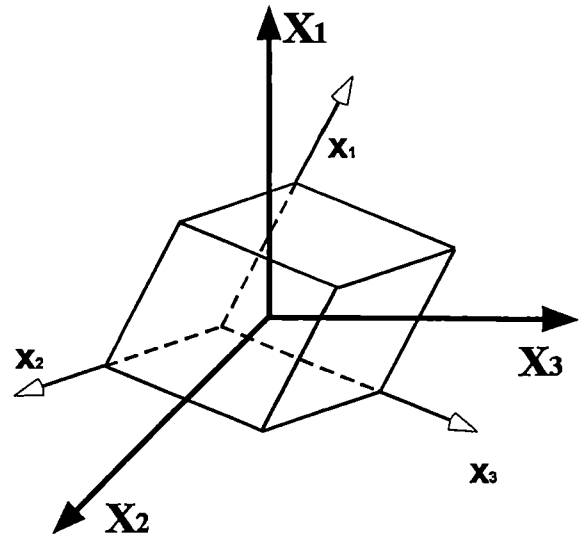
$$[K^*] = \begin{bmatrix} k^* & 0 & 0 \\ 0 & k^* & 0 \\ 0 & 0 & k^* \end{bmatrix} \quad (3.22)$$

where k^* is given in Eq. (3.9).

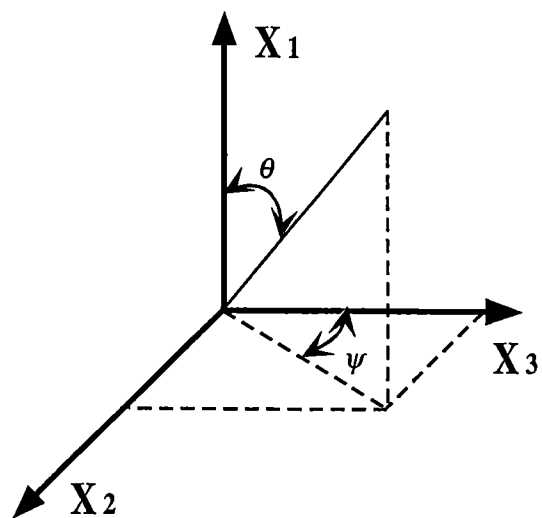
In order to discuss the effect of misoriented carbon cells, the coordinate transformation relationship must be established. The heat transfer flux q in the local (x_1 , x_2 , x_3) and the global (X_1 , X_2 , X_3) coordinate system (see Figure 3.10) are related by



(a)



(b)



(c)

Figure 3 10. Coordinate transformation: (a) Aligned cell, (b) Misoriented cell, (c)

Rotation about the x_1 - and x_3 -axis

$$\{ q_i \}^{local} = [Q_{ij}] \{ q_j \}^{global} \quad (3.23)$$

where the transformation matrix $[Q]$ is given by

$$[Q] = \begin{bmatrix} \cos \theta & -\sin \theta \cos \psi & \sin \theta \sin \psi \\ \sin \theta & \cos \theta \cos \psi & -\sin \psi \cos \theta \\ 0 & \sin \psi & \cos \psi \end{bmatrix} \quad (3.24)$$

where θ is a rotation about the X_3 axis and ψ is a rotation about the X_1 axis (Chou and Nomura, 1981). Similarly, the temperature gradient in the local and global coordinate systems are related by

$$\left\{ \frac{\partial T}{\partial x_i} \right\}^{local} = [Q_{ij}] \left\{ \frac{\partial T}{\partial X_j} \right\}^{global} \quad (3.25a)$$

Substituting Eq (3.23) and (3.25) into Eq. (3.19) gives

$$\{ q_i \}^{global} = [K_{ij}] \left\{ \frac{\partial T}{\partial X_j} \right\}^{global} \quad (3.25b)$$

where the thermal conductivity of the misoriented cell in the global coordinate system, K^{**} , is given by

$$[K^{**}] = [Q]^T [K^*] [Q^T] \quad (3.26)$$

where $[Q]^T$ is the transpose of the transformation matrix

For the three-dimensional effective conductivities, K^{**} , of a misoriented foam cell, the combination of Eq. (3.22), Eq. (3.24), and (3.26) gives

$$\begin{aligned} [K^{**}] &= \begin{bmatrix} k^*(\sin^2 \theta + \cos^2 \theta) & 0 & 0 \\ 0 & k^*(\cos^2 \psi (\sin^2 \theta + \cos^2 \theta) + \sin^2 \psi) & -k^* \sin \psi \cos \psi (\sin^2 \theta + \cos^2 \theta - 1) \\ 0 & -k^* \sin \psi \cos \psi (\sin^2 \theta + \cos^2 \theta - 1) & k^*(\sin^2 \psi (\sin^2 \theta + \cos^2 \theta) + \cos^2 \psi) \end{bmatrix} \\ &= \begin{bmatrix} k^* & 0 & 0 \\ 0 & k^* & 0 \\ 0 & 0 & k^* \end{bmatrix} \end{aligned} \quad (3.27)$$

From Eq. (3.27), it clearly shows that the effective thermal conductivity, K^{**} , of a misoriented cell is identical to that of the aligned cell provided that the cell is cubic ($k_{11} = k_{22} = k_{33} = k^*$).

For a carbon foam consisting of a large number of randomly distributed cubical cells, the effective thermal conductivity, $\bar{K}$, will be given by (Hatta and Taya, 1985)

$$\overline{K} = \frac{1}{4\pi} \int_0^\pi \int_0^{2\pi} K^{**} n(\theta, \psi) \sin \theta d\psi d\theta \quad (3.28)$$

where $n(\theta, \psi)$ is the probability density function of cell orientation and the integration range is from 0 to π for θ and 0 to 2π for ψ to enclose the entire sphere, see Figure 3.11

For the case of a completely random distribution, the probability density function should be independent of θ and ψ , i.e., $n(\theta, \psi) = n_0 = 1$ (Hatta and Taya, 1985). Consequently, $\overline{K}$ is again given by

$$\overline{K} = \begin{bmatrix} k^* & 0 & 0 \\ 0 & k^* & 0 \\ 0 & 0 & k^* \end{bmatrix} \quad (3.29)$$

for the particular case of a perfectly random distribution of cubic cells with the same thermal conductivity in x_1 -, x_2 -, and x_3 -directions

3.2.2 Discussion

The analytical model proposed by Schuetz and Glicksman (1984) predicts that the effective thermal conductivity of foams composed of randomly oriented struts is exactly the same as that for foams consisting of aligned struts regardless of the heat flow direction. The present model yields the same conclusion provided that the cell is cubic symmetric

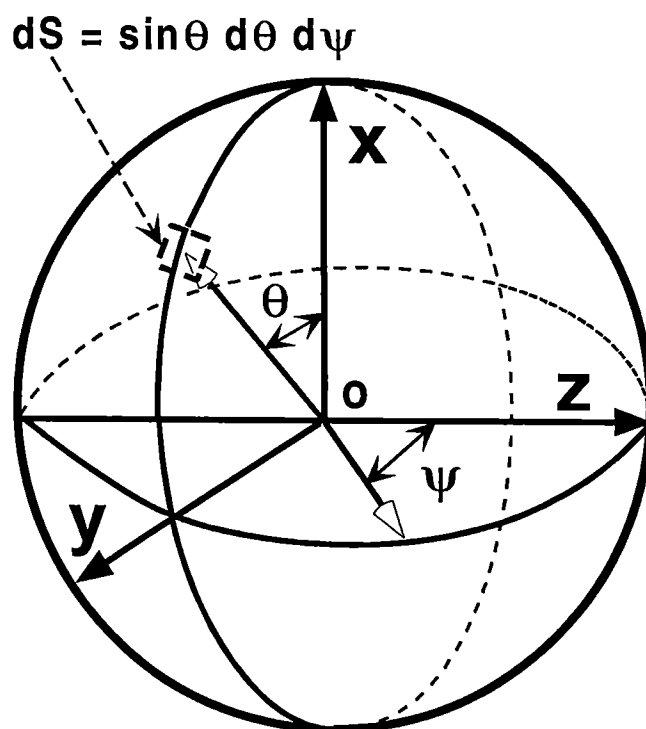


Figure 3 11: Unit sphere describing the orientation of a unit cell.

Furthermore, the present model assumes that the unit cells are randomly oriented and distributed in the carbon foam, which may not be true for foams with macroscopically anisotropic properties. In order to accurately predict the thermal conductivity of carbon foam, the probability distribution function for the strut orientation as well as distribution in the foams may need to be characterized experimentally.

3.3 Tapered Strut Model

3.3.1 Approach

In the present model, all the struts are considered to be tapered with the minimum thickness being in the middle of the struts, see Figure 3.12. All four edges of the strut have the same tapered curve. The junction of the strut is assumed to be a cube. In view of symmetry, only one-eighth of the cubic unit cell is considered in the present analysis. Each individual strut is considered to be anisotropic with distinct longitudinal, transverse, and juncture thermal conductivities. Again, the surfaces parallel to the y-z plane are assumed to be isothermal, whereas all of the other four boundary surfaces that are parallel to the x-y plane or x-z plane are assumed to be adiabatic. Heat is assumed to flow in the x-direction, see Figure 3.12.

To study the effect of the shape of the strut on thermal conduction, the longitudinal strut of the unit cell is considered. The projected longitudinal strut onto the x-y plane is shown in Figure 3.13. By symmetry of the vertical plane along y-axis, only half of the strut is mathematically represented by a parabola equation of

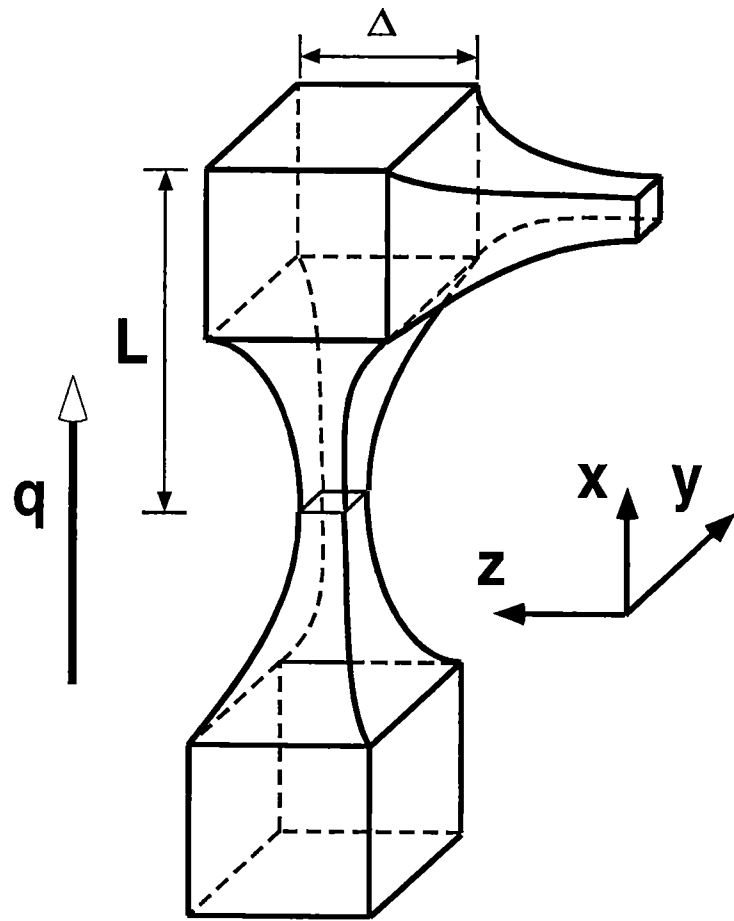


Figure 3.12 Tapered strut model.

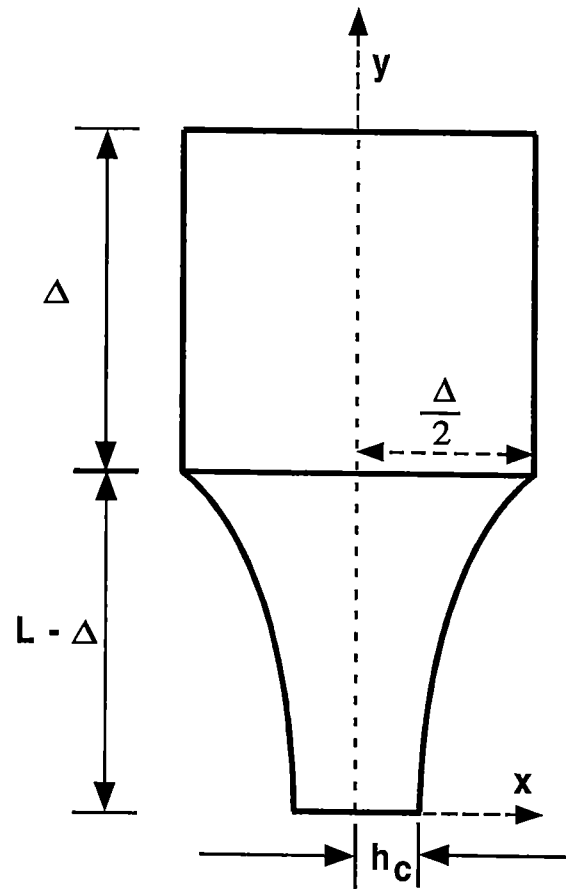


Figure 3 13 Projected two-dimensional view of tapered strut.

$$y^2 = 4 a_c (x - h_c) \quad (3.30)$$

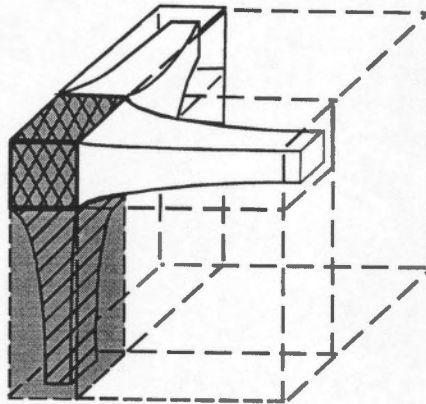
where h_c is half of the minimum strut thickness in the middle of the strut, and a_c is the x-coordinate of the fixed focus point, and is given by

$$a_c = \frac{(L - \Delta)^2}{4 \left(\frac{\Delta}{2} - h_c \right)} \quad (3.31)$$

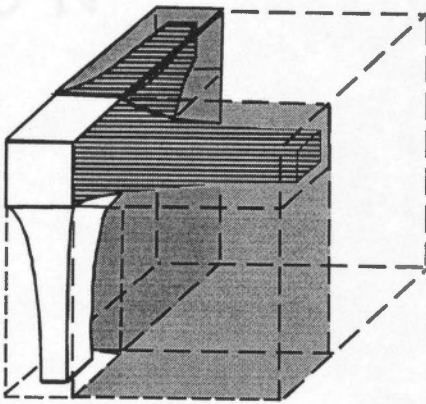
where Δ is the thickness of the strut juncture and L is the half-height of the unit cell

The heat conduction through the unit cell may be simulated by means of four parallel heat resistors. They are (i) one series combination made up of a small cubical strut juncture and the parallel combination of a tapered longitudinal strut and the tapered gas portion under the cubical strut juncture; see Figure 3.14(a), (ii) two series combinations of the transverse struts and the underlying gas, see Figure 3.14(b), and (iii) the rectangular gas portion, see Figure 3.14(c).

The resistor diagram for the overall heat resistance is shown in Figure 3.15, where R_{sl} is the longitudinal resistance of the tapered strut, R_{st} is the transverse resistance of the tapered strut, R_{sj} is the resistance of the cubical junction, R_{g1} is the resistance of the rectangular gas portion, R_{g2} is the resistance of the gas beneath the transverse strut, and R_{g3} is the resistance of the tapered gas beneath the cubical strut juncture. The thermal resistances are given by

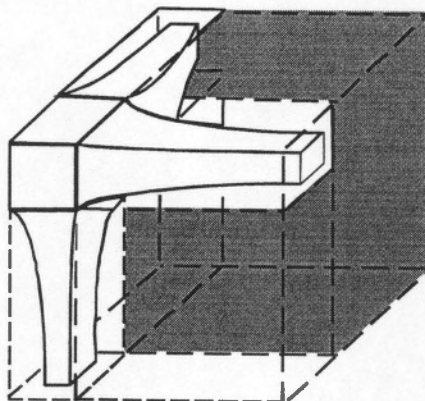


(a)



(b)

Figure 3.14: Heat flow in one-eighth of the cell: (a) Cubical strut juncture, tapered longitudinal strut, and tapered gas portion under the cubical strut juncture; (b) Transverse struts and underlying gas; (c) Rectangular gas portion.



(c)

Figure 3.14: Continue.

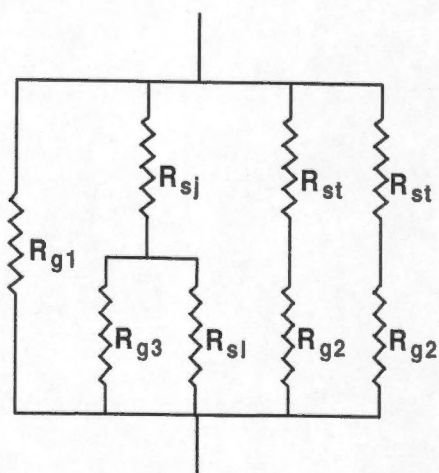


Figure 3.15: Heat resistance circuit diagram.

$$R_{sl} = \frac{1}{k_{sl}} \int_0^{(L-\Delta)} \left[\frac{1}{4 \left(\frac{y^2}{4a_c} + h_c \right)^2} \right] dy \quad (3.32)$$

$$R_{st} = \frac{1}{k_{st} (L - \Delta)} \quad (3.33)$$

$$R_{sj} = \frac{1}{k_{sj} \Delta} \quad (3.34)$$

$$R_{g1} = \frac{L}{k_g (L - \Delta)^2} \quad (3.35)$$

$$R_{g2} = \frac{1}{k_g \Delta} \quad (3.36)$$

$$R_{g3} = \frac{1}{k_g} \int_0^{(L-\Delta)} \left[\frac{1}{\Delta^2 - 4 \left(\frac{y^2}{4a_c} + h_c \right)^2} \right] dy \quad (3.37)$$

where k_{sl} is the longitudinal thermal conductivity of the strut, k_{st} is the transverse thermal conductivity of the strut, and k_{sj} is the thermal conductivity of the small cubical strut juncture, and k_g is the thermal conductivity of the gas

Since the prismatic strut model indicates that the transverse struts do not contribute significantly to the overall effective thermal conductivity of the foam (see subsection 3.1), both transverse struts are thus assumed to be prismatic in calculating the heat resistor. The error due to this assumption is shown to be less than 0.01%.

To compute the effective thermal conductivity of the foam cell, the rules for parallel and series combination are applied. The effective resistance, R^* , of the foam cell can be determined by

$$\frac{1}{R^*} = \frac{1}{R_{g1}} + \frac{1}{\left[\left(\frac{1}{R_{sl}} + \frac{1}{R_{g3}} \right)^{-1} + R_{sj} \right]} + \frac{2}{(R_{st} + R_{g2})} \quad (3.38)$$

Consequently, the effective thermal conductivity, k^* , of the porous foam is given by

$$k^* = k_g (1 - t)^2 + \frac{k_{sj} t (k_g I_1 + k_{sl} I_2)}{k_{sj} I_3 I_2 t + (k_g I_1 + k_{sl} I_2)} + \frac{2k_{st} k_g t (1 - t)}{k_g t + k_{st} (1 - t)} \quad (3.39)$$

where $t = \Delta / L$ is the normalized thickness, I_1 , I_2 , and I_3 are

$$I_1 = \int_0^{(L-\Delta)} \left[\frac{1}{4 \left(\frac{y^2}{4a_c} + h_c \right)^2} \right] dy \quad (3.40a)$$

$$I_2 = \int_0^{(L-\Delta)} \left[\frac{1}{\Delta^2 - 4 \left(\frac{y^2}{4a_c} + h_c \right)^2} \right] dy \quad (3.40b)$$

$$I_3 = \int_0^{(L-\Delta)} \left[\frac{L}{4 \left(\frac{y^2}{4a_c} + h_c \right)^2} \right] dy \quad (3.40c)$$

From the geometry shown in Figure 3.14, the porosity, ϕ , can be expressed by

$$\phi = 1 - \frac{3 \left[\int_0^{(L-\Delta)} 4 \left(\frac{y^2}{4a_c} + r \frac{\Delta}{2} \right)^2 dy \right] + \Delta^3}{L^3} \quad (3.41)$$

where $r = 2h_c / \Delta$ is the ratio of the minimum strut thickness at the middle to the maximum at the junction. After normalizing and integrating, Eq. (3.41) can be rearranged in terms of r and t as

$$5\phi = (8r^2 + 4r - 2)t^3 - (8r^2 + 4r + 3)t^2 + 5 \quad (3.42)$$

where $r = 2h_c / \Delta$ is the ratio of the minimum strut thickness at the middle to the maximum at the junction

To express the normalized thickness, t , in terms of ϕ and r , a dummy variable, y_c , is introduced to remove the t^2 term from Eq (3 42),

$$t = y_c + \frac{1}{3} \left[\frac{(8r^2 + 4r - 2) + 5}{(8r^2 + 4r - 2)} \right] \quad (3.43)$$

Thus, Eq (3 42) can be re-written by

$$y_c^3 - \frac{1}{3} \left[\frac{(8r^2 + 4r - 2) + 5}{(8r^2 + 4r - 2)} \right]^2 y_c - \frac{2}{27} \left[\frac{(8r^2 + 4r - 2) + 5}{(8r^2 + 4r - 2)} \right]^3 + \frac{5(1 - \phi)}{(8r^2 + 4r - 2)} = 0 \quad (3 44)$$

Comparing Eq (3 44) with the following trigonometry identity:

$$\cos^3 \theta - \frac{3}{4} \cos \theta - \frac{1}{4} \cos 3\theta = 0 \quad (3 45)$$

one finds

$$y_c = \cos \left(\frac{1}{3} \cos^{-1} \left(\frac{8}{27} \left[\frac{(8r^2 + 4r - 2) + 5}{(8r^2 + 4r - 2)} \right]^3 - \frac{20(1 - \phi)}{(8r^2 + 4r - 2)} \right) \right) \quad (3.46a)$$

$$y_c = \cos \left(\frac{1}{3} \cos^{-1} \left(\frac{8}{27} \left[\frac{(8r^2 + 4r - 2) + 5}{(8r^2 + 4r - 2)} \right]^3 - \frac{20(1 - \phi)}{(8r^2 + 4r - 2)} \right) + \frac{2}{3} \pi \right) \quad (3.46b)$$

$$y_c = \cos \left(\frac{1}{3} \cos^{-1} \left(\frac{8}{27} \left[\frac{(8r^2 + 4r - 2) + 5}{(8r^2 + 4r - 2)} \right]^3 - \frac{20(1 - \phi)}{(8r^2 + 4r - 2)} \right) + \frac{4}{3} \pi \right) \quad (3.46c)$$

In general, only Eq (3.46c) gives the suitable root. However, Eq. (3.46c) only provides a real root for $r < 0.2$ or $r > 0.8$. For all other values of r , a numerical solution to the cubic equation, Eq (3.42), has to be used. By substituting Eq (3.46c) into Eq. (3.43), one finds the relationship between t , r , and ϕ .

$$t = \cos \left(\frac{1}{3} \cos^{-1} \left(\frac{8}{27} \left[\frac{(8r^2 + 4r - 2) + 5}{(8r^2 + 4r - 2)} \right]^3 - \frac{20(1 - \phi)}{(8r^2 + 4r - 2)} \right) + \frac{4}{3} \pi \right) + \frac{1}{3} \left[\frac{(8r^2 + 4r - 2) + 5}{(8r^2 + 4r - 2)} \right] \quad (3.47)$$

3.3.2 Results

In the case of carbon foam, the ratio of minimum strut thickness to the maximum strut thickness at the junction, r , falls within the range of 0.3 to 1 (Klett, 1999a). To

study the effect of r on effective thermal conductivity of the foam, a list of r is established for isotropic strut with $\phi = 0.8$ at $T = 20^\circ\text{C}$ in Table 3.1 where A_{sj} ($=\Delta^2$) is the cross sectional area of the strut juncture, and A^* , is the equivalent cross sectional area of the longitudinal strut in one-eighth of a unit cell and is given by

$$A^* = \left\{ \int_0^{(L-\Delta)} \left[\frac{1}{4 \left(\frac{y^2}{4a_c} + h_c \right)^2} \right] dy \right\} \frac{1}{(L-\Delta)} \quad (3.48)$$

Table 3.1 The effect of r on k^* for carbon foam with $\phi = 0.8$ at $T = 20^\circ\text{C}$.

r	t	k^* (W/mK)	A_{sj} (m^2)	A^* (m^2)
0.01	0.4989	0.978	0.4989	0.0006
0.1	0.4868	25.785	0.4869	0.0181
0.2	0.4697	60.276	0.4695	0.0447
0.3	0.4492	89.857	0.4491	0.0695
0.4	0.4262	111.114	0.4262	0.0902
0.5	0.4017	124.200	0.4017	0.1049
0.6	0.3768	130.656	0.3768	0.1140
0.8	0.3290	130.858	0.3289	0.1195
0.9	0.3072	127.487	0.3072	0.1183
1.0	0.2871	123.083	0.2873	0.1157

With constant ϕ , t decreases as r increases as shown in Table 3.1. Besides, it is shown in Table 3.1 that the effective thermal conductivity, k^* is minimum at $r = 0.01$ and maximum at $r = 0.8$

The effective thermal conductivity of carbon foam, k^* , is plotted as a function of r , with $\phi = 0.8$ at $T = 20^\circ\text{C}$ in Figure 3.16. It is noted that k^* increases from $r = 0.1$ to approximately 0.8 and decreases as r approaches 1.0 . An increase in k^* resulted from an increase in r is due to the fact that the solid portion in the middle of the struts increases (i.e., the strut is thicker and the juncture becomes smaller as r increases). As a result, the effective thermal conductivity k^* significantly increases. However, the trend stops at $r \approx 0.8$ when k^* reaches its maximum value and starts decreasing. The main reason for this phenomenon may be because of the contribution of the equivalent cross sectional area of the strut, A^* . The equivalent cross sectional area A^* as a function of r with $\phi = 0.8$ at $T = 20^\circ\text{C}$ is shown in Figure 3.17. With an increase in r , the cross sectional area of the strut juncture decreases and the equivalent cross sectional area of the strut increases. However, such an increase in A^* stops and beginning to decrease at $r \approx 0.8$ while the cross-sectional area of the strut juncture continues to decrease as r increase from 0.8 to 1.0 . This is a major factor that affects the effective thermal conductivity of the overall carbon foam.

On the other hand, to study the effect of taper ratio r only a constant normalized thickness, $t = 0.29$ was considered. By holding all the other dimensions constant, three r values of 1.0 , 0.7 , and 0.3 are shown in Figure 3.18. Apparently the strut with a smaller r value would have a higher porosity ϕ . The effective thermal conductivity, k^* , with $t =$

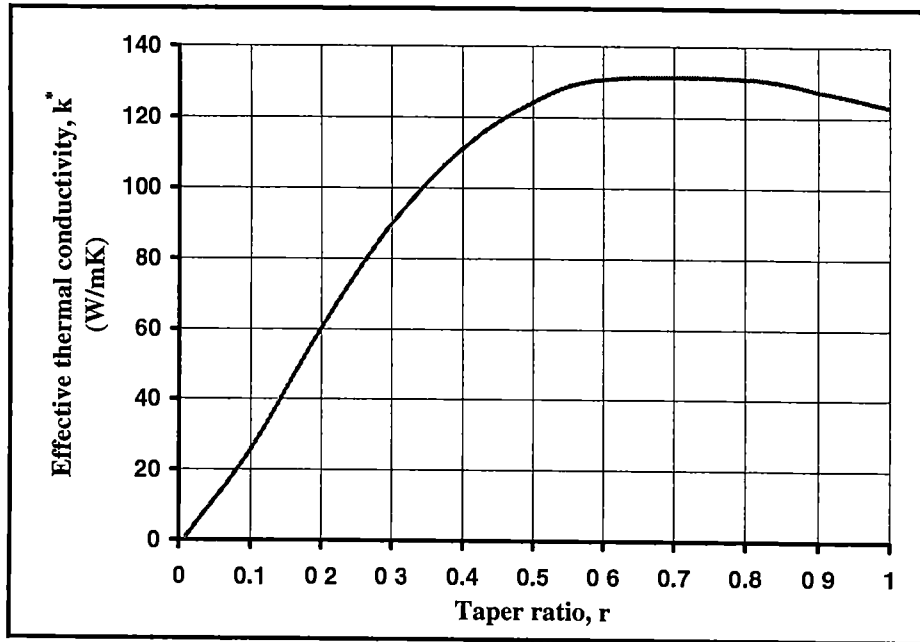


Figure 3 16 Effective thermal conductivity as a function of r , with $\phi = 0.8$ and $T = 20^\circ\text{C}$.

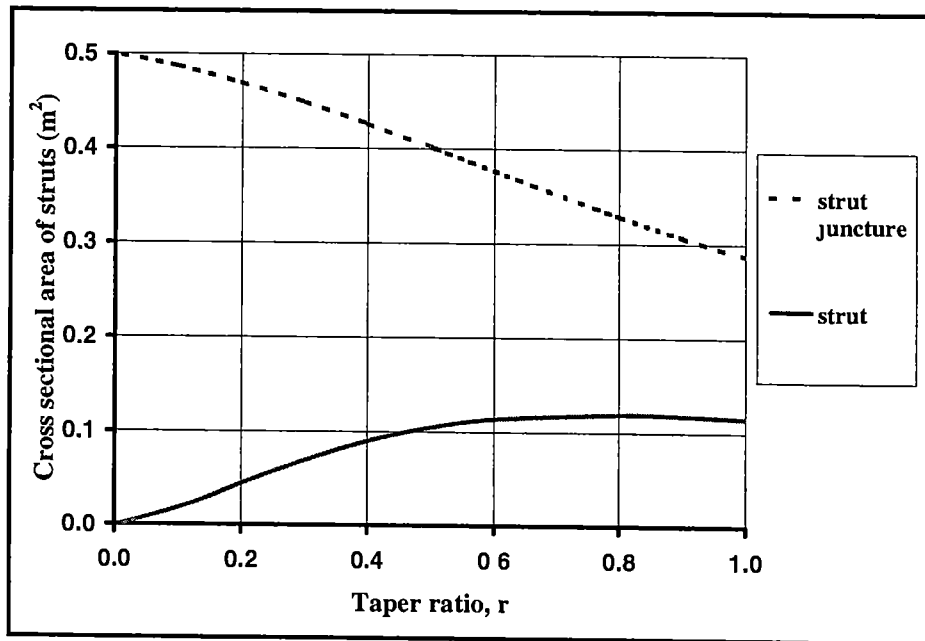


Figure 3 17. Strut cross sectional area as a function of r with $\phi = 0.8$ and $T = 20^\circ\text{C}$.

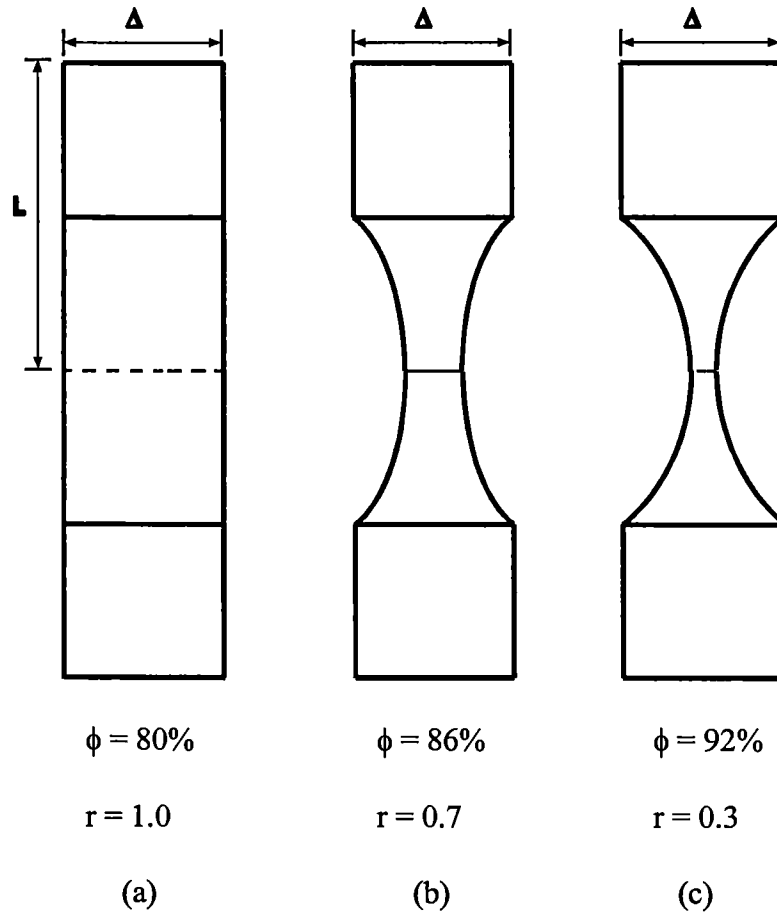


Figure 3.18: Two dimensional view of tapered strut with $t = 0.29$ and: (a) $r = 1.0$ ($\phi = 80\%$), (b) $r = 0.7$ ($\phi = 86\%$), and (c) $r = 0.3$ ($\phi = 92\%$).

0.29 at $T = 20\text{ }^{\circ}\text{C}$ is plotted as a function of r in Figure 3.19 where the relationship between k^* and r is approximately linear

3.3.2.1 Isotropic Struts

The predicted effective thermal conductivity k^* of carbon foam consisting of isotropic struts ($k_{sl} = k_{st} = k_{sj}$) is expressed as a function of temperature as well as porosity in Figure 3.20. The effective thermal conductivities of carbon foams consisting of prismatic struts with porosity of 70% and 80% are compared with those consisting of tapered struts. For the tapered strut model, r is assumed to be 0.3. As it is seen in Figure 3.20, the tapered strut model predicts lower effective thermal conductivities than those predicted by the prismatic strut model.

3.3.2.2 Anisotropic Struts

When the struts are assumed to exhibit different thermal conductivities k_{sl} , k_{st} , and k_{sj} , in longitudinal and transverse directions, and at the strut juncture, the effective thermal conductivity of carbon foam is expressed as a function of temperature in Figure 3.21 for $k_{sj} / k_{sl} = 0.25, 0.5$, and 0.75 . Since k^* is not significantly affected by k_{st} , the ratio of k_{st} / k_{sj} is set to be 0.75 . As it is seen in Figure 3.21, the tapered strut model with $r = 0.3$ predicts a significantly lower effective thermal conductivities than those predicted by the prismatic strut model. The difference between the two model increases when the ratio of k_{sj} / k_{sl} increases.

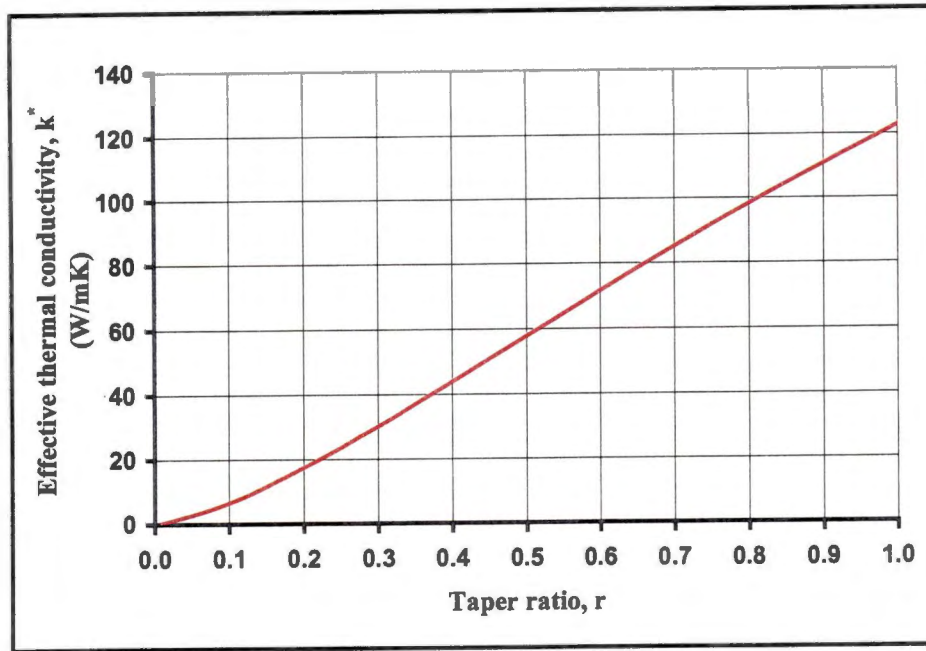


Figure 3.19: Effective thermal conductivity as a function of r with $t = 0.29$ and $T = 20^\circ\text{C}$.

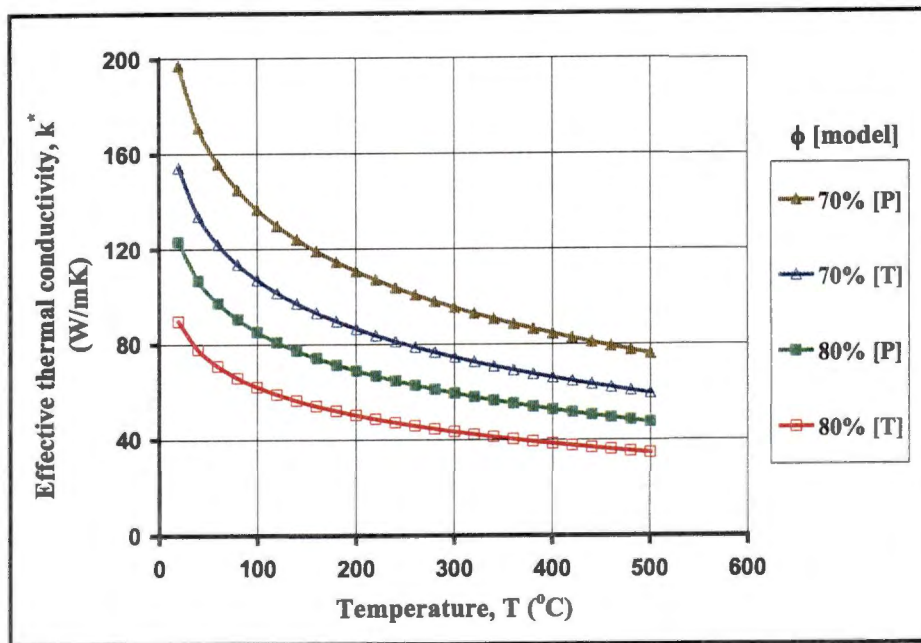


Figure 3.20: Effective thermal conductivity as a function of temperature using prismatic strut model (P) and tapered strut model (T) with $r = 0.3$, $\phi = 70\%$, and 80% ; $k_{sl} = k_{st} = k_{sj}$.

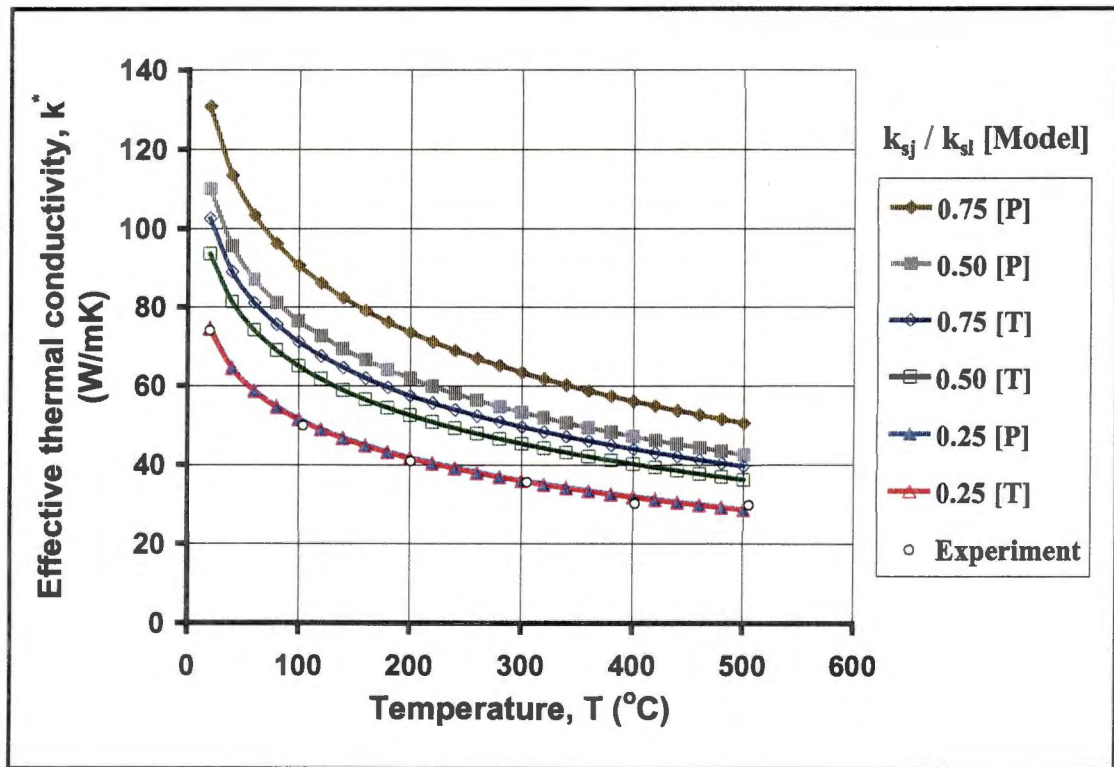


Figure 3.21: Effective thermal conductivity as a function of temperature using prismatic strut model (P) and tapered strut model (T) with $r = 0.3$, $k_{sj} / k_{sl} = 0.25$, 0.50, and 0.75, $k_{st} / k_{sl} = 0.75$, and $\phi = 77\%$.

Since the struts of the carbon foam have a wide variation of the taper ratio, r , from 0.3 to 1; thus, an average r value of 0.7 is chosen. More experimental studies of the actual carbon foam are recommended to verify the tapering effect. Figure 3.22 illustrates k^* as a function of temperature and compares the prediction with the curve for experimental data that is smoothened by data regression (Klett, 1999a) for $r = 0.7$ and $\phi = 77\%$. The result of the tapered strut model matches with the experiment data when the thermal conductivity of the strut juncture is 20% of the longitudinal thermal conductivity of the carbon strut.

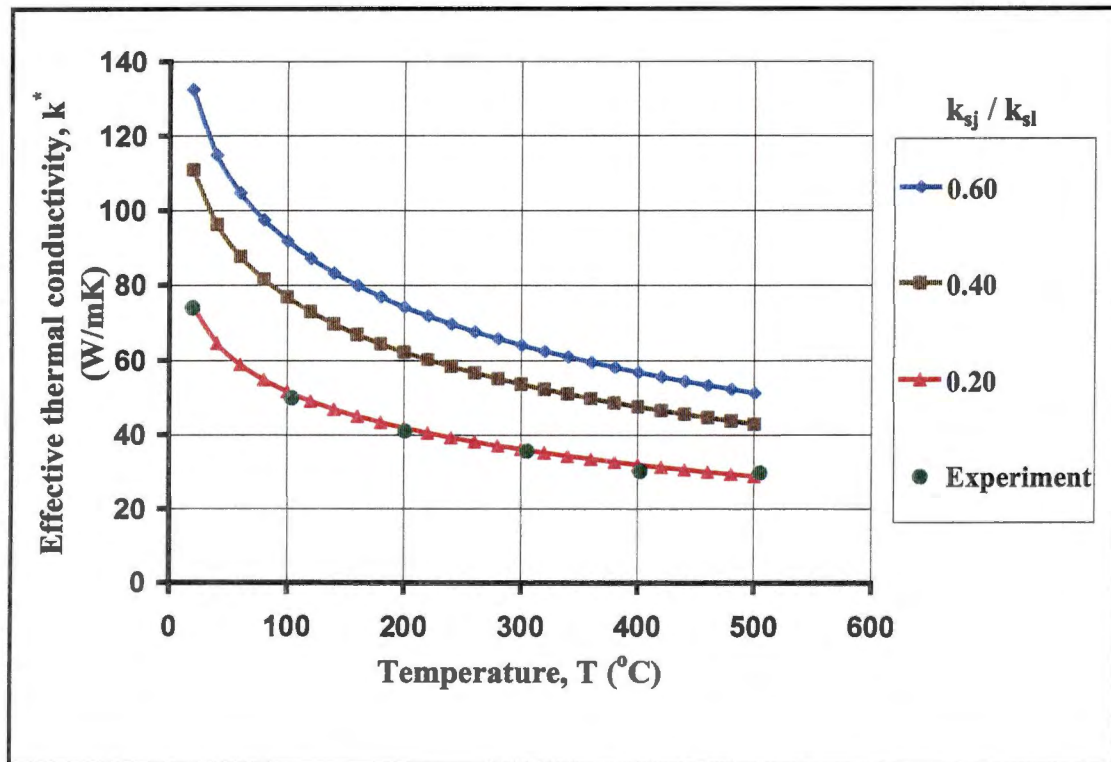


Figure 3.22: Effective thermal conductivity as a function of temperature using tapered strut model with $r = 0.7$, $k_{sj} / k_{sl} = 0.20, 0.40$, and 0.60 , $k_{st} / k_{sl} = 0.75$, and $\phi = 77\%$.

With the recent breakthrough in carbon foam processing, the effective thermal conductivity of the carbon foam, k^* , with $\phi = 77\%$ at room temperature increases to the range of 150 to 180 W/mK (Klett, 1999a). From Table 3.2, the tapered strut model with $r = 0.7$ and $k_{sj} / k_{sl} = 0.6$ falls within the range of the experimental data. The agreement between the experiments and the model may have shown that the juncture thermal conductivity had been improved ($k_{sj} / k_{sl} = 0.6$) with the aid of a better processing method (Klett, 1999a).

3.3.3 Discussion

The carbon strut, in essence, can be described as a thick wall near the strut juncture and a thin ligament near the open pore. Thus, the tapered strut model has a better geometrical representation than the uniform cubic strut model. Furthermore, the effective thermal conductivity of carbon foam consisting of a large number of randomly distributed cubical unit cells, $\overline{K}$, will be equivalent with k^* for both the uniform and tapered cubical strut model.

Table 3 2 Experimental and predicted k^* for carbon foam ($\phi = 77\%$ and $T = 20^\circ\text{C}$).

	Experiment (Klett, 1999a)	Tapered Strut Model
Carbon struts thermal conductivity, k_{sl} (W/mK)	1750	1750
Ratio of k_{sj} / k_{sl}	--	0.6
Taper ratio, r	--	0.7
Effective thermal conductivity, k^* (W/mK)	150 - 180	155

CHAPTER 4

Analytical Modeling on Pressure Drop

4.1 Background

Porous foams receive extensive attention as high-efficiency heat sinks or compact heat exchangers in light of high specific thermal conductivities and heat transfer coefficients. However, the mechanical pumping power necessary to overcome the pressure drop due to cooling fluid friction across the foam is relatively high in comparison with conventional heat exchanger systems at high flow rates. Therefore, the flow fields and the pressure drop across the carbon foam must be understood and analyzed for improvement.

Three simplified geometric models have been developed here to represent the foam microstructure and flow friction characteristics. The three models are the cylindrical strut model, the porous medium model, and the spherical void model. The objective is to determine the relationship among the pressure drop, porosity, foam morphology, and the physical properties of the fluid. Although the models involve approximations to simplify the complexity of forced convection flows across the porous foam, the models capture the approximate functional dependence of microstructural variables, such as porosity and pore size, on pressure drop across the foam.

4.2 Cylindrical Strut Model

4.2.1 Geometrical Representation

A block of foam made up of equal-sized, aligned cubic cells, is considered to be sandwiched between two plates with length L_p and width W_p ; see Figure 4.1. Both plates are assumed to be thin and to have a high thermal conductivity so that the through-thickness heat conduction may be neglected. Uniform temperature T_1 and T_2 are prescribed on the top ($z = 0$) and bottom ($z = b$) plate, respectively. It is assumed that T_1 is much higher than T_2 , and the two ends that are perpendicular to the y -axis at $y = W_p/2$ and $-W_p/2$ are thermally insulated. Cooling fluid with an initial velocity v_o , high pressure p_o , and temperature T_o is forced into the foam at the inlet, $x = 0$ and exits at the outlet, $x = L_p$ with a lower pressure p_e , and a higher temperature T_e . Each cubic unit cell is made up of twelve circular cylinders of a diameter d and a length a ; see Figure 4.2. For simplicity, the cylindrical struts making up the cell edges are assumed to be parallel to the x -, y - or z -coordinate axis and referred as cylinders of type X, Y and Z, respectively. The present analysis considers a single cylinder, a bank of parallel cylinder, and an open-cell foam in cross flow.

The properties of a foam depend strongly on its relative density, ρ , which is the ratio of the density of the foam, ρ^* , to that of the struts, ρ_s , and can be related to the porosity, ϕ , by

$$\rho = \frac{\rho^*}{\rho_s} = 1 - \phi \quad (4.1)$$

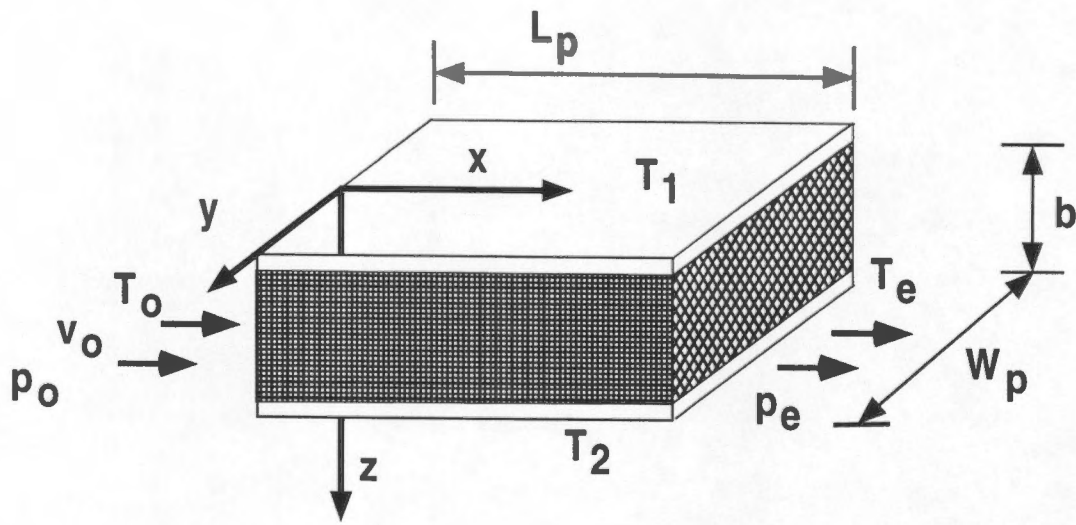


Figure 4.1: Force convection in porous carbon foam.

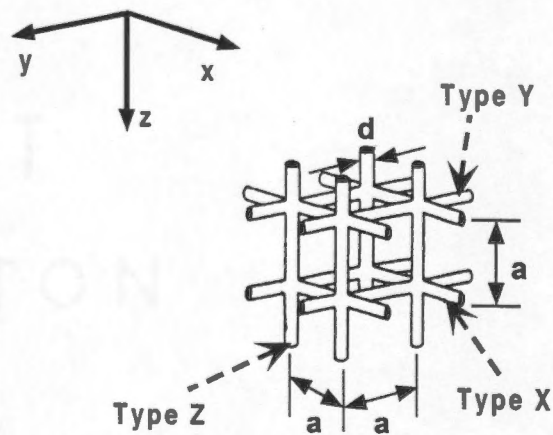


Figure 4.2: Cubic unit cell consisting of struts with a circular cross-sectional area (Lu *et al.*, 1998).

For all of the open-cell foams, the following equation approximately governs the relationship between ρ and d/a (Gibson and Ashby, 1997)

$$\rho = C_1 \left(\frac{d}{a} \right)^2 \quad (4.2)$$

where C_1 is a constant that depends on the microstructure of the cell and can be experimentally determined. It is noted that Eq (4.2) may overestimate the foam density when the relative density, ρ , is high. The overestimation is due to the double counting the corners of the open cells (Gibson and Ashby, 1997). Then a higher-order approximation may be used for example (Gibson and Ashby, 1997),

$$\rho = C_1 \left(\frac{d}{a} \right)^2 \left[1 - C_2 \left(\frac{d}{a} \right) \right] \quad (4.3)$$

where C_2 is a correction factor for a given cell shape. However, the correction is significant only if ρ is more than 0.2. Selected cell shapes and the corresponding constant C_1 are shown in Figure 4.3 and listed in Table 4.1 (Gibson and Ashby, 1997; DeHoff and Rhines, 1968), where a is the base and η is the height of the cell and A_r is the cell aspect ratio, which is defined by η/a . When the cell aspect ratio is equal to 1, the constant C_1 ranges from 1.06 to 4.61 (Gibson and Ashby, 1997). The derivation of C_1 in

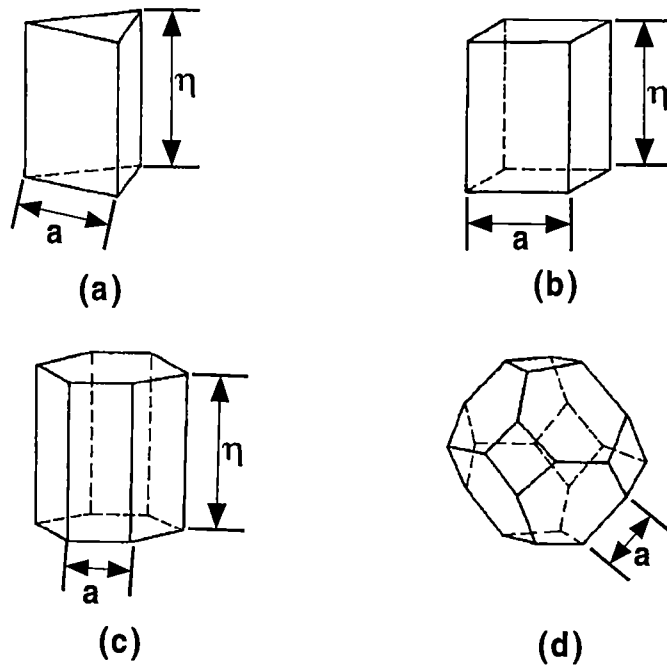


Figure 4 3 Three dimensional cell shapes' (a) triangular prism, (b) rectangular prism, (c) hexagonal prism, (d) tetrakaidecahedron

Table 4.1 The constant C_1 for selected cell shapes (Gibson and Ashby, 1997, DeHoff and Rhines, 1968)

Cell Shape	C_1
Triangular prisms	$\frac{2}{\sqrt{3}} \left\{ 1 + \frac{3}{A_r} \right\}$
Square prisms	$\left\{ 1 + \frac{2}{A_r} \right\}$
Hexagonal prisms	$\frac{4}{3\sqrt{3}} \left\{ 1 + \frac{3}{2A_r} \right\}$
Tetrakaidecahedron	1.06

Table 4.1 assumes that d is much smaller in comparison with a and the error is significant when $\rho > 0.2$.

For an open *cubic* cell consisting of slender cylindrical struts, the normalized diameter of the cylinder, d/a , is approximately related to the relative density by (Lu *et al.*, 1998).

$$\rho \approx \frac{3\pi}{4} \left(\frac{d}{a} \right)^2 \quad (4.4)$$

Besides, the surface area density, α_A , can be defined by an approximate method as (DeHoff and Rhines, 1968)

$$\alpha_A \equiv \frac{A_s}{V} = \frac{4\rho}{d} \quad (4.5)$$

where A_s is the surface area of the foam and V is the volume of the foam. Eq. (4.5) holds for any space filling unit cell having the same relative density and cylindrical strut cross section (DeHoff and Rhines, 1968). Furthermore, Eq. (4.5) has been verified by measuring the surface area density of aluminum foam using computed-tomography (Bastawros *et al.*, 1999). Then, the resulting total surface area per unit width (A_s/W_p) of a segment of foam of height b and length L_p , is approximately given by (Lu *et al.*, 1998).

$$\frac{A_s}{W_p} \approx \rho^{1/2} \left(\frac{2\sqrt{3\pi} L_p b}{a} \right) \quad (4.6)$$

Substituting Eq (4.6) into Eq (4.5) yields

$$\alpha_A \approx \rho^{1/2} \left(\frac{2\sqrt{3\pi}}{a} \right) \quad (4.7)$$

The surface area density is an important parameter for heat exchange systems. A large amount of surface area is a typical characteristic of gas-flow heat exchangers

4.2.2 A Single Cylinder in a Cross Flow

For a circular cylinder of type Y or Z in cross flow, the characteristics of the flow depend very strongly on various parameters such as the size, and orientation of the cylinder, and fluid properties (Munson *et al.*, 1994). As shown in Figure 4.4, the free stream fluid is brought to rest at the forward stagnation point, with an accompanying rise in pressure. The Bernoulli equation written along a streamline is given by (see, for example, Holman, 1986):

$$\frac{dp}{\rho_f} = -\frac{(dv)^2}{2} \quad (4.8)$$

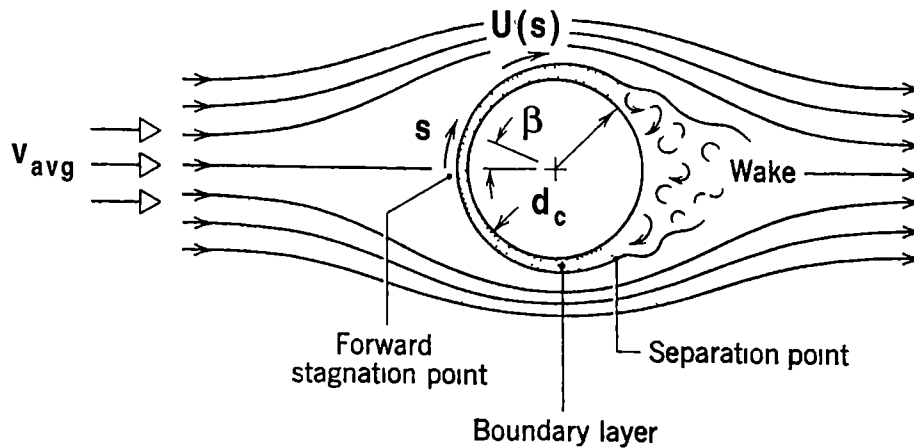


Figure 4.4 Boundary layer formation and separation on a circular cylinder in a cross flow (Incropera and DeWitt, 1990)

where dp is the differential pressure, dv is the differential velocity, and ρ_f is the fluid density

From the forward stagnation point on, the pressure decreases with increasing s , the streamline coordinate. A boundary layer will be developed under the influence of a favorable pressure gradient ($dp/ds < 0$). The basic assumption of the boundary layer theory is that the boundary layer is thin enough so that it does not greatly disturb the flow outside of the boundary layer. However, the pressure eventually reaches a minimum, and further boundary layer development occurs toward the rear of the cylinder in the presence of an adverse pressure gradient ($dp/ds > 0$) and the velocity of the fluid decreases. The fluid particle in the boundary layer experiences a loss of energy due to the viscous effect. The loss of energy means that the fluid near the surface lacks sufficient kinetic energy to overcome the pressure gradient. The fluid within the

boundary layer will stop flowing downstream and a reverse flow takes place at this location, the separation point, where the boundary layer separation starts to occur, see Figure 4.4. A wake is formed in the downstream region (see, for example, Incropera and DeWitt, 1990)

The location of separation, the width of the wake region behind the cylinder, and the pressure distribution on the flow surface depend on the nature of the boundary layer flow. In comparison with a laminar boundary layer, a turbulent boundary layer has more kinetic energy and momentum because its velocity profile is more like an ideal uniform profile and there may be considerable energy associated with the swirling of the velocity (Munson *et al* , 1994). Thus, the separation point for turbulent flow will be further downstream in comparison with the laminar flow.

4 2 3 A Bank of Type Z Cylinders in Cross Flow

The pressure drop of fluid in a cross flow over a bank of Z-type cylinders is considered in this section. The pressure drop depends largely on the flow pattern and the degree of turbulence, which in turn are functions of the fluid properties and the size and arrangement of the cylinders (Kreith and Bohn, 1993) Consider a bank of cylinders being aligned in the direction perpendicular to the free stream with an average velocity v_{avg} ; see Figure 4.5 The configuration of the bank of cylinders is characterized by the cylinder diameter, d , and the spacing between the cylinder, a . The flow field is dominated by boundary layer separation effects and by wake interaction, which in turn influence the convective heat transfer (Kreith and Bohn, 1993)

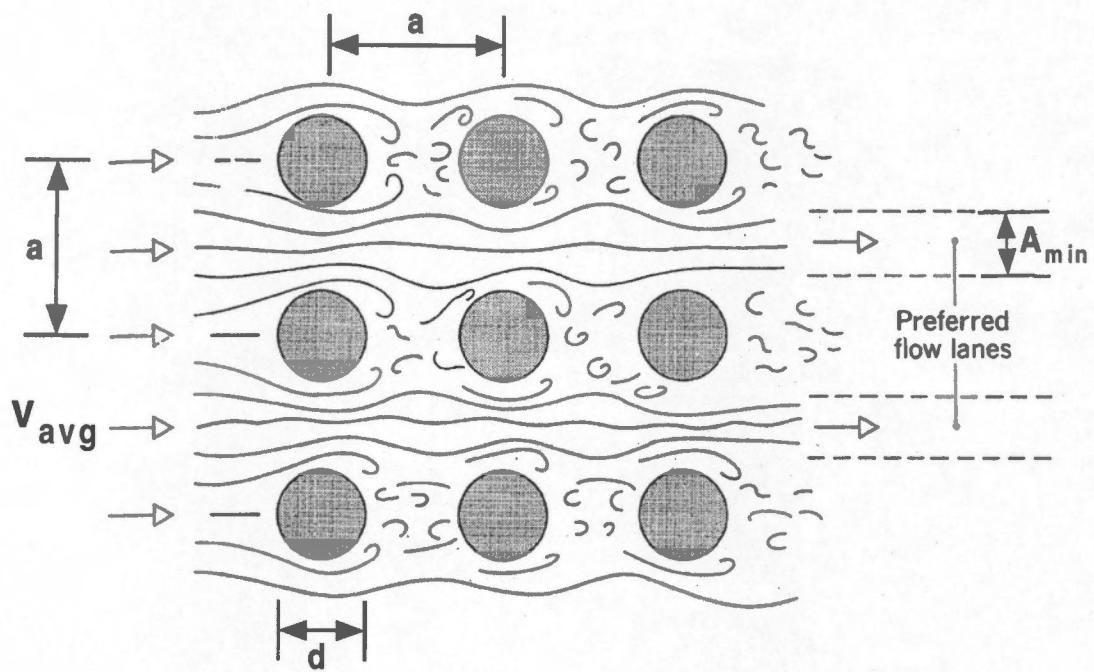


Figure 4.5: A bank of aligned circular cylinders in cross flow (Incropera and DeWitt, 1990).

The flow field around the first row of a bank cylinders is similar to that for a single, isolated cylinder in a cross flow. However, for the subsequent rows, the flow field depends strongly on the arrangement of cylinders (Incropera and DeWitt, 1990). The aligned cylinders beyond the first row of cylinders are in the zone of turbulent wakes as shown in Figure 4.5

The pressure drop for gas flows over a bank of cylinders may be expressed by (Jacob, 1938):

$$\Delta p = 2 N f^* \rho_f v_{\max}^2 \left(\frac{\mu_w}{\mu} \right)^{0.14} \quad (4.9)$$

where $N = L_p/a$ is the number of cylinder rows in the direction of flow, ρ_f is the fluid density, μ_w is the fluid viscosity near the wall, μ is the average fluid viscosity, $v_{\max}$ is the maximum fluid velocity and is assumed to occur in the minimum free flow area, $A_{\min}$, in the fluid flow. The mass conservation for an incompressible fluid flow requires (Incropera and DeWitt, 1990)

$$v_{\max} = \frac{a}{a-d} v_{\text{avg}} \quad (4.10)$$

where v_{avg} is the average free stream velocity. The empirical friction factor, f_{tube} , may be given by (Jacob, 1938)

$$f_{\text{tube}} = \text{Re}_{\text{max}}^{-0.15} \left\{ 0.044 + \frac{0.08 \left(\frac{a}{d} \right)}{\left(\frac{a}{d} - 1 \right)^{0.43 + 1.13(d/a)}} \right\} \quad (4.11)$$

where Re_{max} is the maximum Reynolds number, which is a dimensionless parameter measuring the ratio of inertial forces to viscous forces in the flow and can be expressed by

$$\text{Re}_{\text{max}} = \frac{v_{\text{max}} d}{\nu_f} \quad (4.12)$$

where ν_f is the fluid kinematic viscosity

4.2.4 Foam in Cross Flow

In order to consider the total pressure drop across the foam, the pressure drop due to each bank of X-, Y-, and Z-type cylinders is superimposed. For gas flowing across a foam, the pressure drop produced by cylinders of type Y and Z may be approximated using Eq (4.9) with N being replaced by $N_L = 2L_p/a$ (Lu *et al.*, 1998). The pressure drop is assumed to be uniform in the transverse direction since there are a large number of cylinders in the direction perpendicular to the flow.

Furthermore, the contribution from X-type cylinders and the plates to the pressure drop is complicated but the overall response of such an open cell foam may be characterized by a frictional coefficient, χ , which is to be calibrated from experiments (Lu *et al.*, 1998). An empirical friction factor, f_{foam} , for the foam in a cross flow may be expressed by

$$f_{\text{foam}} = f_{\text{tube}} \chi \approx \text{Re}_{\text{max}}^{-\gamma} \zeta \quad (4.13)$$

where f_{tube} is given in Eq. (4.11), γ is the exponent of the power law and ζ is dependent on the geometrical arrangement of the cylinder bank; typically $5 \leq \zeta \leq 10$ (Žukauskas, 1987).

Since the viscosity of gas does not vary much with temperature, the assumption of $\mu_w/\mu \approx 1$ is considered. Equation (4.9) with N and f_{tube} being replaced by N_L and f_{foam} respectively, becomes

$$\Delta p = \frac{4L_p}{a} f_{\text{foam}} \rho_f v_{\text{max}}^2 \quad (4.14)$$

Since the fluid flow past a foam-filled duct also experiences a boundary frictional drag at the plate wall, excess pressure loss at the entrance region, the bulk frictional drag induced by the foam skeleton, Eq. (4.14), is not complete in terms of total pressure drop. However, according to Kaviani (1985), the total pressure drop experienced by the flow

across the foam-filled duct is dominated by the bulk frictional drag. Hence, the total pressure may be approximated by Eq (4.14) solely. In addition, by substituting Eq. (4.10), (4.12), and (4.13) into Eq (4.14), the total pressure drop per unit length of the foam can be rewritten by

$$\frac{\Delta p}{L_p} = \frac{4\rho_f}{a} v_{avg}^{(2-\gamma)} \left(\frac{a}{v_f} \right)^{-\gamma} \frac{\left(\frac{a}{d} - 1 \right)^\gamma}{\left(1 - \frac{d}{a} \right)^2} \zeta \quad (4.15)$$

Since there is no available data for the empirical friction factor χ for foam materials, the assumption of $\chi \approx 1$ is made. Thus, the friction factor for the foam material and in-line cylinder bank will be identical i.e., $f_{foam} = f_{tube}$, which imply $\gamma = 0.15$ and ζ can be expressed by

$$\zeta \approx 0.044 + \frac{0.08 \left(\frac{a}{d} \right)}{\left(\frac{a}{d} - 1 \right)^{0.43 + 1.13(d/a)}} \quad (4.16)$$

Consequently, the expression for the total pressure drop per unit length, Eq (4.15), can be written by

$$\frac{\Delta p}{L_p} = \frac{4\rho_f v_{avg}^{1.85}}{a} \left(\frac{a}{v_f} \right)^{-0.15} \frac{\left(\frac{a}{d} - 1 \right)^{0.15}}{\left(1 - \frac{d}{a} \right)^2} \left\{ 0.044 + \frac{0.08 \left(\frac{a}{d} \right)}{\left(\frac{a}{d} - 1 \right)^{0.43+1.13(d/a)}} \right\} \quad (4.17)$$

It is noted that the relationship between (d/a) and relative density, ρ , is given in Table 4.1.

4.2.5 Results

In order to have a better understanding of the foam morphology, the correlation of the cylinder diameter, d , and the relative foam density, ρ , for all four cell shapes in Figure 4.3 are analyzed using Eq. (4.2) and are shown in Figure 4.6, where the cell size of the carbon foam, a , is assumed to be $300 \mu\text{m}$. The hexagonal cell model predicts the largest diameter, d , for a given relative density, ρ . On the other hand, the tetrakaidecahedron cell model predicts the lowest d at the same ρ . In order to evaluate the validity of the four different geometrical models, measurements of the cell morphology using optical microscopy is suggested.

Once an accurate model for cell morphology has been found, the relationship between the pressure drop and the porosity can be obtained from Eq. (4.1), (4.2), and (4.17). In Figure 4.7, the pressure drop per unit length, $\Delta p/L_p$, is expressed as a function of average free stream velocity, v_{avg} , for different cell morphology models for air ($v_f = 1.46 \times 10^{-5} \text{ m}^2/\text{s}$ and $\rho_f = 1.23 \text{ kg/m}^3$) flowing through a duct filled with carbon foam of a

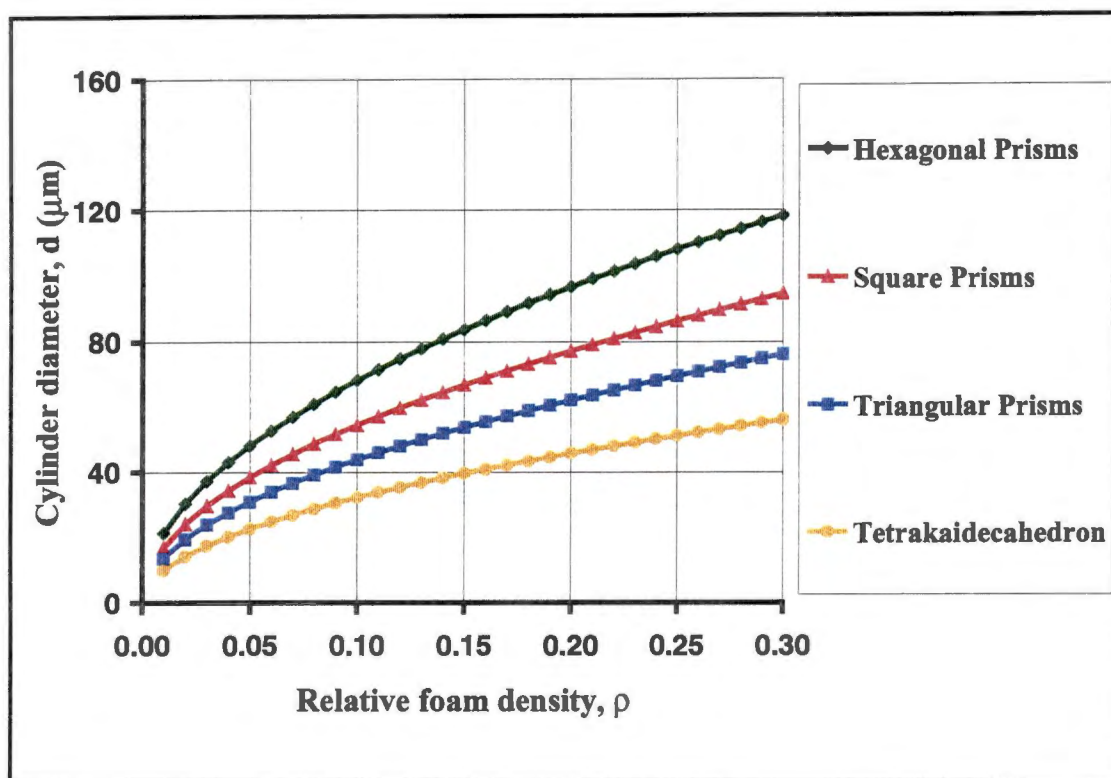


Figure 4.6: Cylinder diameter, d , as a function of relative foam density for different geometrical cell models.

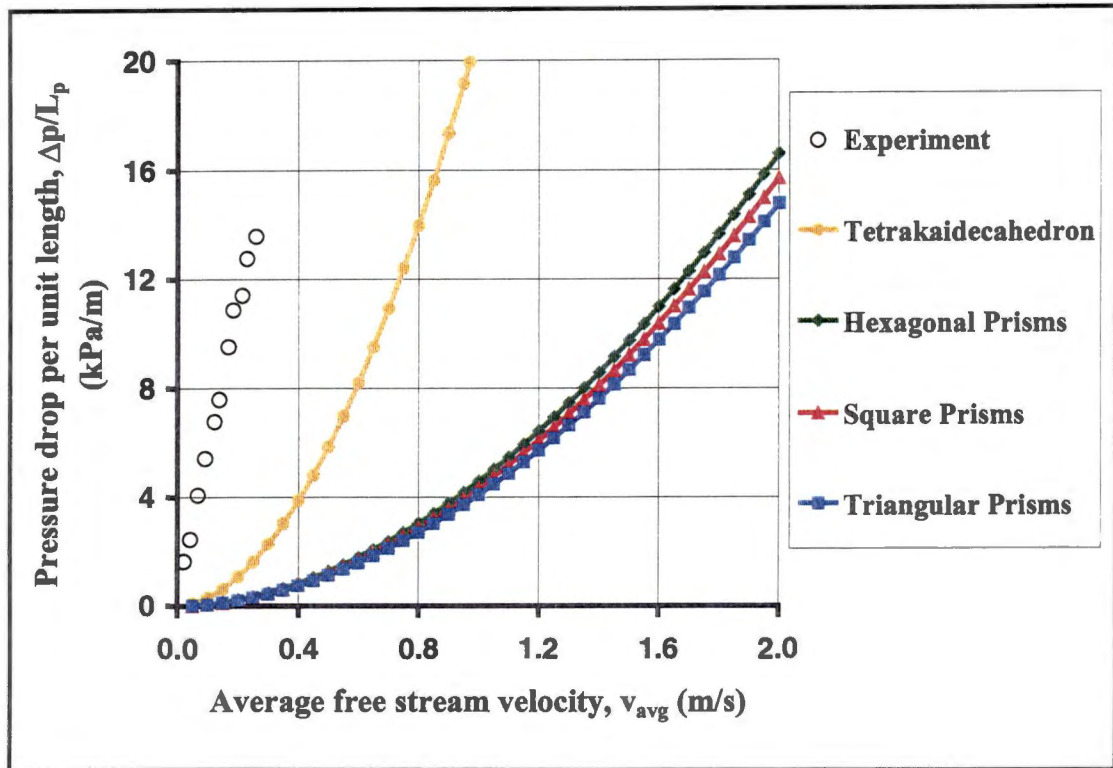


Figure 4.7: Pressure drop per unit length, $\Delta p/L_p$, as a function of fluid velocity, v_{avg} , for air (20°C) flowing across a duct filled with carbon foam ($a = 300\mu\text{m}$; $\phi = 80\%$) using various cell models.

= 300 μm and porosity = 80% at 20 °C. Experimental pressure drop data for Mitsubishi ARA 24 mesophase pitch-derived carbon foam with approximate cell size of 300 μm and porosity of 80% is also listed in Figure 4.7 as a comparison (Klett, 1999a). All the different cell morphology models seem to underestimate the pressure drop of the porous carbon foam.

4.2.6 Discussion

The present model may oversimplify the fluid flow field in carbon foams consisting of non-circular, sharp-edged ligaments, and lead to underestimates of the total pressure drop as the boundary friction drag at the plate wall and the pressure drops at the entrance and exit locations are not considered in the modeling of the fluid flow across a duct filled with foam. Furthermore, the formulation between the relative density, ρ , and (d/a) , Eq (4.2) and Table 4.1, is valid for porosity 80% or above (Gibson and Ashby, 1997). However, the porosity for the foam presented herein is 80%, which is just within the limitation and might introduce some errors to the model. In addition, based on experimental work by Achenbach (1991) on in-line banks of cylinder, the pressure drop through a bank of cylinders with rough surfaces was about 30% less than that for a bank of cylinders with smooth surfaces.

Furthermore, the cylindrical strut model together with a simple cubic unit cell is in reasonable agreement with available experimental data on aluminum foams since the aluminum foam does resemble the assembly of thin ligaments with very high porosity (Lu *et al* , 1998). Since the cylindrical strut model with tetrakaidecahedron unit cell

geometry can be traced to the foam morphology, Bastawros *et al* (1999) concludes that the analogy between the bank of cylinder and foam provides a reasonably accurate representation of the pressure drop in the aluminum foam. Since the cylindrical strut model is not able to accurately predict the pressure drop across the carbon foam, two different models are considered in Section 4.2 and 4.3.

4.3 Porous Medium Model

4.3.1 Approach

The present model is based on the Carman-Kozeny theory of non-Darcian's flow in porous media. It is well known that the pressure drop across a porous medium caused by the frictional drag is linearly proportional to velocity for low-velocity flows, the Darcy's law relating the pressure drop and the fluid velocity in an unbounded porous medium. Most of the existing studies on forced convection in porous media are based on Darcy's law that is only valid for creeping flows through an infinitely extended medium with Reynolds numbers, $Re < 10$ (Bejan and Morega, 1993). In other words, Darcy's flow holds only if inertia effects of the flow are negligible (Philipse and Schram, 1991). In the carbon foam structure, flow paths are tortuous, and the magnitudes and directions of fluid particle velocities change continuously. So, there are inherent inertia effects and deviations from Darcy's law, which is noticeable at lower flow rates in contrast to capillary structures with a similar pore size (Philipse and Schram, 1991). However, the deviations do not imply that the flow in carbon foam must be turbulent. Even when inertia effects are appreciable, flow through a pore structure may still be of a stable

laminar type (Scheidegger, 1974) Furthermore, turbulence at the microscopic (pore) level may not result in turbulence at the macroscopic level (Lage *et al* , 1997). For instance, Dybbs and Edwards (1984), reported that pore level turbulence develops when the fluid velocity exceeds a critical value but there is not any turbulence at the macroscopic level

Since the flow through foams does not satisfy the Darcy's law, the momentum equation is adjusted to include a quadratic velocity term, which addresses the flow regimes that are different from Darcian's flow (Neild and Bejan, 1998) The additional quadratic velocity term, or so-called Forchheimer term, is caused by the inertia effects and can be determined experimentally or numerically (Antohe *et al.*, 1996 and Rahli *et al.*, 1997) For steady, incompressible flow, the modified Darcy's law with Forchheimer term is given by

$$\frac{\Delta p}{L_p} = \frac{\mu_f}{\kappa} v_{avg} + \frac{\rho_f c_F}{\sqrt{\kappa}} v_{avg}^2 \quad (4.18)$$

where κ is the permeability of the foam, μ_f is the fluid viscosity, ρ_f is the fluid density, and c_F is a dimensionless drag coefficient. Eq (4 18) is the modification of an equation associated with Dupuit (1863) and Forchheimer (1901) (Lage, 1998). The second term on the right-hand side of Eq (4 18) is refer as Forchheimer term, and is proposed by Ward (1964) It is found that c_F varies with the nature of the porous medium, and for the low-density metallic foams, c_F is approximately equal to 0.1 (Neild and Bejan, 1998).

According to Nield and Bejan (1998), the transition from Darcy regime to the Forchheimer regime occurs in the Reynolds number ranging from 1 to 10. For a limited range, c_F varies linearly with velocity, which means that the pressure drop is proportional to the cubic of velocity (Nield and Bejan, 1998) and experimental studies reported by Lage *et al* (1997) on porous aluminum medium layer confirm it. However, the flow through carbon foam is well within the Forchheimer regime at high flow rates

The permeability, κ , in Eq. (4.17) is generally regarded as a function of the porosity, ϕ , through the Carman-Kozeny relationship: (Kaviany, 1995)

$$\kappa = \frac{\phi^3}{k_K (1 - \phi)^2 A_o^2} \quad (4.19)$$

where A_o is the volumetric (specific) surface area of the foam, and k_K is the Carman-Kozeny constant, which is a function of both the porosity and the aspect ratio of the fibrous microstructure (Rahli *et al.*, 1997). Then, a mean pore diameter, D , is introduced as the diameter of a hypothetical sphere with the same specific surface area, A_o , and D can be related to the diameter of cylinder, d_c , by

$$D = \frac{6}{A_o} = \frac{3}{2} d_c \quad (4.20)$$

By substituting Eq. (4.20) into Eq (4.19), the permeability can be expressed by

$$\kappa = \frac{D^2 \phi^3}{36 k_K (1 - \phi)^2} = \frac{d_c^2 \phi^3}{16 k_K (1 - \phi)^2} \quad (4.21)$$

For *fibrous beds*, the Carman-Kozeny constant, k_K , is found to be strongly dependent on ϕ as (Kyan *et al*, 1970)

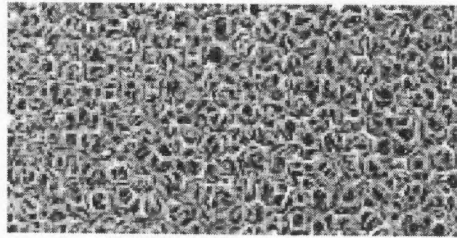
$$k_K = \frac{\phi^6 [62.3 N_e^2 (1 - \phi) + 107.4]}{16 \phi^6 (1 - \phi)^4} \quad (4.22)$$

where N_e and can be expressed by

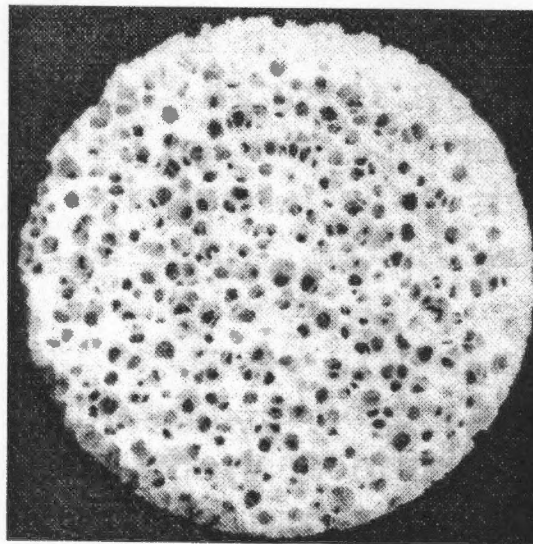
$$N_e = \left(\frac{2\pi}{1 - \phi} \right)^{0.5} - 2.5 \quad (4.23)$$

On the other hand, Bastawros *et al* (1999) experimentally determine that k_K is approximately equal to 10 for all *aluminum foam grades*, having the same aspect ratio at the same density, see Figure 4.8(a). Nevertheless, the nonlinear relationship between pressure drop and average free stream velocity in *ceramic foams* may be expressed as (Scheidegger, 1974)

$$\frac{\Delta p}{L_p} = \frac{\mu_f}{k_1} v_{avg} + \frac{\rho_f}{k_2} v_{avg}^2 \quad (4.24)$$



(a)



(b)

Figure 4.8: Physical 20 ppi (pore per inch) foam materials: (a) Duocel[®] aluminum foam[#], and (b) ceramic foam (Philipse and Schram, 1991).

[#] ERG Product Literature (2000)

where k_1 is the Darcian's permeability, k_2 is the non-Darcian's permeability. Similarly, the first term is the contribution of the viscous effects and the second term is the contribution of inertia effects. For ceramic foam with 82% porosity (see Figure 4.8b), $k_1 \approx 3.2 \times 10^{-9} \text{ m}^2$ and $k_2 \approx 2.1 \times 10^{-5} \text{ m}$ are determined experimentally by Philipse and Schram (1991).

4.3.2 Results and Discussions

Due to lack of experimental data on carbon foams, the parameters for the fibrous medium, ceramic foams or aluminum foams from the literature are used to obtain the pressure drop across carbon foam as a function of average free stream velocity for air flow with a mean pore diameter of $300 \mu\text{m}$ at 20°C ($v_f = 1.46 \times 10^{-5} \text{ m}^2/\text{s}$ and $\rho_f = 1.23 \text{ kg/m}^3$), see Figure 4.9. For fibrous medium, Eq. (4.22) is used for k_K and $c_F \approx 0.1$ in Eq. (4.18). For aluminum foam, $k_K \approx 10$ and $c_F \approx 0.1$. Finally, $k_1 \approx 3.2 \times 10^{-9} \text{ m}^2$, and $k_2 \approx 2.1 \times 10^{-5} \text{ m}$ are used in Eq. (4.25) for ceramic foams. In addition, the experimental pressure drop data from the Mitsubishi ARA 24 mesophase pitch derived carbon foam is also shown in Figure 4.9 (Klett, 1999a).

As it is seen in Figure 4.9, all three porous medium models under predict the pressure drop across the carbon foam. The porous medium model using the parameters of ceramic foam has a better prediction of the experimental data than the other two. Although the porous medium model provides a better prediction of the internal flow field than the cylindrical strut model, it is still highly arbitrary or uncertain since most of the parameters used are not related to carbon foam. Thus, experimental determination of the

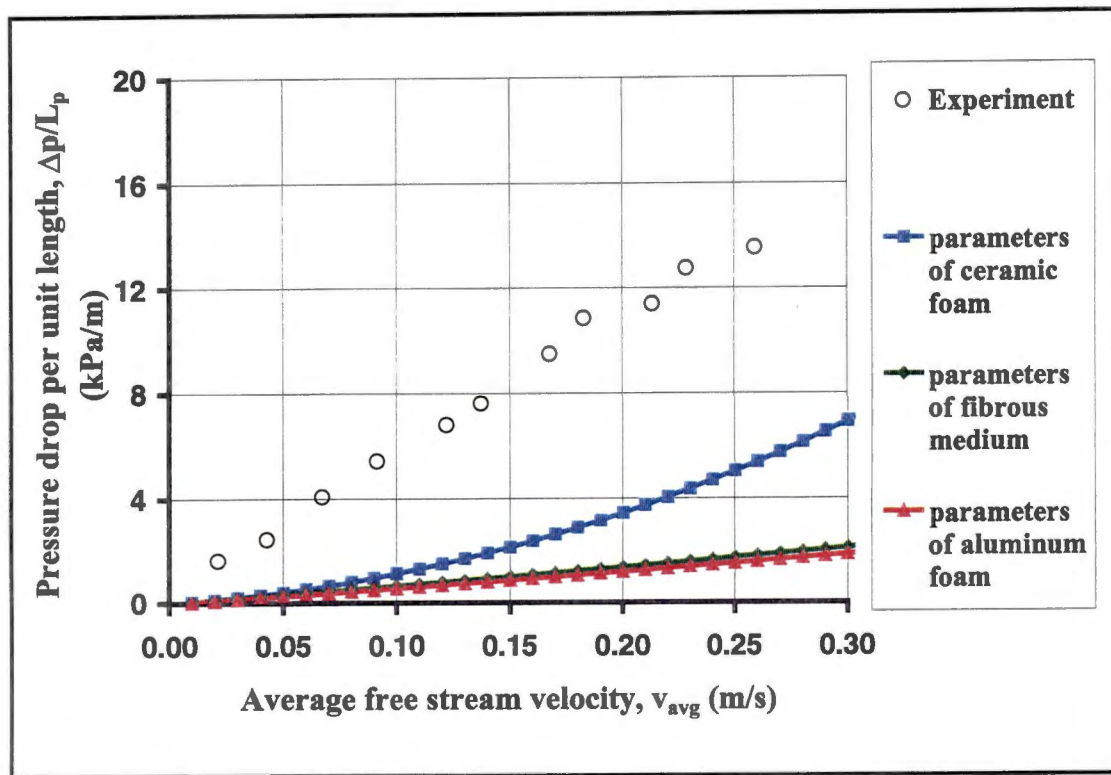


Figure 4.9: Pressure drop per unit length, $\Delta p/L_p$, as a function of cell shape and fluid velocity, v_{avg} , for air (20°C) flowing across a duct filled with carbon foam ($D = 300\mu\text{m}$; $\phi = 80\%$).

necessary parameters is necessary for using the porous medium model. It should be noted that although Eq. (4 18) of the porous medium model encompasses different flow regimes, nevertheless the drag coefficient, c_F , must be calibrated for each foam morphology (Bastawros *et al* , 1999).

4.4 Spherical Void Model

4 4 1 The Unit Cell

The present model assumes that a block of carbon foam is composed of three-dimensional arrays of tetrakaidecahedron cells. The tetrakaidecahedron cells are packed to fill the entire space; see Figure 4 10.

Figure 4.11 is the geometrical representation of the tetrakaidecahedron unit cell where L is the edge length of each side of the tetrakaidecahedron cell, a_{sq} is the distance between two mutually parallel *square* faces of the cell, a_{hex} is the distance between two mutually parallel *hexagonal* faces of the cell, D_{sq} is the diameter of the circle on the square face, and D_{hex} is the diameter of the circle on the hexagonal face. Each cell has 36 edges and 14 faces with 6 square faces and 8 hexagonal faces. Each cell contains a body-centered spherical void located at the center of the cell. Since the cells are open, the voids in general are interconnected with the voids in the neighboring cells. The dihedral angle between two hexagon faces is equal to $109^{\circ}28'$, and the dihedral angle between the square and hexagon face is equal to $125^{\circ}16'$ (Pearce and Pearce, 1978).

A two-dimensional projection of the mid-plane of the tetrakaidecahedron cell is shown in Figure 4.12, where D is the diameter of the spherical void. If the pore

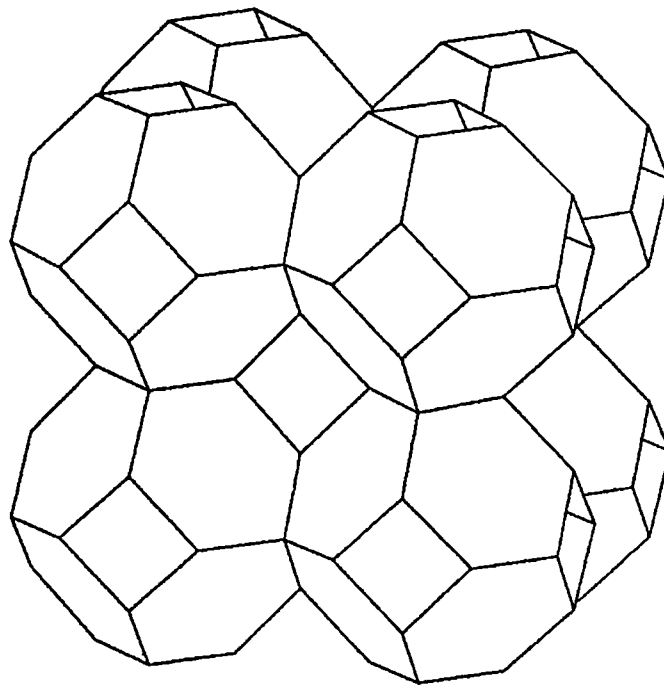


Figure 4.10: Stacked tetrakaidecahedron cells.

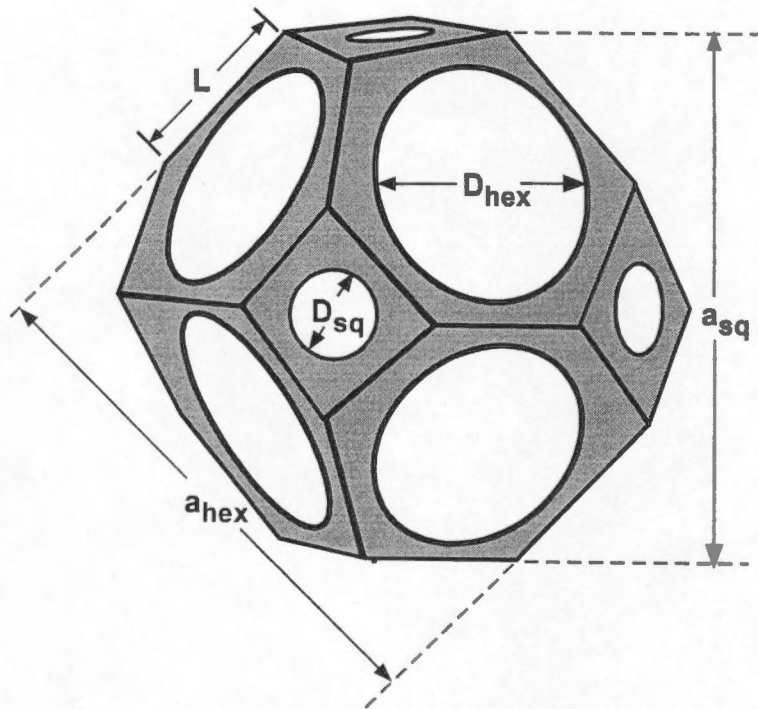


Figure 4.11: Tetrakaidecahedron unit cell.

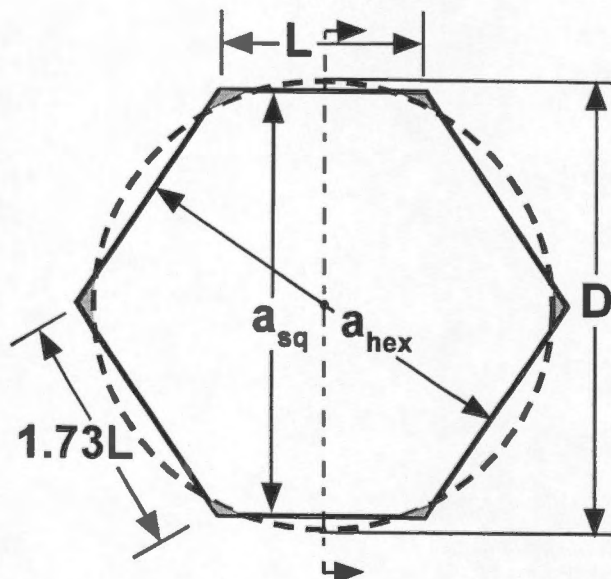


Figure 4.12: Two-dimensional projection of the mid-plane of the unit cell.

diameter, D is larger than $a_{\text{hex}} (< a_{\text{sq}})$, the voids are interconnected on the hexagonal faces. Depending on the porosity of the carbon foam, ϕ , and the pore diameter, D , the solid which remains in the cell is either in the form of rods at the cell edges or perforated plates at the faces and cell edges

As it can be seen in Figure 4 12, the distance between any two mutually parallel sides of the hexagonal face is $1.73L$. The distance between two square faces, a_{sq} , can be expressed in term of edge length, L , as

$$a_{\text{sq}} = 2.83L \quad (4.25)$$

Whereas the distance between two mutually parallel hexagonal faces, a_{hex} , is given by

$$a_{\text{hex}} = 2.45L \quad (4.26)$$

The diameters of the circles on the square and the hexagonal face can be formulated as the function of D as well as a_{sq} and a_{hex} , respectively, by using the Pythagorean theorem as

$$\left(\frac{D_{\text{sq}}}{2}\right)^2 = \left(\frac{D}{2}\right)^2 - \left(\frac{a_{\text{sq}}}{2}\right)^2 \quad (4.27a)$$

$$\left(\frac{D_{\text{hex}}}{2}\right)^2 = \left(\frac{D}{2}\right)^2 - \left(\frac{a_{\text{hex}}}{2}\right)^2 \quad (4.27b)$$

By substituting Eq (4.25) in Eq. (4.27a) and Eq (4.26) in Eq (4.27b), D_{hex} and D_{sq} can be expressed in terms of L and D by

$$D_{\text{sq}} = 2 \left[\sqrt{\left(\frac{D}{2}\right)^2 - \left(\frac{2.83L}{2}\right)^2} \right] \quad (4.28a)$$

$$D_{\text{hex}} = 2 \left[\sqrt{\left(\frac{D}{2}\right)^2 - \left(\frac{2.45L}{2}\right)^2} \right] \quad (4.28b)$$

The void volume in the cell is equal to the volume of the sphere subtract the six spherical end caps on the square faces, V_{cs} , and eight spherical end cap on the hexagonal faces, V_{ch} , that is

$$V_{\text{void}} = \frac{\pi D^3}{6} - 6V_{\text{cs}} - 8V_{\text{ch}} \quad (4.29)$$

where

$$V_{cs} = \frac{\pi}{12} (D - 2.83L)^2 \left(D + \frac{2.83L}{2} \right) \quad (4.30a)$$

$$V_{ch} = \frac{\pi}{12} (D - 2.45L)^2 \left(D + \frac{2.45L}{2} \right) \quad (4.30b)$$

The volume of tetrakaidecaheron unit cell is given by (Gibson and Ashby, 1997):

$$V_{cell} = 11.31L^3 \quad (4.31)$$

Thus, the foam porosity, ϕ , can be expressed by

$$\phi = \frac{\left[\frac{\pi D^3}{6} - 6 \left\{ \frac{\pi}{12} (D - 2.83L)^2 \left(D + \frac{2.83L}{2} \right) \right\} - 8 \left\{ \frac{\pi}{12} (D - 2.45L)^2 \left(D + \frac{2.45L}{2} \right) \right\} \right]}{11.31L^3} \quad (4.32)$$

The surface area of the cell wall may be approximated as the surface area of the spherical void. The resulting errors of overestimating the surface area of the cell may be justified by the large amount of graphite multi-layers on the surface of the carbon foam (Klett and Burchell, 1998). Therefore, the total surface area of the tetrakaidecahedron cell is estimated by

$$A_s = \pi D^2 \quad (4.33)$$

The surface area density of the foam, α_A , defined as the surface area per unit volume of the unit cell is thus given by

$$\alpha_A = \frac{\pi D^2}{11.31 L^3} \quad (4.34)$$

It is noted that Eq. (4.33) represents an estimate of the average surface area density of the carbon foam using the mean pore and mean size of the cell.

4.4.2 Fluid Velocity Profile

The fluid motion is usually a rather complex phenomenon and despite the resources of mathematical analysis we have relies heavily on observation and measurement of actual flow patterns. The principle of dynamical similarity and the use of dimensionless parameters provide us with a very useful technique for translating measured results from one flow configuration to another.

One important dimensionless parameter is the Reynolds number, which is defined in Eq (4.12). The physical significance of the Reynolds number is that it provides a measure of the relative magnitude of the inertial forces, associated with the acceleration or deceleration of a fluid element, to the viscous forces occurring within the flow. A more appropriate Reynolds number for foams is define by

$$Re_{\max} = \frac{v_{\max} L_c}{\nu_f} \quad (4.35)$$

where $v_{\max}$ is the maximum fluid velocity occur within the foam, and ν_f is the fluid kinematics viscosity. The difference between Eq (4.35) and Eq (4.12) is the replacement of the cylinder diameter, d , by a general characteristic length of the body, L_c . In general, the larger the Reynolds number, the smaller the region of the flow field in which viscous effects are important. A commonly used characteristic length for porous media is the hydraulic diameter, D_h , which is defined as four times the flow area per wetted perimeter, for flows through a non-circular cross-sectional pipe (Rohsenow *et al* , 1985). Therefore, the Reynolds number is expressed by

$$Re_{\max} = \frac{v_{\max} D_h}{\nu_f} \quad (4.36)$$

According to Carman-Kozeny theory (Dullien, 1992), which is based on a semiheuristic model of flow through solid matrices, the hydraulic diameter, D_h is defined as four times the void volume per surface area:

$$D_h = \frac{4V_{\text{void}}}{A_s} \quad (4.37)$$

where V_{void} is the void volume and A_s is the surface area of the unit cell. By dividing both the numerator and denominator by unit cell volume, V_{cell} , Eq (4.37) can be rewritten by

$$D_h = \frac{4\phi}{\alpha_A} \quad (4.38)$$

where α_A is the surface area density, and ϕ is the porosity. For porous crossed-rod and woven-screen matrices, Coppage and London (1956) show that the hydraulic diameter can be expressed by

$$D_h = \frac{4L_p A_c}{A_{\text{total}}} \quad (4.39)$$

where A_{total} is the total surface area of the foam, A_c is the minimum free flow cross-sectional area, and L_p is the total length of the core structure in the flow direction.

In order to establish the velocity profile for the fluid passing through the carbon foam in terms of the dimensionless ratio, Re_{max} , the maximum velocity within the foam, v_{max} , which is a function of free stream velocity and foam geometry, is used as the reference velocity. By using the conservation of mass, v_{max} of the spherical void model can be formulated based on the minimum flow area occurring at the hexagonal face as

$$v_{\max} = \frac{A_{\text{fr}}}{A_c} v_{\text{avg}} = \frac{2.598L^2}{\frac{\pi}{4} D_{\text{hex}}^2} v_{\text{avg}} \quad (4.40)$$

where v_{avg} is the average free stream velocity, $A_{\text{fr}} (= 2.598L^2)$ is the (frontal) area of the hexagonal surface, and A_c is the minimum flow area occurring at the hexagonal face of the unit cell as shown in Figure 4.13.

4.4.3 Fluid Friction

The foam immersed in a fluid experiences forces due to the interaction between the foam and the fluid surrounding it. The forces at the fluid and foam interface can be described in terms of the wall shear stresses, τ_w , due to viscous effects and the normal stresses (pressure), p_w . The resultant force in the direction of the upstream velocity is termed the drag whereas the resultant force that is normal to the upstream velocity is termed the lift

Friction factor is in general defined as the equivalent shear force in the flow direction per unit of heat transfer or friction area (Kay and London, 1984). For foams, it is, however, considered to be a combination of viscous shear force (skin friction) and normal force (form drag) since there is no reason in design work to attempt to separate these effects. Besides, such a general definition of friction factor can be used as a common treatment to all surfaces. Thus, the friction factor, f , can be defined by the ratio of resultant stress to kinetic energy of the flow as

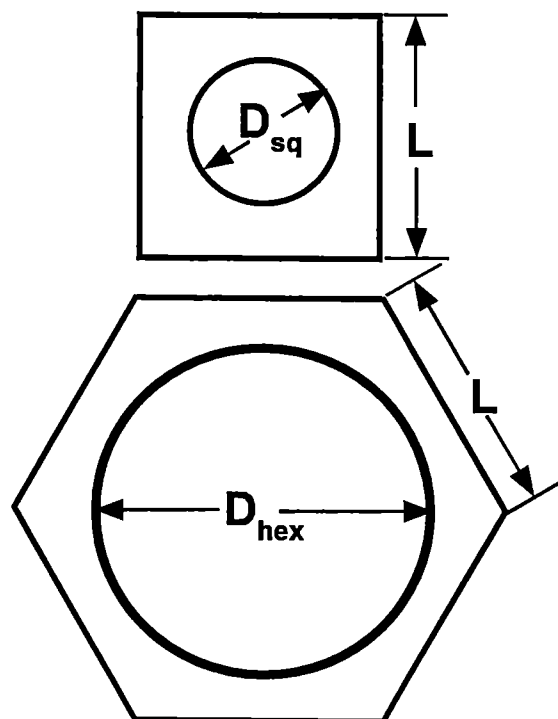


Figure 4 13 Square and hexagonal faces of the tetrakaidecaheron unit cell.

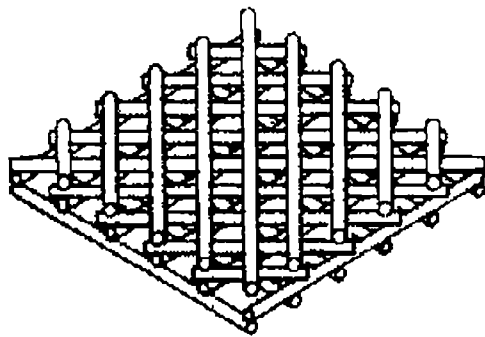
$$f = \frac{(\tau_w + p_w)}{0.5 \rho_f v_{avg}^2} \quad (4.41)$$

where ρ_f is the density of the fluid. In addition, for the flow through cylindrical tubes, f in Eq. (4.41) is the same as the conventional Fanning friction factor (Munson *et al.*, 1994).

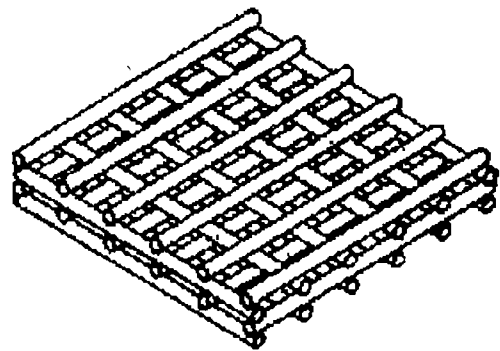
The friction factor, f , for carbon foam structure is not available. Therefore, the friction factor experimentally determined for crossed-rod, woven-screen, and glass-ceramics matrices by Kays and London (1984) are used herein.

Figure 4.14 shows three types of crossed rods: (a) random, (b) inline, and (c) staggered rods. The diameter of the rod is equal to 0.375 in. They are perfectly stacked with one row touching another. Figure 4.15 shows the woven-screen matrices with rod diameter being equal to 0.0615 in. The last two types of matrices investigated by Kays and London (1984) are the glass-ceramic matrices with straight triangular passages and hexagonal passages shown in Figure 4.16 (a) and (b), respectively.

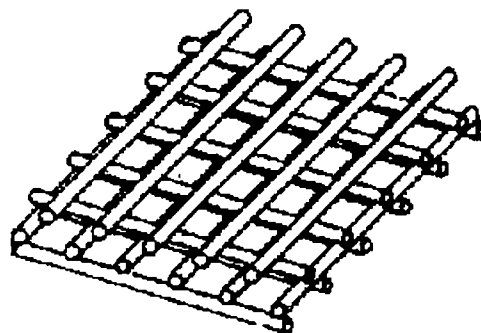
All the friction factors vs. Reynolds number graphs for the aforementioned matrices are presented in Kays and London (1984). The Reynolds numbers presented range from 5 to 100,000, which covers both the laminar flow region and the transition region (Coppage and London, 1956). However, due to the coverage of both laminar and transition region, the attempt to develop a more general expression for the flow friction data for the entire range of Reynolds number is not possible. To develop a relationship between friction factor, f , and Reynolds number, curve fitting has been performed on the



(a)



(b)



(c)

Figure 4 14 Stacked cross-rod matrices with (a) random, (b) inline; and (c) staggered configuration (Kays and London, 1984)

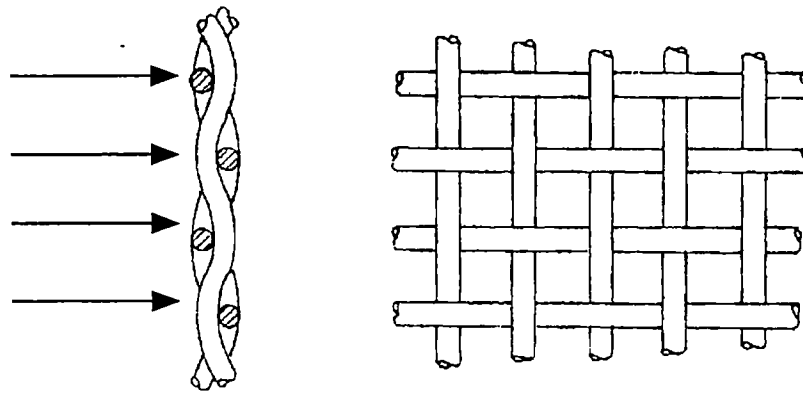


Figure 4.15. Woven-screen matrices (Kays and London, 1984).

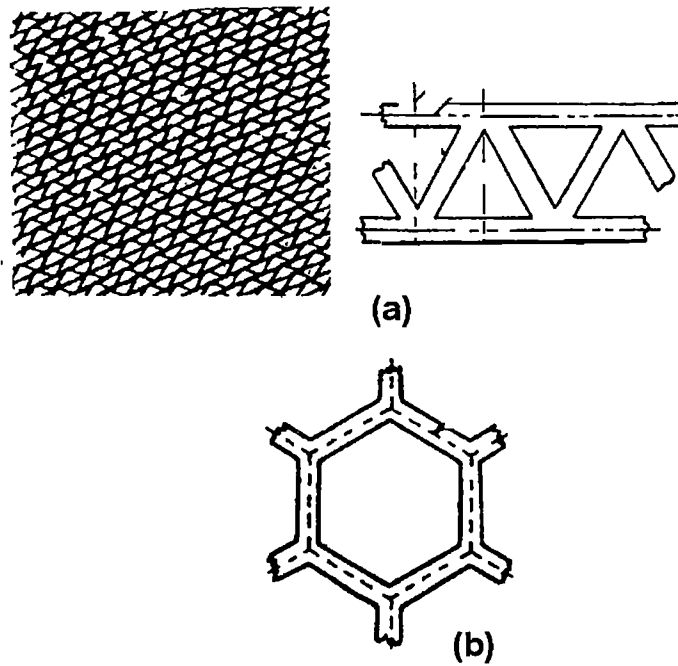


Figure 4.16. Glass-ceramic matrices with (a) straight triangular passages, (b) hexagonal passages (Kays and London, 1984)

experimental friction data for the range of Reynolds number lower than 10,000, which is of our current interest.

Figure 4.17 is a log-log graph showing the friction data for the Reynolds number ranging from 0.01 to 10,000. The dots are the experimental friction data while the solid line except for those for glass-ceramic are the trends obtained by curve fitting. Besides, B and m are constants and exponents, respectively. The friction factor of glass-ceramic matrices are expressed by (Kays and London, 1984)

$$f = 13.33 \text{Re}_{\max}^{-1} \quad (\text{straight triangular passage}) \quad (4.42a)$$

$$f = 15.05 \text{Re}_{\max}^{-1} \quad (\text{hexagonal passage}) \quad (4.42b)$$

In addition, the power law curve fitting for the random, inline, and staggered crossed-rod matrices, and woven-screen matrices gives

$$f = 0.65 \text{Re}_{\max}^{-0.08} \quad (\text{random crossed-rods}) \quad (4.43a)$$

$$f = 0.56 \text{Re}_{\max}^{-0.09} \quad (\text{inline crossed-rods}) \quad (4.43b)$$

$$f = 1.45 \text{Re}_{\max}^{-0.15} \quad (\text{staggered crossed-rods}) \quad (4.43c)$$

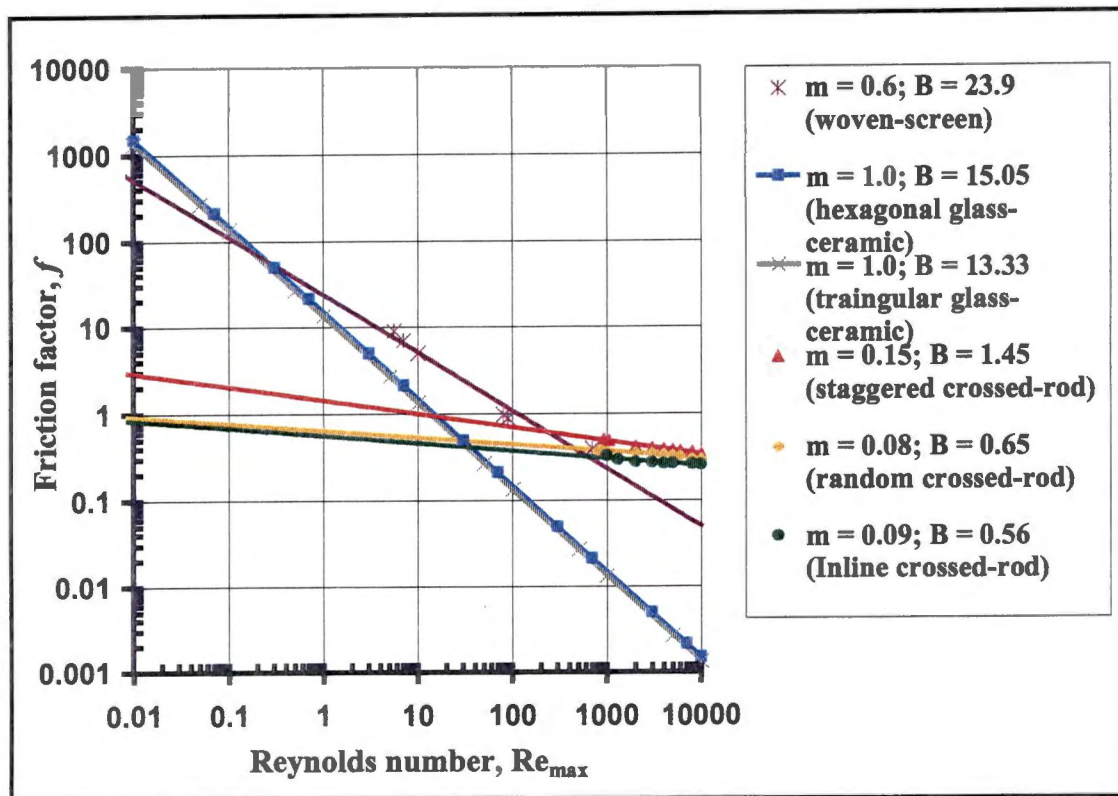


Figure 4.17: Friction factor, f , as a function of Reynolds number for various matrices with $\phi = 79\%$ and 81% for triangular and hexagonal passage glass-ceramic matrices, and 79.6% for all other matrices.

$$f = 23.9 \text{Re}_{\max}^{-0.67} \quad (\text{woven-screen}) \quad (4.43d)$$

The curve fitting used in the prediction of friction characteristic of carbon foam is valid for maximum Reynolds number, $\text{Re}_{\max}$, ranging from 0.001 to 0.1. Equations (4.42a, b) and (4.43a, b, c, d) can be written in a general form by

$$f = B \text{Re}_{\max}^{-m} \quad (4.44)$$

where B and m are constants. It is noted that the cylindrical struts model in section 4.2 suggests that $f_{\text{foam}} \sim \zeta \text{Re}_{\max}^{-\gamma}$ with $\gamma = 0.15$ for foams having cubic unit cells, which is consistent with Eq. (4.44). It is found that m for ceramic foams has a weak dependence on cell size, a , with $m = 0.18, 0.12, 0.09$ and 0.07 corresponding to foams of size 50, 30, 20 and 10 ppi (pores per inch), respectively (Schlegel *et al*, 1993). Moreover, aluminum and other metallic foams have been tested and it is shown that $m \approx 0.4$ for different morphologies (Bastawros *et al*, 1999). Table 4.2 summarizes all the coefficient B and exponent m values for different type of matrices, ceramic foams, and aluminum foams.

4.4.4 Pressure Drop Formulation

The configuration illustrated in Figure 4.18 is considered for the calculation of pressure drop, where L_p is the total length of the foam in the flow direction, and v_{avg} is

Table 4 2: The coefficient B and exponent m for various matrices and foams.

Type	B	m
Glass-ceramic (hexagonal passage)	15.05	1.00
Glass-ceramic (straight triangular passage)	13.33	1.00
Woven-screen matrices	23.90	0.67
Staggered crossed-rod matrices	1.45	0.15
Random crossed-rod matrices	0.65	0.08
Inline crossed-rod matrices	0.56	0.09
Aluminum foam (Bastawros <i>et al.</i> , 1999)	--	0.40
Aluminum foam (Lu <i>et al.</i> , 1998)	--	0.15
Ceramic foam (Schlegel <i>et al.</i> , 1993)	--	0.07 - 0.18

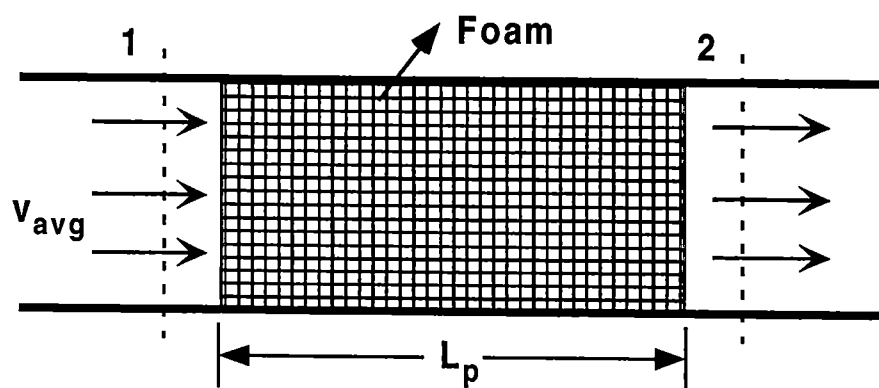


Figure 4 18 Pressure drop across carbon foam

the average free stream velocity before entering the foam. A uniform free stream velocity is assumed for the flow before and after it passes through the foam.

The following general equation for pressure drop in heat exchangers is obtained by integrating the momentum equation through the core (Kays and London, 1984)

$$\frac{\Delta p}{p_1} = \frac{G^2}{2p_1\rho_f} \left[(K_c + 1 - \phi^2) + 2 \left(\frac{\rho_1}{\rho_2} - 1 \right) + f^* \frac{A_{\text{total}}}{A_c} \frac{\rho_1}{\rho_f} - (1 - \phi^2 - K_e) \frac{\rho_1}{\rho_2} \right] \quad (4.45)$$

where Δp is the pressure drop across the foam, p_1 is the inlet pressure, G is the flow stream mass velocity, ρ_f is the average fluid density, ρ_1 is the fluid inlet density, ρ_2 is the fluid outlet density, K_c and K_e are the entrance and exit loss coefficients respectively, f^* is the core friction factor, A_c is the minimum flow area, and A_{total} is the total surface area of the foam. The first term on the right hand side of Eq. (4.45) addresses the flow entrance effect, the second term is the flow acceleration, and the third term is for the core friction losses, and the last term deals with the flow exit effect. The flow stream mass velocity, G , can be expressed by

$$G = \frac{W}{A_c} = \rho_f v_{\text{max}} \quad (4.46)$$

where W is the mass flow rate for the system and $v_{\max}$ is used as the reference velocity for the velocity profile within the foam. Then, by substituting Eq (4.46) into Eq (4.45) with the aid of Eq (4.39), Eq (4.45) becomes

$$\Delta p = \frac{\rho_f v_{\max}^2}{2} \left[(K_c + 1 - \phi^2) + 2 \left(\frac{\rho_1}{\rho_2} - 1 \right) + f^* \frac{4L_p}{D_h} \frac{\rho_1}{\rho_f} - (1 - \phi^2 - K_e) \frac{\rho_1}{\rho_2} \right] \quad (4.47)$$

For simplicity, the core friction, entrance and exit loss effects are combined into a single friction factor, f , since only the overall friction effect is interested in the studies on pressure drop across the foam. Furthermore, the fluid density, ρ_f is assumed to remain constant ($\rho_f = \rho_1 = \rho_2$), the pressure drop across the foam can thus be expressed by

$$\Delta p = \frac{\rho_f v_{\max}^2}{2} \left[f \frac{4L_p}{D_h} \right] \quad (4.48)$$

Since the fluid density is assumed to be constant, the core friction but not the flow acceleration has influence on pressure drop. However, this simplification can be relaxed for the heat convection process where the inlet and outlet temperatures and densities are significantly different. For multipass configurations, losses in the return headers need to be accounted for separately, as must any losses in inlet and exit headers and associated ducting (Kays and London, 1984)

In order to express the pressure drop per unit length as a function of porosity, ϕ and surface area density α_A , Eq (4.38) is substituted into Eq. (4.48) giving

$$\frac{\Delta p}{L_p} = \frac{\rho_f v_{\max}^2}{2} \left[f \frac{\alpha_A}{\phi} \right] \quad (4.49)$$

Substitution of Eq. (4.36) and (4.44) into Eq. (4.49) gives

$$\frac{\Delta p}{L_p} = \frac{\rho_f}{2\phi} \alpha_A v_{\max}^2 B \left(\frac{v_{\max} D_h}{v_f} \right)^{-m} \quad (4.50)$$

The combination of Eq (4.38), (4.40) and (4.50) gives

$$\frac{\Delta p}{L_p} = \frac{\rho_f}{2\phi} \alpha_A \left[\frac{4(2.598 L^2)}{\pi D_{\text{hex}}^2} v_{\text{avg}} \right]^{2-m} B \left(\frac{1}{v_f} \frac{4\phi}{\alpha_A} \right)^{-m} \quad (4.51)$$

With the aid of Eq. (4.28b) and (4.34), Eq. (4.51) is re-written to give the pressure drop across the carbon foam

$$\frac{\Delta p}{L_p} = \frac{\rho_f B}{2\phi^{1+m}} \left(\frac{4}{v_f} \right)^{-m} \left[\frac{2598L^2}{\pi \left[\left(\frac{D}{2} \right)^2 - \left(\frac{245L}{2} \right)^2 \right]} v_{avg} \right]^{2-m} \left[\frac{\pi D^2}{11.31L^3} \right]^{1+m} \quad (4.52)$$

By knowing the ρ_f and v_f , which are the fluid physical properties, v_{avg} of the free stream flow, the coefficient B and the exponent m from the power law relationship of the experimental friction data, the foam porosity, ϕ , mean pore diameter of sphere, D , and the unit cell edge length, L , which can be determined from Eq (4.32) numerically, the pressure drop across the carbon foam based on the spherical void model is estimated in Eq. (4.52).

4.4.5 Results

By applying all the coefficients B and exponents m in Table 4.2 to Eq (4.52), the pressure drop per unit length, $\Delta p/L_p$, as a function of average free stream velocity, v_{avg} , for air at 20 °C ($v_f = 1.46 \times 10^{-5}$ m²/s and $\rho_f = 1.23$ kg/m³) flowing through a duct filled with carbon foam ($D = 300$ μm and $\phi = 80\%$) is shown in Figure 4.19. The experimental pressure drop data for Mitsubishi ARA24 mesophase pitch derived carbon foam is also shown in Figure 4.19 as a comparison (Klett, 1999a)

The model using the friction parameters of hexagonal glass-ceramic matrices ($B = 15.05$; $m = 1.0$) agrees well with the experimental data. The model using the frictional parameters of the random, inline, and staggered crossed-rod matrices significantly

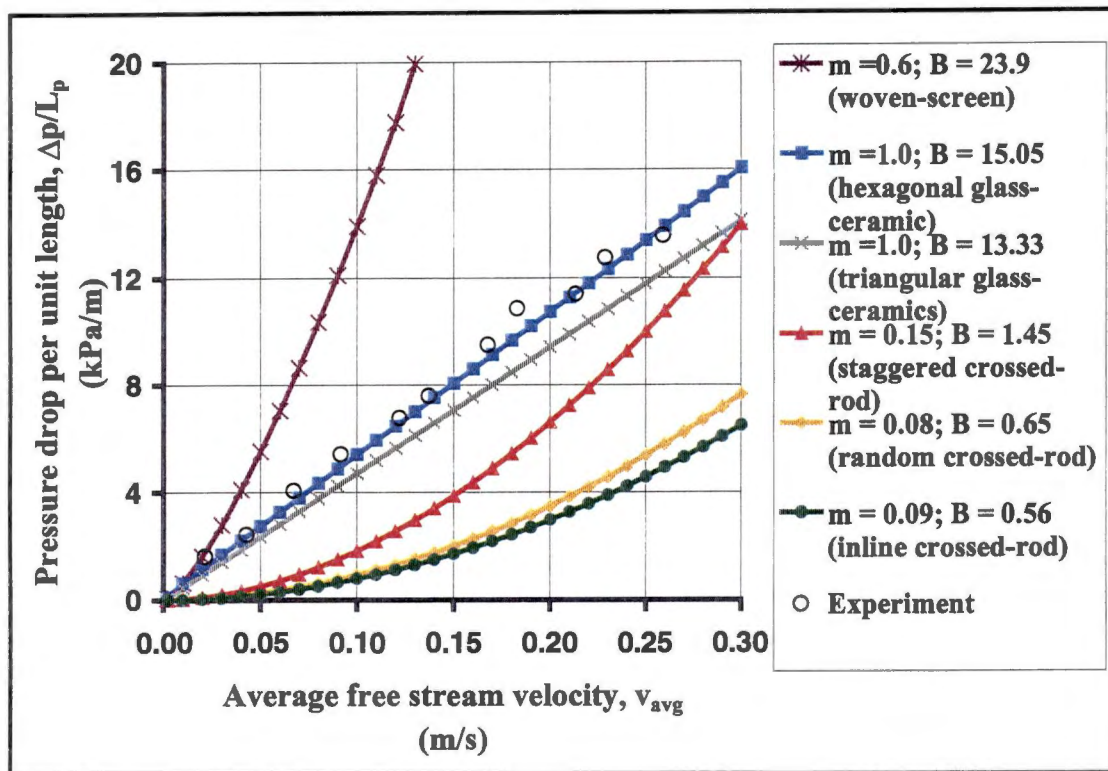


Figure 4.19: Pressure drop per unit length, $\Delta p/L_p$, as a function of fluid velocity, v_{avg} , for air (20°C) flowing across a duct filled with carbon foam ($D = 300\mu\text{m}$; $\phi = 80\%$) by using various flow friction prediction.

underestimate the pressure drop of the carbon foam, while that using the parameters for the woven-screen matrices significantly overestimates the pressure drop of the carbon foam. The mismatch of the f of all the crossed-rod matrices confirms the statement made by Tong and London (1957) regarding risky prediction when the experimental friction data is extrapolated to a lower Reynolds number for a different geometry. Nevertheless, the glass-ceramic matrices which has a much similar geometry and is associated with lower Reynolds number agrees reasonably well with the experimental data of carbon foam. However, more experiments should be conducted in order to draw a better conclusion about the present model's prediction.

The final expression of the pressure drop prediction of the ARA24 foam can be written by

$$\frac{\Delta p}{L_p} = \frac{15.05 \rho_f v_f}{8 \phi^2} \frac{2.598 L^2}{\pi \left[\left(\frac{D}{2} \right)^2 - \left(\frac{2.45 L}{2} \right)^2 \right]} v_{avg} \left[\frac{\pi D^2}{11.31 L^3} \right]^2 \quad (4.53)$$

Eq. (4.53) shows an approximately linear relationship between pressure drop and the average free stream velocity. Therefore, Darcy's law will in fact still be valid to some extent under the low flow rates and foam with small pores. However, the linear relationship between the pressure drop and the average free stream velocity might not hold if the flow rate increases significantly.

As for the Conoco mesophase pitch-derived carbon foam with a smaller pore diameter of 60 μm , $\Delta p/L_p$ as a function of v_{avg} is shown in Figure 4.20 for the pore diameter of 60, 200 and 300 μm . Other flow properties and foam morphology are the same as one might expect, the pressure drop across the foam increase with decreasing pore size due to the fact that less fluid is able to get through the smaller pores.

The pressure drop across the carbon foam as a function of mean pore diameter D for air and water at $v_{\text{avg}} = 0.01, 0.1$, and 1 m/s are predicted in Figure 4.21 and 4.22, respectively. It is clearly shown that an increase in the mean pore diameter significantly decreases the pressure drop. Nevertheless, the heat transfer performance might be degraded by an increase in the mean pore diameter.

Figure 4.23 shows $\Delta p/L_p$ across a duct filled with carbon foam ($D = 300 \mu\text{m}$; $\phi = 80\%$) as a function of v_{avg} for different cooling fluid type – water, Freon 12, ammonia, and air. The pressure drop of Freon 12 and ammonia are considerably lower than water, which results in the highest pressure drop. On the other hand, the air gives the lowest pressure drop since it has the lowest density.

4.4.6 Discussion

We conclude that the spherical void model provides a reasonably good presentation of the pressure drop across the carbon foam although the friction factor might not be the actual physical hydraulic drag the foam and the fluid experience. It is noted that the velocity of the fluid flow considered herein is less than 0.3 m/s ($Re_{\text{max}} = 16$) and should be in the transitional of the laminar and creeping flow. Besides, the

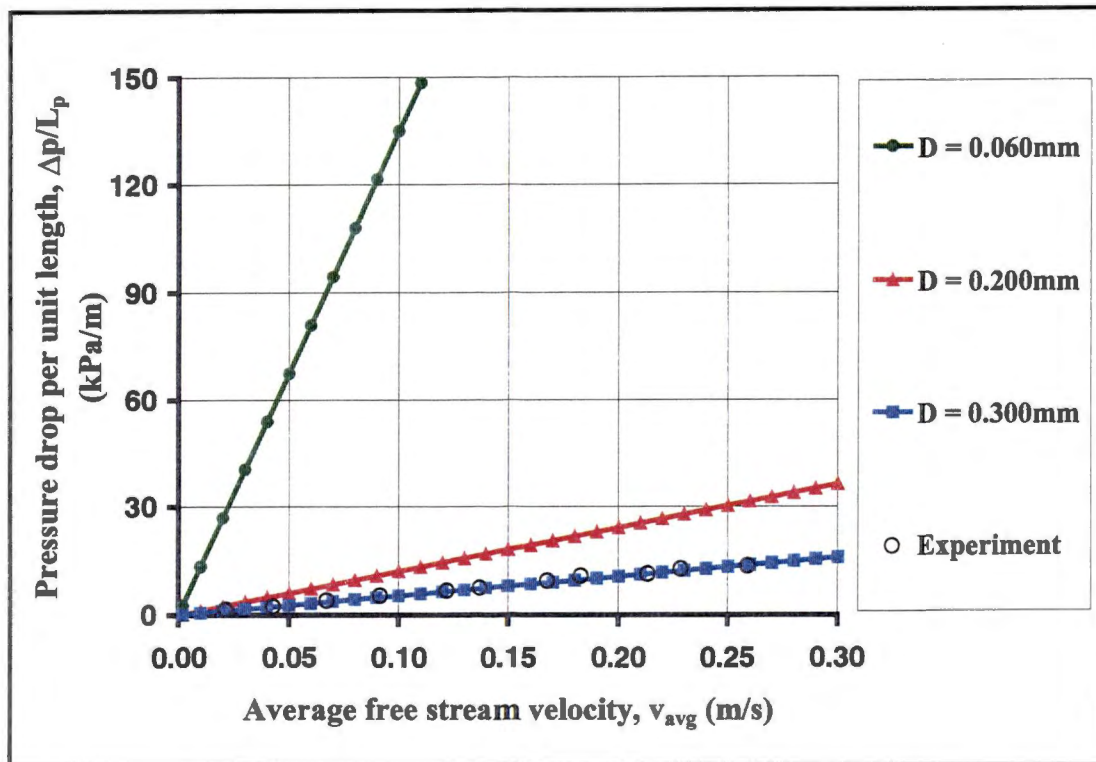


Figure 4.20: Pressure drop per unit length, $\Delta p/L_p$, as a function of fluid velocity, v_{avg} , for air (20°C) flowing across a duct filled with carbon foam with different mean pore diameter ($\phi = 80\%$).

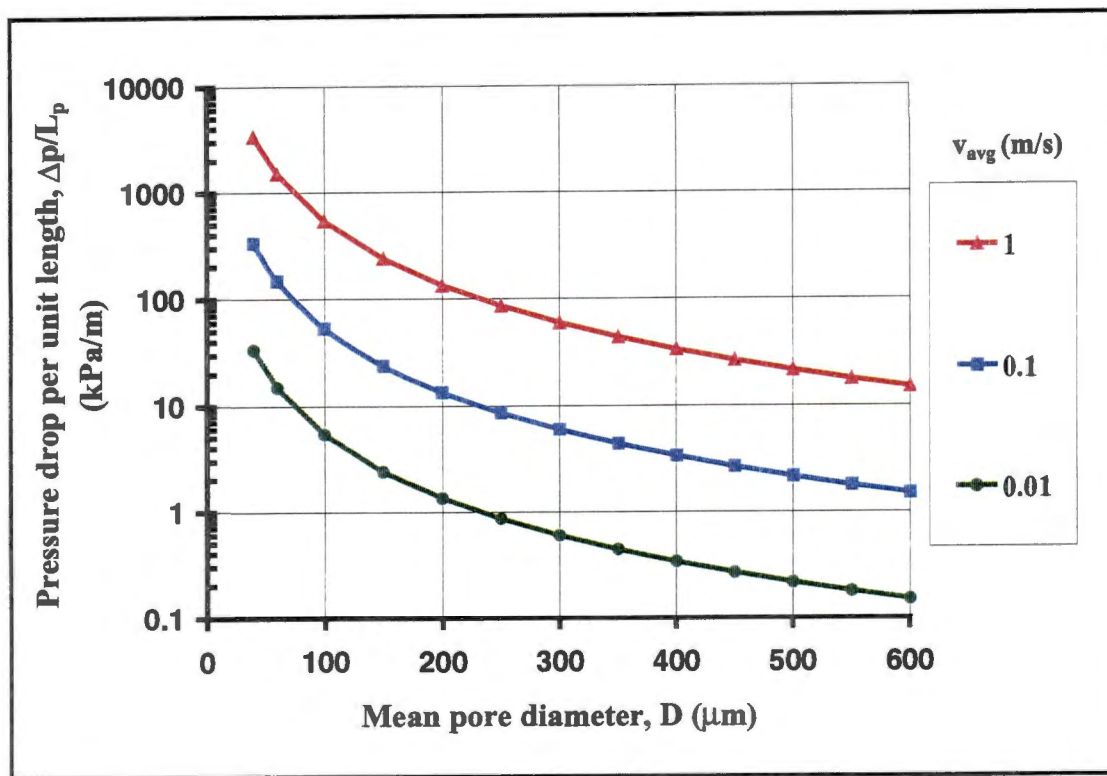


Figure 4.21: Pressure drop per unit length, $\Delta p/L_p$, as a function of mean pore diameter for air (20°C) with different velocities flowing across a duct filled with carbon foam ($\phi = 80\%$).

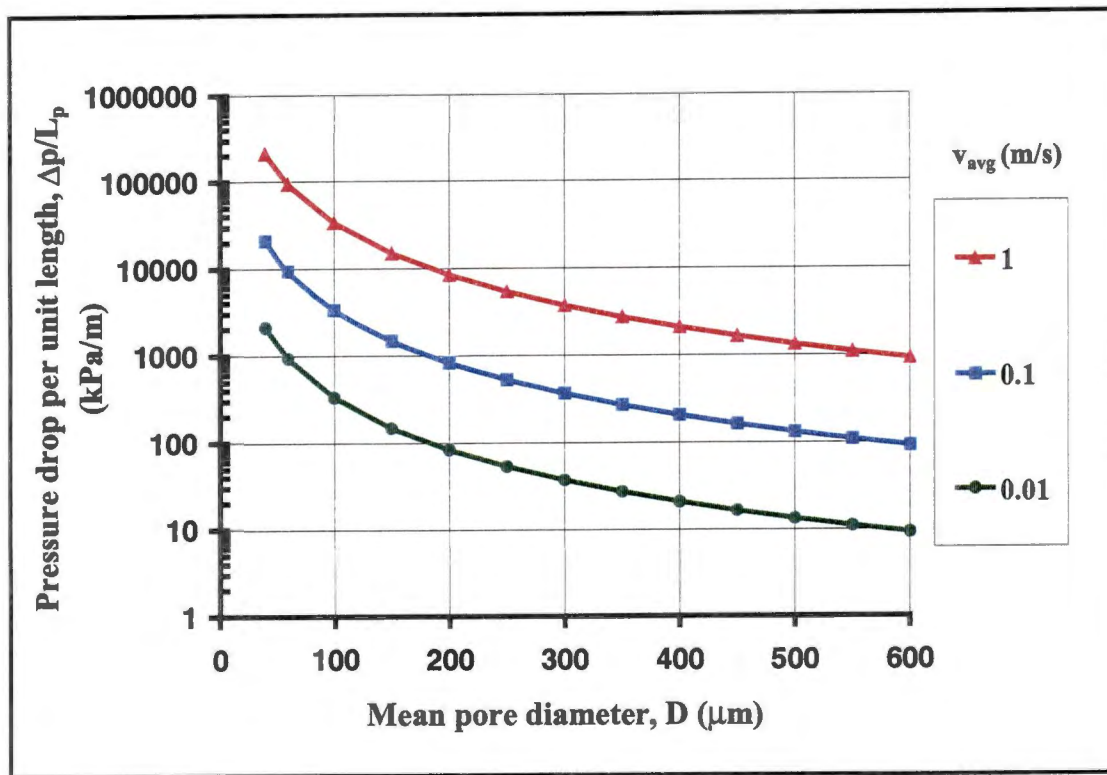


Figure 4.22: Pressure drop per unit length, $\Delta p/L_p$, as a function of mean pore diameter for water (20°C) with different velocities flowing across a duct filled with carbon foam ($\phi = 80\%$).

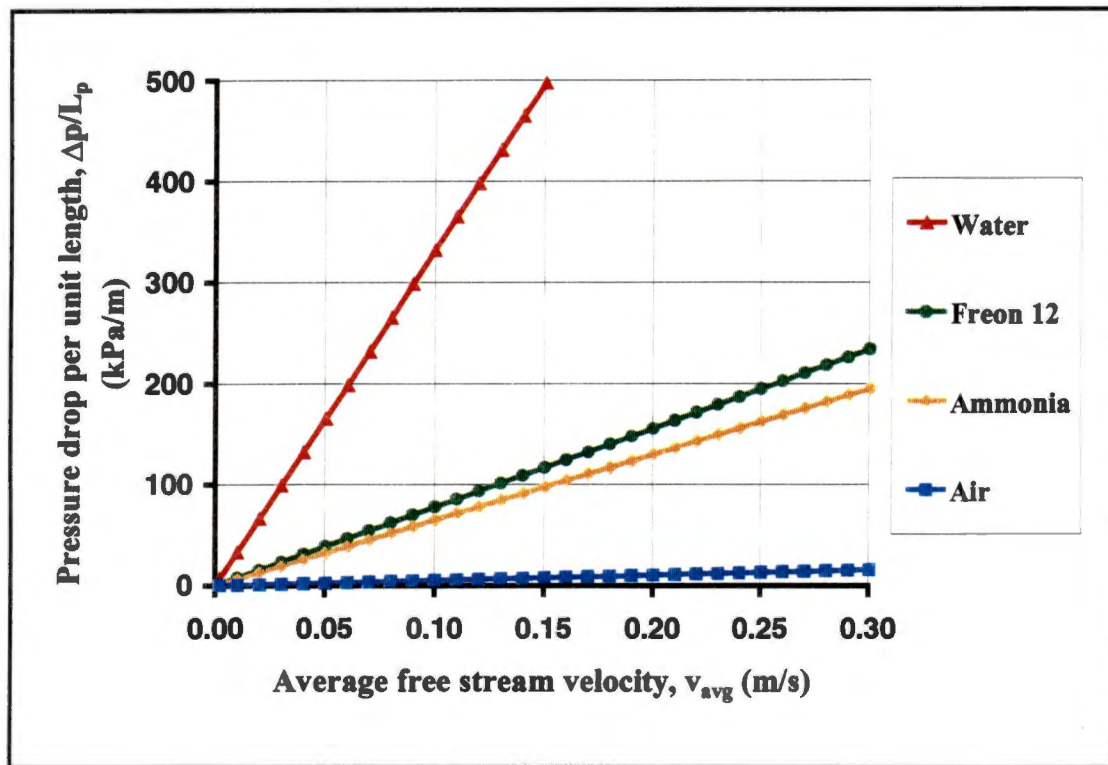


Figure 4.23: Pressure drop per unit length, $\Delta p/L_p$, as a function of fluid velocity, v_{avg} , for different fluids at 20°C flowing across a duct filled with carbon foam ($D = 300\mu\text{m}$; $\phi = 80\%$).

carbon foam morphology of this model is accurately represented in the spherical void model in comparison with the porous medium model and the cylindrical strut model that is appropriate for open cell aluminum foam. The main difference is that carbon foam has thicker walls (struts) in comparison with the aluminum foam, which involves slender struts.

In a heat exchanger system, the heat transfer rate, q , and pressure drop (pumping power) often had the approximate proportionality with velocity, v , respectively as (Bell, 1990)

$$\Delta p \propto v^\lambda \quad (4.54)$$

$$q \propto v^j \quad (4.55)$$

where λ range from 1.0 to 1.8 and j ranges from 0.3 to 0.6 for a typical heat exchanger. Therefore, optimizing of the foam performance as a core material is essential for the optimum heat transfer rate while maintaining relatively low the pressure drop across the carbon foam. Thus, the theoretical model can be used as a guiding tool for the design of optimum foam structures that would maximize heat transfer per unit pumping power.

CHAPTER 5

Mechanical and Thermal Testing

5.1 Background

The mechanical and thermal properties of the novel pitch-derived carbon foam need to be fully characterized before the carbon foam can be considered for commercial applications. In order to determine the physical properties, mechanical and thermal testing had been performed.

Tensile and flexural tests have been conducted to determine the strengths of carbon foam infiltrated composite materials. In addition, a prototype heat sink and several heat exchanger devices have been tested to evaluate the overall heat transfer coefficient and the pressure drop of the carbon foam used as the core material for thermal management applications.

5.2 Mechanical Testing

5.2.1 Overview

The specific gravity, compressive strength, and Young's modulus of the carbon foam and foam-based composites together with four other thermal management materials are listed in Table 5.1 (Klett, 1999b). The compressive strength of the ARA carbon foam, which is about 3 MPa, is rather low in comparison with aluminum honeycombs but is higher than those of reticulated vitreous carbon (RVC) glassy carbon foams and aluminum foams. When the carbon foam is impregnated with an epoxy resin, its

Table 5.1. Mechanical properties of the ARA carbon foams compared with other thermal management materials (Klett, 1999b)

Material	Specific Gravity	Compressive Strength (MPa)	Young's Modulus (Compression) (MPa)
ARA carbon foam	0.54	3.40	180
ARA carbon foam/Epoxy	1.26	34.3	560
ARA carbon foam/CVD Carbon	1.30	31.6	850
Reticulated vitreous carbon (RVC) glassy carbon foam*	0.05	0.38	48
EWC-300/Cyanate ester resin†	1.72	18.5	--
Aluminum honeycomb‡	0.19	18.6	1030
Aluminum foam (Steiner <i>et al</i> , 1997)	0.50	~1.0	~1.0

* ERG Product Literature (2000)

† Amoco Product Literature (1997)

‡ Hexcel Product Data Sheet (1997)

compressive strength increases by an order of magnitude to 34.3 MPa, and approaches that of commercial thermal management panels (Klett and Burchell, 1998). Although the compressive strengths of 31.6 MPa is also achieved when the foam is densified with chemical vapor deposited (CVD) carbon, the mode of failure of these two foam-based materials are different. While both raw carbon foam and the resin-filled foam exhibit a high work of fracture, the CVD/foam material has a more brittle failure (Klett, 1999b).

The mechanical properties of the novel pitch-derived carbon foam are of prime interest since they can be used for deciding the size or shape of carbon foam-made components to meet design specifications. Besides, the mechanical test results may be used for material designs and quality control. Tensile and flexural tests on carbon foam and its infiltrated composites are performed to determine and characterize the strength and stiffness of the material.

5.2.2 Foam Based Composite Materials

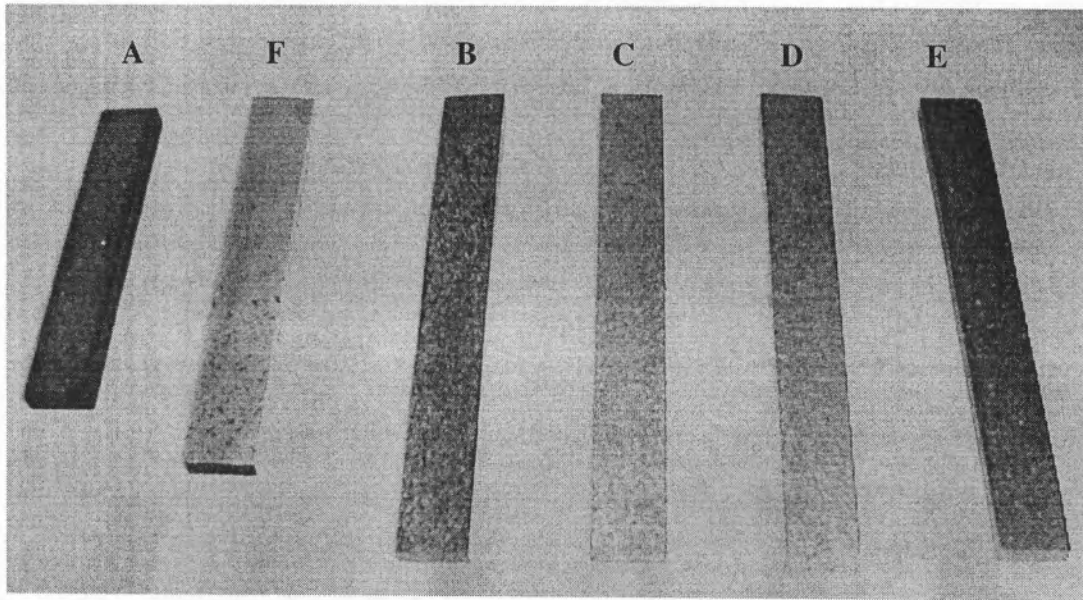
The foam based composite specimens have been fabricated by vacuum impregnating ARA24 pitch-derived carbon foams with selected resins together with the suitable hardener. The vacuum impregnation process is performed in a closed chamber at room temperature (Klett, 1999a).

Several types of commercially available matrices are used to bind carbon foams together by virtue of its cohesive characteristic, to transfer load to and from the foams, and to protect the foams from environments and handling. Besides, the resins can keep the carbon foam in the proper orientation so that they can carry the design loads, provide

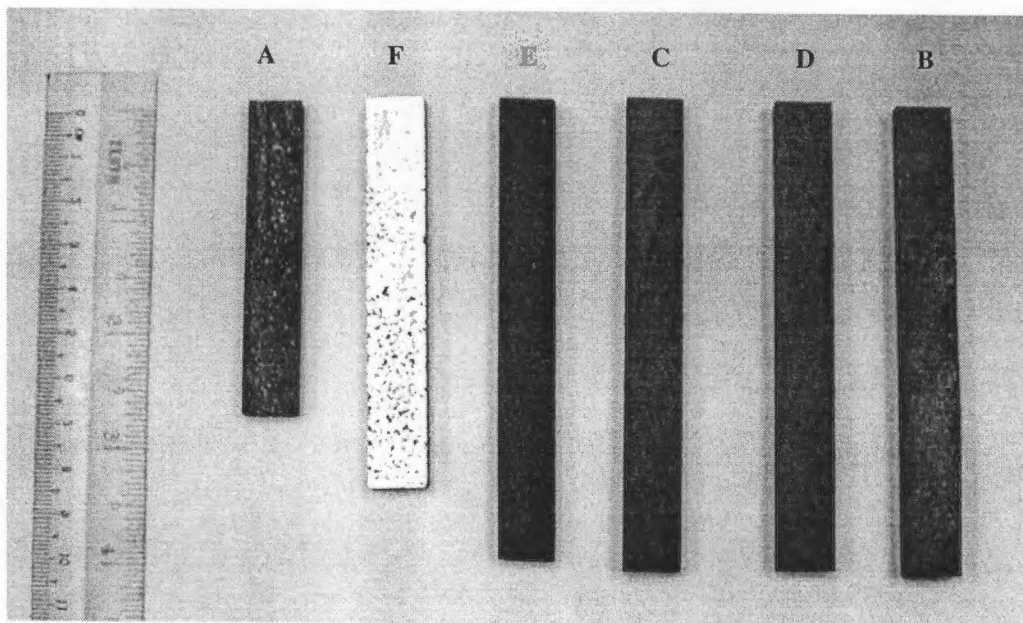
resistance to crack propagation and damage evaluation, and enhance the interlaminar shear strength of the composites (Reinhart and Clements, 1987)

The selected resins are (1) polyester resin, (2) polycyanate resin (RS-14), (3) epoxy, (4) SMJ carbon and (5) aluminum. The polyester resin used is the Bondo Fiberglass automotive grade resin. It has moderate viscosity and cures at room temperature. The polycyanate resin has a low viscosity, high toughness, high temperature stability, and low shrinkage during cure. The epoxy resin has a low viscosity, can be cured at room temperatures, and provides excellent interfacial bonding to graphite. SMJ carbon is a proprietary pitch-type deposit that has a high thermal conductivity and is used for high temperature applications. The aluminum/silicon alloys used exhibit high toughness. The foam used to impregnate with aluminum is first densified with carbon using chemical vapor deposition (Klett, 1999a)

Samples of carbon foam infiltrated composites used for mechanical tests are shown in Figure 5.1(a), where they are labeled as A-SMJ carbon, B-polyester resin, C-polycyanate resin, D-polycyanate resin with surface finishing, E-epoxy resin, and F-aluminum. All of the specimens except F-CVD foam/aluminum and A-foam/SMJ carbon specimens are machined into rectangular bar of 0.5 inch in width, 0.2 inch in thickness, and 4 inch in length, see Figure 5.1(b). With the same cross-sectional area, the F-CVD foam/aluminum specimens and A-foam/SMJ carbon specimens have a length of 3.36 inch and 2.75 inch, respectively. The dimensions and densities of all of the specimens are listed in Table 5.2 and all of them were used for both the tensile and flexural testing.



(a)



(b)

Figure 5.1: Carbon foam infiltrated composites specimen: A-SMJ carbon; B-polyester resin; C-polycyanate resin; D-polycyanate resin (surface finished); E-epoxy resin; and F-aluminum.

Table 5.2. Carbon foam infiltrated composite specimens

Matrix	Specimen	Length (inch)	Width (inch)	Thickness (inch)
Foam / SMJ carbon	A1	2.741	0.4785	0.1950
	A2	2.740	0.4865	0.1890
	A3	2.748	0.4790	0.1800
	A4	2.750	0.4790	0.1895
	A5	2.751	0.4855	0.1815
Foam / polyester resin	B1	4.055	0.4800	0.1975
	B2	4.050	0.4990	0.1965
	B3	4.040	0.4790	0.2000
	B5	4.006	0.4970	0.2010
Foam / polycyanate resin	C1	4.000	0.4965	0.1985
	C2	4.008	0.4955	0.2015
	C3	4.010	0.4955	0.2040
	C5	4.005	0.4995	0.2010
	C6	4.000	0.5040	0.2040
Foam / polycyanate resin with surface finishing	D1	4.061	0.5000	0.2020
	D2	4.062	0.5030	0.2035
	D3	4.071	0.4955	0.2025
	D5	4.030	0.4960	0.2100
Foam / epoxy resin	E1	3.960	0.4815	0.1900
	E2	3.959	0.4880	0.1920
	E3	3.960	0.4915	0.1960
	E4	3.961	0.4890	0.1910
	E5	3.962	0.4905	0.1900
CVD foam / aluminum	F1	3.359	0.5010	0.2090
	F2	3.364	0.5000	0.2010
	F3	3.366	0.5020	0.2010
	F4	3.367	0.5000	0.2030

Both the tensile and flexural tests are carried out using the Instron Model 4301 testing machine, an electromechanical device consisting of a standard load frame and drive unit, a load weighing system, and a microprocessor based control console, all of which are built into one compact assembly; see Figure 5.2. A moving crosshead applies a load to the specimen at commanded speeds. The load cell is a calibrated transducer that measures the applied load. The upper grip coupling is a self-aligning connection between the upper grip and the load cell, while the base grip adaptor is a rigid coupling between the lower grip and the base of the machine frame. Finally, the control console provides an interface between the operator and the loading frame.

5.2.3 Tensile Tests

Fundamentally important material behavior, such as Young's modulus, ultimate strength, and proportional limit, can be determined by uniaxial tensile tests. The uniaxial tensile test samples are prepared according to ASTM D3039-95, which is the standard tensile test method for polymer matrix composite materials. For simplicity, all tensile test specimens have a uniform cross-sectional area and a bonded tab of aluminum shim (1 inch x 0.5 inch) is glued on both side of the specimen where the grip is in contact with the specimen. The thin shim layers are able to protect the surfaces of the end sections of the specimen against damage in the grips, to improve the transmission of tensile forces to the specimen, and to prevent the specimen from failing at the end sections near the grip.

Once the tensile specimen is positioned between the upper and lower grips of the Instron Model 4301 machine, a load is applied at a constant rate of 0.02 in/min until

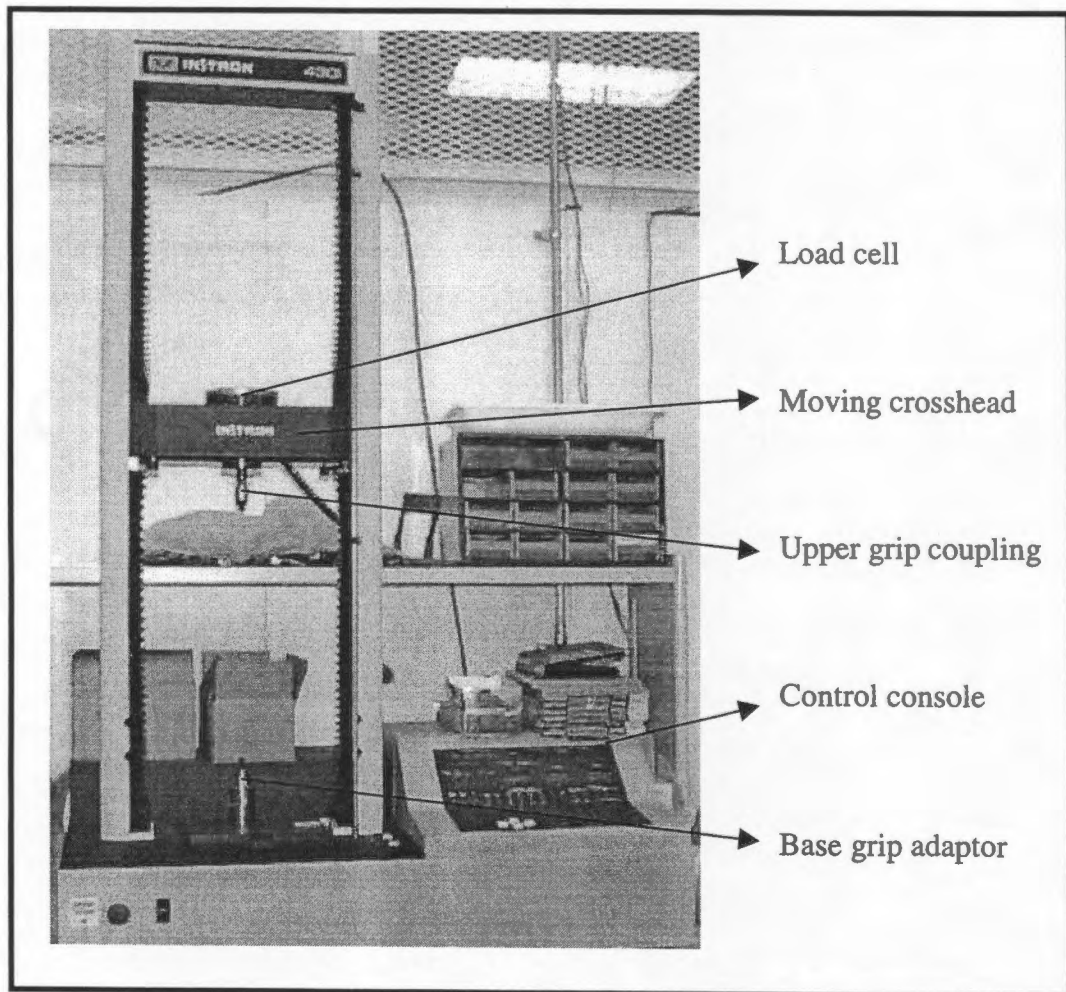


Figure 5.2: Instron Model 4301 machine used for tensile and flexural testing.

failure occurs. The load cell records the load for every displacement measured at a sampling rate of 5 data points per second

From the tensile test results, the ultimate tensile strength of the specimen, σ_{UTS} , can be calculated determined by

$$\sigma_{UTS} = \frac{P_{max}}{A_n} \quad (5.1)$$

where P_{max} is the maximum load prior to failure, and A_n is the nominal cross-sectional area of the specimen.

5.2.4 Flexural Tests

The flexural test method can be used to determine the flexural properties of the carbon foam-infiltrated composites. Since there is no standard flexural test method for foam-reinforced composites, the current flexural tests are conducted according to ASTM C1341-96, which is the standard flexural test method for continuous fiber-reinforced ceramic composites, and ASTM D790-93, which is the standard flexural test method for unreinforced and reinforced plastics, and electrical insulating materials.

As illustrated in Figure 5.3, a rectangular specimen with depth, d_f , rests on two supports and is subjected by two point forces of $P/2$. The support span is denoted by L_f , and the load span is one-third the support span.

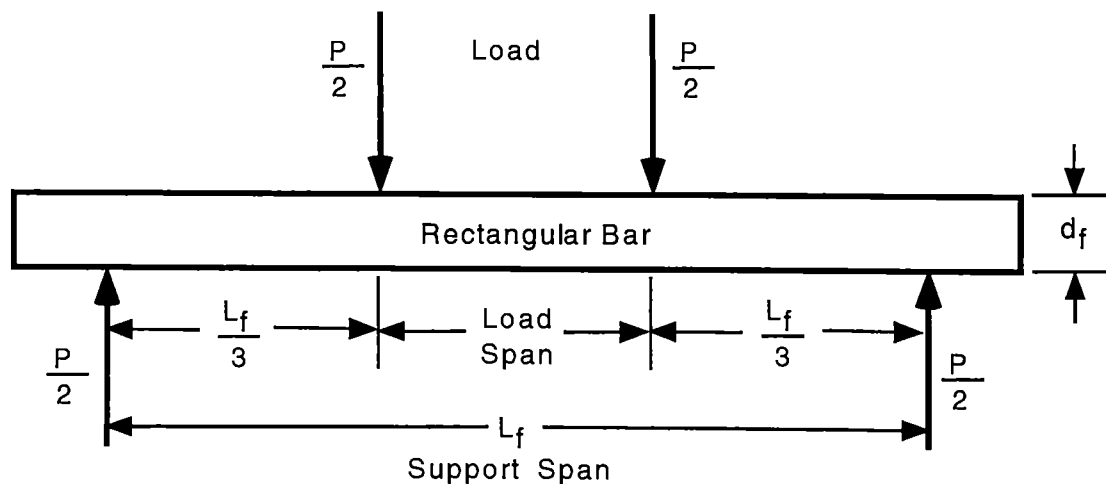


Figure 5.3 Four-point bend.

All of the rectangular bars used for the flexural tests have a depth of 0.2 inch, whereas the support span, L_f , of 3 inch is selected for all of the specimens except the foam/SMJ carbon composite specimens, which has $L_f = 2.361$ inch. Therefore, the rectangular bar used for the flexural testing has a support span-to-depth ratio, L_f/d_f , of 12 for the foam/SMJ carbon specimens and 15 for the rest of the composite specimens. Although ASTM C1341-96 and ASTM D790-93 recommends the L_f/d_f ratio be greater than 16 to prevent shear failure, the test specimens with L_f/d_f ratio less than 16 are able to produce a valid test since tensile or compressive failure in the outer surfaces of the specimen.

The loading noses and supports used in four-point bend have cylindrical surfaces in order to avoid excessive indentation, or failure due to stress concentration directly under the loading noses. Once the specimen is set up according to Figure 5.3, loads are

applied at a constant crosshead motion rate of 0.08 in/min for all of the specimen except the foam/SMJ carbon specimen for which the rate of 0.0516 in/min is used. The crosshead motion rate $R_{\text{crosshead}}$ is selected based on the following expression in ASTM D790-93:

$$R_{\text{crosshead}} = \frac{0.185L_f^2 Z}{d_f} \quad (5.2)$$

where Z is the rate of straining of the outer layer. The load cell records the loads and displacement readings at a sampling rate of 10 data points per second.

The objective of the flexural tests is to determine the flexural strengths of the specimens. According to ASTM D790-93, the flexural strength is equal to the maximum stress in the outer fibers at material failure. From Figure 5.3, the maximum stress, σ_{BEND} , occurs in the outer fiber at the midpoint of the load span and can be formulated by using the elementary beam theory

$$\sigma_{\text{BEND}} = \frac{Mc}{I} \quad (5.3)$$

where M is the maximum resultant internal moment and is equal to $P_{\text{max}}L_f / 6$, which occurs at the midpoint of the load and support spans, c is the distance from the neutral axis to the point farthest from the neutral axis, i.e., the outer or inner surface, and I is the

moment of inertia of the cross-sectional area computed about the neutral axis, i.e., for a rectangular cross section

$$I = \frac{b_f d_f^3}{12} \quad (5.4)$$

where b_f is the width of the specimen, and d_f is the depth of the specimen. By substituting Eq (5.4) into Eq (5.3), the flexural strength of the specimens, σ_{BEND} , can be rewritten by

$$\sigma_{\text{BEND}} = \frac{\left(\frac{P_{\text{max}} L_f}{6} \right) \left(\frac{d_f}{2} \right)}{\left(\frac{b_f d_f^3}{12} \right)} = \frac{P_{\text{max}} L_f}{b_f d_f^2} \quad (5.5)$$

where P_{max} is the maximum load prior to failure

5.2.5 Results and Discussions

Table 5.3 shows the maximum load prior to failure, P_{max} , the maximum displacement at the maximum load, δ , and the ultimate tensile strength, σ_{UTS} , of all of the tensile test specimens. The flexural testing results are listed in Table 5.4. The flexural strength is higher than the tensile strength for all of the specimens and there are some variations in both the tensile and the flexural strengths data, i.e., the Weibull distribution. The main reason for the variations may be due to the brittle nature of the carbon foam phase.

Table 5 3 Tensile test results of carbon foam infiltrated composite specimens

Matrix	Specimen	Maximum Load prior to failure, P_{max} (lbs)	Displacement at maximum load, δ (inch)	Tensile strength, σ_{UTS} (psi)
Foam / SMJ carbon	A1	62	0.040	660
	A2	48	0.038	520
Foam / polyester resin	B1	57	0.051	610
	B2	57	0.054	580
Foam / polycyanate resin	C1	61	0.054	620
	C2	63	0.064	630
	C3	50	0.057	500
Foam / polycyanate resin with surface finishing	D1	68	0.058	670
	D2	69	0.056	680
Foam / epoxy resin	E1	108	0.061	1180
	E1-1 [#]	177	0.065	1930
	E2	85	0.051	900
	E3	170	0.075	1800
CVD foam / aluminum	F1	380	0.099	3600
	F1-1 [#]	430	0.118	4100

[#] Tested on the intact portion of the specimen after the first test

Table 5 4 Flexural test results of carbon foam infiltrated composite specimens.

Matrix	Specimen	Maximum Load prior to failure, P_{max} (lbs)	Displacement at maximum load, δ (inch)	Flexural strength, σ_{BEND} (psi)
Foam / SMJ carbon	A3	9.43	0.029	1430
	A4	15.57	0.036	2140
	A5	9.61	0.032	1420
Foam / polyester resin	B3	11.82	0.162	1850
	B5	12.10	0.146	1810
Foam / polycyanate resin	C5	10.00	0.073	1480
	C6	10.37	0.071	1490
Foam / polycyanate resin with surface finishing	D3	9.72	0.062	1440
	D5	10.85	0.075	1490
Foam / epoxy resin	E4	11.25	0.057	1890
	E5	12.35	0.066	2090
CVD foam / aluminum	F3	41.08	0.052	6080
	F4	24.60	0.048	3580

Furthermore, brittle failure can be observed from the typical load vs displacement diagram for tensile and flexural tests, shown in Figure 5 4 and 5 5, respectively. In Figure 5 4, the tensile load varies linearly with displacement until failure without much plastic deformation. Similarly, the flexural test specimens fail in a brittle mode even if the load is not linearly proportional to the displacement for most of the foam-based composites. Although brittle fracture is not usually observed in carbon fiber-reinforced polymer matrix composites, the failure modes for carbon foam-polymer matrix composites may be very different as both the reinforcement and the matrix are three-dimensionally continuous (Klett, 1999a).

The average tensile and flexural strengths of all of the carbon foam infiltrated composite materials together with the raw carbon foam are summarized in Table 5 5. The strength of the carbon foam infiltrated composite materials is significantly improved in comparison with the raw carbon foam. It is noted that the CVD foam/aluminum composite has the highest tensile and flexural strength. Based on the fracture surface analysis, the high strength of the CVD foam/aluminum composite is due to the good foam/aluminum interface (Klett, 1999a). Among the foam/resin composites, the epoxy resin has higher strengths than the polyester and polycyanate resins because epoxy resin exhibits better interfacial bonding at interface (Klett, 1999a). The main reason for the poor interface of the polycyanate resin is due to coefficient of thermal expansion mismatch during curing (cooling) process (Klett, 1999a).

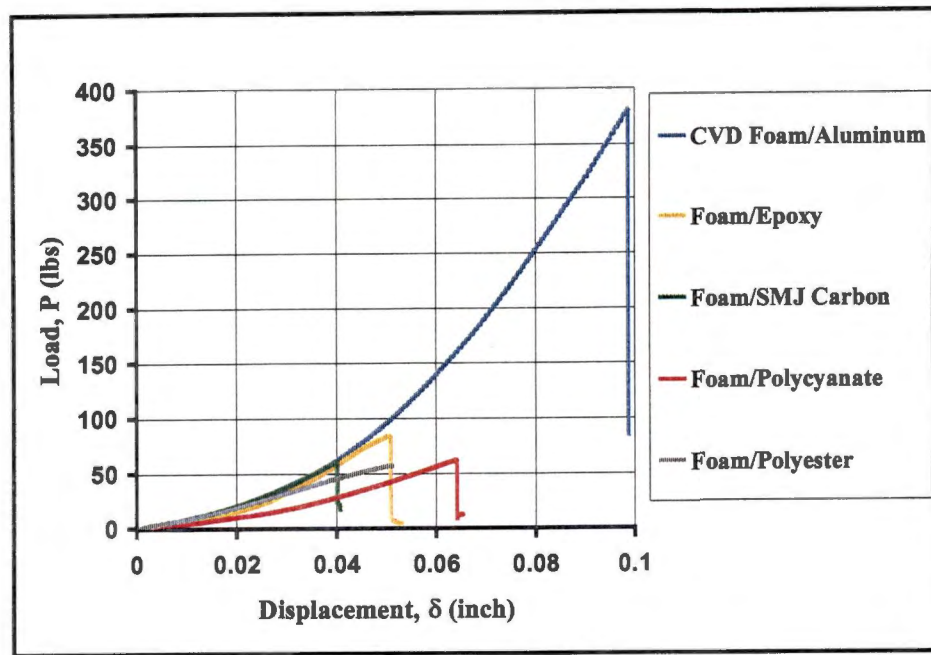


Figure 5.4: Load as a function of displacement for carbon foam infiltrated composite specimens under tensile testing.

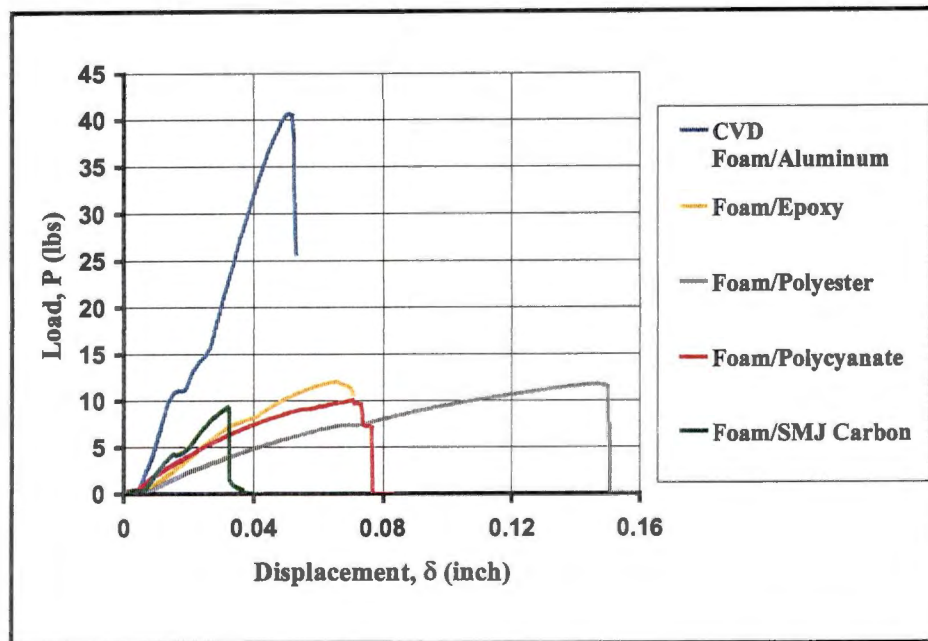


Figure 5.5: Load as a function of displacement for carbon foam infiltrated composite specimens under flexural testing.

Table 5.5 Summary of tensile and flexural strengths of foam infiltrated composites.

Matrix	Density (g/cm ³)	Tensile strength, σ_{UTS} MPa (psi)	Flexural strength, σ_{BEND} MPa (psi)
Raw carbon foam	0.54	0.69 (100)	--
Foam / SMJ carbon	1.34	4.07 (590)	11.5 (1660)
Foam / polyester resin	1.33	4.10 (595)	12.6 (1830)
Foam / polycyanate resin	1.37	4.27 (620)	10.2 (1480)
Foam / epoxy resin	1.31	10.10 (1460)	13.7 (1990)
CVD foam / aluminum	2.34	26.50 (3850)	33.3 (4830)

5.3 Thermal Testing

5.3.1 Background

Recently, light-weight, porous carbon materials have received extensive attention as a potential heat sinks or compact heat exchangers in light of their high thermal efficiency and light weight. By using the porous carbon foam as the core material in a heat exchanger used for the electronic module in power inverters, the overall heat transfer coefficient under forced convective flow is enhanced significantly.

Electronically powered components generally produce more heat than they can dissipate naturally. Therefore, the heat sink is used to remove heat and to protect the electronic device from the heat it produces as a by-product of normal operation. With the heat being removed, the device will be able to operate without exceeding its operating temperature. A generally accepted fact is that, for every 10°C reduction in the

temperature, operating life doubles (Lorenzetti, 1999) Laboratory tests of heat sinks with carbon foam cores for electronic inverter modules have been designed, built, and tested In order to improve the design of electronic inverter modules by maximizing heat transfer yet reducing the pressure drop, heat exchanger tests have been performed to provide valuable knowledge in terms of foam morphology, geometry selection of core materials, optimum flow rate, and type of cooling fluid

5 3 2 Prototype Heat Sink

A prototype heat sink for electronic modules in power inverter applications has been fabricated and tested The prototype heat sink is a forced convection heat exchanging system with single pass air-flow through carbon foam core within an aluminum casing The objective of the prototype heat sink testing is to obtain the overall heat transfer coefficient at different air-flow rates and electronic module power supplies.

5 3 2 1 Experimental Setup

The experimental setup consists of a rectangular aluminum casing containing a carbon foam block, one inlet and one outlet air-flow pipe, five thermocouples, one temperature indicator, one heating pad, one voltage-controlled transformer, one flow meter, and one pressure gage. The entire experimental setup is shown in Figure 5 6. The heating pad, which is a flexible insulated heater, is used to simulate the isothermal heat source and is attached to the bottom of the 14 75 x 7.625 x 1 25 inch aluminum casing,

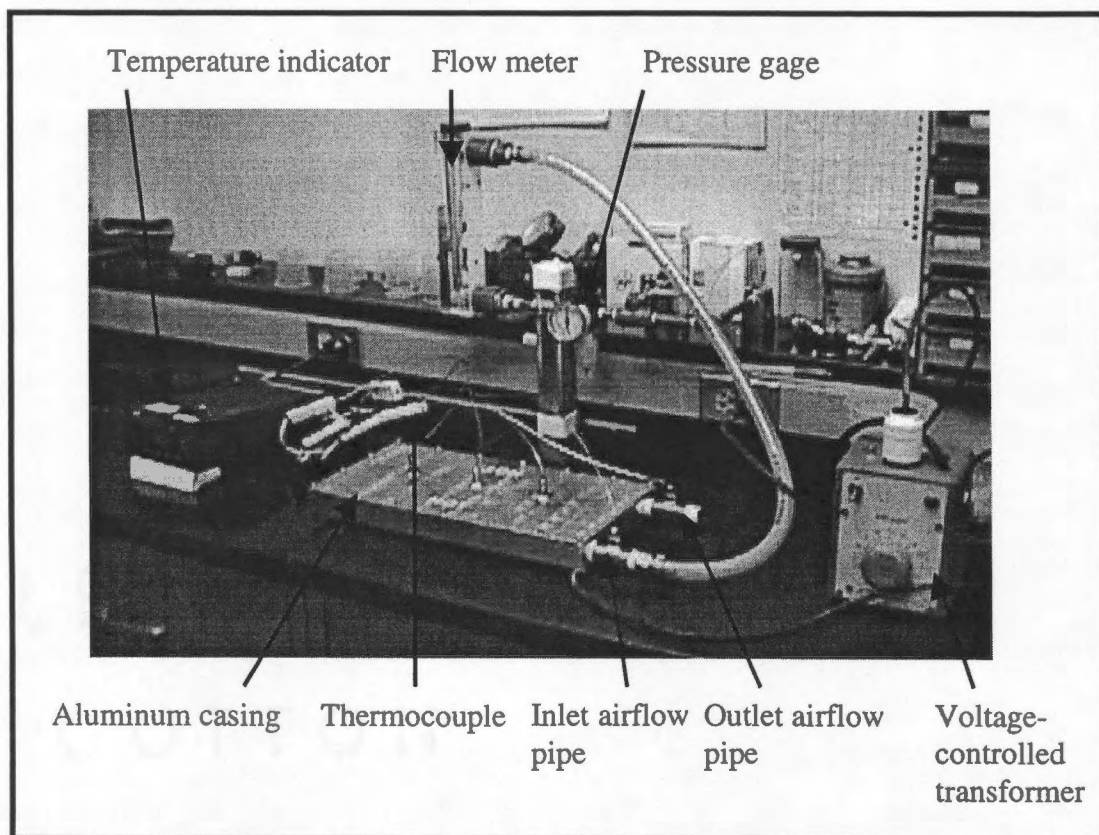


Figure 5.6: Experimental setup for prototype heat sink.

see Figure 5.7 Air is forced through the system to remove the heat from the power electronic module. A 12.75 x 3.875 x 0.8125 inch of rectangular carbon foam block is placed at the center of the aluminum casing, leaving two rectangular channels on each side to serve as the inlet and outlet ducts, which are connected to the pipe fittings. The carbon foam block is mounted on the side of the aluminum casing with the heat source. As illustrated in Figure 5.8, there are twenty-four circular blind holes with a diameter of 0.0625 inch on both sides of the carbon foam block exposed to the air flow channel. These blind holes are only 2.9 inch in depth, which is $\frac{3}{4}$ of the foam width, and they are for minimizing the pressure drop of airflow across the foam block. Furthermore, the blind holes will ensure that the air will flow through the foam block effectively. Five thermocouples are used to measure the temperatures at the pipe entrance, pipe exit, and three equally spaced locations within the carbon foam block, as shown in Figure 5.8.

A schematic of the test apparatus is shown in Figure 5.9. Cold air flows from the inlet pipe into the rectangular channel on one side of the aluminum casing. Then, the cold air is forced through the blind holes and diffuses through the open pore carbon foam to remove the heat from the simulated heat source. After diffusing through the foam, the air exits at the blind holes and flows into the atmosphere through the rectangular channel on the opposite side of the casing and through the outlet pipe, as shown in Figure 5.10.

The air-flow rate studied ranges from 0 to 510 lpm (liter/minute) and the heat transfer rate simulated ranged from 100 to 180 Watts. The experimental data, including the inlet pressure, inlet volumetric flow rate, and the temperature of the inlet air, outlet air, and three equally spaced locations within the carbon foam block, are collected

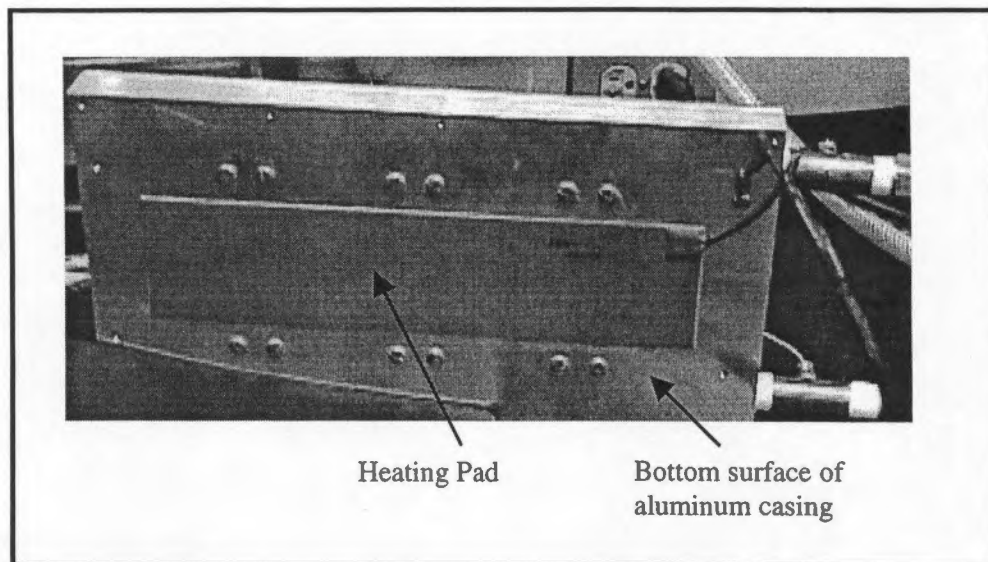


Figure 5.7: Heating pad attached to the bottom of aluminum casing.

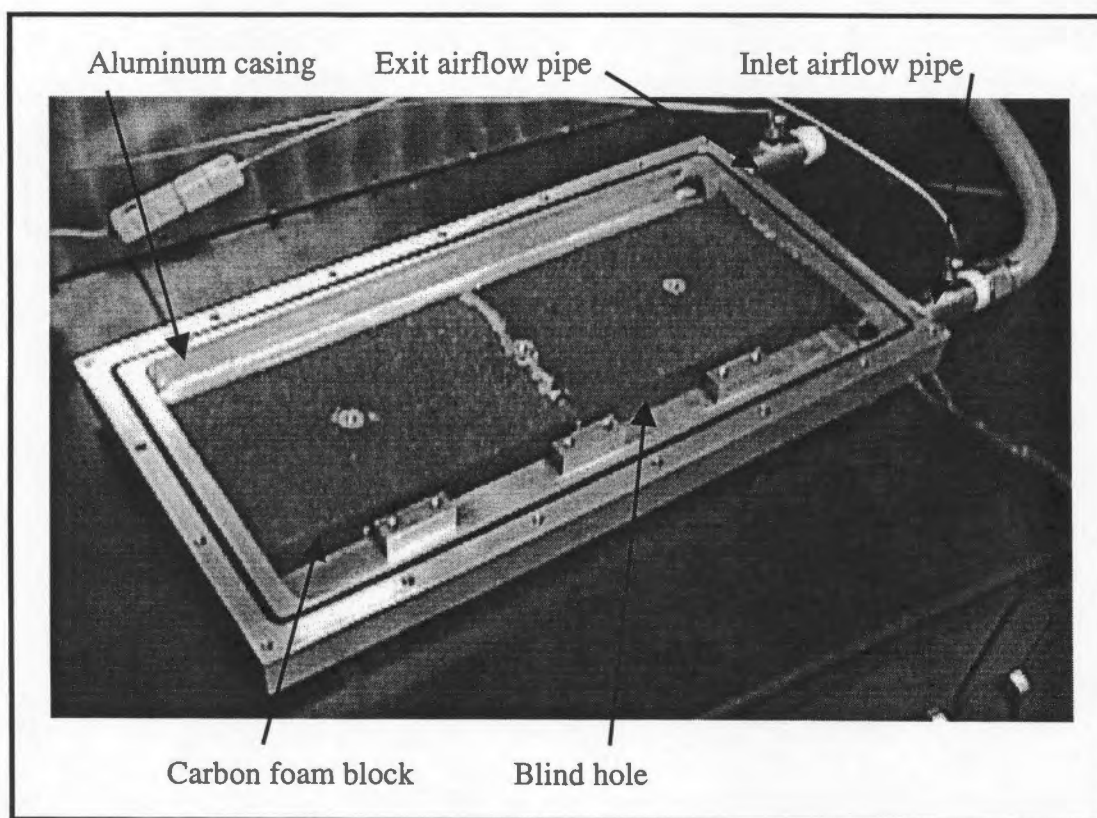


Figure 5.8: Carbon foam block situated in the center of aluminum casing.

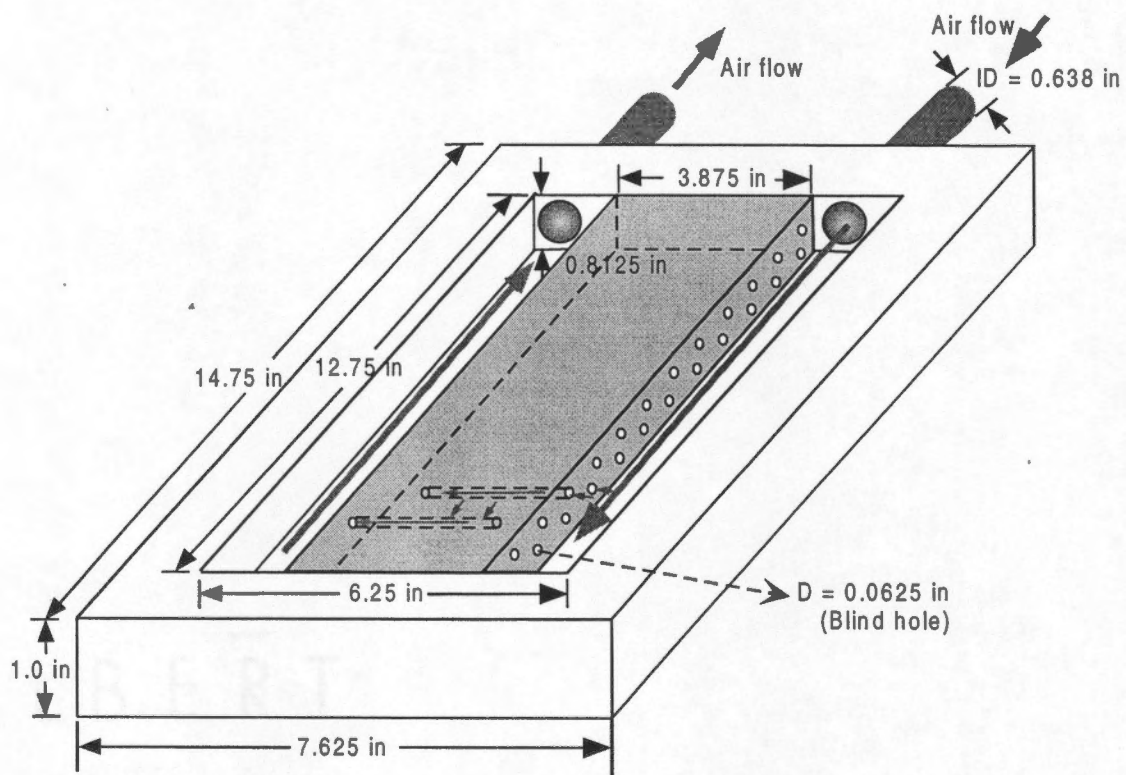


Figure 5.9: Prototype heat sink using a carbon foam block as the core material.

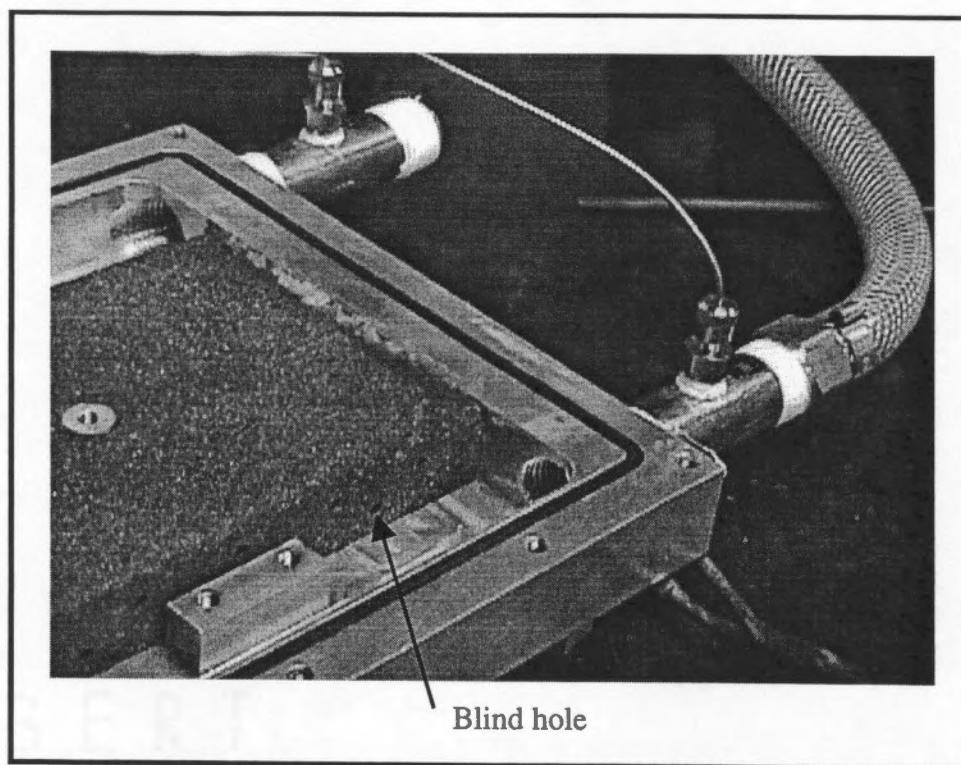


Figure 5.10: Blind holes on both sides of the carbon foam block.

after the entire system reaches steady state for each constant air-flow and heat transfer rate

Like the heat exchanger systems used in practice, the overall heat transfer coefficient for the prototype heat sink system, U_o , can be given by (Kreith and Bohn, 1993)

$$q = U_o A_{\text{foam}} (\Delta T_{\text{lm}}) \quad (5.6)$$

where q is the heat transfer rate, A_{foam} is the heat transfer surface area of the foam (i.e., the surface that is attached to the aluminum casing), and ΔT_{lm} is the log mean temperature difference (LMTD) that can be defined by (Kreith and Bohn, 1993)

$$\Delta T_{\text{lm}} = \frac{\Delta T_{\text{in}} - \Delta T_{\text{out}}}{\ln \left[\frac{\Delta T_{\text{in}}}{\Delta T_{\text{out}}} \right]} = \frac{(T_s - T_{f,\text{in}}) - (T_s - T_{f,\text{out}})}{\ln \left[\frac{(T_s - T_{f,\text{in}})}{(T_s - T_{f,\text{out}})} \right]} \quad (5.7)$$

with ΔT_{in} and ΔT_{out} being the inlet and outlet temperature differences, respectively, $T_{f,\text{in}}$ is the inlet cooling fluid temperature, $T_{f,\text{out}}$ is the outlet cooling fluid temperature, and T_s is the heating temperature. The LMTD is generally only an approximation in practice because the temperature of the cooling fluid in a heat exchanger is usually not constant but varies from point to point (Kreith and Bohn, 1993).

5.3.2 2 Results and Discussions

Table 5.6 gives the air-flow rate, inlet pressure, power, inlet air temperature, $T_{f,in}$, outlet air temperature, $T_{f,out}$, heating temperature, T_s , the log mean temperature difference, ΔT_{lm} , and the overall heat transfer coefficient, U_o of the prototype heat sink under forced convection. The maximum overall heat transfer coefficient is $806 \text{ W/m}^2\text{K}$ at the volumetric flow rate of 510 lpm and the power supply of 103 W.

The overall heat transfer coefficient, U_o , is plotted as a function of volumetric flow rate at different power supply levels in Figure 5.11. One finds that U_o increases with increasing volumetric flow rate. Although U_o should be a constant for various power supplies, the results show a consistently decreasing U_o as the power supply level increases. This may be due to the fact that the prototype heat sink was tested at a low power range and the assumption of no heat loss might not be reasonable since heat losses took place on all surfaces of the aluminum casing that were exposed to the atmosphere. Therefore, the energy delivered by heater is only partially picked up by the air flow.

5.3.3 Heat Exchanger

In addition to the prototype heat sink for power inverter module, several different heat exchanger designs using carbon foam as the core material have been fabricated and tested. The heat exchanger systems are flexible in replacing carbon foam core geometries and are capable of high power supply or heat transfer rate processes. The main objective of the heat exchanger testing is to obtain the overall heat transfer

Table 5 6: Experimental data of the prototype heat sink under forced convection.

Air-flow rate (lpm)	Inlet pressure (kPa)	Power (W)	T _{f,in} (°C)	T _{f,out} (°C)	T _s (°C)	ΔT _{lm} (°C)	U _o (W/m ² K)
142	5.17	103	23.1	43.0	43.8	6.1	528
		116	23.4	47.0	48.2	7.8	466
		129	23.7	53.2	54.5	9.3	433
		141	23.9	57.6	59.2	10.9	407
		154	24.2	63.6	65.6	13.0	372
		167	24.4	70.2	72.4	14.9	353
		180	24.8	76.1	78.7	16.9	334
283	13.8	103	21.9	33.9	34.6	4.1	779
		116	22.0	35.6	36.7	5.2	692
		129	22.1	38.3	39.6	6.2	647
		141	22.2	41.3	42.9	7.5	595
		154	22.4	45.1	46.8	8.5	568
		167	22.4	48.6	50.6	9.9	530
		180	22.5	52.6	54.8	11.2	504
425	31.0	103	21.8	29.8	31.1	4.1	794
		116	21.8	31.4	33.0	4.9	736
		129	21.8	33.5	35.3	5.8	695
		141	21.9	36.1	38.1	6.8	654
		154	21.9	38.5	40.9	8.0	603
		167	22.0	41.2	43.9	9.2	572
		180	22.1	44.3	47.3	10.4	541
510	48.3	103	21.8	28.7	30.2	4.0	806
		116	21.7	30.1	31.8	4.7	770
		129	21.7	31.5	33.6	5.6	714
		141	21.8	34.2	36.4	6.6	677
		154	21.8	36.4	38.9	7.6	637
		167	21.8	38.4	41.3	8.7	602
		180	21.8	40.6	44.1	10.2	556

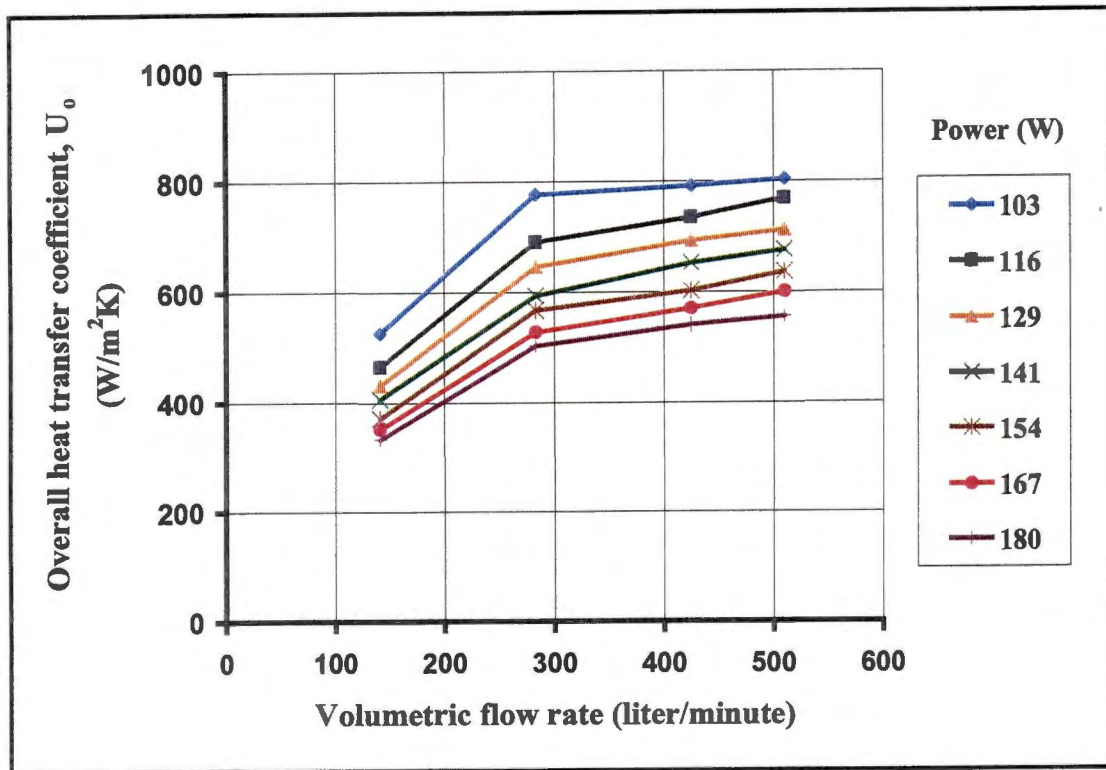


Figure 5.11: Overall heat transfer coefficient as a function of volumetric flow rate for varying power supply.

coefficient of and the pressure drop across the carbon foam at different flow rates, heat transfer rates, and foam core geometries.

5.3.3.1 Experimental Setup

The entire experimental setup is shown in Figure 5.12 and consists of an aluminum plate and a carbon foam block encapsulated within a nylon test chamber, which provides the thermal insulation. As it is shown in Figure 5.13, the inlet and the outlet pipes are aligned and connected into the flow channel whose dimension is 2 x 4 x 1.25 inch. The nylon test chamber is sealed with o-rings and pressure tap holes are drilled into one side of the nylon test chamber in order to measure the pressure drop across the carbon foam block. The 2 x 2 x 1.25 inch carbon foam block is mounted onto the 4 x 6 x 0.375 inch aluminum plate by brazing with Superbraz[®], a high thermal conductivity braze manufactured by Materials Resources International, Inc, see Figure 5.14. Two cartridge heaters, which are contained within an aluminum block of 2 x 2 x 0.75 inch, is used to create isothermal heating and is mounted on the top of the aluminum plate. Four thermocouples are used to measure the temperature of the inlet air, outlet air, aluminum plate, and heater.

The schematic of the heat exchanger systems tested is shown in Figure 5.15. As the heat source is simulated by using the cartridge heaters, the cooling fluid is forced into the carbon foam block to remove heat from the system. It is noted that there is no gap between the flow channel and the carbon foam block. Within the flow channel, the air is

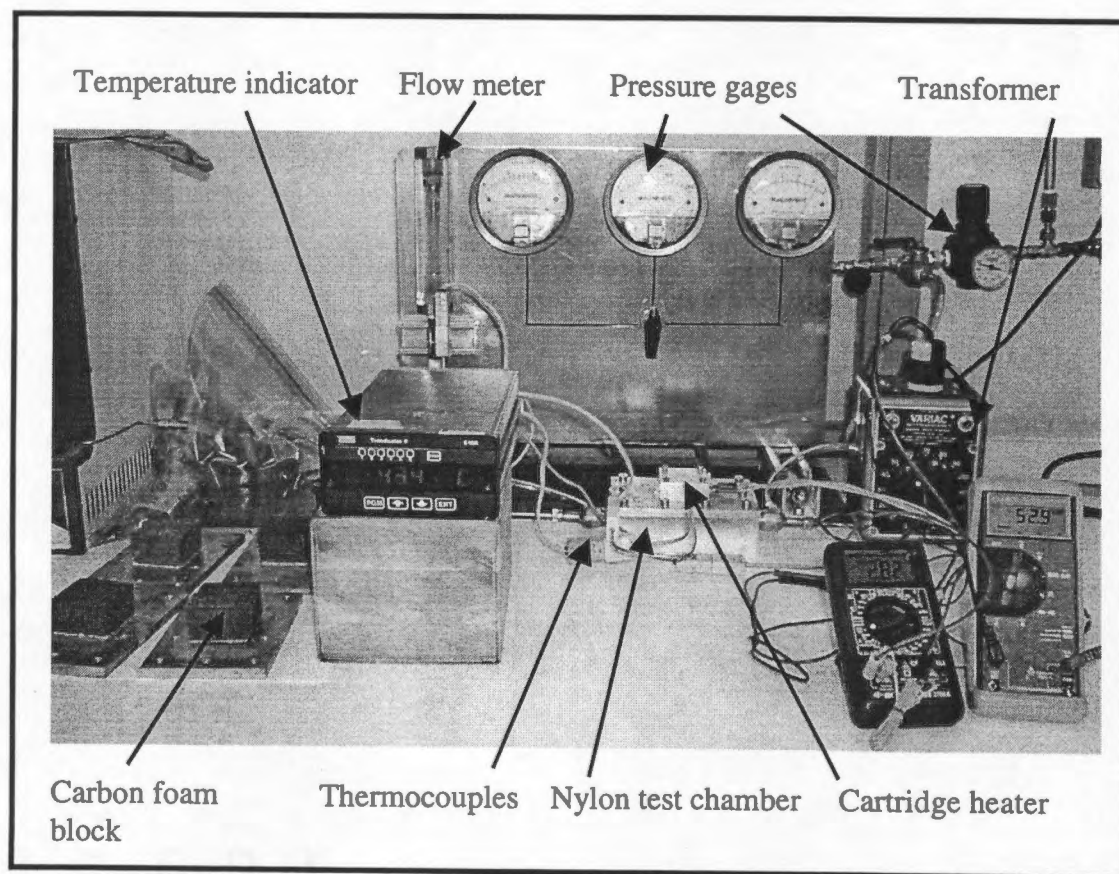


Figure 5.12: Setup for heat exchanger experiments.

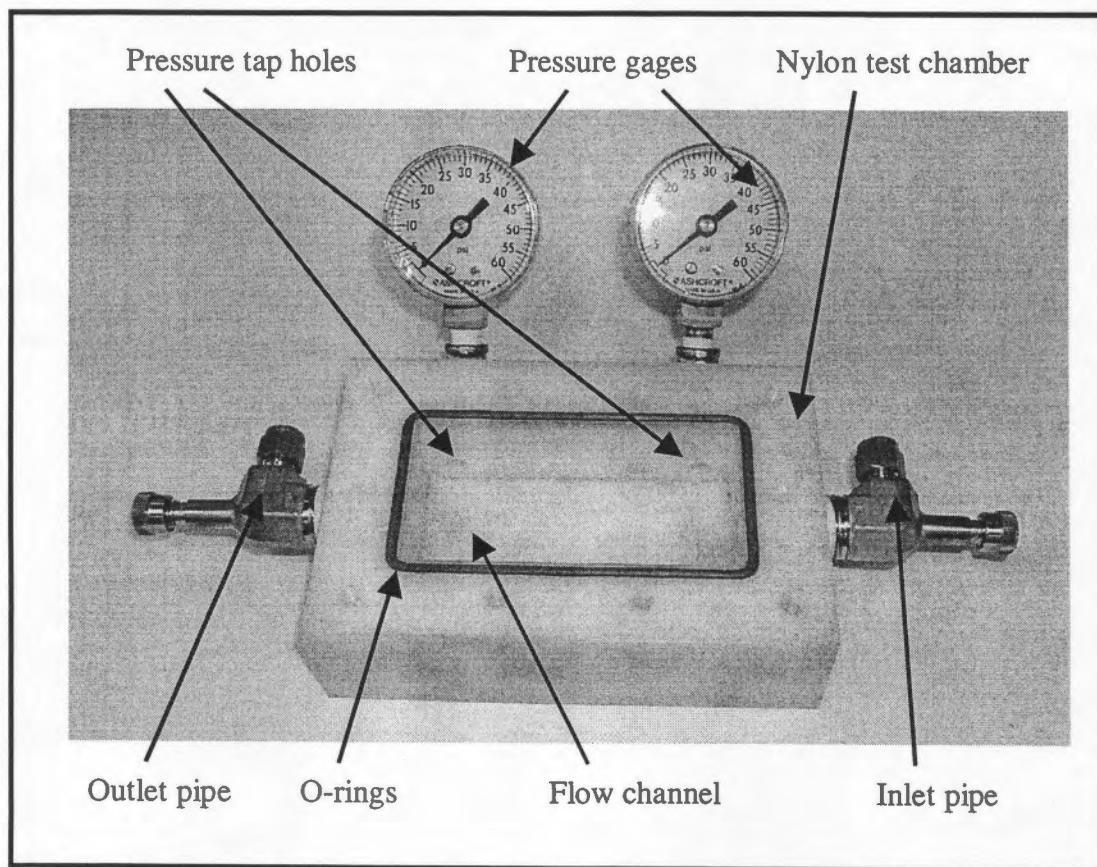


Figure 5.13: Nylon test chamber with 2 x 4 x 1.25 inch flow channel.

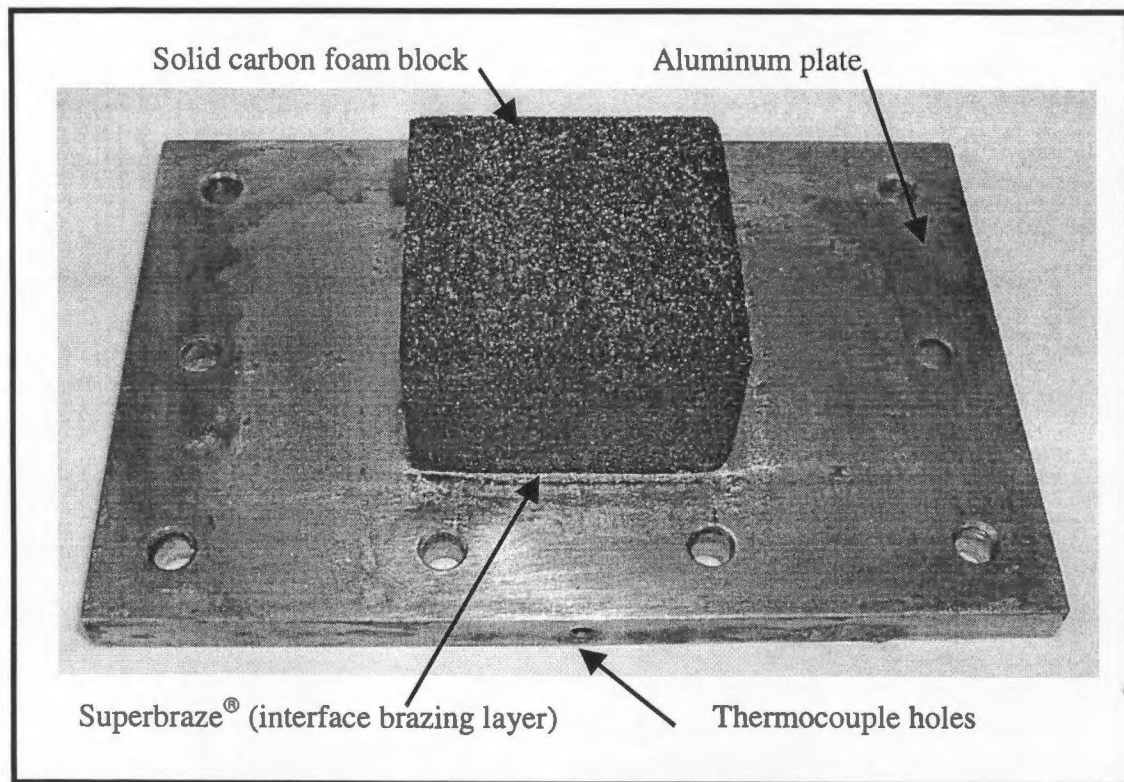


Figure 5.14: Carbon foam block mounted on an aluminum plate.

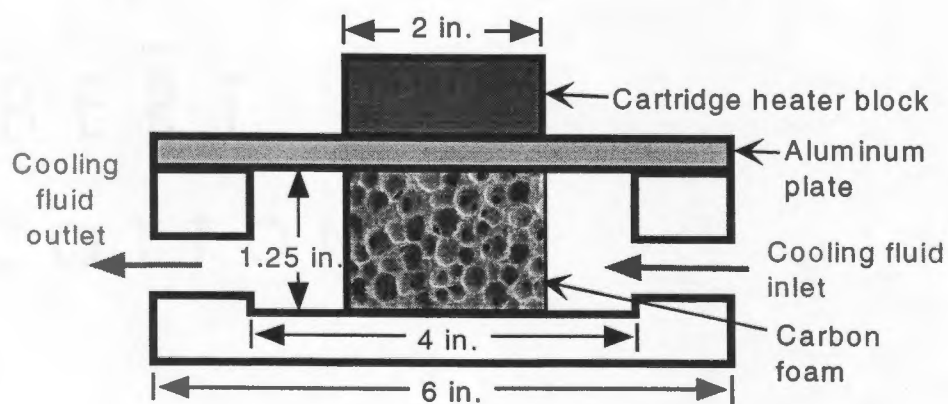


Figure 5.15: Schematic diagram of the heat exchanger system.

forced through the open pore carbon foam block and exits through the opposite side of the nylon chamber into the atmosphere

The cooling fluid used in the experiment is ambient air. The volumetric flow rate ranges from 0 to 425 lpm and the heat transfer rate ranged from 0 to 600 Watts. The experimental data, including the inlet pressure, outlet volumetric flow rate, voltage, current, and temperature of the inlet air, outlet air, aluminum plate, and heater, is collected for each constant air-flow and heat flux in a steady state temperature system. Like the prototype heat sink, Eq (5.6) and Eq (5.7) are utilized to compute the overall heat transfer coefficient, U_o , and the log mean temperature difference, ΔT_{lm}

5.3.3.2 Results and Discussions

Thermal testing is carried out using five carbon foam block samples in the same heat exchanger test chamber. The samples used have varying fabrication processing conditions, foam densities and geometrical designs. The samples are labeled 1 to 5 with all of the detailed dimensions and foam densities listed in Table 5.7. It is noted that sample #4 has 20 vertical fins aligned parallel to the air flow (see Figure 5.16). All fins have a thickness of 0.172 cm and a depth of 1 inch. The gap between two fins is 0.086 cm.

Figure 5.17 presents the overall heat transfer coefficient, U_o , as a function of power supply at different volumetric flow rates for sample #2 carbon foam block. U_o shows an approximately constant relationship for varying power supply. The heat loss to atmosphere on the surface of the heater is estimated to be insignificant. In order to

Table 5.7: Dimensions and densities of the carbon foam blocks tested in the heat exchanger chamber.

Carbon Foam Block	Density (g/cm ³)	Height cm (inch)	Length cm (inch)	Width cm (inch)
Sample #1	0.43	3.16 (1.24)	5.04 (1.99)	5.12 (1.98)
Sample #2	0.47	3.20 (1.26)	5.14 (2.02)	5.07 (2.00)
Sample #3	0.48	3.17 (1.25)	5.24 (2.06)	5.05 (1.99)
Sample #4	0.49	3.16 (1.24)	5.02 (1.97)	5.06 (1.99)
Sample #5	0.46	3.21 (1.26)	5.21 (2.05)	5.04 (1.98)

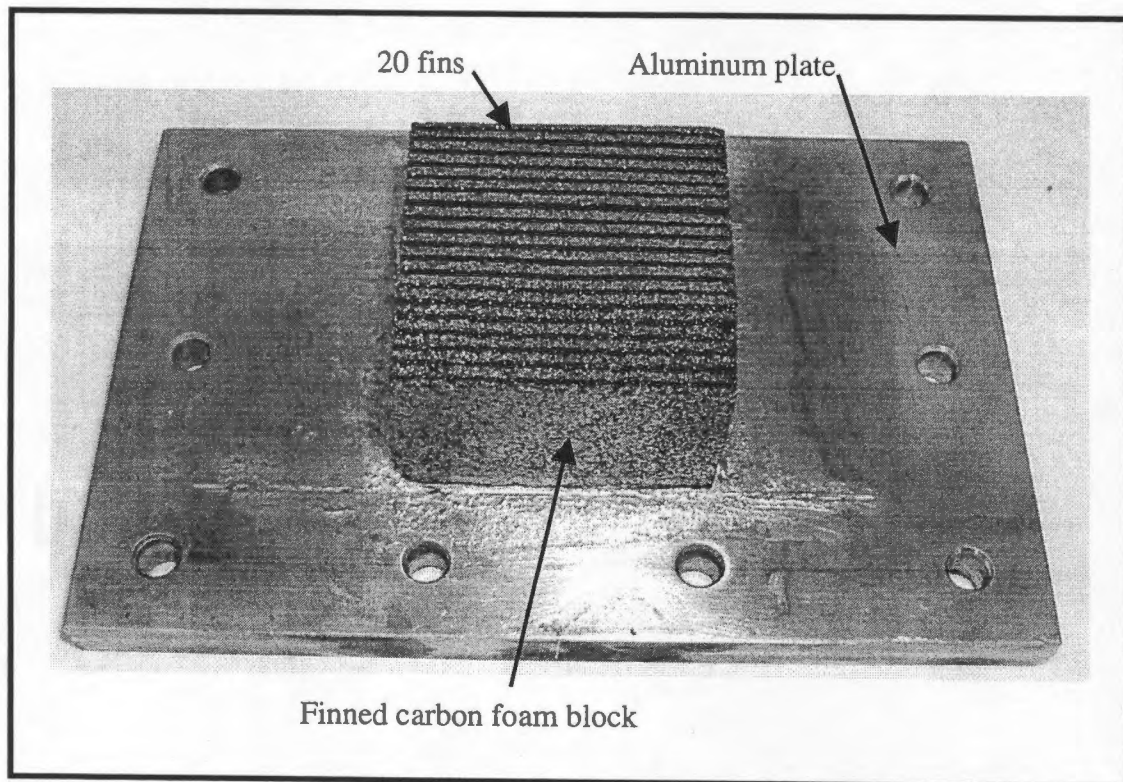


Figure 5.16: Sample #4, finned carbon foam block.

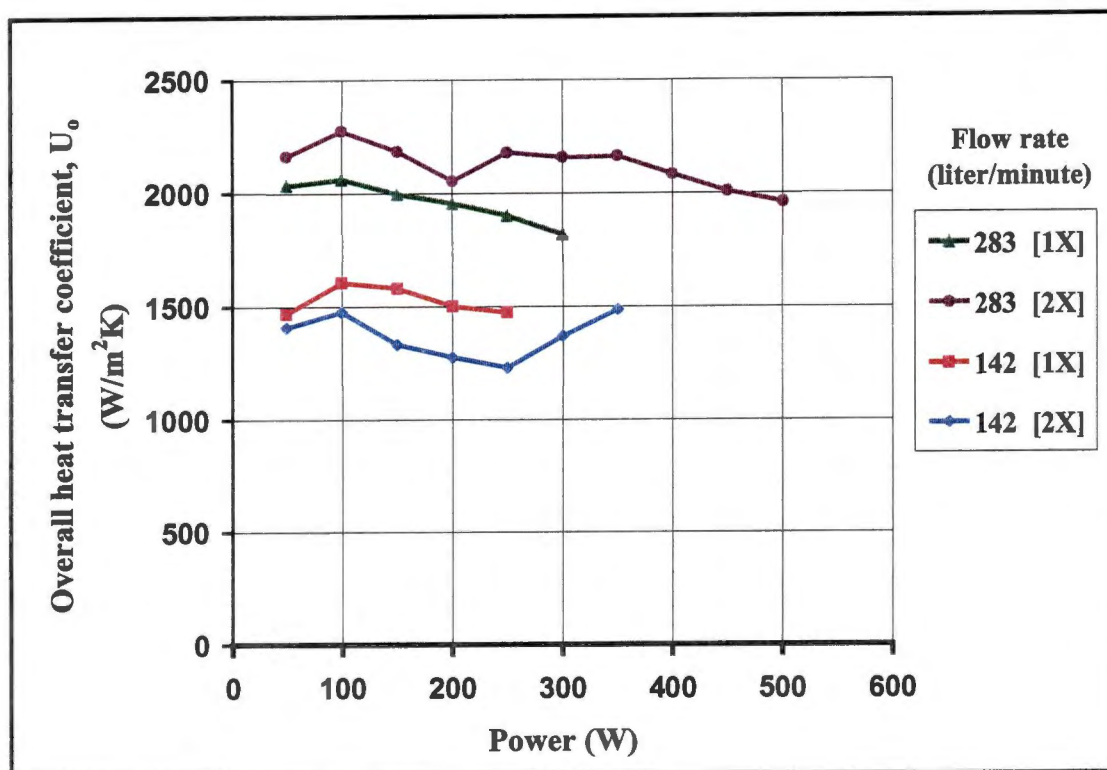


Figure 5.17: Overall heat transfer coefficient as a function of power for two repeated experiments on sample #2 carbon foam block (density = 0.47 g/cm^3).

check the consistency of the experiment, the test of sample #2 is repeated and the results are also shown in Figure 5 17. At a constant power supply and volumetric flow rate, the comparison between the first experiment [1X] and the second experiment [2X] clearly shows that there is a difference of 60 - 500 W/m²K in U_o . The main reason for the variation in the test results may be the varying ambient air inlet temperature and humidity (Klett, 1999a)

A typical temperature of the heater versus power for different volumetric flow rates is shown in Figure 5 18. At a constant power, the temperature of the heater, T_s , decreases as the flow rate increase. At a power of 50 W, T_s is cooled down from about 110°C at 0 lpm to about 40°C with air flow of 396 lpm. Furthermore, the temperature of the heater, T_s , without air flow is extrapolated to be about 750°C; while T_s with 396 lpm air flow is only 150°C at power of 400 W (power density of 100 W/in²).

The overall heat transfer coefficient, U_o , is plotted as a function of volumetric air flow rate at various power supply levels for sample #2, as shown in Figure 5 19. Similar curves are plotted in Figure 5.20 for sample #3 foam block with 0.48 g/cm³ density. It is seen that U_o increases with increasing volumetric flow rate in both of the figures. Both foam blocks, which had the same processing condition and approximately the same density, exhibit very different overall heat transfer coefficient, U_o . The U_o of the foam block with 0.47 g/cm³ density (sample #2) is much higher at a range of 1,980 – 2,470 W/m²K at air-flow rate of 14 cfm. The reason for the difference is not clear. It may be due to the variance in ambient air inlet temperature, humidity, or uncertainty in the experimental setup. It may also be due to the slightly difference in the dimensions of the

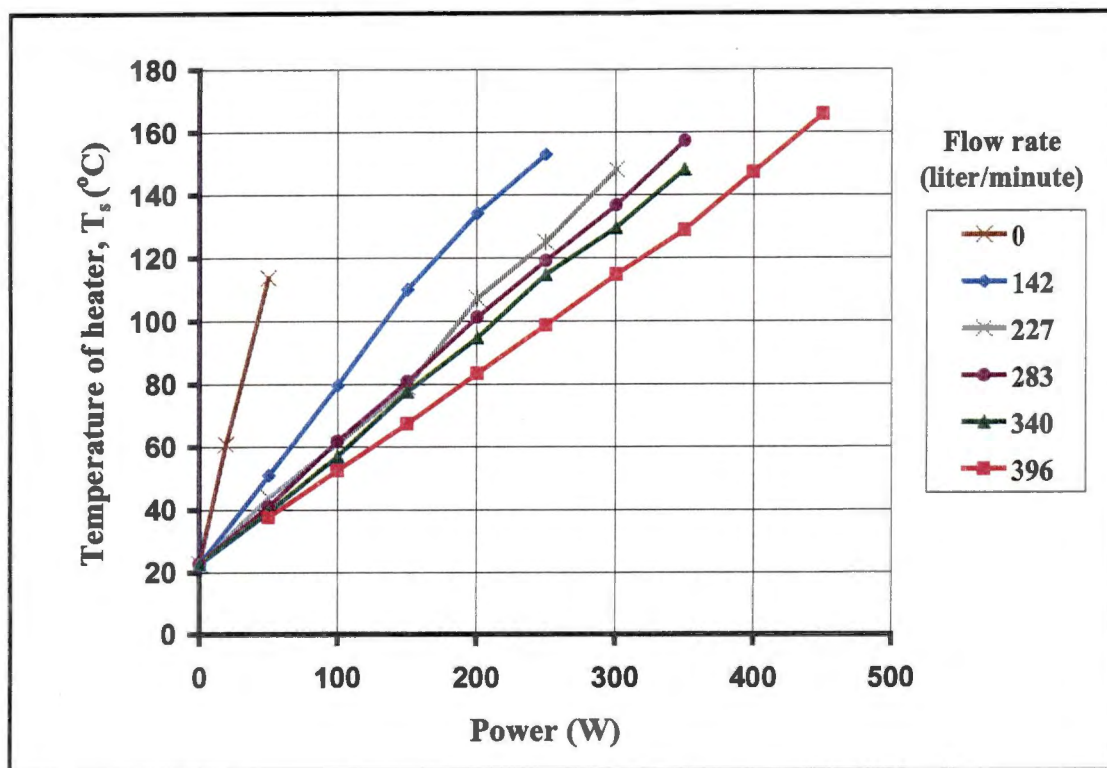


Figure 5.18: Temperature of heater as a function of power supply for a typical heat exchanger utilizing carbon foam block.

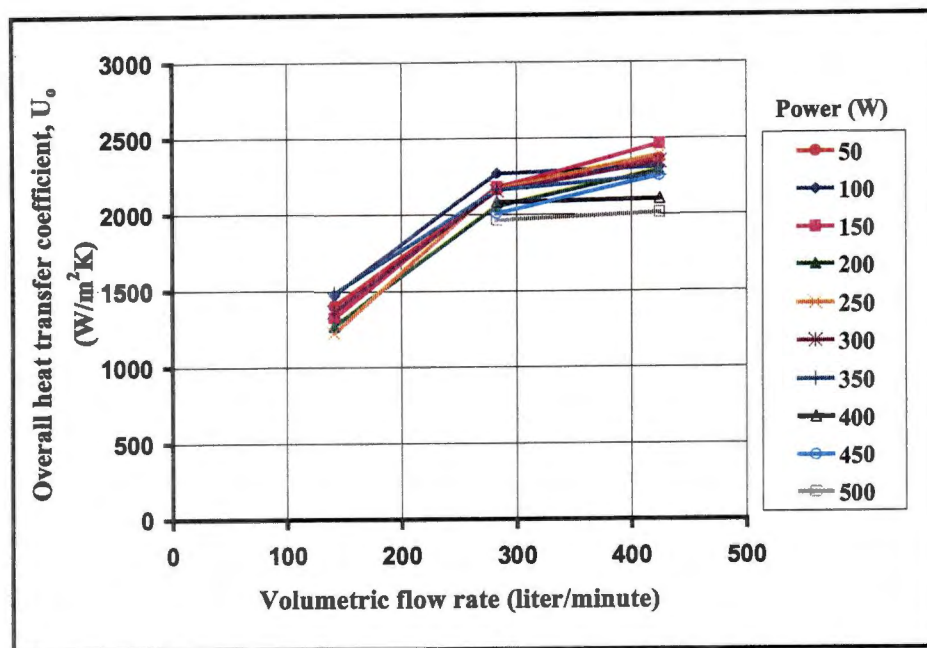


Figure 5.19: Overall heat transfer coefficient as a function of flow rate at varying power for heat exchanger using sample #2 foam (density = 0.47 g/cm³).

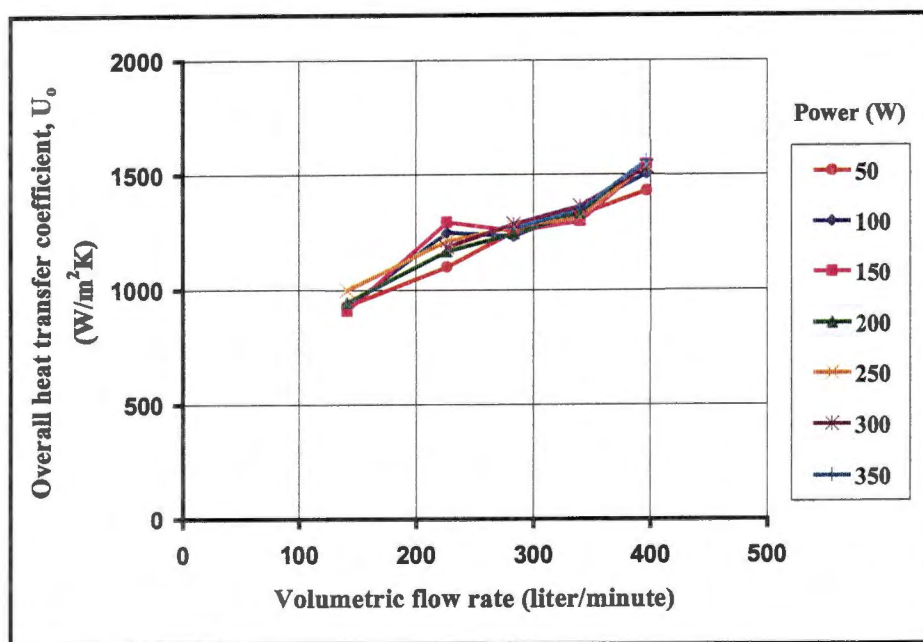


Figure 5.20: Overall heat transfer coefficient as a function of flow rate at varying power for heat exchanger using sample #3 foam (density = 0.48 g/cm³).

sample foam blocks. Sample #2 foam block has a slightly larger height and width than sample #3, which reduces the gap between the edge of the foam block and the test chamber. Although a thin layer of grafoil is used to seal the gap, a more effective way is necessary to reduce the uncertainty of the experiments.

Likewise, the overall heat transfer coefficient, U_o , versus volumetric flow rate for two carbon foam blocks under different processing conditions. sample #1 (density = 0.43 g/cm^3) and sample #5 (density = 0.46 g/cm^3) are shown in Figure 5.21 and 5.22, respectively. Since the foam block of density 0.43 g/cm^3 (sample #1) has a significantly lower density, U_o is much lower compare to the other carbon foam blocks. On the other hand, the carbon foam block of density 0.46 g/cm^3 (sample #5) has the highest U_o (between 2710 and $3040 \text{ W/m}^2\text{K}$ at 396 lpm).

Furthermore, the overall heat transfer coefficient, U_o , as a function of volumetric flow rate for sample #4, the finned foam block of density 0.49 g/cm^3 , is shown in Figure 5.23. The rate of increase in U_o is lower with increasing air flow rate in comparison with those of the solid foam blocks. Moreover, the overall heat transfer coefficient for the finned foam design is about one-half of that for the solid blocks. On the other hand, the finned design essentially eliminates the pressure drop across the foam block since the air does not have to be forced through the foam block. The pressure drop across the foam block is too low to be detected from the pressure gage of the current experiment setup and is estimated to be less than 0.018 psi/in .

Since the overall heat transfer coefficient, U_o , for different heat transfer rates falls within a narrow range at a constant air flow rate, the average overall heat transfer

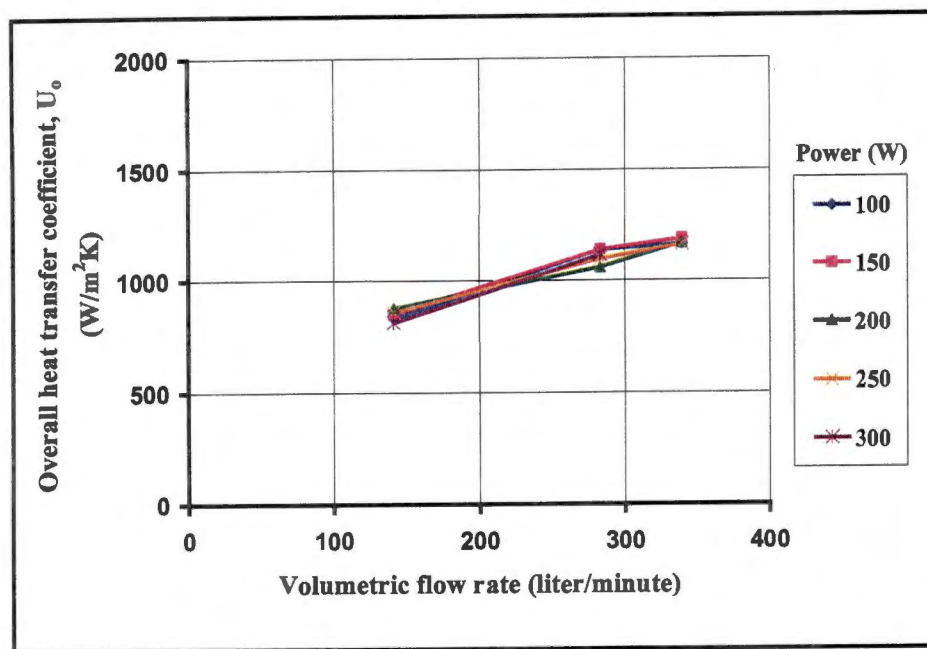


Figure 5.21: Overall heat transfer coefficient as a function of flow rate at varying power for heat exchanger using sample #1 foam (density = 0.43 g/cm³).

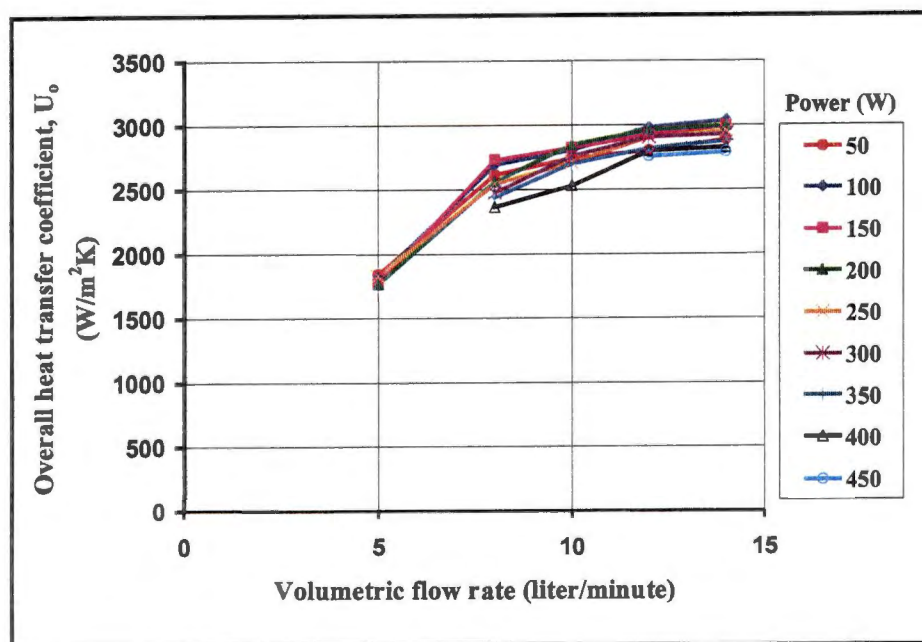


Figure 5.22: Overall heat transfer coefficient as a function of flow rate at varying power for heat exchanger using sample #5 foam (density = 0.46 g/cm³).

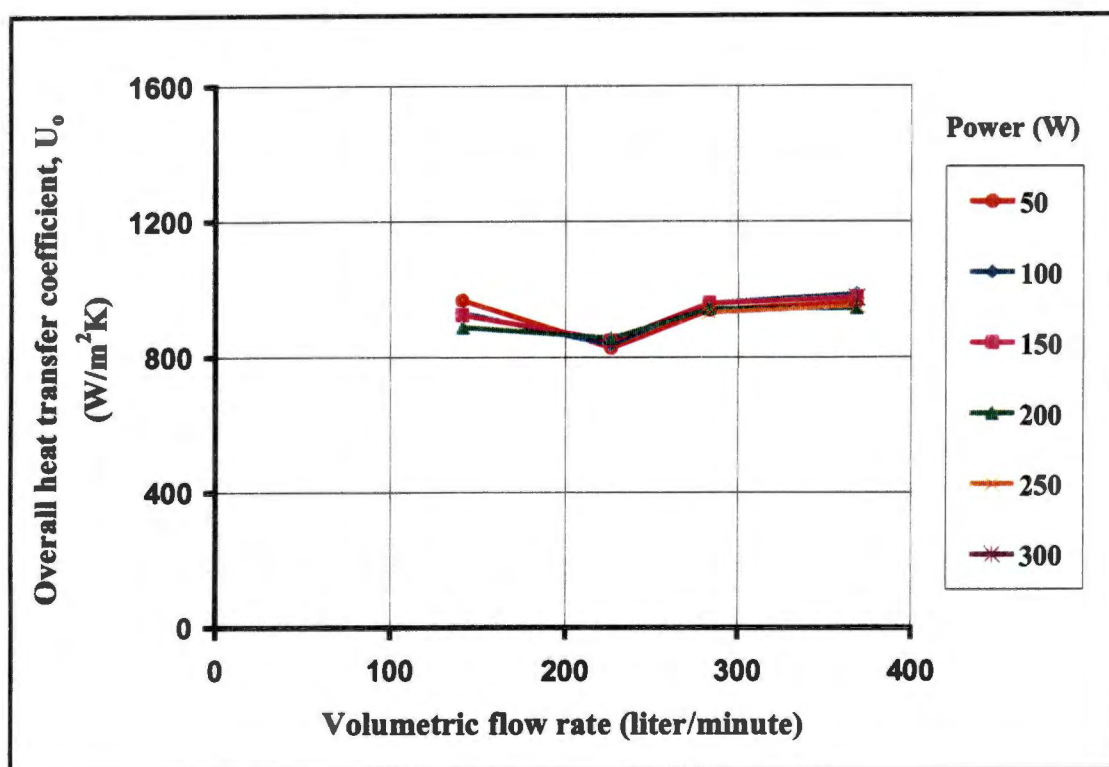


Figure 5.23: Overall heat transfer coefficient as a function of flow rate at varying power for heat exchanger using sample #4 finned foam (density = 0.49g/cm³).

coefficient is computed for all the foam blocks. Figure 5.24 represents the average overall heat transfer coefficient as a function of air flow rate for all the foam blocks tested. The finned foam block offers nearly zero pressure drop and is still able to provide an overall heat transfer coefficient of $900 \text{ W/m}^2\text{K}$. Moreover, the high U_o ($830 - 2880 \text{ W/m}^2\text{K}$) is very superior to that of a standard automobile radiator of $30 \text{ W/m}^2\text{K}$ (Klett, 1999a). Therefore, the overall heat transfer coefficient of carbon foam-based heat exchanger can be two-orders of magnitude greater than the conventional heat exchanger and may be dramatically lighter and smaller (Klett *et al.*, 2000b).

The pressure drop per unit length as a function of free stream velocity for sample #2 and #3 with densities of 0.47 g/cm^3 and 0.48 g/cm^3 , respectively, is illustrated in Figure 5.25. The pressure drop rises significantly with increasing free stream velocity. The foam block with the slightly higher density of 0.48 g/cm^3 has a slightly higher pressure drop than the other. In addition, the pressure drop changes with temperatures since the density of air is lower at high temperatures. Figure 5.26 shows the pressure drop versus temperature of the heater at different air flow rates for a typical heat exchanger test utilizing a solid foam block. For all air flow rates, the pressure drop decreases as the temperature of the heater increases.

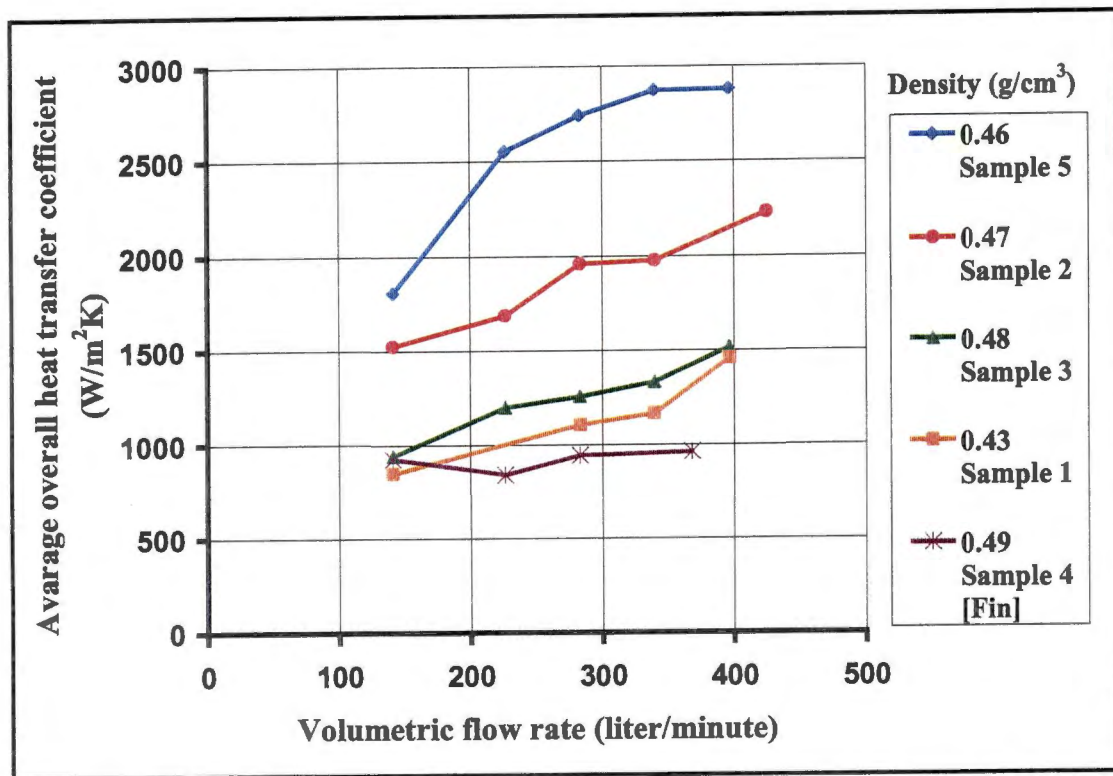


Figure 5.24: Average overall heat transfer coefficient as a function of flow rate for all the heat exchangers tested.

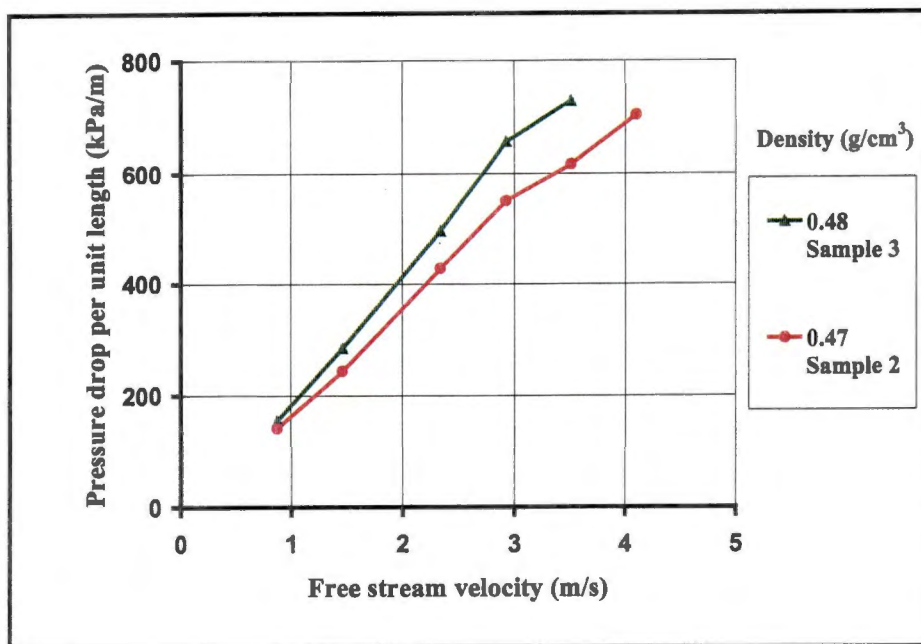


Figure 5.25: Pressure drop per unit length as a function of free stream velocity for heat exchangers utilizing carbon foam.

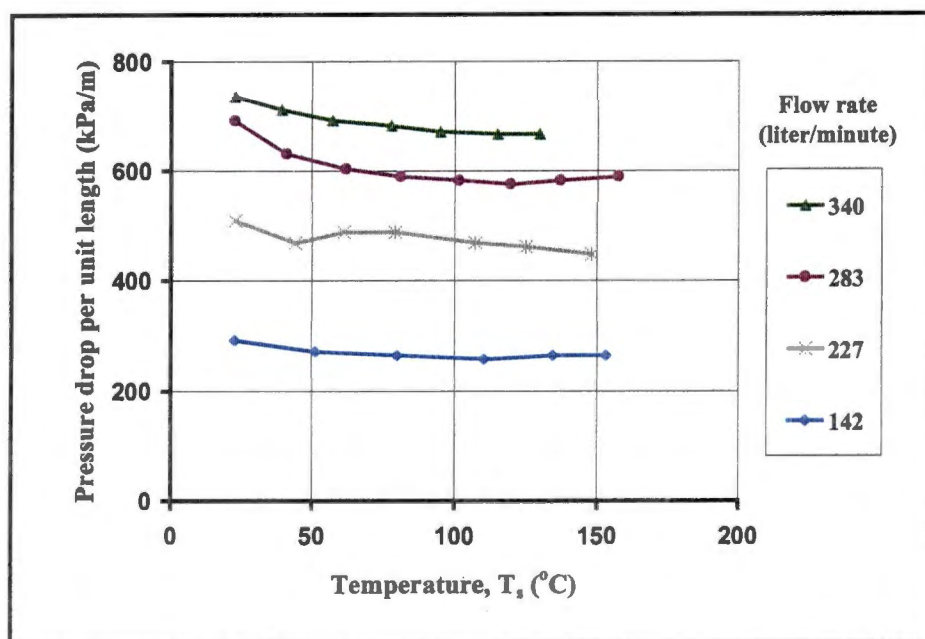


Figure 5.26: Pressure drop per unit length as a function of temperature of heater at different air flow rates for a typical heat exchanger utilizing carbon foam.

CHAPTER 6

Conclusions and Recommendations

6.1 Conclusions

The modeling and testing of mesophase pitch-derived carbon foams with high thermal conductivities are reported in the present work. The lightweight, isotropic thermal conductivity as well as mechanical properties, and the continuous, open-cell graphitic networks within the foam may lead to novel and interesting methods of thermal management (Klett, 1999b)

For the modeling of effective thermal conductivity, the tapered, anisotropic strut model provides closed-form, accurate predictions of the overall thermal conductivity of the porous carbon foam. The struts perpendicular to the heat flow direction are found to provide very little contribution to the overall heat conduction in the porous carbon foam. In addition, the disrupted strut junctures significantly reduce the overall thermal conductivity of carbon foam. Furthermore, it is concluded that the overall thermal conductivity of foam composed of randomly oriented cubic cells is identical to that of the foam consisting of aligned cubic cells regardless of the heat flow direction

A closed-form relationship between the pressure drop and vital microstructural parameters, such as porosity and pore size, for a carbon foam core under forced convection has been developed using the spherical void model and has been validated for low-velocity flows. The geometric model is fairly representative of the carbon foam morphology and the pressure drop across the foam is linearly proportional to the average free stream velocity, which is further supported by the Darcy's law for an unbounded

porous medium with small pore size at low flow rates. At high flow velocities, the relationship between the pressure drop and free stream velocity of the fluid becomes nonlinear due to microscopically turbulent flow and inertia effects, and the modification to the momentum equation to include a quadratic velocity term is necessary as proposed in the porous medium model. As it is predicted by the spherical void model, an increase in the foam pore size significantly decreases the pressure drop across the foam. In addition, the pressure drop across the foam using air as the cooling fluid is considerably lower than water in spite of the decrease in the overall heat transfer coefficient for convective heat transfer.

Among the foam-based composites tested, the CVD foam/aluminum composite had the highest tensile and flexural strength, while the foam/epoxy composites had the highest strength among the foam/*polymer* composites due to good interfacial bonding (Klett, 1999a). In addition, the failure mode of all the foam-based composites is highly brittle without much inelastic deformation.

Thermal testing on a prototype heat sink utilizing carbon foam has been carried out in order to simulate the heat transfer in high power electronic modules and to determine the overall heat transfer coefficient of the heat sink. Several heat exchangers using foams with different designs are tested as well. The overall heat transfer coefficient as well as pressure drop data are obtained and the effects of density and geometry of the foam are examined. Based on the present experimental results, the overall heat transfer coefficient of $3000 \text{ W/m}^2\text{K}$ for heat exchanger using carbon foam core is significantly higher than that of a standard automobile radiator of $30 \text{ W/m}^2\text{K}$.

Although the heat transfer coefficient for the finned foam block geometry is reduced by approximately 50%, the pressure drop is eliminated.

For a typical heat exchanger, the heat transfer rate, q , and pressure drop (pumping power) can be approximately expressed in term of velocity, v , by the following proportionally (Bell, 1990)

$$\Delta p \propto v^\lambda \quad (6.1)$$

$$q \propto v^j \quad (6.2)$$

where $\lambda > j$ typically. Obviously, it is essential to design an optimum foam performance as the core material for customize heat exchanger design theoretically and experimentally in order to optimize the overall heat transfer coefficient while maintaining relatively low the pressure drop across the carbon foam. Hence, the spherical void model can be used as a tool for designing the optimum foam structures that would maximize heat transfer per unit pumping power

6.2 Recommendations

In view of the random microstructure of the foam, the probability distribution for the orientation, location, and distribution of the struts (cell wall) may need to be characterized experimentally in order to have a better prediction of the thermal conductivity of carbon foam. Moreover, detailed microstructural characterization of the

struts are recommended to ensure the tapering effect is accurately accounted for in the model.

Experiments on flow friction data for the carbon foam may need to be performed in order to have a better understanding of the friction characteristics and prediction of pressure drop for various flow rates using the closed-form, spherical void model. Additionally, the model may need to be extended to higher flow rates in order to utilize the model to a turbulent flow regime. For instance, one can experimentally obtain the relationship between friction factor and Reynolds number for higher flow rates or can analytically include a nonlinear velocity term in the momentum equation. The power law relationship between the friction factor and Reynolds number might vary for the foam as it is experimentally proven by Kays and London (1984) in the transient and turbulent flow regime for the crossed-rod and woven-screen matrices.

The permeability and Carman-Kozeny constant may need to be determined experimentally in order for a better utilization of the porous medium model, which might be a suitable model for the high-velocity flow through carbon foams.

Additional thermal testing with different heat exchanger designs utilizing carbon foam as the core materials is recommended for a better understanding the performance of the foam. In addition, a different high temperature brazing material and test chamber will solve the temperature limitation of the thermal testing, since the currently used Superbraz® and nylon test chamber has a melting temperature at about 230°C and 200°C, respectively. Furthermore, a test chamber with a sapphire window using infrared imaging might be helpful in visualizing the temperature maps within the foam core.

Hence, the local temperature profile within the test chamber can be known. Finally, each sample should be tested repeatedly in order to obtain statistically meaningful data.

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Appendix

Thermal Testing Data

Prototype Heat Sink

inlet vol		air flow	inlet pressure	voltage	Temperature (°C)					T	ΔT _{lm}	Power	UoA	Uo
(cfm)		(psi)	(volt)		1	2	3	4	5	(°C)	(K)	(W)	(W/K)	(W/m²K)
0		0	80		50.3	47.8	63.3	59.1	54.8	63.3	50.3	103	--	--
0		0	90		57.3	55.3	74.6	69.1	63.3	74.6	57.3	116	--	--
0		0	100		66.1	62.4	87.1	80.3	73.4	87.1	66.1	129	--	--
0		0	110		75.1	70.7	100.1	92.4	84.0	100.1	75.1	141	--	--
0		0	120		83.7	83.3	113.2	104.5	94.6	113.2	83.7	154	--	--
0		0	130		94.4	87.7	127.5	117.9	106.3	127.5	94.4	167	--	--
0		0	140		100.1	93.5	137.5	127.3	113.7	137.5	100.1	180	--	--
5		0.75	80		23.1	43.0	43.8	42.3	41.6	43.8	6.1	103	16.8	528
5		0.75	90		23.4	47.0	48.2	46.1	44.8	48.2	7.8	116	14.8	466
5		0.75	100		23.7	53.2	54.5	52.1	50.5	54.5	9.3	129	13.8	433
5		0.75	110		23.9	57.6	59.2	56.3	54.3	59.2	10.9	141	13.0	407
5		0.75	120		24.2	63.6	65.6	62.2	59.6	65.6	13.0	154	11.9	372
5		0.75	130		24.4	70.2	72.4	68.6	65.6	72.4	14.9	167	11.3	353
5		0.75	140		24.8	76.1	78.7	74.3	70.6	78.7	16.9	180	10.6	334
10		2.0	80		21.9	33.9	34.6	33.4	33.0	34.6	4.1	103	24.8	779
10		2.0	90		22.0	35.6	36.7	34.7	33.8	36.7	5.2	116	22.1	692
10		2.0	100		22.1	38.3	39.6	37.1	35.8	39.6	6.2	129	20.6	647
10		2.0	110		22.2	41.3	42.9	39.7	38.0	42.9	7.5	141	19.0	595
10		2.0	120		22.4	45.1	46.8	43.2	41.1	46.8	8.5	154	18.1	568
10		2.0	130		22.4	48.6	50.6	46.4	43.9	50.6	9.9	167	16.9	530
10		2.0	140		22.5	52.6	54.8	50.2	48.0	54.8	11.2	180	16.1	504
15		4.5	80		21.8	29.8	31.1	29.9	29.6	31.1	4.1	103	25.3	794
15		4.5	90		21.8	31.4	33.0	31.4	30.8	33.0	4.9	116	23.5	736
15		4.5	100		21.8	33.5	35.3	33.3	32.6	35.3	5.8	129	22.1	695
15		4.5	110		21.9	36.1	38.1	35.7	34.7	38.1	6.8	141	20.8	654
15		4.5	120		21.9	38.5	40.9	38.0	36.7	40.9	8.0	154	19.2	603
15		4.5	130		22.0	41.2	43.9	40.5	39.0	43.9	9.2	167	18.2	572
15		4.5	140		22.1	44.3	47.3	43.4	41.6	47.3	10.4	180	17.3	541
18		7.0	80		21.8	28.7	30.2	29.2	28.8	30.2	4.0	103	25.7	806
18		7.0	90		21.7	30.1	31.8	30.6	29.9	31.8	4.7	116	24.5	770
18		7.0	100		21.7	31.5	33.6	32.1	31.0	33.6	5.6	129	22.8	714
18		7.0	110		21.8	34.2	36.4	34.7	33.7	36.4	6.6	141	21.6	677
18		7.0	120		21.8	36.4	38.9	36.8	35.5	38.9	7.6	154	20.3	637
18		7.0	130		21.8	38.4	41.3	38.7	37.2	41.3	8.7	167	19.2	602
18		7.0	140		21.8	40.6	44.1	41.0	38.9	44.1	10.2	180	17.7	556

Heat Exchanger System for Sample 1 (Density = 0.43 g/cm³)

outlet vol flowrate (cfm)	inlet pressure (psi)	pressure drop (psi)	voltage (volts)	current (amps)	Temperature (°C)				T (°C)	ΔT _{lm} (K)	Power (W)	UoA (W/K)	Uo (W/m ² K)	AVG Uo (W/m ² K)
					1	2	3	4						
0	0	0	18.7	1.0	25.8	23.7	51.2	52.2	51.2	26.4	19	—	—	
0	0	0	43.2	2.31	39.6	34.1	168.1	173.1	168.1	131.2	100	—	—	
13	25	4.10	18.7	1.0	19.6	24.8	28.1	29.4	28.1	5.5	19	3.4	1316	
13	25	4.10	37.2	2.0	19.7	32.4	44.4	49.1	44.4	17.6	74	4.2	1639	
13	25	4.00	55.7	3.0	19.7	46.2	75.1	84.9	75.1	40.7	167	4.1	1590	
13	25	3.80	65.4	3.5	19.8	55.2	100.8	113.8	100.8	61.6	229	3.7	1440	
13	25	3.75	74.6	4.0	19.8	63.9	120.6	137.3	120.6	76.6	298	3.9	1509	
13	25	3.45	84.1	4.5	19.8	75.6	155.6	175.9	155.6	105.5	378	3.6	1391	
13	25	3.20	94.8	5.0	19.4	85.1	191.7	216.7	191.7	136.8	474	3.5	1342	1461
5	28	1.30	43.2	2.31	19.8	51.5	83.7	89.4	83.7	46.3	100	2.2	836	
5	26	1.50	52.9	2.84	19.8	60.7	110.3	119.1	110.3	68.0	150	2.2	856	
5	26	1.50	61.1	3.28	19.9	71.7	136.4	147.6	136.4	88.1	200	2.3	882	
5	27	1.45	68.3	3.66	20.2	85.7	168.3	180.9	168.3	112.2	250	2.2	863	
5	27	1.40	74.9	4.01	19.2	75.3	191.6	202.1	191.6	142.5	300	2.1	817	851
10	25	3.10	43.2	2.31	19.1	37.3	63.0	69.4	63.0	34.0	100	2.9	1138	
10	25	3.10	52.9	2.84	19.1	44.2	83.8	92.9	83.8	51.1	150	2.9	1139	
10	25	3.00	61.1	3.28	19.3	53.9	111.3	122.9	111.3	73.3	200	2.7	1059	
10	25	3.00	68.3	3.66	19.3	59.7	129.3	143.2	129.3	88.3	250	2.8	1097	
10	25	2.95	74.9	4.01	19.4	66.4	149.1	164.4	149.1	104.4	300	2.9	1114	1109
12	25	3.55	43.2	2.31	19.0	35.3	61.1	67.5	61.1	33.3	100	3.0	1162	
12	25	3.50	52.9	2.84	19.1	41.9	80.4	89.4	80.4	49.0	150	3.1	1188	
12	25	3.45	61.1	3.28	19.2	49.1	101.8	113.4	101.8	66.5	200	3.0	1167	
12	25	3.35	68.3	3.66	19.2	55.9	122.3	136.4	122.3	83.4	250	3.0	1161	1169

Heat Exchanger System for Sample 2 [1X] (Density = 0.47 g/cm³)

outlet vol flowrate (scfm)	Inlet pressure (psi)	pressure drop (psi)	voltage (volt)	current (Amps)	Temperature (°C)				T (°C)	ΔT _{lm} (K)	Power (W)	UoA (W/K)	Uo (W/m ² K)	AVG Uo (W/m ² K)
					1	2	3	4						
0	0	0	19.1	1.02	28.4	25.2	56.8	57.5	56.8	30.0	19	0.7	252	
0	0	0	30.5	1.63	39.6	33.9	100.7	102.2	100.7	63.9	50	0.8	302	277
5	27	2.30	30.5	1.63	19.9	37.6	43.8	46.5	43.8	13.1	50	3.8	1470	
5	27	2.30	43.2	2.31	19.8	49.8	61.9	66.9	61.9	24.1	100	4.1	1607	
5	27	2.25	52.9	2.84	19.9	62.9	82.3	89.7	82.3	36.8	150	4.1	1582	
5	27	2.15	61.1	3.28	20.0	77.7	105.8	115.1	105.8	51.7	200	3.9	1502	
5	27	2.05	68.3	3.66	20.0	89.9	126.7	137.7	126.7	65.7	250	3.8	1475	1527
8	29	4.05	30.5	1.63	19.8	32.1	37.8	40.6	37.8	10.7	50	4.7	1803	
8	29	3.95	43.2	2.31	19.9	43.5	56.3	61.6	56.3	22.6	100	4.4	1712	
8	29	3.80	52.9	2.84	20.0	54.6	75.1	82.6	75.1	35.0	150	4.3	1664	
8	29	3.65	61.1	3.28	20.2	65.4	93.8	103.4	93.8	47.5	200	4.2	1636	
8	29	3.50	68.3	3.66	20.2	75.0	111.1	122.3	111.1	59.3	250	4.2	1632	1689
10	27	-	30.5	1.63	19.4	28.9	34.4	37.0	34.4	9.5	50	5.3	2037	
10	27	-	43.2	2.31	19.7	37.9	49.0	54.1	49.0	18.8	100	5.3	2062	
10	27	-	52.9	2.84	19.7	46.4	64.2	71.6	64.2	29.1	150	5.2	1998	
10	27	-	61.1	3.28	19.7	54.7	79.4	88.9	79.4	39.7	200	5.1	1958	
10	27	-	68.3	3.66	19.6	62.2	94.7	106.1	94.7	50.9	250	4.9	1905	
10	27	4.94	74.9	4.01	19.8	71.2	112.8	126.4	112.8	63.9	300	4.7	1822	1964
12	26	-	30.5	1.63	19.7	28.7	34.7	37.4	34.7	9.8	50	5.1	1963	
12	26	-	43.2	2.31	19.9	36.0	47.5	52.6	47.5	18.4	100	5.4	2103	
12	26	-	52.9	2.84	19.9	43.6	61.6	69.1	61.6	28.2	150	5.3	2064	
12	26	-	61.1	3.28	19.9	50.9	75.7	85.3	75.7	38.2	200	5.2	2031	
12	26	-	68.3	3.66	19.9	59.1	92.9	104.8	92.9	50.9	250	4.9	1903	
12	26	-	74.9	4.01	19.8	65.2	105.4	119.2	105.4	60.1	300	5.0	1938	
12	26	-	81.0	4.32	20.0	72.9	122.3	137.9	122.3	72.7	350	4.8	1866	1981

Heat Exchanger System for Sample 2 [2X] (Density = 0.47 g/cm ³)														
outlet vol flowrate (cfm)	Inlet pressure (psi)	pressure drop (psi)	voltage (volt)	current (Amps)	Temperature (°C)				T (°C)	ΔT _{lm} (K)	Power (W)	UoA (W/K)	Uo (W/m ² ·K)	AVG Uo (W/m ² ·K)
5	36	190	0	0	20.7	21.4	22.1	22.3	22.1	1.0	0	—	—	
5	36	170	30.5	1.63	20.6	40.0	46.2	49.7	46.2	13.7	50	3.6	1410	
5	36	165	43.2	2.31	20.6	54.4	67.2	72.8	67.2	26.2	100	3.8	1478	
5	36	165	52.9	2.84	20.7	71.4	94.5	102.2	94.5	43.6	150	3.4	1334	
5	36	160	61.1	3.28	20.9	87.7	121.1	129.9	121.1	60.8	200	3.3	1277	
5	36	165	68.3	3.66	21.2	103.0	147.7	158.2	147.7	78.6	250	3.2	1232	
5	36	165	74.9	4.01	21.3	107.9	156.7	167.8	156.7	84.9	300	3.5	1371	
5	36	170	81.0	4.32	21.2	111.1	164.5	177.6	164.5	91.1	350	3.8	1489	1370
10	34	420	0	0	20.7	21.4	22.1	22.3	22.1	1.0	0	—	—	
10	34	415	30.5	1.63	20.2	30.3	35.1	38.1	35.1	8.9	50	5.6	2163	
10	34	415	43.2	2.31	20.1	36.5	46.6	52.1	46.6	17.0	100	5.9	2274	
10	34	410	52.9	2.84	20.1	46.2	61.9	69.9	61.9	26.7	150	5.6	2184	
10	34	400	61.1	3.28	20.2	56.2	78.8	89.4	78.8	37.8	200	5.3	2055	
10	34	395	68.3	3.66	20.3	61.5	88.5	99.6	88.5	44.5	250	5.6	2179	
10	34	390	74.9	4.01	20.3	69.0	102.2	115.4	102.2	53.9	300	5.6	2158	
10	34	385	81.0	4.32	20.4	75.8	114.8	129.6	114.8	62.7	350	5.6	2164	
10	34	380	86.5	4.62	20.6	83.4	130.7	147.3	130.7	74.3	400	5.4	2083	
10	34	375	91.8	4.90	20.4	91.6	147.6	165.6	147.6	86.8	450	5.2	2008	
10	34	365	96.8	5.17	20.4	99.5	164.0	183.4	164.0	98.8	500	5.1	1962	2123
15	31	513	30.5	1.63	20.2	29.4	33.8	37.2	33.8	8.2	50	6.1	2365	
15	31	502	43.2	2.31	20.2	36.9	46.6	52.8	46.6	16.7	100	6.0	2318	
15	31	485	52.9	2.84	20.3	41.7	56.2	64.6	56.2	23.6	150	6.4	2486	
15	31	470	61.1	3.28	20.5	50.3	71.4	82.4	71.4	33.8	200	5.9	2295	
15	31	460	68.3	3.66	20.4	55.5	80.9	93.7	80.9	40.4	250	6.2	2395	
15	31	455	74.9	4.01	20.4	61.4	93.2	106.4	93.2	49.5	300	6.1	2351	
15	31	450	81.0	4.32	20.5	67.9	107.3	123.4	107.3	60.0	350	5.8	2259	
15	31	445	86.5	4.62	20.8	75.9	125.4	143.1	125.4	73.6	400	5.4	2103	
15	31	430	91.8	4.90	20.6	79.8	132.1	151.2	131.1	77.2	450	5.8	2259	
15	31	410	96.8	5.17	20.6	90.3	156.0	175.7	156.0	96.4	500	5.2	2012	
15	31	410	101.5	5.42	20.7	98.3	171.9	194.2	171.9	107.8	550	5.1	1978	
15	31	415	106.1	5.66	20.7	102.6	181.6	204.5	181.6	115.1	600	5.2	2021	2235
15	31	-	96.8	5.17	20.6	87.4	143.5	164.8	143.5	85.2	500	5.9	2277	
15	31	-	101.5	5.42	20.6	95.1	161.1	183.8	161.1	98.6	550	5.6	2162	2219

Heat Exchanger System for Sample 3 (Density = 0.48 g/cm³)

outlet vol (cfm)	inlet pressure (psi)	pressure drop (psi)	voltage (volt)	current (Amps)	λ_{eff}	Temperature (°C)				T (°C)	ΔT_{lm} (K)	Power (W)	UoA (W/K)	Uo (W/m ² K)	AVG Uo (W/m ² K)	$\Delta P / L$ (psi/in)
0	0	0	0	0	198	1	2	3	4	228	19	0	0.0	0	0.000	0.000
0	0	0	19.1	1.00	29.4	21.7	22.8	22.9	22.9	60.9	32.4	19	0.6	228	238	0.000
0	0	0	30.5	1.63	37.2	34.5	113.7	117.2	117.2	113.7	77.8	50	0.6	248	238	0.000
5	35	2.15	0	0	198	21.7	22.8	22.9	22.9	22.8	19	0	—	—	—	1.075
5	35	2.00	30.5	1.63	20.0	38.2	51.1	56.9	56.9	51.1	20.7	50	2.4	932	1000	1.000
5	35	1.95	43.2	2.31	20.2	51.9	79.7	89.0	89.0	79.7	41.7	100	2.4	928	1000	0.975
5	35	1.90	52.9	2.84	20.2	66.6	110.1	122.7	122.7	110.1	63.9	150	2.4	911	950	0.950
5	35	1.95	61.1	3.28	20.3	77.8	134.3	149.8	149.8	134.3	81.9	200	2.4	948	200	0.975
5	35	1.95	68.3	3.66	20.4	85.1	153.1	172.3	172.3	153.1	96.8	250	2.6	1001	944	0.975
8	28	3.75	0	0	198	21.7	22.8	22.9	22.9	22.8	19	0	—	—	—	1.875
8	28	3.45	30.5	1.63	19.4	32.2	44.1	50.1	50.1	44.1	17.5	50	2.8	1100	1725	1.725
8	28	3.60	43.2	2.31	19.4	38.9	61.1	71.3	71.3	61.1	30.9	100	3.2	1250	1800	1.800
8	28	3.60	52.9	2.84	19.4	46.4	79.2	93.4	93.4	79.2	45.0	150	3.3	1295	1800	1.800
8	28	3.45	61.1	3.28	19.5	58.2	107.2	123.9	123.9	107.2	66.5	200	3.0	1168	1725	1.725
8	28	3.40	68.3	3.66	19.6	66.2	125.0	144.8	144.8	125.0	79.8	250	3.1	1213	1700	1.700
8	28	3.30	74.9	4.01	19.7	75.3	148.2	170.9	170.9	148.2	98.1	300	3.1	1187	1202	1.650
10	33	5.09	0	0	198	21.7	22.8	22.9	22.9	22.8	19	0	—	—	—	2.545
10	33	4.65	30.5	1.63	19.7	30.5	41.1	47.4	47.4	41.1	15.4	50	3.2	1254	2325	2.325
10	33	4.45	43.2	2.31	19.7	39.3	61.9	72.8	72.8	61.9	31.4	100	3.2	1232	2225	2.225
10	33	4.35	52.9	2.84	19.7	46.8	80.9	95.3	95.3	80.9	46.3	150	3.2	1256	2175	2.175
10	33	4.30	61.1	3.28	19.7	54.8	101.3	119.1	119.1	101.3	62.4	200	3.2	1244	2150	2.150
10	33	4.25	68.3	3.66	20.3	62.1	119.2	140.0	140.0	119.2	76.1	250	3.3	1273	2125	2.125
10	33	4.30	74.9	4.01	19.9	68.6	136.8	160.7	160.7	136.8	90.4	300	3.3	1288	2150	2.150
10	33	4.35	81.0	4.32	20.1	75.7	157.3	184.5	184.5	157.3	107.0	350	3.3	1267	1259	2.175
12	33	5.38	0	0	198	21.7	22.8	22.9	22.9	22.8	19	0	—	—	—	2.690
12	33	5.41	30.5	1.63	19.5	29.2	39.4	45.6	45.6	39.4	14.5	50	3.4	1329	2705	2.705
12	33	5.23	43.2	2.31	19.4	36.1	57.1	67.8	67.8	57.1	28.5	100	3.5	1355	2615	2.615
12	33	5.09	52.9	2.84	19.6	44.0	77.7	92.0	92.0	77.7	44.8	150	3.4	1300	2545	2.545
12	33	5.02	61.1	3.28	19.7	50.6	94.8	111.9	111.9	94.8	58.3	200	3.4	1332	2510	2.510
12	33	4.94	68.3	3.66	19.7	58.9	114.8	135.2	135.2	114.8	73.8	250	3.4	1313	2470	2.470
12	33	4.91	74.9	4.01	19.7	64.8	129.6	153.4	153.4	129.6	85.4	300	3.5	1363	2455	2.455
12	33	4.91	81.0	4.32	19.8	70.8	148.2	174.8	174.8	148.2	100.8	350	3.5	1346	1334	2.455
14	33	—	30.5	1.63	19.7	27.9	37.7	43.9	43.9	37.7	13.5	50	3.7	1430	—	—
14	33	—	43.2	2.31	19.6	33.0	52.6	63.6	63.6	52.6	25.7	100	3.9	1503	—	—
14	33	—	52.9	2.84	19.7	38.3	67.5	82.5	82.5	67.5	37.7	150	4.0	1543	—	—
14	33	—	61.1	3.28	19.7	44.2	83.5	101.7	101.7	83.5	50.6	200	4.0	1536	—	—
14	33	—	68.3	3.66	19.8	49.6	98.8	119.8	119.8	98.8	62.9	250	4.0	1539	—	—
14	33	—	74.9	4.01	19.7	55.4	114.9	138.8	138.8	114.9	76.0	300	4.0	1532	—	—
14	33	—	81.0	4.32	19.8	60.8	128.9	156.3	156.3	128.9	87.0	350	4.0	1559	—	—
14	33	—	86.5	4.62	19.8	67.7	147.3	177.6	177.6	147.3	101.7	400	3.9	1523	—	—
14	33	—	91.8	4.90	19.9	73.1	165.8	198.3	198.3	165.8	117.3	450	3.8	1486	1517	—

Heat Exchanger System for Sample 4 (Density = 0.49 g/cm³)

outlet vol flowrate (cfm)	inlet pressure (psi)	pressure drop (psi)	voltage (volt)	current (Amps)	Temperature (°C)				T (°C)	ΔT _{lm} (K)	Power (W)	UoA (W/K)	Uo (W/m ² K)	AVG Uo (W/m ² K)
					1	2	3	4						
0	0	0	19.1	1.00	30.0	26.4	60.0	61.7	60.0	31.8	19	0.6	233	
0	0	0	30.5	1.63	33.9	31.8	106.7	110.8	106.7	73.8	50	0.7	261	247
5	29	0	30.5	1.63	19.1	34.6	47.8	52.7	47.8	20.0	50	2.5	966	
5	29	0	43.2	2.31	19.2	48.7	77.3	86.9	77.3	41.6	100	2.4	929	
5	29	0	52.9	2.84	19.3	62.6	106.4	121.3	106.4	63.0	150	2.4	924	
5	29	0	61.1	3.28	19.6	78.7	139.9	160.6	139.9	87.4	200	2.3	888	927
8	28	0	30.5	1.63	18.7	30.9	48.6	53.8	48.6	23.3	50	2.1	829	
8	28	0	43.2	2.31	18.7	41.5	77.3	86.5	77.3	46.3	100	2.2	836	
8	28	0	52.9	2.84	18.8	51.3	104.9	118.6	104.9	68.6	150	2.2	849	
8	28	0	61.1	3.28	18.9	61.0	132.3	151.4	132.3	90.7	200	2.2	856	842
10	27	0	30.5	1.63	18.7	29.1	44.9	49.8	44.9	20.6	50	2.4	938	
10	27	0	43.2	2.31	18.6	37.1	68.9	78.1	68.9	40.3	100	2.5	958	
10	27	0	52.9	2.84	18.7	44.9	93.5	107.3	93.5	60.8	150	2.5	958	
10	27	0	61.1	3.28	18.9	53.6	119.9	139.5	119.9	82.4	200	2.4	942	
10	27	0	68.3	3.66	18.9	61.8	145.8	171.6	145.8	104.0	250	2.4	932	946
13	26	0	30.5	1.63	18.8	28.5	44.1	48.2	44.1	20.1	50	2.5	961	
13	26	0	43.2	2.31	18.7	35.9	67.2	82.6	67.2	39.3	100	2.5	985	
13	26	0	52.9	2.84	18.8	43.3	91.6	111.5	91.6	59.7	150	2.5	975	
13	26	0	61.1	3.28	18.9	50.2	117.8	136.8	117.8	82.3	200	2.4	944	
13	26	0	68.3	3.66	18.9	58.1	141.7	167.1	141.7	101.9	250	2.5	950	
13	26	0	74.9	4.01	18.9	63.4	162.2	192.6	162.2	119.7	300	2.5	973	965

Heat Exchanger System for Sample 5 (Density = 0.46 g/cm³)

outlet vol flowrate (cfm)	inlet pressure (psi)	pressure drop (psi)	voltage (volt)	current (amps)	Temperature (°C)				T (°C)	ΔT _{lm} (K)	Power (W)	UoA (W/K)	Uo (W/m ² ·K)	AVG Uo (W/m ² ·K)	ΔP/L (psi/in)
					1	2	3	4							
0	0	0.00	19	1	261	276	547	566	547	278	19	--	--	--	0.000
0	0	0.00	30.5	1.63	339	293	1029	1082	1029	713	50	--	--	--	0.000
5	36	2.60	0	0	198	217	228	229	228	19	0	--	--	--	1.300
5	36	2.40	30.5	1.63	201	318	348	430	348	74	0	--	--	--	1.200
5	36	2.35	43.2	2.31	206	406	472	637	472	143	100	7.0	2585	2615	1.175
5	36	2.35	52.9	2.84	201	484	593	842	593	213	150	7.1	2734	2635	1.175
5	36	2.35	61.1	3.28	202	608	752	1098	752	303	200	6.6	2563	2580	1.175
5	36	2.30	68.3	3.66	202	680	877	1303	877	377	250	6.4	2548	2545	1.150
5	36	2.35	74.9	4.01	202	784	1021	1533	1021	469	300	6.4	2480	2510	1.175
5	36	2.35	81.0	4.32	203	876	1161	1730	1161	555	350	6.1	2443	2470	1.175
5	36	2.35	86.5	4.62	204	968	1313	1969	1313	654	400	6.1	2367	2556	1.175
10	33	--	0	0	198	217	228	229	228	19	0	--	--	--	2.600
10	33	--	30.5	1.63	201	311	340	424	340	70	50	7.1	2747	2600	2.600
10	33	--	43.2	2.31	200	393	456	624	456	138	100	7.2	2809	2615	2.615
10	33	--	52.9	2.84	201	473	572	826	572	206	150	7.3	2828	2635	2.635
10	33	--	61.1	3.28	202	542	679	1026	679	273	200	7.4	2849	2635	2.635
10	33	--	68.3	3.66	201	631	815	1273	815	357	250	7.0	2715	2635	2.635
10	33	--	74.9	4.01	202	692	915	1469	915	422	300	7.1	2761	2635	2.635
10	33	--	81.0	4.32	202	767	1038	1683	1038	502	350	7.0	2704	2635	2.635
10	33	5.41	86.5	4.62	203	879	1214	1862	1214	612	400	6.5	2530	2743	2.705
12	33	--	0	0	198	217	228	229	228	19	0	--	--	--	2.705
12	33	--	30.5	1.63	200	291	324	388	324	89	50	7.2	2805	2635	2.635
12	33	--	43.2	2.31	200	362	427	554	427	130	100	7.7	2985	2635	2.635
12	33	--	52.9	2.84	199	443	544	743	544	199	150	7.6	2931	2635	2.635
12	33	--	61.1	3.28	200	509	646	909	646	262	200	7.7	2966	2635	2.635
12	33	--	68.3	3.66	199	581	759	1094	759	333	250	7.5	2906	2635	2.635
12	33	--	74.9	4.01	201	649	866	1264	866	400	300	7.5	2909	2635	2.635
12	33	--	81.0	4.32	201	726	991	1463	991	481	350	7.3	2821	2635	2.635
12	33	--	86.5	4.62	201	793	1102	1634	1102	553	400	7.2	2799	2635	2.635
12	33	--	91.8	4.90	202	861	1219	1817	1219	631	450	7.1	2762	2876	2.876
14	32	--	0	0	198	217	228	229	228	19	0	--	--	--	2.876
14	32	--	30.5	1.63	201	286	319	384	319	67	50	7.5	2891	2635	2.635
14	32	--	43.2	2.31	202	357	422	548	422	127	100	7.8	3042	2635	2.635
14	32	--	52.9	2.84	202	433	534	731	534	194	150	7.7	2999	2635	2.635
14	32	--	61.1	3.28	202	493	633	891	633	259	200	7.7	3001	2635	2.635
14	32	--	68.3	3.66	202	558	739	1066	739	327	250	7.6	2959	2635	2.635
14	32	--	74.9	4.01	201	622	845	1236	845	397	300	7.6	2932	2635	2.635
14	32	--	81.0	4.32	202	688	957	1418	957	471	350	7.4	2879	2635	2.635
14	32	--	86.5	4.62	202	748	1067	1589	1067	547	400	7.3	2829	2635	2.635
14	32	--	91.8	4.90	203	818	1184	1777	1184	624	450	7.2	2794	2635	2.635
14	32	--	96.8	5.17	202	874	1283	1924	1283	691	500	7.2	2805	2635	2.635
14	32	--	101.5	5.42	203	938	1414	2192	1414	787	550	7.0	2708	2635	2.635
14	32	--	106.1	5.66	203	990	1503	2317	1503	846	600	7.1	2747	2882	2.882

Heat Exchanger System for Sample 2 (Density = 0.47 g/cm ³)												
outlet vol flowrate (cfm)	Inlet pressure (psi)	pressure drop (psi)	voltage (volt)	current (Amps)	Temperature (°C)				T (°C)	Power (W)	ΔP / L (psil/m)	exit velocity (ft/s)
3	37	1.05	0	0	22.0	22.0	22.0	22.0	22.0	0	0.525	2.9
5	36	1.80	0	0	22.0	22.0	22.0	22.0	22.0	0	0.900	4.8
8	34	3.15	0	0	22.0	22.0	22.0	22.0	22.0	0	1.575	7.7
10	33	4.05	0	0	22.0	22.0	22.0	22.0	22.0	0	2.025	9.6
12	32	4.55	0	0	22.0	22.0	22.0	22.0	22.0	0	2.275	11.5
14	32	5.20	0	0	22.0	22.0	22.0	22.0	22.0	0	2.600	13.4
3	37	0.95	0	0	22.0	22.0	22.0	22.0	22.0	0	0.475	2.9
5	36	1.95	0	0	22.0	22.0	22.0	22.0	22.0	0	0.975	4.8
8	35	3.40	0	0	22.0	22.0	22.0	22.0	22.0	0	1.700	7.7
10	34	4.35	0	0	22.0	22.0	22.0	22.0	22.0	0	2.175	9.6
12	34	4.85	0	0	22.0	22.0	22.0	22.0	22.0	0	2.425	11.5
13	34	5.27	0	0	22.0	22.0	22.0	22.0	22.0	0	2.635	12.5
3	37	1.05	0	0	20.7	21.4	22.1	22.3	22.1	0	0.525	2.9
5	36	1.90	0	0	20.7	21.4	22.1	22.3	22.1	0	0.950	4.8
8	35	3.15	0	0	20.7	21.4	22.1	22.3	22.1	0	1.575	7.7
10	35	4.20	0	0	20.7	21.4	22.1	22.3	22.1	0	2.100	9.6
12	34	4.65	0	0	20.7	21.4	22.1	22.3	22.1	0	2.325	11.5
14	33	5.20	0	0	20.7	21.4	22.1	22.3	22.1	0	2.600	13.4

Heat Exchanger System for Sample 3 (Density = 0.48 g/cm ³)												
outlet vol flowrate (cfm)	inlet pressure (psi)	pressure drop (psi)	voltage (volt)	current (Amps)	1	2	3	4	T (°C)	Power (W)	ΔP / L (psi/in)	exit velocity (ft/s)
3	36	1.15	0	0	22.4	21.8	22.3	22.4	22.3	0	0.575	2.9
5	35	2.10	0	0	22.4	21.8	22.3	22.4	22.3	0	1.050	4.8
8	34	3.65	0	0	22.4	21.8	22.3	22.4	22.3	0	1.825	7.7
10	33	4.85	0	0	22.4	21.8	22.3	22.4	22.3	0	2.425	9.6
12	33	5.38	0	0	22.4	21.8	22.3	22.4	22.3	0	2.690	11.5
3	36	1.10	0	0	19.8	21.7	22.8	22.9	22.8	0	0.550	2.9
5	35	2.15	0	0	19.8	21.7	22.8	22.9	22.8	0	1.075	4.8
8	34	3.75	0	0	19.8	21.7	22.8	22.9	22.8	0	1.875	7.7
10	33	5.09	0	0	19.8	21.7	22.8	22.9	22.8	0	2.545	9.6
12	33	5.38	0	0	19.8	21.7	22.8	22.9	22.8	0	2.690	11.5

Heat Exchanger System for Sample 5 (Density = 0.46 g/cm ³)												
outlet vol flowrate (cfm)	inlet pressure (psi)	pressure drop (psi)	voltage (volt)	current (Amps)	1	2	3	4	T (°C)	Power (W)	ΔP / L (psi/in)	exit velocity (ft/s)
3	37	1.45	0	0	20.6	20.2	22.2	22.1	22.2	0	0.725	2.9
5	35	3.00	0	0	20.6	20.2	22.2	22.1	22.2	0	1.500	4.8
8	34	5.20	0	0	20.6	20.2	22.2	22.1	22.2	0	2.600	7.7
3	37	1.85	0	0	19.8	21.7	22.8	22.9	22.8	0	0.925	2.9
5	36	2.6	0	0	19.8	21.7	22.8	22.9	22.8	0	1.300	4.8
8	34	5.20	0	0	19.8	21.7	22.8	22.9	22.8	0	2.600	7.7

Vita

Chin Chien Tee was born in Kuala Lumpur, Malaysia on January 13, 1977. He attended elementary school and high school there. In fall 96, he transferred from Inti College, Subang Jaya, Malaysia to the University of Oklahoma, Norman, to further his education. He was awarded the membership in the National Honor Society of Golden Key, the Engineering Honor Society of Tau Beta Pi, and the Mechanical Engineering Honor Society of Pi Tau Sigma. He graduated from the University of Oklahoma with a B S degree in Mechanical Engineering in spring 1998.

During summer of 1998, he joined the BP-Amoco Chemical Co. in Decatur, Alabama, as an Intern Mechanical Engineer. Starting fall 1998, he enrolled in the graduate program in the Department of Mechanical and Aerospace Engineering and Engineering Sciences at the University of Tennessee, Knoxville, and accepted a graduate research assistantship and a teaching assistantship. He presented selected results from the research supervised by Dr. Yu and collaborated with the Oak Ridge National Laboratory at two international conference, i.e the 1999 ASME Mechanics and Materials Conferences at Virginia Tech, on June 27 – 30, 1999, and the Twenty-Fourth American Carbon Society Biennial Conferences at Charleston, South Carolina, on July 11-16, 1999. He received his M S degree in Mechanical Engineering from the University of Tennessee, Knoxville in May 2000.