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CARBON FOAM – NEW GENERATION OF ENHANCED SURFACE COMPACT RECUPERATORS FOR GAS TURBINES

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ABSTRACT

The potential of porous carbon foam is explored in the context of compact recuperators for microturbine applications. Porous carbon foam has an open, interconnected pore structure and an extremely high solid-phase conductivity, which render the material a viable alternative in compact heat exchanger design. The material is also mechanically stable, non-corrosive and relatively inert to temperatures up to approximately 500^{o}C, which make it particularly attractive for high-temperature non-oxydizing and moderate temperature oxidizing applications. Hydrodynamic and thermal engineering models are proposed based on recent work applied to air-water heat exchangers. The models are developed based on a unit-cube geometric model for carbon foam, a heat transfer model and well-established convective correlations that are extended to account for the effects of the carbon foam. The present calculations suggest that the use of carbon foam in a relatively simple configuration results in a significant reduction in thermal resistance accompanied by a rise in the hydrodynamic resistance. These preliminary results suggest that very compact heat transfer devices could be developed. With further investigation it is felt that the hydrodynamic resistance could be reduced while preserving the heat transfer performance resulting in very high-performance, compact heat transfer devices.

Keywords: *Carbon foam, Recuperator for micro-turbine, Thermal engineering model*

INTRODUCTION

The search for new heat transfer and structural materials is an ongoing effort connected to the development of recuperators for micro-turbines because of the high temperature, corrosive environment that such devices normally operate under. The heat transfer materials must ideally provide large surface area for heat transfer, a low mean friction factor, and must have stable chemical, thermal and mechanical properties that enable it to endure the operating conditions. The most widely used material in modern recuperators for micro-turbines is stainless steel and ceramics have only recently been considered for use. These materials are typically used to develop plate-fin and primary surface recuperators, both of which provide enhanced surface area using extended, impermeable surfaces. The present paper considers the use of porous carbon foam for the development of a new generation of enhanced surface compact gas-to-gas recuperators. Porous carbon foam has an open, interconnected pore structure and an extremely high conductivity, which combined, render the material an interesting alternative for heat exchanger design. To take full advantage of the porous structure of the foam, the recuperator must be designed to allow gas to flow through the foam where area enhancements of 5,000-50,000 m^2/m^3 are available, depending upon the porosity and pore diameter of the foam. Such increases in surface area would serve to reduce the thermal resistance of the heat transfer device, thereby raising its effectiveness.

Porous Carbon Foam, recently developed at Oak Ridge National Laboratory (ORNL), exhibits high quality geometric and thermal properties that may increase the heat transfer capacity for gas convection applications such as for recuperators on gas turbines. These unique properties include: 1) an extremely high bulk or stagnant effective thermal conductivity (with air) between 40 and 180 W/m K [1]. This

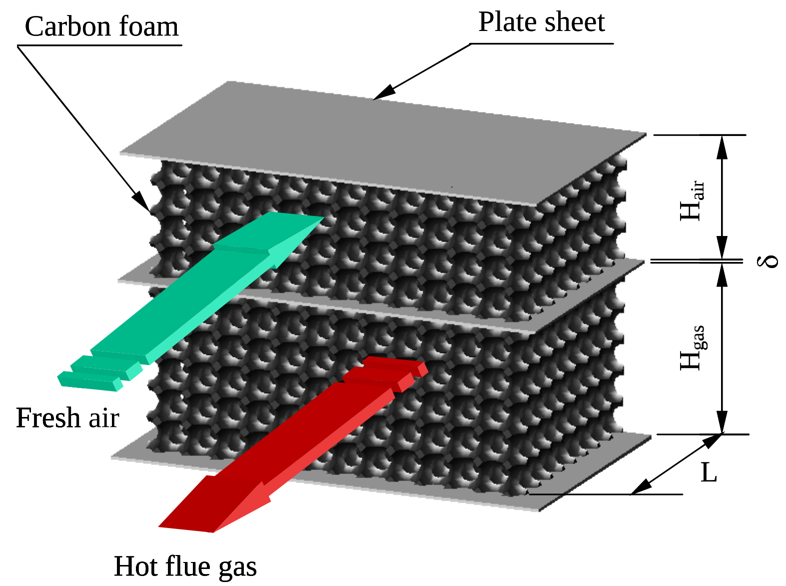

Figure 1: Flow passage configuration for recuperator made from carbon foam.

high effective conductivity results from the extremely high solid-phase conductivity of the graphitized material (k= 900-1700 W/m K). In contrast, similar porosity aluminum foams have effective conductivities of approximately 5-20 W/m K (with air), which result from a solid-phase conductivity of about 230 W/m K (for pure aluminum). As such, the carbon foam has a much higher capability to conduct heat into its internal structure so that infiltrated fluid can convect heat away; 2) an open, inter-connected void structure that enables fluid infiltration such that enormous increases in surface area for heat transfer are available (5,000 to 50,000 m^2/m^3); 3) a low density (from 0.2 to 0.6 g/cm^3, depending upon porosity), which makes the material suitable for compact and lightweight applications. In contrast, aluminum foam has a density of 0.3 to 0.8 g/cm^3 depending upon porosity; 4) An increase in exposed surface area and a rough open structure, which leads to increased mixing at the external fluid interface; and 5) an inert composition that is resistant to structural changes and corrosion for operating temperatures up to 500^{o}C.

Recent investigations of the thermal performance of air-water radiators made using carbon foam finned tubes suggests that the thermal performance of the carbon foam fins with optimized pore structure and fin configuration gives a 10-15% improvement over the aluminum fins [2, 3]. It is also suggested by [3] that more significant improvements can be obtained if the heat transfer devices are designed to take better advantage of the internal structure of the porous carbon foam. The unique combination of (extremely) high conductivity and open, interconnected internal structure enables the material to be used to construct effective extended volumes for gas passage such that both the fresh air and flue gas can be exposed to very high surface areas thereby reducing the convective resistances and improving the performance of the heat exchanger. The present paper presents an investigation into the use of porous carbon foam in a compact, enhanced surface recuperator. The approach used in [3] is employed here to develop hydrodynamic and thermal engineering models to determine the hydrodynamic and thermal resistances, respectively, of the recuperator such that the potential benefits of carbon foam can be explored. The implications of using porous carbon foam for compact recuperators are then discussed giving an overall practical assessment.

The remaining sections of this paper describe: the alternate flow passage configuration of the recuperator made from carbon foam, the hydrodynamic and the thermal models, and preliminary calculations to illustrate the potential benefits of using carbon foam for enhanced surface recuperators.

FLOW PASSAGE CONFIGURATION

Figure 1 shows the flow pass configurations of a recuperator made from carbon foam. The cold fresh air and the hot flue gas are arranged in counter flow and both streams are forced through a volume of porous carbon foam (porous channel), based on the suggestion of [3]. The thickness of porous channel on the hot gas side, H_{gas}, is greater than that on the cold air side, H_{air}, due to the fact that both the mass flow rate and the temperature of the hot gas are higher than the cold air. L is the length of the cold air and hot gas channels along the flow direction. δ is the thickness of the plate or parting sheet between the cold air and the hot gas.

HYDRODYNAMIC MODEL

A hydrodynamic model is required to calculate the pressure drops on both the cold air and the hot gas sides for a given geometry and flow condition. In the present case the geometry is dependent upon the flow passages' height (H_{gas} and H_{air}) and length (L), and the porosity and void diameter of the porous carbon foam used to construct the cold air and hot gas porous

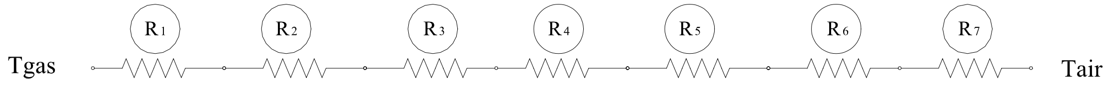

Figure 2: Thermal resistance circuit of carbon foam recuperator.

flow channels. The pressure losses across the porous channels are determined by a formulation combining the Darcy-Weisbach [4] and Darcy-Forchheimer extended equations [5] expressed as:

$$\Delta P = \sum C \frac{\rho_{avg}}{2} V_{avg}^2 + \left(\frac{\mu_{avg}}{K} V_{avg} + \frac{c_f}{\sqrt{K}} \rho_{avg} V_{avg}^2 \right) L \qquad (1)$$

where $\sum C$ is the sum of contraction and expansion coefficients at the foam channel entrance and exit. ρ_{avg} and μ_{avg} are the fluid density and dynamic viscosity at the fluid average temperature. V_{avg} is the fluid velocity in the foam channel, also called the filter velocity, at the fluid average temperature. K and c_f are the permeability and the Forchheimer coefficient of the carbon foam and can be determined using a unit-cube geometry model for the carbon foam [6]:

$$K = \frac{36\varepsilon^3}{147\beta^2} \qquad (2)$$

$$c_f = 0.0928 \frac{1}{\varepsilon^{3/2}} \qquad (3)$$

where ε is the porosity of the foam. β is the surface area to volume ratio and is determined by [6]:

$$\beta = \frac{\pi D}{H^3} (3H - 4D), \qquad (4)$$

and D is the void diameter of the foam. H is the height of a characteristic unit-cube and is determined by solving a cubic equation derived from the definition of the porosity of porous media [6].

THERMAL MODEL

Two heat transfer modes are active in the exchange of heat in the configuration of the carbon foam recuperator as shown in Fig. 1. The first mode is the convection heat transfer between the fluid and the internal surface of the foam. The other is conduction heat transfer through the plate or parting sheet of the primary surface, the bond contact layers and the fouling layers on the both the air and gas sides. The governing equation used in the design of heat exchangers can be expressed in the general form:

$$q = \frac{\Delta T_m}{R_t} \qquad (5)$$

where ΔT_m is the mean temperature difference between the two fluids, and R_t is the total thermal resistance, which can be obtained from a thermal-electrical analogy of the heat exchanger. The thermal circuit for the present case, carbon foam recuperator, is shown in Fig. 2. R_1 and R_7 are the fouling (i.e. buildup and blockage) resistances on the hot flue gas side and the cold fresh air side. R_2 and R_6 are the convection thermal resistances on the hot gas and cold air side. R_3 and R_5 are the bond contact thermal resistance on the hot gas and cold air side. R_4 is the conduction resistance through the plate or parting sheet between the cold air and the hot gas. The total thermal resistance is the sum of R_1 to R_7 according to the resistance circuit shown in Fig. 2 and is expressed as:

$$R_t = R_1 + R_2 + R_3 + R_4 + R_5 + R_6 + R_7 \qquad (6)$$

The fouling resistance on the hot flue gas side, R_1 is modeled using $R_1 = r_{gf}/A_{tg}$ where r_{wg} is the fouling factor [7] and A_{tg} is the total heat transfer surface area on the hot gas side. The fouling resistance on the cold air side, R_7 is modeled using the same expression, but accounting for the different fouling factor and surface area on the cold air side. The bond contact resistance on the hot gas side is obtained from $R_3 = t_{bc}/k_{bc}A_{tb}$, where t_{bc} is the bond thickness, k_{bc} is the thermal conductivity of the bond material and A_{tb} is the area of the bond surface. The bond contact resistance on the cold air side is the same as that on the hot gas side if the same bonding process is used, thus $R_5 = R_3$. The plate sheet conduction resistance R_4 is expressed as $R_4 = \delta/kA_{tb}$, where k is the conductivity of the plate sheet material. The convective resistance of the cold fresh air or the hot flue gas flow channel is approximated by:

$$R_x = \frac{D_E}{\eta_o A_t k_{avg} \overline{Nu_{sf}}} \qquad (7)$$

where the subscript x stands for either the hot gas or the air. k_{avg} is the thermal conductivity of fluid (either the hot gas or cold air) evaluated at its average temperature. A_t is the heat transfer surface area on either the cold air or the hot gas side and is determined by:

$$A_t = \Lambda_t \beta \qquad (9)$$

here Λ_t is the total foam volume for either the hot gas or the cold air flow channel, and β is the interior surface area to volume ratio of the foam determined by Eq. 4 .

The overall surface efficiency on the extended surface made of carbon foam Λ_o is determined by:

$$\eta_o = 1 - \frac{A_f}{A_f + A_b} (1 - \eta_f) \qquad (10)$$

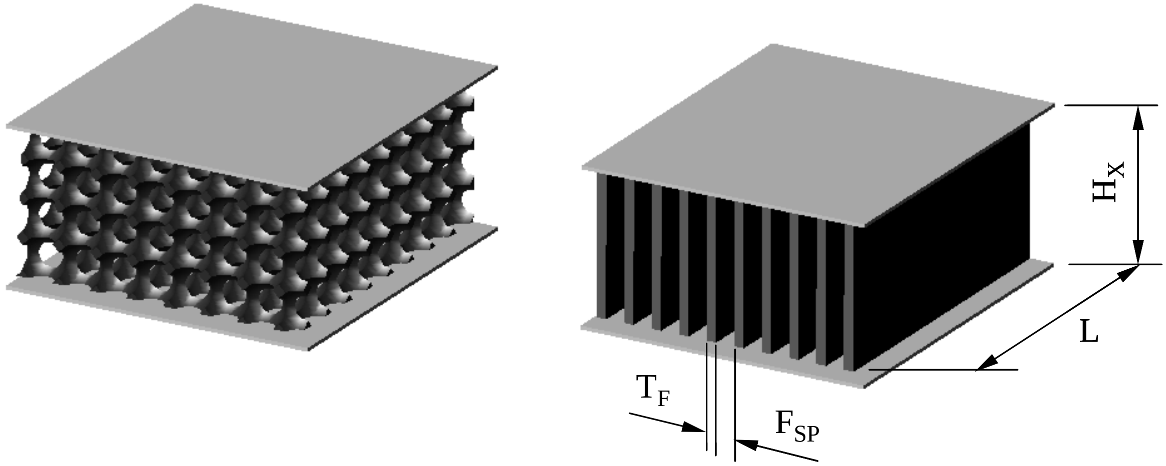

Figure 3: Equivalent micro-fin configuration for the flow channel of carbon foam.

where A_f, A_b and η_f are the fin surface area, the prime surface area and fin efficient of an equivalent micro-fin as shown in Fig. 3. T_F and Fsp are the equivalent fin thickness and space between two fins, and are obtained by [8]:

$$T_F = \sqrt{12\varepsilon K} \tag{11}$$

$$F_{SP} = \sqrt{\frac{12K}{\varepsilon}} \tag{12}$$

K is the permeability of the foam determined by Eq. (2). The equivalent fin efficient η_f in Eq. (10) then is determined by

$$\eta_f = \frac{\tanh\sqrt{2\overline{Nu_{sf}}\frac{k_{avg}}{k_s}\frac{H_F}{D_E}\frac{H_F}{T_F}\left(1+\frac{T_F}{L}\right)}}{\sqrt{2\overline{Nu_{sf}}\frac{k_{avg}}{k_s}\frac{H_F}{D_E}\frac{H_F}{T_F}\left(1+\frac{T_F}{L}\right)}} \tag{13}$$

here H_F is equivalent effective fin height and is half of the channel height, $H_{Fe}= H_x/2$. k_s and k_{avg} are the thermal conductivity of the solid phase of the foam at the film temperature and the thermal conductivity of the fluid an its average temperature. $\overline{Nu_{sf}}$ is the pore-level average Nusselt number on either the gas or the air side, which is dependent upon the pore-level Reynolds number, $Re_d= V_{avg}D_E/\nu_{avg}$. Here D_E is the equivalent particle diameter obtained from [9]:

$$D_E = \frac{6(1-\varepsilon)}{\beta} \tag{14}$$

For $Re_d \leq 75$, the average pore-level Nusselt number is obtained from [10, 11]:

$$\overline{Nu_{sf}} = 0.004\left(\frac{d_v}{D_E}\right)^{0.35} \mathrm{Re}_d^{\,1.35}\,\mathrm{Pr}^{1/3} \tag{15}$$

where d_v is the equivalent diameter of the void phase determined by [6]:

$$d_v = \sqrt[3]{\frac{6\varepsilon}{\pi}}H\ . \tag{16}$$

For $Re_d \geq 350$ the average pore-level Nusselt number is obtained from [12]:

$$\overline{Nu_{sf}} = 1.064\,\mathrm{Re}_d^{\,0.59}\,\mathrm{Pr}^{1/3} \tag{17}$$

For $75 < Re_d < 350$, linear interpolation between Eqs. 15 and 17 is to be used [13].

PRELIMINARY RESULTS

The hydrodynamic and thermal models are used to carry out preliminary calculations to illustrate the effects of the porous carbon foam on the performance of compact recuperators. Recall that Ott et al. [2] constructed an air-water heat exchanger using porous carbon foam fins and conducted a series of tests to quantify the thermal and hydrodynamic performance. While the heat exchanger considered in [2] is for water-to-air heat transfer, it is still possible to use the results of the study to investigate the potential benefits of the foam channel configuration proposed for the compact recuperator. Any improvement observed by modifying the air-side of the air-water heat exchanger in [2] can be applied to both sides of the air/gas recuperator to further reduce the overall thermal resistance. Since it is the potential benefits that are explored herein, it is appropriate to make initial comparisons to a similar, though not identical device.

To enable a comparison in performance, calculations are done for a case where the finned structure in [2] is simply replaced by a block of porous foam, while maintaining the frontal area of the heat exchanger. In this manner, all of the air is forced through the interconnected porous channel thereby exposing the air to the internal surface area of the foam. Table 1 gives a summary of the results for these calculations. The first column in the table shows the results from [2] using a finned configuration. The second column shows the results from the proposed engineering models for the case where the finned section is simply replaced by a foam block of the same overall volume. In this case, the effectiveness becomes 100% owing to the increase in surface area and the reduction in air-side thermal resistance, but the pressure drop across the air-side is nearly 110 times higher than the finned case. This suggests that the amount of foam used can be reduced

Table 1: Comparisons of thermal performance of carbon foam fin and block channels

	Carbon foam fin	Full length foam block	Shortened foam block
1. Porous carbon foam properties			
1.1 Pore diameter: μm	350	350	**400**
1.2 Porosity: %	75	75	**90**
1.3 Solid thermal conductivity: W/m.K	1200	1200	1200
2. Carbon foam flow channel			
2.1 Flow channel width: mm	50.8	50.8	50.8
2.2 Flow channel height: mm	3.158	3.158	3.158
2.3 Flow channel depth: mm	38.1	38.1	**1.444**
3. Thermal conditions			
3.1 Air inlet temperature: ºC	31.6	31.6	31.6
3.2 Air flow rate: kg/min	0.189	0.189	0.189
3.3 Base plate temperature: ºC	98.8	98.8	98.8
4. Geometrical information			
4.1 Internal heat transfer surface area: m^2	0.00822	**0.05023**	**0.00121**
4.2 Weight of carbon foam: g	0.9168	**3.056**	**0.0463**
5. Thermal performance			
5.1 Pore-level heat transfer coefficient: $W/m^2.K$	162.1165	**1900.6**	**1391.23**
5.2 Overall surface efficient: %	97.86	**80.765**	**78.215**
5.3 Air side thermal resistance: $m^2.K/W$	0.76126	**0.01297**	0.76094
5.4 Heat exchanger effectiveness: %	33.87	**100.00**	33.87
5.5 Heat load dissipated: W	72.3269	**213.55**	72.3283
5.6 Air pressure drop: kPa	0.5563	**60.8863**	**0.799**
6. Ratio of air pressure drop	1	**109.45**	**1.42**

considerably. The last column in the table shows the results obtained by reducing the length of the foam block to the extent that the heat load from the finned case is recovered. A length reduction from 38.1 mm to 1.44 mm was required to match the thermal resistance of the finned case. For this case, the pressure drop has been reduced substantially, but is still approximately 1.4 times higher than the finned case. Note, however, that the volume occupied by the foam-block heat exchanger is more than 20 times smaller than the foam-finned case, which implies that very compact heat transfer devices could be designed using porous carbon foam. Since it would be impractical to consider building a 1.44 mm thick sheet of carbon foam, the porous channel concept considered herein may be a better approach, although much more development must be done to devise a flow path that gives a practical balance between the thermal and hydrodynamic resistances.

It is also of interest to note that the heat exchanger constructed by [2] does not necessarily represent the ideal carbon foam-finned configuration and thus, the improvements given above may be artificially high. Yu et al. [3] conducted a parametric study and showed that the air-side thermal resistance of a carbon foam finned-tube heat exchanger can be reduced by reducing the foam porosity and pore diameter, increasing the fin height and density, and reducing the fin thickness. While a more efficient heat exchanger design was proposed, it is not known whether a core of the proposed geometry could be constructed, or whether the resulting core would have the structural integrity required for a practical application. Similar comparisons to the carbon foam-finned tube heat exchanger considered by [3] still suggest factor of 5 size reductions while maintaining the thermal performance. Thus, preliminary calculations suggest that there are potential benefits that could be derived using porous carbon foam to reduce the thermal resistance in compact recuperators.

Despite the high thermal performance and the above mentioned advantages of the carbon foam, there remain concerns about the utility and reliability of porous carbon foam for recuperator applications. These concerns include: 1) the current temperature limit; a higher operating temperature limit would allow higher inlet flue gas temperatures; 2) fouling effects and erosion caused by particles in the flue gas; 3) bond contact quality under high operating temperature; and 4) differences in the heat expansion of the carbon foam and plate materials due to the high temperature difference between the intake cold fresh air and the exhausting hot flue gas. All of these practical issues are the subject of ongoing research.

CONCLUSIONS

A preliminary investigation into the use of porous carbon foam for compact recuperator design has been carried out. Hydrodynamic and thermal models have been proposed so that calculations for a simple porous channel configuration could be made. Calculations made to compare a finned configuration to a porous block configuration suggest that significant size reductions can be obtained while maintaining the heat load of the heat exchanger. The reduction in thermal resistance is, however, accompanied by a slight rise in the hydrodynamic resistance. The results suggest that there is potential for the use of carbon foam in compact recuperator design, but the flow passages must be arranged to give a more practical balance between the thermal and hydrodynamic resistances.

ACKNOWLEDGMENTS

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