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Design and Experiment of Compact and Effective Carbon Foam Recuperative Heat Exchangers

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A conceptual design for a compact, lightweight, recuperative heat exchanger with an effectiveness of 98% is presented. This heat exchanger consists of discrete pair of carbon foam blocks packed between thin sheets of stainless steel. The flowpaths were piled alternately in a modular manner so that the hot and cold streams counterflow in the recuperative heat exchanger. Measures were taken to minimize the axial conduction in the heat exchanger. The anisotropic property of carbon foam was exploited to achieve higher effectiveness. The paper shows how the overall effectiveness of the heat exchanger can reach beyond 98% by placing many pair of carbon foam blocks in series. Experiments with four pair of carbon foam blocks were conducted to validate the design concept. Results show that carbon foam can effectively increase heat transfer between the hot and cold streams. With four pair of carbon foam blocks, an effectiveness $\varepsilon_{\text{total}}$ greater than 80% was achieved. An effectiveness $\varepsilon_{\text{total}}$ of 98% can be reached by using 50 pair of carbon foam blocks. This new development has advantages in size and weight and can be easily scaled up for larger heat transfer requirements.

Nomenclature

A_b	=	heat transfer area, m^2
c_p	=	heat capacity of the air, $\mathrm{J/kg \cdot K}$
D_c	=	characteristic dimension of the pore structure, m
D_h	=	hole size between two pores, m
D_p	=	pore size, m
F_{SP}	=	space between the fins, m
H_x	=	height of the carbon foam block, m
k	=	thermal conductivity, $\mathrm{W/m \cdot K}$
k_b	=	bulk thermal conductivity of carbon foam, $\mathrm{W/m \cdot K}$
k_g	=	thermal conductivity of air, $\mathrm{W/m \cdot K}$
k_s	=	thermal conductivity of fin, $\mathrm{W/m \cdot K}$
L	=	length of the carbon foam block, m
$\dot{m}$	=	mass flow rate of air, kg/s
Q_{air}	=	heat transfer from hot to cold fluids, W
$T_{C,i}$	=	cold air inlet temperature, K
$T_{C,o}$	=	cold air outlet temperature, K
T_F	=	thickness of fin, m
$T_{H,i}$	=	hot air inlet temperature, K
$T_{H,o}$	=	hot air outlet temperature, K
U	=	overall heat transfer coefficient based on log mean temperature difference, $\mathrm{W/m^2 \cdot K}$
V	=	air velocity, m/s
ΔT_m	=	log mean temperature difference, K
ε	=	effectiveness of one pair of heat exchangers
$\varepsilon_{\text{total}}$	=	total effectiveness of the heat exchanger
ϕ	=	porosity

I. Introduction

NASA has identified cryogenic fluid management as a key technology to be developed for future space exploration architectures. Storage of oxygen is essential for a sustainable consumables transfer station for use on the lunar surface. NASA Kennedy Space Center [1] proposed a novel cryogenic system that incorporates integrated refrigeration with a reverse Brayton cycle and oxygen storage to meet the envisioned architectural requirements as well as to increase the mission capabilities. A critical component of a reverse Brayton cryocooler is the recuperative heat exchanger. The effectiveness of the heat exchanger is a performance measure of the degree of heat exchange between the hot and cold gas streams in the reverse Brayton cycle. An effectiveness of 100% indicates that there is complete heat exchange between the two streams. A high-effectiveness (greater than 98%) recuperative heat exchanger is required to maintain the cryocooler at a high operational coefficient of performance (COP), thereby reducing energy consumption. Zhou et al. [2] investigated the effect of the effectiveness of the recuperator on the COP of a reverse Brayton cryocooler for liquefaction of oxygen. A pressure ratio of 1.8 was used, and the isentropic efficiencies of the compressor and turbine were assumed to be 65 and 75%, respectively. The result indicated that the COP increases from 0.07 to 0.11 when the heat exchanger effectiveness increases from 0.96 to 0.98. Therefore, it is very critical to develop a highly effective recuperative heat exchanger. To achieve the removal of 48 W of thermal energy required for liquefaction and zero-loss storage of oxygen, the heat exchanger must have a 2000 W capacity for heat transfer [1].

There have been some developmental efforts for recuperative heat exchangers for cryocoolers. The microtube strip heat exchanger was developed for applications in reverse-Brayton-cycle systems, but it is difficult to form a counterflow-type heat exchanger with this arrangement, resulting in a reduction in effectiveness [3]. McCormick et al. [4] developed a radial-flow heat exchanger (RFHX) for their turbo-Brayton cryocoolers. The RFHX has an effectiveness of 0.89, with a diameter of 21 cm and a height of 12 cm. The mass of the recuperator is 5.1 kg. Hill et al. [5] described the slotted-plate heat exchanger (SPHX), which was developed for the Hubble Space Telescope as part of the Near-Infrared Camera and Multi-Object Spectrometer cryocooler. The heat exchanger is composed of a stack of copper plates containing concentric rings of slotted flow passages. The plates in the recuperator are separated by low-conductivity stainless steel spacers brazed between the concentric rings of slots. The effectiveness can reach 0.997 at a 1 g/s mass flow rate of neon in the system. Hill et al. also replaced the

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copper and stainless steel SPHX with a silicon micromachined recuperator to reduce the weight of the heat exchanger. These heat exchangers were designed for a cryocooler system at a 5 W capacity level. Hoch et al. [6] reported a modular design of a high-effectiveness recuperative heat exchanger for a pulse-tube/reverse-Brayton hybrid cryocooler. The design uses etched cooper plates interleaved with a stainless steel axial conduction barrier. The measured effectiveness can only reach 0.93 at a 1 g/s mass flow rate of helium. Another design was suggested for a counterflow micro heat exchanger with low pressure loss [7]. The square grooves were etched in polymer wafers and the wafers were stacked one by one to form a rectangular heat exchanger. Special consideration was made to reduce the axial conduction to achieve high effectiveness. It is difficult to scale up this micro heat exchanger design for large-capacity systems.

This study presents a concept and validation for a compact, lightweight, and high-effectiveness heat exchanger for liquid-oxygen storage for sustainable lunar applications. The purpose was to develop and test a design for a compact and lightweight recuperative heat exchanger that has high effectiveness and can be easily scalable for larger heat capacity. The effect of the axial conduction on effectiveness was considered, as well as heat transfer enhancement between the hot and cold streams. Carbon foam was selected for heat transfer enhancement to reduce the temperature difference between the hot and cold gas streams. Instead of using a continuous piece of carbon foam, multiple pair of carbon foam blocks were used in the design. This cut off the thermal path of the carbon foam in the axial direction. The anisotropic feature of carbon foams was exploited in the design for each pair of foam blocks. The relation between the overall effectiveness of the recuperator and the effectiveness of one pair and the number of pair of foam is given. Experiments were conducted on four pair of stacked carbon foams to verify the feasibility of the approach. The effects of a stacked configuration on heat transfer were investigated. The data were correlated at different volumetric flow rates. Air was used to test the performance of the heat exchanger in this study. Further studies will be conducted in the future using a working fluid that is more suitable for a cryocooler (e.g., nitrogen, neon, or helium).

II. Concept of a Stacked Counterflow Air Heat Exchanger

New heat exchanger designs must be developed that are modular, easily scalable, highly effective, compact, and lightweight. Figure 1 shows a schematic of the cross section of the concept of an air-counterflow-type heat exchanger. The basic unit was composed of separate blocks of carbon foam packed between thin sheets of solid material. As mentioned later in this section, it is important to minimize axial conduction between the hot and cold ends of the recuperator to achieve high effectiveness. The flowpaths were piled alternately and the hot and cold gas paths were arranged for counterflow. The stacked parallel-plate arrangement allows symmetrical counterflow passages that balance flow across each heat transfer interface. This design is modular and can easily be scaled up for large capacity with more stacks.

To reduce the size of the heat exchanger, the heat transfer between the thin separating sheets with both the hot and cold gases must be significantly improved by using carbon foam blocks. The Poco carbon foam blocks used in this work have a density of 0.6 g/cm^3, porosity of 0.75, and bulk thermal conductivity of 135 $W/m \cdot K$ in the out-of-plane direction [8]. Carbon foam has an effective thermal conductivity of 100–230 $W/m \cdot K$ [8], which is much higher than the effective thermal conductivity of aluminum foam (5–9 $W/m \cdot K$) [9]. Gallego and Klett [10] reported that the heat transfer between a heated substrate and a moving airstream when carbon foam was used could be 10 times higher than the heat transfer when aluminum foam was used. The low density of carbon foam and the small thickness of the thin sheets separating the hot and cold fluids will lead to the light weight of the recuperator. Yu and Thompson [11] reported an application of carbon foam in compact recuperators for gas turbine systems. Their investigation was focused on heat transfer enhancement using carbon foam. Based on their thermal analysis using full-length foam block without consideration of axial conduction along the carbon foam, they reported that heat exchanger effectiveness of 100% can be achieved. Their claim is questionable because it is well known that axial conduction in the recuperative heat exchanger can degrade its performance considerably [12].

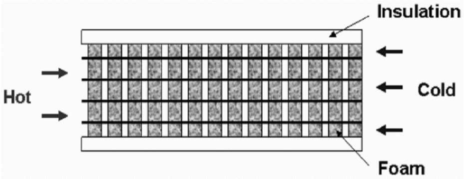

Fig. 1 Schematic of the heat exchanger.

Although it is important to reduce the thermal resistance in the transverse direction between the hot and cold fluids as much as possible, the axial (or longitudinal) heat conduction between the hot and cold ends of the heat exchanger must be kept to a minimum [12]. To reduce this axial heat conduction, several measures were taken. First, the separation plates were made thin to reduce the cross-sectional area in the axial direction. Second, the material used should have a low thermal conductivity k. Commercially available 100-μm-thick stainless steel sheets were chosen ($k = 11$ $W/m \cdot K$) for the experiment. Finally, a series of discrete carbon foam blocks were used instead of a continuous piece. This cut off the thermal path of the carbon foam in the axial direction. Note that the carbon foam blocks also act as a support for the thin stainless steel sheet and could allow a large pressure differential between the hot and cold fluids if needed. Huang and Vafai [13] numerically investigated convection heat transfer in a partially blocked channel using porous blocks on one side of the channel. Yucel and Guven [14] reported an investigation on forced convection in a partially blocked channel using porous blocks on both sides of a channel. Both papers reported results of crossflow over the porous blocks in the channels. In the configuration of the current study, the channel was fully blocked by carbon foam blocks. Air is forced to penetrate the carbon foam blocks uniformly to take advantage of the highly conducting carbon foam to achieve maximum heat exchange effectiveness. Flow and heat transfer in the partially and fully blocked configurations have different applications and performance.

III. Performance of One Pair of Carbon Foam Blocks

A performance model was used to assess how much a pair of carbon foam blocks could raise the temperature of the cold stream and reduce the temperature of the hot stream. Figure 2 shows one pair of carbon foam blocks. Heat transfer from hot to cold is enhanced through the carbon foam blocks. Heat transfer in carbon foam involves the combined process of heat conduction within the carbon ligaments and convective heat transfer between the ligaments and air in the open-cell graphite foam. There were several investigations on fluid flow and heat transfer in metal foams. Krishnan et al. [15] simulated thermal transport in open-cell metal foams based on a periodic unit cell structure. They predicted the effective thermal conductivity of aluminum foam and local Nusselt number and compared their results with available experimental results. Mahjoob

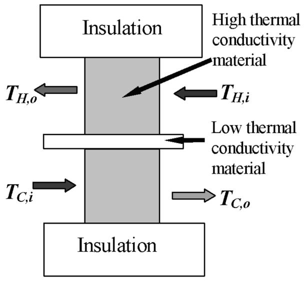

Fig. 2 Heat transfer between one carbon foam pair.

and Vafai [16] investigated pertinent correlations in the literature for flow and thermal transport in metal foam heat exchangers. Carbon and metal foams have large differences in foam configuration, pore size, porosity, effective thermal conductivity, and specific area per unit volume. The simulation results and correlations for metal foams may not be applicable to carbon foam due to the significant difference in foam structure. Yu et al. [17] conducted simulations on fluid flow and heat transfer in porous carbon foam based on a unit cube cell model. Tee et al. [18] did a similar simulation based on a cubic strut model. These studies were able to predict the effective conductivity as a function of porosity in out-of-plane direction. However, the models did not include the anisotropic feature of the carbon foam.

The carbon foams have high thermal conductivity and very large specific area per volume with small pores. As discussed in [19], air can reach thermal equilibrium with the solid in carbon foam very rapidly. The temperature difference between fluid and solid in carbon foams is very small. The major factor for enhancement of heat transfer in this case is the extended surface effect. The dispersion effect [20] is not important in this case.

Carbon foam blocks were modeled as extended surfaces of straight fins rather than a porous medium in this study. The model includes the heat transfer enhancement by the carbon foam. The effect of anisotropic characteristic of carbon foam was included by allowing different thermal conductivity values for the fins in different directions. Figure 3 shows the schematic of the model. H_x is the height and L is the length of the carbon foam blocks, T_F is the thickness of the fins, and F_{SP} is the space between the fins. A similar fin model for a carbon foam recuperative exchanger was used by Yu and Thompson [11]. They used Taylor's [21] formulas to determine the thickness of the fins and the space between them. They also used solid thermal conductivity of 1200 W/mK in their simulation. They did not consider axial conduction in the simulation.

In this study, the dimensions of T_F and F_{SP} are determined based on the characteristic dimensions and porosity ϕ of carbon foam. Figure 4 is a scanning electron microscope (SEM) image of carbon foam. Carbon foam has the feature of a porous medium, with 350 μm pores connected with 110 μm holes between pores. The distance between two connecting holes represents the characteristic length of the pore structure in the carbon foam. Figure 5 shows a 2-dimensional drawing of one sphere connected with two other spheres. The circles represent the pores and the intersection between two circles represents the holes between two pores. D_p is the pore size, D_h is the hole size between two pores, and D_c represents the characteristic dimension of the pore structure. Using the dimensions given previously for D_p and D_h, D_c can be shown to be 333 μm. T_F and F_{SP} are selected by setting D_c equal to $T_F + F_{\mathrm{SP}}$ and choosing T_F and F_{SP} so that the flow channels shown in Fig. 3 match the porosity of the carbon foam. Following this, for a porosity of $\phi = 0.75$, the dimensions of T_F and F_{SP} can be determined to be 84 and 249 μm, respectively.

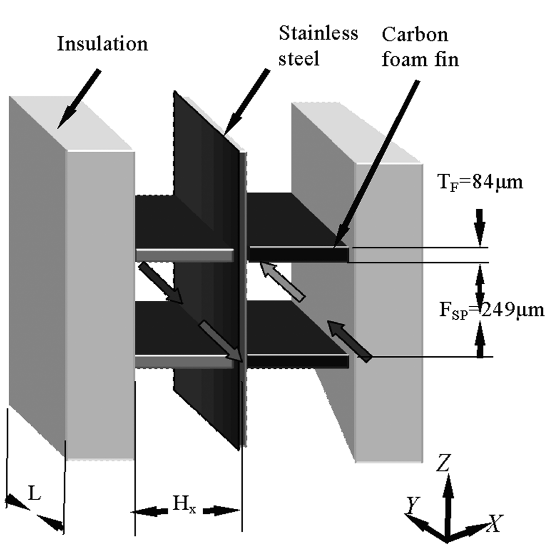

Fig. 3 Model of one pair of carbon foam blocks.

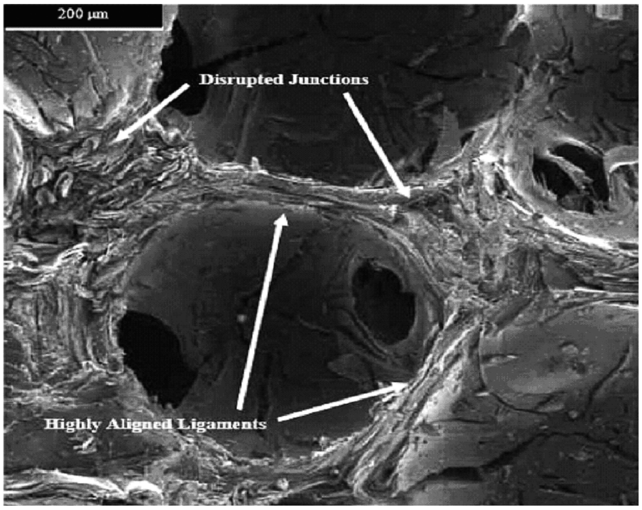

Fig. 4 SEM picture of carbon foam.

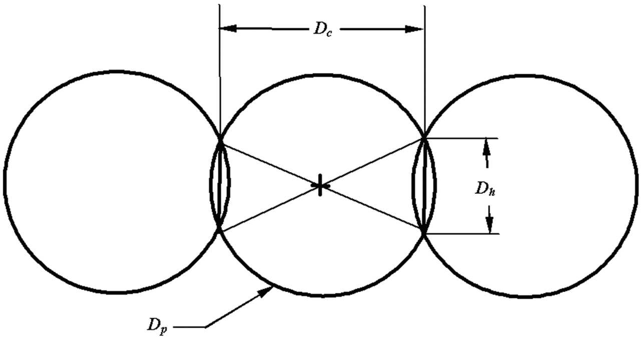

Fig. 5 Dimensions of carbon foam.

The carbon foams used in this study have an anisotropic thermal conductivity. The bulk thermal conductivity k_b is 135 $\mathrm{W/m \cdot K}$ in the out-of-plane direction and is 45 $\mathrm{W/m \cdot K}$ in the other two directions. The out-of-plane direction (the direction of high thermal conductivity) is in the heat transfer direction from the hot and cold fluids in this study and one of the directions with low thermal conductivity is in the axial flow direction of the heat exchanger. The highly effective conductivity results from the high solid-phase conductivity of graphitized material. Carbon foam has a high capability to conduct heat into its internal structure and then transfer heat to infiltrating fluids. The thermal conductivity of the fins k_s used in the model was chosen to be the thermal conductivity of the solid part of the carbon foam structure. Because the fin structure model treats carbon foam as a parallel structure of solid carbon ligament and air (with thermal conductivity k_g), the porosity ϕ and bulk thermal conductivity ($k_b = k_s(1 - \phi) + k_g\phi$) can be used to determine the thermal conductivity of the solid k_s for the fin structure in all three directions based on the measured bulk thermal conductivity. Because of the anisotropic bulk thermal conductivity k_b of the carbon foam, the thermal conductivity k_s of the fins along the fin direction (x direction in Fig. 3) is 550 $\mathrm{W/m \cdot K}$, and in the axial y direction and the transverse z direction, the thermal conductivity k_s is 180 $\mathrm{W/m \cdot K}$. These values are consistent with the thermal conductivity of the carbon ligaments proposed by Vrable [22].

Hot and cold flows were assumed to be in the laminar regime due to the small Reynolds number based on the dimension of the spacing between fins (the Reynolds number is about 166 at $V = 1$ m/s). The inlet temperatures of the hot and cold streams are chosen as 298 and 290 K, respectively. The commercial COMSOL Multiphysics software (incompressible Navier–Stokes and conduction and convection packages) was used to simulate the conjugated heat exchange between the hot and cold fluids while considering the fin

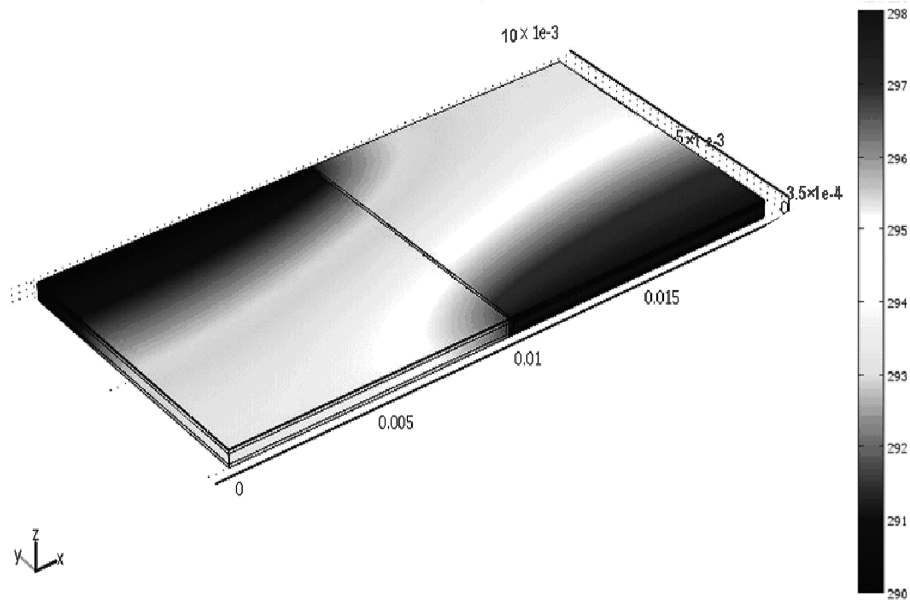

Fig. 6 Temperature profiles for one carbon foam pair, $V = 0.38$ m/s, units in Kelvin and meters.

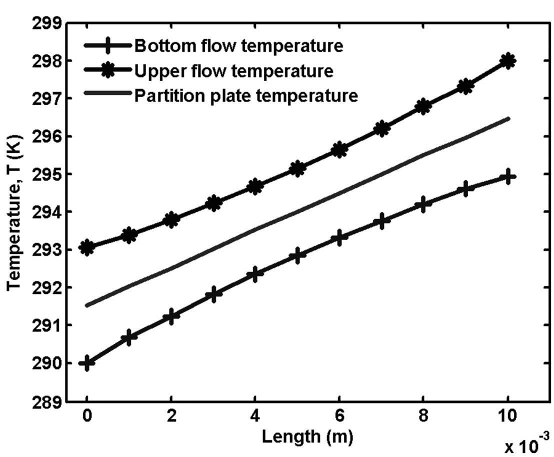

Fig. 7 Temperatures of hot and cold streams, one carbon foam pair, $V = 0.38$ m/s.

effect as well as anisotropic solid conduction. The element type was Lagrange–quadratic and the number of elements used was up to 380,000. The relative tolerance was set to be 10^{-8}. The simulation results were checked to be independent of mesh size.

Figure 6 shows the temperature distribution of the fins. Figure 7 shows that the hot gas can be cooled from 298 to 293 K and the cold gas can be heated from 290 to 295 K. The effectiveness for one pair is defined as the ratio of actual heat transfer rate to the maximum possible heat transfer rate. When the heat-capacity flow rate of the hot stream is equal to that of the cold stream, the effectiveness is given by

$$\varepsilon = \frac{T_{H,i} - T_{H,o}}{T_{H,i} - T_{C,i}} \tag{1}$$

Based on the simulation result, the effectiveness for one single pair of foam blocks is 62% at an airflow speed of 0.38 m/s in the channel.

IV. Approach to Achieve 98% Effectiveness

The present requirement for the recuperative heat exchanger is for the temperature of the hot stream to decrease from 298 to 98 K (a 200 K difference), whereas the temperature of the cold stream has to increase from 94 to 294 K (also a 200 K difference). This can be accomplished with 50 pair of carbon foam blocks, with each pair responsible for a 4 K temperature change. The effectiveness of each pair is limited because of axial heat conduction due to heat transfer from the hot end to the cold end for each pair of carbon foam blocks. The overall high effectiveness of the heat exchanger is achieved through a series of carbon foam pair. Figure 8 shows the stepwise change of the temperature in the heat exchanger. The number of pair required in the design of the heat exchanger depends on the performance of each pair. The relation between the overall effectiveness $\varepsilon_{\text{total}}$ and the effectiveness of each pair ε can be expressed as follows:

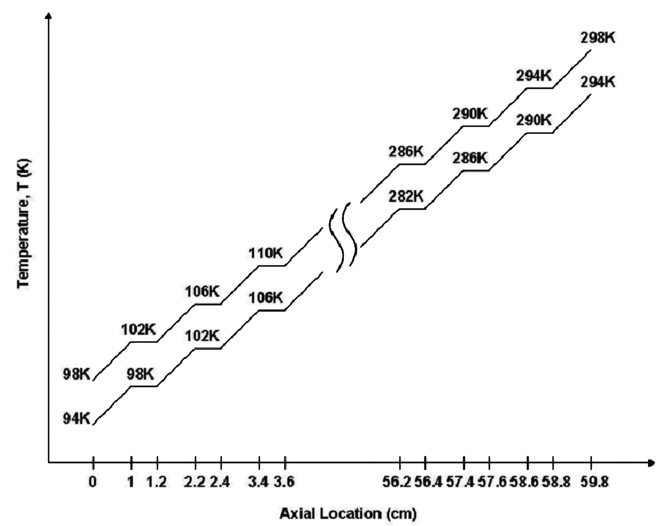

Fig. 8 Axial temperature distributions within recuperator.

$$\varepsilon_{\text{total}} = \frac{N \cdot \varepsilon}{1 + (N - 1)\varepsilon} \tag{2}$$

where N is the number of carbon foam pair. Even if the effectiveness for each pair of carbon foam blocks is only 50%, the overall effectiveness can still be as high as 98% through 50 pair of them in series. From the preceding analysis, it appears the design concept of using discrete 1 cm carbon foam blocks can provide an overall effectiveness of 98%.

V. Experimental Validation of Concept

Figure 9 shows the configuration of a recuperator in which the hot and cold flow channels have 4 carbon foam blocks each. The

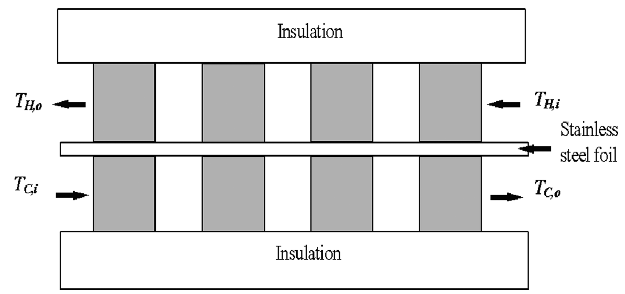

Fig. 9 Experimentally measured $T_{H,i}$, $T_{H,o}$, $T_{C,i}$, and $T_{C,o}$ (i.e., hot/cold air inlet/outlet temperature).

dimensions of the flow channels are $10 \times 1.7 \times 1$ cm (length by width by height). The cross sections of the air inlet and outlet are both 1.7×1 cm (length by width). The carbon foam blocks of dimensions $1 \times 1.7 \times 1$ cm (length by width by height) are glued to a 0.1-mm-thick stainless steel plate using a silver-loaded epoxy (Pyro-Duct™) [23]. This epoxy has a higher thermal conductivity than other epoxies and adhesives. The spaces between foam blocks are designed to cut off the axial thermal path of the carbon foam from the hot end to the cold end of the recuperator. It was calculated if the total length of the air gaps is equal to 40% of that of the total length of foam blocks, axial conduction would be less than 0.1% of the amount of heat transfer between the hot and cold streams. The total length of the air gaps is not expected to have much effect on the overall effectiveness of the recuperator. The carbon foam blocks were mounted and sealed inside a Plexiglas chamber. The carbon foam occupied the entire cross section of the channel. The entire test chamber was insulated and sealed to prevent air leaks. The cold airstream was provided by passing room-temperature air from the lab supply system through a cold thermal bath. The air from the cold outlet $T_{C,o}$ (see Fig. 9) was then heated by another hot thermal bath to become the hot airstream at the inlet temperature $T_{H,i}$. In this way, both the cold and hot airstreams have the same mass flow rate.

The purpose of the experiment was to evaluate the heat transfer performance of carbon foam blocks when used in a recuperative air heat exchanger. The air temperatures between the foam blocks were measured by 0.3-mm-diam thermocouples that were placed in the space between the carbon foam blocks. The temperatures at the inlet and outlet ($T_{H,i}$, $T_{H,o}$, $T_{C,i}$, and $T_{C,o}$; that is, hot/cold air inlet/outlet temperature) were also measured. The flow rate was measured with a flow meter that was calibrated with a high-precision mass flow meter. All data collection was automated using a Keithley 2700 data acquisition system and a personal computer. The thermocouples used in experiments were carefully calibrated and the error in temperature measurement was within $\pm 0.1°C$. The errors for pressure drop and flow rate were estimated to be less than ± 10 Pa and 1.2%, respectively.

The heat transfer performance of the heat exchanger with carbon foam blocks was evaluated by the overall heat transfer coefficient, which is defined as

$$U = \frac{Q_{\text{air}}}{A_b \Delta T_m} \tag{3}$$

where Q_{air} is the heat given up by the hot air that passes through the carbon foam. It was calculated by

$$Q_{\text{air}} = \dot{m} c_p (T_{H,i} - T_{H,o}) \tag{4}$$

A_b is the heat transfer area between the hot and cold fluids; it coincides with the surface area of the dividing stainless steel sheet. The log mean temperature difference ΔT_m is defined as

$$\Delta T_m = \frac{(T_{H,i} - T_{C,o}) - (T_{H,o} - T_{C,i})}{\ell n[(T_{H,i} - T_{C,o})/(T_{H,o} - T_{C,i})]} \tag{5}$$

where $T_{H,i}$, $T_{H,o}$, $T_{C,i}$, and $T_{C,o}$ are the hot and cold inlet and outlet air temperatures.

The effectiveness of the heat exchanger can be calculated based on the temperature drop of the hot air or the temperature rise of the cold air:

$$\varepsilon_{\text{total}H} = \frac{(T_{H,i} - T_{H,o})}{(T_{H,i} - T_{C,i})} \tag{6}$$

$$\varepsilon_{\text{total}C} = \frac{(T_{C,o} - T_{C,i})}{(T_{H,i} - T_{C,i})} \tag{7}$$

Because the flow rates were the same for both the hot and the cold sides, and the specific heat could be assumed to be constant in the temperature range of the experiment, the effectiveness of the hot and cold sides should be the same, in theory. The small difference (error) in the effectiveness calculated from the hot and cold streams is due to measurement error in the experiments:

$$\text{error} = \frac{|\varepsilon_{\text{total}H} - \varepsilon_{\text{total}C}|}{\varepsilon_{\text{total}H}} \tag{8}$$

The average effectiveness of the heat exchanger is calculated as

$$\varepsilon_{\text{total}} = \frac{|\varepsilon_{\text{total}H} + \varepsilon_{\text{total}C}|}{2} \tag{9}$$

The airflow speed V and the inlet and outlet air temperatures are measured in the experiment. The measured data were processed using Eqs. (3–9).

VI. Results and Discussion

The total pressure drop across the four 1 cm carbon foam blocks was measured at velocities of 0.25 to 1 m/s. The results are shown in Fig. 10. This pressure drop compares well with the pressure drop across a 5-cm-long solid carbon foam measured by Oak Ridge National Laboratory [24] when their pressure drop was scaled down linearly from 5 to 4 cm. Though the pressure drop of 5 kPa at $V = 1$ m/s could be high for certain applications such as in cryocoolers, it should be noted that carbon foams in a corrugated configuration could yield a pressure drop as much as 20 times lower than solid carbon foams while maintaining an equal or better heat transfer performance [10].

Table 1 shows the heat transfer results for the four pair of carbon foam blocks at airflow speeds from 0.25 to 1 m/s. The measured overall heat transfer coefficient was 568 $W/m^2 \cdot K$ at the speed of 1 m/s. Without the 4 carbon foam pair, the overall heat transfer coefficient was only 10.2 $W/m^2 \cdot K$ for the empty flow channels at $V = 1$ m/s. Clearly, the heat transfer enhancement by the carbon foam pair is critical. Table 1 also includes five sets of data for different cold and hot inlet temperatures at the 0.5 m/s airflow speed.

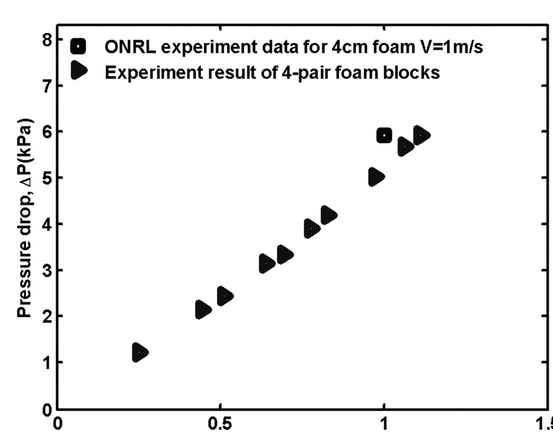

Fig. 10 Comparison of pressure drop with Oak Ridge National Laboratory data .

Table 1 Experimental results

$T_{H,i}$(°C)	$T_{H,o}$(°C)	$T_{C,i}$(°C)	$T_{C,o}$(°C)	ΔT_m(°C)	V(m/s)	$\varepsilon_{\text{total}H}$	$\varepsilon_{\text{total}C}$	Error(%)	$\varepsilon_{\text{total}}$	U(W/m$^2\cdot$K)
22.1	19.8	19.3	21.8	0.4	0.25	0.84	0.87	3.1	0.86	216
29.9	16.5	14.3	27.2	2.4	0.38	0.86	0.83	3.5	0.84	305
24.9	14.7	12.5	23.0	2.1	0.5	0.83	0.84	2.0	0.83	366
26.4	15.1	12.9	24.1	2.3	0.5	0.83	0.83	0.1	0.83	365
42.7	22.4	17.9	38.9	4.2	0.5	0.82	0.85	3.2	0.83	359
49.5	20.9	14.6	44.0	5.9	0.5	0.82	0.84	2.7	0.83	358
63.5	33.5	27.0	57.7	6.2	0.5	0.82	0.84	2.1	0.83	356
25.1	14.4	12.1	22.9	2.3	0.67	0.82	0.83	1.3	0.83	465
24.7	16.0	13.6	22.5	2.3	1.0	0.79	0.80	1.5	0.80	568

The overall heat transfer coefficient varied from 366 to 356 W/m$^2\cdot$K, even when the log mean temperature difference varied from 2.1 to 6.2°C. This means that the overall heat transfer coefficient U is fairly independent of the inlet temperatures of the hot and cold airstreams. Figure 11 shows the measured overall heat transfer coefficient of the recuperative heat exchanger with four pair of carbon foam blocks. These results demonstrate that carbon foam is a good medium for heat transfer enhancement because of its high thermal conductivity.

The uncertainty for U at $V = 0.25$ m/s is 25%, due to small temperature differences, but uncertainties at high velocities are small. The average errors for all the experiments for the Q_{air}, ΔT_m, and U are 4.2, 3.1, and 5.9%, respectively.

Table 1 also shows the overall effectiveness with 4 pair of carbon foam blocks. The measured effectiveness decreases slightly from 86 to 80% when the air speed V increases from 0.25 to 1 m/s. The difference in the effectiveness of the heat exchanger between the hot and cold sides was under 4%. The error for $\varepsilon_{\text{total}}$ was estimated to be 4.7% for the worst case, with $V = 0.25$ m/s, and the average error for $\varepsilon_{\text{total}}$ was 3.2%. The overall effectiveness for the empty flow channels without the carbon foam blocks was always under 6%, which is substantially lower than those with 4 pair of carbon foam blocks.

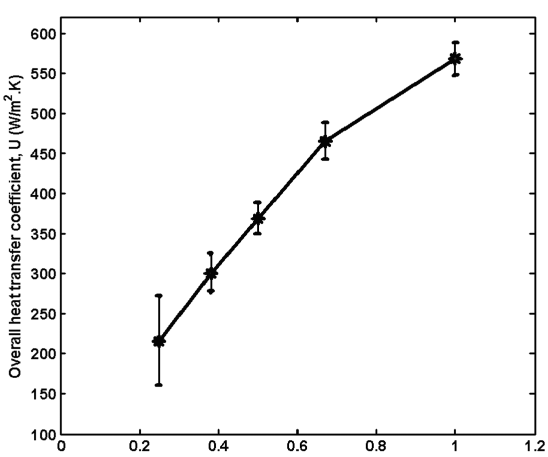

Fig. 11 Overall heat transfer coefficient vs airflow speed.

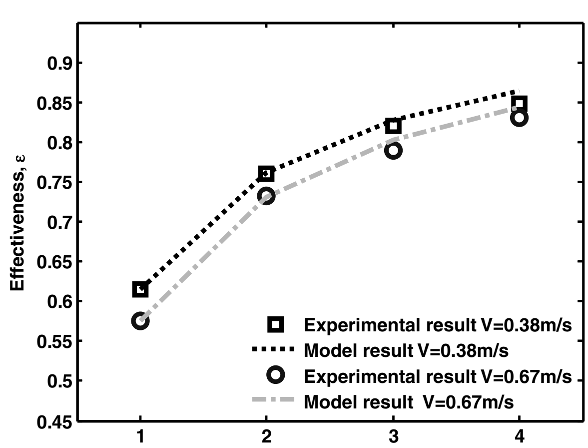

Fig. 12 Comparison between theoretical and measured effectiveness.

Using the measured air temperatures across any pair of carbon foam blocks, one could calculate the effectiveness for the pair and compare it with the effectiveness from the fin model described earlier that simulates one pair of carbon foam blocks. Following Eq. (1) and Fig. 2, the effectiveness for one pair is

$$\varepsilon_{1\ \text{pair},H} = \frac{(T_{H,i} - T_{H,o})}{(T_{H,i} - T_{C,i})} \tag{10}$$

With a similar expression for $\varepsilon_{1-\text{pair},C}$, the average effectiveness for one pair is

$$\varepsilon = \frac{|\varepsilon_{1\ \text{pair},H} + \varepsilon_{1\ \text{pair},C}|}{2} \tag{11}$$

For one carbon foam pair, the theoretical effectiveness from the fin model is $\varepsilon = 0.58$ and 0.62 when the flow velocity is $V = 0.67$ and 0.38 m/s, respectively. These theoretical results were extrapolated to 2, 3, and 4 carbon foam pair using Eq. (1). The average effectiveness measured across one carbon foam pair is $\varepsilon = 0.57$ and 0.61 at these velocities. This excellent agreement between model and experiment for one pair of carbon foam blocks is shown in Fig. 12.

In a similar fashion, by using the air temperatures across two and three pair of carbon foam blocks, one could estimate the average effectiveness for two and three pair of carbon foam blocks. The average measured effectiveness for one, two, three, and four pair of carbon foam blocks are plotted in Fig. 12. It is very interesting to note the effectiveness versus number of foam pair follows Eq. (1). Figure 12 also shows that though the effectiveness for one pair is only 0.6, using four pair in series can increase the overall effectiveness to about 80%.

VII. Conclusions

A highly effective recuperative heat exchanger is described that can have an effectiveness of more than 98%. Such a heat exchanger would consist of separate blocks of carbon foam packed between thin sheets of low-thermal-conductivity stainless steel. Carbon foam is used to enhance heat transfer between the hot and cold fluids to reduce the size of the heat exchanger. The flowpaths are piled alternately in a modular manner so that the hot and cold streams counterflow in the heat exchanger. This new configuration was designed to minimize the axial conduction. Experimental results showed that a carbon-foam-based heat exchanger could effectively transfer heat from the hot to cold streams. With four pair of carbon foam blocks, an effectiveness $\varepsilon_{\text{total}}$ over 80% was achieved. An effectiveness $\varepsilon_{\text{total}}$ over 0.98 could be attained by using 50 pair of carbon foam blocks. This development has advantages in size and weight and can be scaled up for larger systems.

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