OPTIMIZATION OF AN EXHAUST HEAT EXCHANGER USING METAL FOAM BAFFLES FOR ORC WASTE HEAT RECOVERY SYSTEM

Tianyu Chen[1], Gequn Shu[1]*, Hua Tian[1]*, Xiaonan Ma[1], Tingting Zhao[1], Hongfei Zhang[1], Zhao Zhang[1]

[1] State Key Laboratory of Engines, Tianjin University
Tianjin, China
sgq@tju.edu.cn

* Corresponding Author
Gequn Shu, Hua Tian

ABSTRACT

Heat exchanger is an important component in ORC waste heat recovery systems (WHRs), and shell and tube heat exchanger is one of the most common heat exchangers. Compared with traditional metal baffle heat exchanger, the heat transfer area of metal-foam baffle heat exchanger (MF-STHE) is enhanced due to the geometry of metal-foam baffles, which results in a significantly increase in heat transfer performance. Properly designed segmental metal foam baffles can reduce pressure loss and enhance heat transfer in MF-STHE. In present investigations, a 3D numerical model of an exhaust heat exchanger using metal-foam segmental baffles is established to optimize its performance. Firstly, the vector distribution is analyzed. Then, some MF-STHE models with various porosities and pore densities are simulated under same mass flow rate (1 kg/s) for parametric study. The area goodness factor $j/f^{1/2}$ is considered as one evaluation factor to reveal the comprehensive performance of different MF-STHEs. The results show that the comprehensive performance of the MF-STHE with $\varepsilon = 0.71$ and PPI = 20 is the optimum one in all MF-STHEs considered in this paper.

Keywords: Exhaust heat exchanger, Metal-foam baffle, ORC waste recovery system

1. INTRODUCTION

Energy conservation and environmental protection have attracted widespread concern in various regions and countries over the past decade (Miró et al. 2016). The research reports by diverse researchers and institutes (Berntsson et al. 2015, Johnson et al. 2008, Zhang et al. 2009) reveal that considerable heat is wasted to surroundings in industrial production process. Thus, as an effectual measure, ORC waste heat recovery system is widely used in industry activities (Shu et al. 2016).

In many ORC systems, exhaust gas is considered as a high-grade energy (Wang et al. 2017), and is commonly recovered by exhaust heat exchangers in most applications. To achieve the most effective results of ORC system, the heat transfer and pressure drop performance of heat exchangers should be optimized as much as possible (Yang et al. 2016). The shell-and-tube heat exchanger (STHE) is the most common heat exchanger in diverse industrial production processes (Peng et al. 2007), and baffles are employed in STHEs to improve the heat transfer performance. Nevertheless, owing to the structure limitations, baffles also cause some disadvantages like stagnant zones and large pressure drop (Wang et al. 2018a). To ameliorate the comprehensive performance of STHEs, various improved baffles, such as trefoil-hole baffles (Ma et al. 2017) and helical baffles (Yang et al. 2015) are studied by numerous scholars. Zhang et al. (2013) have applied segmental baffles and helical baffles in oil cooler to optimize its performance, and conclude that the performance of oil cooler with helical baffles can obviously be better than that of oil cooler with segmental baffles by about 51.8%-76.4%.

However, these innovate baffles cannot totally eliminate stagnant zones. The metal foam baffles can be exploited to address this problem. As a porous medium, part of working fluid can flow through metal foam baffles to avoid the appearance of stagnant zones. Furthermore, using metal foam will noticeably improve the heat transfer performance of exhaust heat exchangers due to its large area-to-volume ratio (Mahdi et al. 2017). Alvandifar et al. (2018) performed a numerically investigation to study the performance of partially metal-foam wrapped pipe bundle. They pointed out that the area goodness factor of metal-foam wrapped pipe bundle is superior to that of bare pipe bundle by about 114%.

Due to the effectiveness, the optimization of metal-foam baffles is feasible and necessary. As a porous medium, the disturbance on fluid flow caused by metal-foam baffles is far less than that caused by traditional metal baffles. However, the disturbance on fluid flow also offers extra heat transfer. The objective of the current study is to optimize the performance of the exhaust heat exchangers using metal-foam baffles for ORC waste heat recovery systems. To achieve this purpose, various metal-foam samples with different porosities and pore densities are analyzed to offer appropriate flow resistance.

2. MODELING

2.1 Computational domain

Fig. 1 illustrates the physical model and computational domain of MF-STHE studied in this paper, and the detailed geometric parameters are listed in Table 1. In most exhaust heat exchangers, the thermal resistance of exhaust side is much larger than that of coolant side (Wang et al. 2018b). Therefore, only exhaust side of MF-STHE is considered in this paper, and an isothermal border of 360K is employed on pipe walls. It should be noted that, all metal-foam baffles used in this paper are made of Cu metal foams. Besides, for the sake of the accuracy of the numerical model, the material properties of gas such as density and viscosity are set in the form of polynomial functions (Chen et al. 2019). The boundary conditions and fluid properties of the exhaust gas are listed in Table 2.

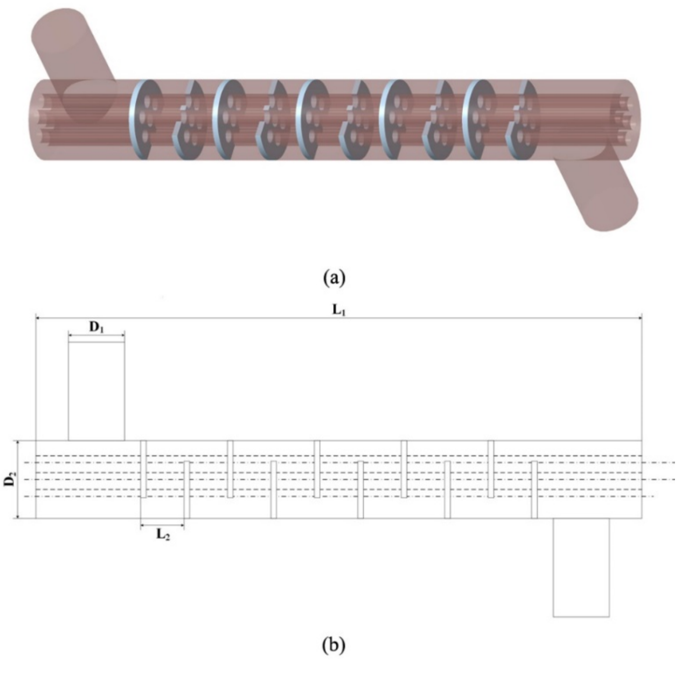

Figure 1: (a) Physical model and (b) computational domain of MF-STHE

Table1: Detailed geometric parameters of MF-STHEs

MF-STHE	Dimensions and description
Shell parameters	
Diameter of inlet/outlet D_1 (mm)	49
Diameter of shell D_2 (mm)	68
Effective length L_1 (mm)	530
Tube parameters	
Numbers	7
Diameter D_3 (mm)	12
Layout	Equilateral triangle
Spacing L_2 (mm)	17
Baffle parameters	
Numbers	10
Spacing L_3 (mm)	38
Material	Cu metal-foam

Table 2: The boundary conditions and fluid properties of the exhaust gas

Parameters	Values
Boundary conditions	
Mass flow rate (kg/s)	0.1
Inlet temperature of exhaust (K)	747.42
Outlet pressure of exhaust (Pa)	0
Fluid properties	
Density (kg/m^3)	2.166914-4.105207 $\times 10^{-3}$x+2.488116 $\times 10^{-6}$x^2
specific heat capacity (J/kg/K)	959.9381+0.2591265x
thermal conductivity (w/m/K)	6.465345 $\times 10^{-3}$+6.35358 $\times 10^{-5}$x
Viscosity (kg/m/s)	7.16745$\times$10^{-6}+3.6535$\times$10^{-8}x

To complete the simulation, three assumptions are performed as follows:

- Metal foams employed in this paper are homogeneous and isotropic with constant tortuosity and porosity.
- The material properties of gas and Cu metal-foam such as density, thermal conductivity are regarded to be pressure-independent.
- Natural convection and radiative heat transfer are not considered.

2.2 Numerical method

The governing formulas comprise continuity, Darcy-Forchheimer-Brinkman momentum equation and energy equation. To evaluate the thermal condition in metal foams, the local thermal non-equilibrium (LTNE) is considered in this paper (Xu et al. 2015). The formulas for fluid and solid phase are listed respectively:

$$\left(\rho c_p\right)_f v \cdot \nabla T_f = \nabla \cdot k_e^f \nabla T_f - h_{sf} a_{sf}\left(T_f - T_s\right) \quad (1)$$

$$0 = \nabla \cdot k_s^e \nabla T_s + h_{sf} a_{sf}\left(T_f - T_s\right) \quad (2)$$

It should be noted that the permeability and inertia coefficient of porous media are computed as claimed by Yang et al. (2014) and Calmidi (1998):

$$K = \frac{\varepsilon\left[1-(1-\varepsilon)^{1/3}\right]}{108\left[(1-\varepsilon)^{1/3}-(1-\varepsilon)\right]} d_p^2 \quad (3)$$

$$C_F = 0.00212(1-\varepsilon)^{-0.132}(\frac{d_f}{d_p})^{-1.63} \quad (4)$$

An unstructured grid is established by ANSYS ICEM for computation. To seek out the optimum grid, three different numbers of grids (3729094, 4420939, 5022804) are considered as reference. The result reveals that difference of heat transfer coefficient between 3729094 and 4420939 quads is about

0.8 %, and the difference between 4420939 and 5022804 quads is just 0.4 %. Thus, the grid with 4420939 quads is adopted for numerically simulation ultimately.
The commercial software (ANSYS fluent) is used for numerical simulation, and the detail about numerical method can be found in Chen et al. (2019). The mass flow rate inlet and pressure outlet are regarded as appropriate boundary conditions, and the accuracy of the numerical method for metal foams is validated in our previous study, details can be found in Wang et al [27].

3. RESULTS AND DISCUSSION

3.1 Contours

The velocity vectors of MF-STHE in the symmetry plane when m = 0.1 kg/s are depicted in Fig. 2. In traditional metal baffle shell and tube heat exchangers (MB-STHEs), the stagnant zones occurred near the baffles (Ozden et al. 2010). However, as shown in Fig. 2, part of exhaust gas flows through the metal-foam baffles rather than being blocked, which is beneficial for reducing pressure loss. In addition, the average velocity of exhaust gas in flow direction is gradually reduced. Because, the exhaust density increases significantly with decreasing temperature.

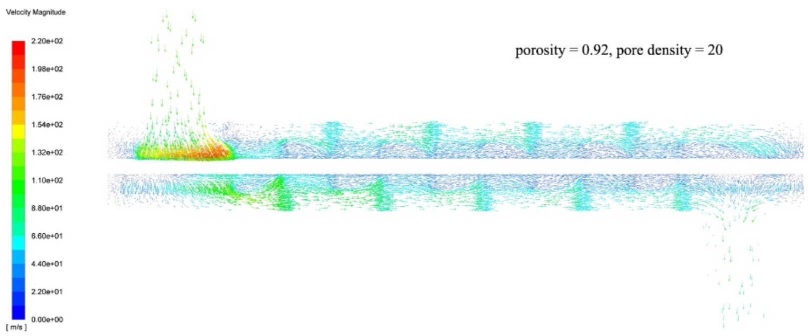

Figure 2: The velocity vectors of MF-STHE in the symmetry plane when m = 0.1 kg/s

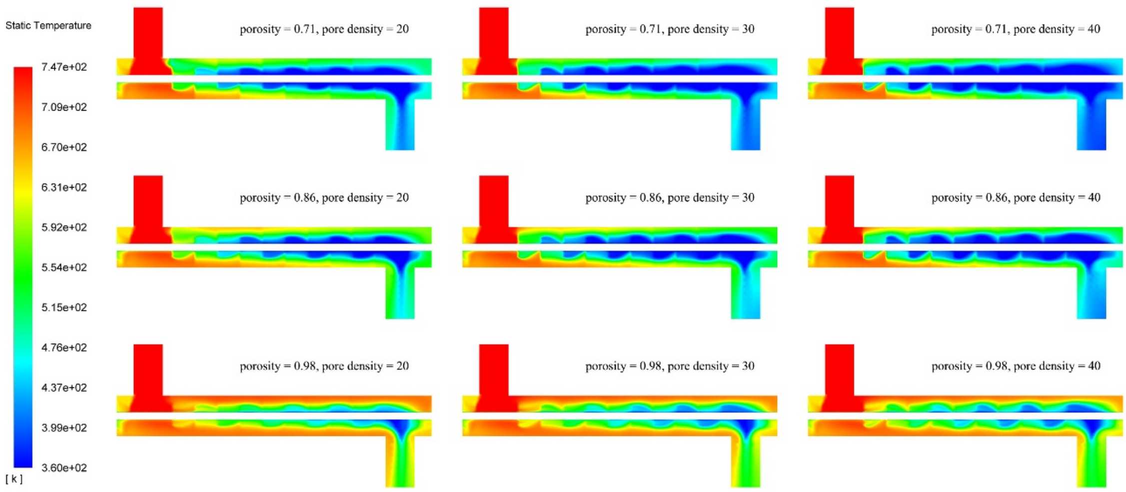

Figure 3: The temperature distribution of MF-STHEs with different physical parameters in the symmetry plane when m = 0.1 kg/s

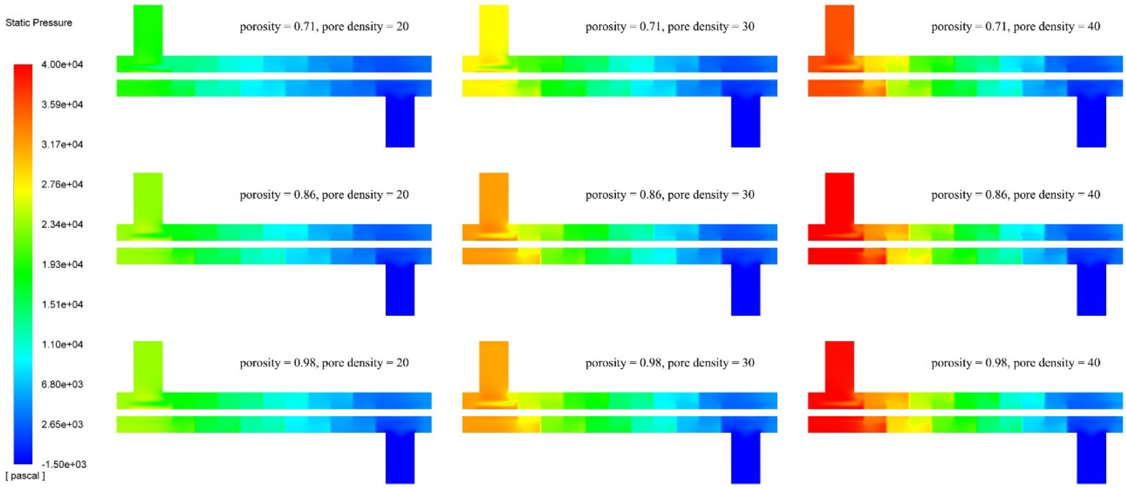

Figure 4: The pressure distribution of MF-STHEs with different physical parameters in the symmetry plane when m = 0.1 kg/s

Temperature and pressure distribution of MF-STHEs with different porosities and pore densities are shown schematically in Fig. 3 and 4. From the simulation results, all temperature distributions follow the same trend. It should be noted that the exhaust temperature changes notably near each baffle. This phenomenon reveals that the structure of metal-foam is extremely beneficial for heat transfer performance of exhaust heat exchangers. According to the pressure distributions, compared with porosity, variation in pore density has a more remarkable impact on pressure loss of MF-STHEs.

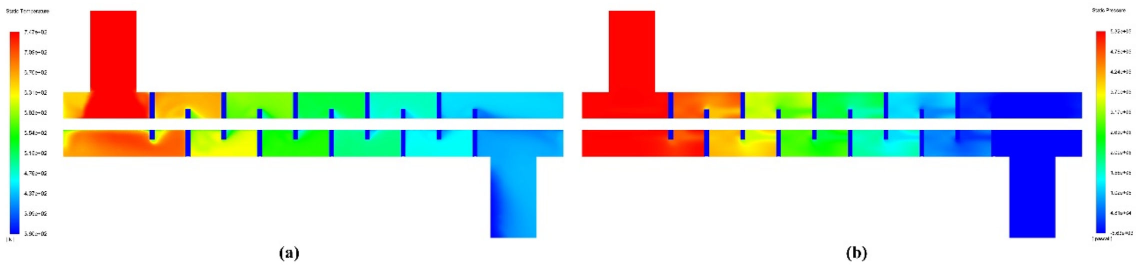

Figure 5: The (a) temperature and (b) pressure distribution of heat exchanger with traditional Cu metal baffles in the symmetry plane when m = 0.1 kg/s

The heat exchanger with traditional Cu metal baffles (MB-STHE) when m = 0.1 kg/s is also simulated in this paper for further comparison, and the temperature and pressure distribution in the symmetry plane are shown in Fig. 4. According to Fig. 4 (a), the outlet temperature of the MB-STHE is 424.53 K, and the heat transfer coefficient is defined as:

$$h = \frac{Q}{A \cdot \Delta T_m}$$

where Q is the total heat transfer, A is the heat transfer area and ΔT_m is the log mean temperature difference, so the heat transfer coefficient of MB-STHE is 1027.70. Besides, as shown in Fig.4 (b), the pressure drop of MB-STHE is about 5.28×10^5 Pa, which is much higher than MF-STHEs. In summary, compare with MB-STHE, the MF-STHEs can achieve similar heat transfer with much lower pressure drop under the same work condition.

3.2 Heat transfer and pressure drop

Fig. 6 (a) illustrates the heat transfer coefficient of MF-STHEs at various porosities and pore densities when m = 0.1 kg/s. As indicated in Fig. 6 (a), the heat transfer coefficient increases with increasing pore density and decreasing porosity, and the impact of pore density is more significantly with lower porosity.

Fig. 6 (b) schematically shows the pressure drop versus porosity and pore density. The results show that the pressure drop increases firstly and then decrease with the increase of porosity. This phenomenon can be interpreted as follows: on one hand, the heat transfer of MF-STHEs increases with the decrease of porosity, thus the average density of exhaust gas decreases and the average velocity increases; on the other hand, the flow resistance decreases with increase porosity due to increasing permeability. Especially, as described in the model performed by Bhattacharya et al. (2002), inertial coefficient also decreases with porosity when $\varepsilon > 0.97$. In detail, before the pressure drop reaches to the apex value, the increase of flow velocity of exhaust gas is dominant which causes the enhanced pressure drop. After that, the decrease of flow resistance is dominant which causes the less pressure drop. In addition, when pore density of metal-foam baffles varies from 20 to 40, pressure drop of MF-STHEs monotonously increases.

3.3 Comprehensive performance

The area goodness factor $j/f^{1/2}$ is adopted as an index to evaluate the comprehensive performance of MF-STHEs. The friction factor j and Colburn factor f are defined as:

$$j = \frac{Nu}{Re \cdot Pr^{1/3}} \tag{5}$$

$$f = \frac{2D\Delta P}{L_1 \rho u^2} \tag{6}$$

The area goodness factor $j/f^{1/2}$ of MF-STHEs at various porosities and pore densities when m = 0.1 kg/s is depicted in Fig. 6 (c). The results reveal that the impact of porosity has more greatly impact on comprehensive performance than that of pore density, and the MF-STHE with smaller porosity and smaller pore density has better performance. To summarize, the MF-STHE with $\varepsilon = 0.71$ and PPI = 20 is the optimum one in all MF-STHEs considered in this paper. Besides, compared with the poor-performing MF-STHE ($\varepsilon = 0.98$, PPI = 40), the area goodness factor of optimum MF-STHE increases by about 190.85%. Thus, adopting proper metal-foam baffles is highly beneficial to the comprehensive performance of exhaust heat exchangers.

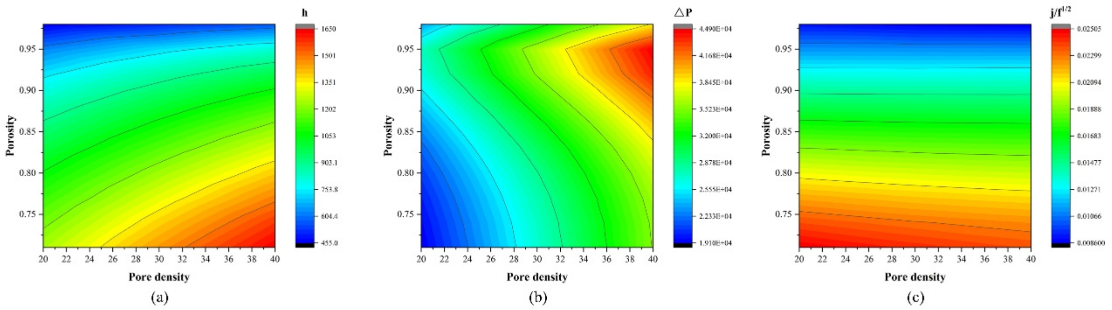

Figure 6: (a) Heat transfer coefficient, (b) pressure drop and (c) area goodness factor of MF-STHEs at various porosities and pore densities when m = 0.1 kg/s

4. CONCLUSIONS

In this study, a numerical model is established to optimize an exhaust heat exchanger using metal foams. The thermal-hydraulic performance of MC-STHEs with different porosities (0.71 - 0.98) and pore densities (20 - 40) are analyzed. The results show that appropriate choices can noticeably improve the comprehensive performance of the heat exchanger. The following major conclusions can be drawn out:

- Compared with tradition metal baffles, adopting metal-foam baffles can completely prevent the occurrence of stagnant zones. Besides, the metal-foam baffles are very beneficial to thermal performance of MF-STHEs

- The impact of porosity and pore density on heat transfer and pressure drop are analyzed. The heat transfer coefficient increases with increasing pore density and decreasing porosity. The pressure drop increases firstly and then decrease with the increase of porosity, and pressure drop of MF-STHEs monotonously increases with pore density.
- The comprehensive performance of MF-STHEs are evaluated by area goodness factor $j/f^{1/2}$. The result shows that the MF-STHE with ε = 0.71 and PPI = 20 is the optimum one in all MF-STHEs, and proper metal-foam baffles can significantly improve the comprehensive performance of MF-STHEs.

NOMENCLATURE

A	heat transfer area	(m^2)
a_{sf}	surface area density	(–)
c_p	specific heat	(J/kg·K)
C_F	Forchheimer coefficient	(–)
D	diameter	(mm)
d_p	pore diameter	(mm)
d_f	fibre diameter	(mm)
f	friction factor	(–)
h	heat transfer coefficient	($W/m^2 \cdot K$)
j	Colburn factor	(–)
k	thermal conductivity	(W/m·K)
K	permeability	(m^2)
m	mass flow rate	(kg/s)
Nu	Nusselt number	(–)
PPI	Pores per linear inch	(–)
Pr	Prandtl number	(–)
Q	heat transfer capacity	(W)
T	temperature	(K)
u	velocity	(m/s)
ε	porosity	(–)
ρ	density	(kg/m^3)
Subscript		
f	fluid	
s	solid	

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