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# A performance analysis of porous graphite foam heat exchangers in vehicles

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## HIGHLIGHTS

► Graphite foam wavy corrugated fins present good thermal and flow characteristics.
► The graphite foam wavy corrugated fin heat exchanger presents a high power density.
► A high compactness factor is provided by the graphite foam heat exchanger.
► The graphite foam heat exchanger has a low coefficient of performance (duty/pumping loss).
► Graphite foam heat exchangers have great potential in vehicle cooling applications.

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ABSTRACT

Due to the increasing cooling power and space limitation in vehicles, a new compact heat exchanger – graphite foam heat exchanger is proposed for vehicle cooling application. The graphite foam has high thermal conductivity (the effective thermal conductivity is 40–150 W/m K) and low density (0.2–0.6 g/cm$^3$), but it has high flow resistance which is a problem in heat exchanger applications. In order to find a graphite foam heat exchanger with low flow resistance, four different configurations (baffle, pin-finned, corrugated, and wavy corrugated) of graphite foam fins are analyzed in terms of thermal performance and pressure drop by using a computational fluid dynamics approach. The simulation results show that the wavy corrugated foam presents high thermal performance and low pressure drop. Moreover, a comparative study between the wavy corrugated foam heat exchanger and a conventional aluminum louver fin heat exchanger is carried out to evaluate the performance of graphite foam heat exchangers in terms of coefficient of performance (removed heat/air pumping loss), power density (removed heat/mass of heat exchangers), and compactness factor (removed heat/volume of heat exchangers). Finally, this paper concludes that graphite foam heat exchangers should be further developed in vehicles, and presents several recommendations for how such development can be promoted.



## 1. Introduction

Due to the high thermal conductivity of metal materials, aluminum or copper heat exchangers are very popular in vehicles. However, with the increased power production and reduced under-bonnet space, vehicle cooling becomes a more serious problem than before. In order to increase the thermal performance of heat exchangers in vehicles, it is important to apply extended surfaces on the air side to compensate for the low heat transfer coefficient. Thus, the cooling surface of heat exchangers has to be increased to dissipate the tremendous cooling power. However, because of space limitations in vehicles, there is not much available space to increase the size of heat exchangers, which has led to an urgent need to develop a new compact heat exchanger with high thermal performance for vehicle cooling.

Due to its big specific surface area, a porous medium at a small size might be a good choice for the development of new compact heat exchangers. Compared to a metal foam [1–4], a graphite foam developed by Oak Ridge National Laboratory [5] has extremely high thermal conductivity. Several research studies on the characteristics of graphite foams have been carried out [6–8]. These studies show that the characteristics of graphite foams are as follows:

I. High thermal conductivity: The effective thermal conductivity of graphite foam, which is a weighted average of the solid material and the pores where a fluid is passing, is between 40 and 150 W/m K [8]. This is much higher than the

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effective thermal conductivity of aluminum foam (between 2 and 26 W/m K [1]).

II. Low density: The density of graphite foam ranges from 200 to 600 kg/m$^3$, which is about 20% of that of aluminum.

III. Large specific surface area: Because of the open pores and inter-connected void structure, the specific surface area of graphite foam is between 5000 and 50,000 m$^2$/m$^3$ when the pore size is around from 500 μm to 10 μm respectively [8].

IV. Weak mechanical properties: The tensile strength of graphite foam is much less than that of a metal foam. The weak mechanical properties block the development of the graphite foam heat exchanger. Adding additional material into the graphite foam or changing the fabrication process might improve the foam's mechanical properties.

Based on these characteristics, the graphite foam has become a very promising material for heat exchangers. For example, Klett et al. [9] designed a radiator with carbon foam. In their study, the cross section of the automotive radiator was reduced from 48 cm × 69 cm to 20 cm × 20 cm. The reduced size enabled a substantial decrease of the overall weight, cost and volume of the cooling system. Furthermore, Yu et al. [10] proved that the thermal performance of a carbon foam finned tube radiator could be improved by 15% compared to a conventional aluminum finned tube radiator without changing the frontal area or the air flow rate or pressure drop. Also Garrity et al. [11] carried out an experimental comparison between a carbon foam heat exchanger and a multi-louvered fin heat exchanger. They found that the carbon foam samples brought away more heat than the multilouvered fin when the volume of the heat exchangers was the same.

Even though there is a huge heat transfer enhancement in the graphite foam, the graphite foam is still associated with other problems. The most important issue is that there is a high pressure drop due to the large hydrodynamic loss associated with the cell windows connecting the pores [12]. In a study concerning reduction of the pressure drop, Gallego and Klett [13] presented six different configurations of graphite foam heat exchangers. That study showed that the solid foam had the highest pressure drop while the finned configuration had the lowest pressure drop. In another study, Leong et al. [14] found that the baffle foam presented the lowest pressure drop among four configurations of graphite foams at the same heat transfer rate. Lin et al. [15] revealed that a corrugated foam could reduce the pressure drop while maintaining a high heat transfer coefficient compared to the solid foam. All together, these studies illustrate that the configuration has an important effect on the pressure drop through the graphite foam.

The present study concerns a computational fluid dynamics (CFD) analysis, with the aim to evaluate what graphite foam fin configuration is presenting the lowest pressure drop and highest thermal performance among baffle, pin-finned, corrugated and wavy corrugated graphite foam fins. Moreover, in order to predict the performance of graphite foam heat exchangers in vehicles, the graphite foam fin with low pressure drop and high thermal performance is compared with a conventional aluminum louver fin in terms of (1) coefficient of performance (COP, how much heat can be removed by a certain input pumping power), (2) power density (PD, how much heat can be removed by a certain mass of the fins), and (3) compactness factor (CF, how much heat can be removed in a certain volume).

## 2. Physical model and assumptions

### 2.1. Physical model

A simplified model of a plate-fin heat exchanger is shown in Fig. 1. Four different configurations (baffle, pin-finned, corrugated, and wavy corrugated) of the graphite foam, which are equivalent fins, are placed between two water tubes. As shown in Fig. 1, the hot water flows inside the flat tubes, and the cold air flows through the porous carbon foam. The heat is transmitted through the tube wall and the graphite foam porous cell surface and finally it is dissipated to the air. There are many parameters to describe the configuration of the graphite foam heat exchanger. The overall size of the four configurations of the graphite foam core is 1.2 cm ($z$-direction) × 4.5 cm ($y$-direction) × 5 cm ($x$-direction). The details of the configurations and geometries are shown in Fig. 2. The important parameters of the graphite foam are described in Table 1, which is based on Ref. [12]. The fluid is assumed to be incompressible with constant properties and in steady-state. The water tubes are made of aluminum. Due to the large heat transfer coefficient between the hot water and the inner wall of the tube, as well as the high thermal conductivity of the pipe wall, the water tube is assumed to be at constant temperature. The connection between the tube wall and the graphite foam is assumed perfect without any air gap inside. Thus, the thermal resistance at the interface between the tube wall and the graphite foam is neglected.

### 2.2. Modeling assumptions

Before the numerical computations, a discussion of the computational model (laminar or turbulent) in adoption of the flow regime is carried out. In the comparison among the four configurations of the graphite foam, the inlet air speed is selected to be in the range from 12 m/s to 20 m/s based on the speed of vehicles. Correspondingly, the Reynolds number based on the frontal velocity and the hydraulic diameter ($D_h$) is ranging from 15,120 to 25,200. Thus turbulent flow prevails on the air side. However, inside the graphite foam the flow is laminar. This is so, because it is difficult to generate turbulent eddies in the small open pores of the foam.

The effect of turbulence on the flow field is implemented by the "renormalization group" (RNG) $k$–$\varepsilon$ turbulence model. On the other hand, due to the laminar flow inside the graphite foam, the RNG $k$–$\varepsilon$ turbulence model might be useful to take into account of low-Reynolds number effect near the foam walls.

### 2.3. Computational domain

In order to make sure the graphite foam fin is located in the fully developed flow region, the computational domain is extended upstream 1.5 times the graphite foam fin length to eliminate the entrance length effect. Similarly, the computational domain is extended downstream 5 times the length of the graphite foam fin to achieve the one-way coordinate assumption at the domain outlet. Thus, the whole stream length of the computational domain is 7.5 times the actual graphite foam fin length, as shown in Fig. 3.

## 3. Mathematical formulation and numerical method

### 3.1. Governing equations

According to the above presented assumptions, the governing equations for continuity, momentum and energy may be expressed as follows, see Refs. [16–18]:

#### 3.1.1. Air zone governing equations (turbulent flow)

Continuity equation

$$\frac{\partial(\rho_{\mathrm{air}} u_i)}{\partial x_i} = 0 \tag{1}$$