UNIVERSITY OF CALIFORNIA
RIVERSIDE

An Investigation and Characterization of Metal Foam Filled Double-Pipe Heat Exchangers

A Thesis submitted in partial satisfaction
of the requirements for the degree of

Master of Science

in

Mechanical Engineering

by

Xi Chen

August 2014

Thesis Committee:
Dr. Kambiz Vafai, Chairperson
Dr. Marko Princevac
Dr. Guanshui Xu

The Thesis of Xi Chen is approved:

Committee Chairperson

University of California, Riverside

ACKNOWLEDGEMENTS

I would like to thank my advisor Prof, Kambiz Vafai, who has been a truly exceptional mentor, for his encouragement and guidance during the graduate study. I would also like to thank my committee members of Prof. Marko Princevac and Prof. Guanshui Xu, each of whom have taught me a great deal about the similarities and differences of science and engineering. I appreciate their suggestions and comments during the graduate study.

I would like to thank my parents, family, friends and colleagues for their support during this period. I extend my thanks to my lab mates and class mates Shujuan Wang, Xing Yuan, Ana Quezada, Arman Haghighi, Tolou Tavakoli, Xiaolong Ouyang and Pei Wang for helping me with my thesis and other school work at UCR. Finally, I would like to thank Mechanical Engineering administrative staff s Sarah Nosce and Paul Talevera for answering my many questions about the study and living in UCR.

DEDICATION

This document is dedicated to my parents and girlfriend Wenjing Xu, who have always given me supports to pursue my dreams.

ABSTRACT OF THE THESIS

An Investigation and Characterization of Metal Foam Filled Double-Pipe Heat Exchangers

by

Xi Chen

Master of Science, Graduate Program in Mechanical Engineering
University of California, Riverside, August 2014
Dr. Kambiz Vafai, Chairperson

The effect of using metal foams in double-pipe heat exchangers is investigated in this work. The advantages and drawbacks of using metal foams in these types of heat exchangers is characterized and quantified. The analysis starts with an investigation of forced convection in metal foam filled heat exchangers using Brinkman-Forchheimer-extended Darcy model and the Local Thermal Equilibrium (LTE) energy model. An excellent agreement is displayed between the present results and established analytical results. The presented work enables one to establish the optimum conditions for the use of metal foam filled double-pipe heat exchangers.

Keywords: Metal Foam, Double-Pipe, Heat Exchangers

TABLE OF CONTENTS

Acknowledgements iv

Dedication v

Abstract vi

Table of Contents vii

List of Tables viii

List of Figures ix

Nomenclature x

Chapter 1 Introduction 1

Chapter 2 Physical problem 4

Chapter 3 Mathematical Formulation and Boundary Conditions 5

3.1 Mathematical Formulation 5

3.2 Boundary Conditions 5

3.3 Parameter Specifications 6

Chapter 4 Validation 9

Chapter 5 Results and discussions 11

5.1 Case 1: The Metal Foam Filled Pipes 11

5.2 Case 2: Double-Pipe Heat Exchanger Filled with a Metal Foam 12

Chapter 6 Conclusions 15

References: 26

LIST OF TABLES

Table 1 Pertinent parameters for metal foam filled heat exchangers 16

Table 2 Comprehensive performance characteristics of metal foam heat exchangers compared to the plain tube heat exchanger 17

LIST OF FIGURES

Fig. 1 Schematic of the double-pipe heat exchanger filled with metal foams: a) A single pipe filled with a metal foam, (b) Cross sectional view of the double-pipe heat exchanger. 18

Fig. 2 Comparison of the fully developed Flow and Temperature fields in a rectangular duct: a) Velocity field, b) Temperature field, c) the Nusselt number. 19

Fig. 3 Comparison of the fully developed flow and temperature fields in a circular pipe: a) Velocity field, b) Temperature field. 20

Fig. 4 Comparison of the velocity fields in a metal foam filled double-pipe heat exchanger. 21

Fig. 5 Variation of the pressure drop with Darcy number in a metal foam filled pipe: a) Pressure drop, b) Friction factor. 22

Fig. 6 Effect of Reynolds number, Darcy number and thermal conductivity variations on the Nusselt number for a metal foam filled pipe: a) Effect of Re_D and Da variations, b) Effect of k_s variations. 23

Fig. 7 Variation of pressure drop and friction factor variations with Darcy number in a metal foam filled double-pipe heat exchanger: a) Pressure drop, b) Friction factor. 24

Fig. 8 Comparison of the heat transfer performance in a metal foam filled double-pipe heat exchanger with a plain double-pipe heat exchanger: a) Total heat transfer coefficient, b) Ratio of total heat transfer coefficient. 25

Nomenclature

A = surface area, m^2

c_f = specific heat, J/(kg K)

D = pipe diameter, m

Da = Darcy number

d_p = pore size, m

d_f = fiber diameter of metal foam, m

F = inertial coefficient

h = convective heat transfer coefficient, W/(m^2 K^1)

K = Permeability, m^2

k = thermal conductivity, W/(m K)

m = mass flow, kg/s

Nu = Nusselt number

P = pressure, Pa

Q = Power, W

q = heat flux, W/m^2

R = pipe radius, m

T = temperature, K

U = overall heat transfer coefficient, W/(m^2 K)

u = velocity, m/s

ε = porosity of the porous medium

μ = viscosity, kg/(m s)

ρ = density, kg/m^3

Subscripts

e = effective value

f = fluid phase

s = solid phase

m = mean value

w = wall

Chapter 1

Introduction

Metal foam is a porous medium that has a solid metal matrix with empty or fluid-filled pores. Because metal foams have both functional and structural properties, they have been utilized in a wide range of sectors and industries, including transportation, defense, aerospace, architectural designs and energy industries. Metal foams can be classified as closed or open cell type. Open-cell metal foams have the following characteristics: 1) large surface area, which improves energy absorption and heat transfer[1]; 2) extended foam ligaments normal to the flow direction, which results in fluid mixing, boundary layer disruption and an increase in fluid turbulence[2]; and 3) lightweight, high strength and rigidity. Therefore, metal foams have a high potential to enhance heat transfer. For example, Lin et al. [3] demonstrated that the solid foam radiator is significantly more efficient than the fin radiator in enhancing the heat transfer performance. Boomsma et al. [4] have shown that the compressed aluminum foam heat exchanger has nearly half the thermal resistance and significantly higher heat transfer efficiency as compared to the regular heat exchanger. Mahjoob and Vafai [5] have pointed out that although there is an increase in the pressure drop, the use of metal foam leads to a substantial heat transfer enhancement that can make up for the increased pressure loss.

There has been an increased interest in establishing metal foam properties such as the effective thermal conductivity, permeability, and inertial coefficient. Calmidi and Mahajan [6] developed a two-dimensional model to obtain the effective thermal conductivity of the metal foams. A one-dimensional analytical heat conduction model was developed based on a three-dimensional geometry of the metal foam cells by Boomsma and Poulikakos [7] which displayed a good agreement with the experimental data. Bhattacharya et al. [8] researched the same topic based on the analysis of Calmidi and Mahajan [6] and found that the effective thermal conductivity depends more on the porosity than the pore density. Their model was more realistic for high porosity metal foams. The permeability and the inertial coefficient, which are two key parameters of open-cell metal foams, have been studied by various researchers. These two parameters are highly structural dependent, and much of the prior works on packed beds and granular porous media with a porosity of 0.3-0.6 has been utilized to investigate their dependence on the pore size and porosity. However, packed bed attributes may not be directly applicable to metal foams. As such, Calmidi [9] proposed a model based on his study of Porvair foams to derive specific formulation that relates the permeability and the inertial coefficient in terms of the porosity and pore size of metal foams. Zhao et al. [10] also investigated determination of permeability and inertial coefficient for highly porous metal foams.

As mentioned earlier, metal foams with open cells can be regarded as a porous medium. So the fluid flow through these foams can be described by Darcy's Law. However, Darcy's Law is limited to laminar flow and when the Reynolds number based on the pore size is in the range of 1 to 10. An increase in the velocity will lead to deviation from Darcy's law [11]. To solve this problem, Vafai and Tien [12] have proposed a generalized model to incorporate the boundary and inertial effects. For modeling the heat transfer in porous medium, the energy equation model proposed by Vafai and Tien [12] is utilized. It should be noted that some additional assumptions are required when this energy equation is used: 1) small temperature differences between the fluid and solid phase on a local basis, i.e the solid and fluid phases are in local thermal equilibrium (LTE); 2) natural convection and radiation in the porous medium can be neglected; The LTE model has been utilized by various investigators in the analysis of forced convection through a porous medium [13-15].

Heat transfer performance of open-cell metal foams has been investigated by several investigators. These works have mainly focused on the fundamental rectangular channel geometry. Forced convection in metal foam filled pipes have also been investigated [16]. However, the double-pipe heat exchangers filled with metal foams which are the most pertinent metal foam heat exchangers used in various industrial applications have rarely been considered. This could be due to the complexity of the coupling process between the fluids which leads to a temperature and heat flux distribution which is not constant at the interface of the two pipes. To simplify the model and reduce the computational time, prior works have considered a constant heat flux or a constant temperature at the interface. But this does not reflect the actual conditions in the operation of double-pipe heat exchangers. Therefore, in this work, the Brinkman-Forchheimer-extend-Darcy and local thermal equilibrium equations are utilized to simulate metal foam filled heat exchangers while incorporating the coupling effect between the inner and outer pipes and the intervening fluids. The influence of different parameters such as different Darcy numbers on fluid and thermal performance of metal foam filled pipes is addressed. The heat transfer performance of practical double-pipe heat exchangers filled with metal foams is also analyzed and compared with that of plain tube heat exchangers in this study, to quantify several aspects of optimum operating conditions for these types of devices.

Chapter 2

Physical Problem

The problem under consideration is based on forced convective flow through a pipe filled with metal foams shown in Fig. 1a. The liquid flows through the metal foam filled pipe with a diameter of 2R and length L. The pipe wall is impermeable and subject to a variable heat flux $q_w(\mathrm{z})$ which is determined based on the coupling conditions.

The inner pipe is inside an outer pipe constituting a simple double-pipe heat exchanger, as shown in Fig. 1b. The inner and outer pipes, with diameters R_1 and R_2 respectively, are both filled with metal foams. The cold and hot liquids flow in opposite directions in a counter-flow arrangement through the inner and the annular sections. The heat is transferred from the hot water in the outer annular pipe to the cold water in the inner pipe (or the other way around) through the inner pipe's wall subject to the coupling conditions. The outer pipe is assumed to be insulated.

Chapter 3

Mathematical Formulation and Boundary Conditions

3.1 Mathematical Formulation

As mentioned earlier, the Brinkman-Forchheimer-extended-Darcy flow model and the local thermal equilibrium based energy equation are utilized in our analysis. It is assumed that the properties of the metal foam and the fluid are homogeneous and isotropic. The radiation and natural convection are considered to be negligible. The governing equations can be written as [12, 15, 17]:

$$\frac{\mu_f}{\varepsilon}\left(\frac{\partial^2 u}{\partial r}+\frac{1}{r}\frac{\partial u}{\partial r}\right)-\frac{\mu_f}{K}u-\rho_f\,\frac{F\varepsilon}{K^{1/2}}u^2-\frac{dp}{dz}=0 \tag{1}$$

$$\rho_f c_f u\frac{dT}{dz}=k_e\left(\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}\right) \tag{2}$$

3.2 Boundary Conditions

3.2.1 Boundary conditions for metal foam filled pipes

For a metal foam pipe subjected to a constant heat flux, the gradients of the velocity and temperature in the r direction vanish at the centerline. The appropriate boundary conditions are as follows:

$$\frac{du}{dr}=0\,,\ \frac{\partial T}{\partial r}=0 \text{ at } r=0 \tag{3}$$

$$u=0\,,\ q_w=k_e\frac{\partial T}{\partial r} \text{ at } r=R \tag{4}$$

3.2.2 Boundary conditions for double-pipe heat exchangers filled with metal foams

(1) Inlet boundary conditions

a. Inner pipe: $T=T_{f1,in}$; and the velocity ($u_{f1,in}$) is calculated based on the specified inner Reynolds number, Re_{D1}.

b. Outer pipe: $T=T_{f2,in}$; and the velocity ($u_{f2,in}$) is calculated based on the specified outer Reynolds number, Re_{D2}.

(2) Boundary conditions at the walls

a. Inner pipe: The wall thickness of the inner pipe is considered to be negligible and the coupling conditions are applied across the inner wall.

b. Outer pipe: The outer pipe is considered to be adiabatic

3.3 Parameter Specifications

Several parameters, such as the permeability (K), effective thermal conductivity (k_e), and inertial coefficient (F), need to be calculated to initiate the numerical simulation. The permeability K, for the open-cell metal foams, can be obtained through a formulation proposed by Calmidi [9] based on his experimental data, given as

$$\frac{K}{d_p^2} = 0.00073(1-\varepsilon)^{-0.224}\left(d_f / d_p\right)^{-1.11} \tag{5}$$

where d_p is the pore size; and d_f is the fiber diameter of the metal foam. These quantities are in turn obtained from

$$d_p = \frac{0.0254}{PPI} \tag{6}$$

and

$$\frac{d_f}{d_p} = 1.18\sqrt{\frac{1-\varepsilon}{3\pi}}\left(\frac{1}{1-e^{-((1-\varepsilon)/0.04)}}\right) \tag{7}$$

where PPI signifies pores per inch. The effective thermal conductivity of a fluid filled porous medium can be estimated by [1]

$$k_e = \varepsilon k_f + (1-\varepsilon)k_s \tag{8}$$

To determine the inertial coefficient, the following correlation, which is based on the experimental data for metal foams, was proposed by Zhao et al.[10].

$$F = C\left[(1-\varepsilon)^n / d_p\right] \cdot K^{1/2} \tag{9}$$

where $C = 29.613$ and $n = 1.5226$ for the alumina alloy foams while $C = 7.861$ and $n = 0.5134$ for the copper foams.

The Reynolds number for the inner and annular pipes is represented as:

$$\mathrm{Re}_D = \frac{D_h \cdot u_m \cdot \rho_f}{\mu_f} \tag{10}$$

where D_h is the hydraulic diameter equaling $2R_1$ for the inner section and $2\left(R_2 - R_1\right)$ for the annular section.

The following are two accepted definitions of the Reynolds number for a porous medium. The first one is the permeability based Reynolds number given by Eq. (11) [12, 18, 19].

$$\mathrm{Re}_K = \frac{\rho u \sqrt{K}}{\mu} \tag{11}$$

The other is the pore size based Reynolds number given by Eq. (12) [5].

$$\mathrm{Re}_{dp} = \frac{\rho u d_p}{\mu}(1-\varepsilon) \tag{12}$$

The Darcy number Da is defined as

$$Da = \frac{K}{H^2} \tag{13}$$

All the constant parameters needed in simulations are presented in Table 1.

Chapter 4

Validation

The dimensionless velocity u^* is defined as:

$$u^* = \frac{u}{u_\infty} \tag{14}$$

where u_∞ is the velocity outside the momentum boundary layer. The dimensionless temperature θ is expressed as:

$$\theta = \frac{T_w - T}{T_w - T_m} = \frac{T_w - T}{q_w / h} \tag{15}$$

where T_w, T_m, q_w and h are the wall temperature, mean temperature of the fluid, heat flux at the wall and the mean convective heat transfer coefficient, respectively.

The numerical results for the rectangular duct were compared with the analytical results of Vafai and Kim [15]. Fig. 2 shows the comparisons between the present dimensionless velocity, temperature distribution and the Nusselt number ($Nu = \frac{h \cdot D_h}{k_e}$) and those given in [15]. The numerical results were found to be in very good agreement with the analytical solutions of Vafai and Kim [15]. A further comparison was done for a circular pipe. Our results were compared with the results of Hooman et al. [14, 20]. Fig. 3 shows the comparisons between the present dimensionless velocity and temperature distributions and those given in [14, 20]. The maximum deviation was found to be less than 2%.

There is a scarcity of numerical and experimental data for forced convection in double-pipe heat exchangers filled with metal foams. Fig. 4 shows the comparison between our results and prior works for the velocity distributions for the inner pipes. As can be seen, we have an excellent agreement.

Chapter 5

Results and Discussions

5.1 Case 1: The Metal Foam Filled Pipes

5.1.1 Pressure drop

Fig. 5a presents the variation of pressure drop per unit length in metal foam pipes at different Reynolds and Darcy numbers. As expected, the metal foam pipe has a substantially larger pressure drop than a plain pipe under the same operating conditions. As expected, the pressure drop increases substantially with a decrease in the Darcy number due to the greater resistance to the flow at the lower Darcy numbers.

Fig. 5b shows the relationship between the friction factor ($f = \dfrac{\Delta\langle p\rangle 2H}{L\rho_f u_m^2/2}$) and the permeability based Reynolds number (Re_K). As can be seen in this figure, the friction factor (f) is nearly invariable after $\mathrm{Re}_K > 20$. This is because the inertial force (Forchheimer term which is proportional to square of velocity) becomes the dominant part of the pressure drop when Re_K becomes larger than 20.

5.1.2 Heat transfer performance

To examine and assess the heat transfer performance, the overall Nusselt number, defined as $Nu_f = \dfrac{hD}{k_f}$, is used. Fig. 6a shows the effects of the Reynolds number (Re_D) and the Darcy number (Da) on the overall heat transfer coefficient (Nu_f) for a metal foam filled pipe. As can be seen, the Nusselt number increases gradually with an increase in the fluid velocity. While a decrease in the Darcy number (the permeability) leads to an improvement in the Nusselt number. Comparisons of the heat transfer performance of a plain pipe and a metal foam filled pipe are also presented in Fig. 6a. As expected, metal foam filled pipes have a substantially higher Nusselt number (up to 50 times) as compared to a plain pipe.

To investigate the effect of different materials, metal foams, such as titanium foam ($k = 7$ W/m-K), stainless steel foam ($k = 16$ W/m-K), nickel foam ($k = 91$ W/m-K), alumina foam ($k = 202$ W/m-K), and copper foam ($k = 399$ W/m-K), are simulated at $\mathrm{Re}_{dp} = 5$ and 20. Results are presented in Fig. 6b which illustrates the almost linear increase in heat transfer with an increase in the thermal conductivity of the solid skeleton.

5.2 Case 2: Double-Pipe Heat Exchanger Filled with a Metal Foam

5.2.1 Pressure drop

Fig. 7a displays the effect of the Reynolds number on the pressure drop per unit length for metal foam pipes with $Da = 10^{-4}$ and $Da = 10^{-2}$. The relationship between the friction factor (f) and the permeability based Reynolds number (Re_K) is shown in Fig.7b.

As expected, the pressure drop increases with an increase in the Reynolds number and a decrease in the Darcy number. The viscous term (Darcy term) is the dominant factor when $\mathrm{Re}_K < 20$, and the inertia force (Forchheimer term) becomes dominant for the pressure drop after $\mathrm{Re}_K > 20$.

5.2.2 Heat transfer performance

The logarithmic mean temperature difference, which is the appropriate average temperature difference for the heat exchanger design, is employed. For the counter-flow arrangement, we can present:

$$\Delta T_1 = T_{f2\_in} - T_{f1\_out} \tag{16}$$

$$\Delta T_2 = T_{f2\_out} - T_{f1\_in} \tag{17}$$

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{In \dfrac{\Delta T_1}{\Delta T_2}} \tag{18}$$

and

$$Q = U \cdot A \cdot \Delta T_m \tag{19}$$

where A is the total surface area of the inner pipe.

The heat transfer Q can also be presented as:

$$Q = \dot{m}_1 c_{p1} \left(T_{f1,\mathrm{out}} - T_{f1,in} \right) = \dot{m}_2 c_{p2} \left(T_{f2,in} - T_{f2,out} \right) \tag{20}$$

where $\dot{m}_1$ and $\dot{m}_2$ are the mass flow rates of the cold and hot fluids (Kg/s); c_{p1} and c_{p2} are the specific heat of cold and hot fluids (J/Kg-K); and $T_{f1,in}$, $T_{f1,out}$, $T_{f2,in}$ and $T_{f2,out}$ are the average temperatures of cold and hot fluids flowing in and out of the pipes.

Fig. 8a displays the heat transfer performance for a plain double-pipe and a double-pipe which is filled with metal foams. Again, as can be seen, the total heat transfer increases when the Darcy number decreases and the Reynolds number increases.

Fig. 8b presents the ratio of the metal foam heat exchanger heat transfer coefficient for a double-pipe to that of the plain double-pipe heat exchanger. It can be seen that metal foams have a substantial effect on the heat transfer enhancement at lower Reynolds numbers. For example, the overall heat transfer coefficient is increased by 18 times when the Reynolds number is close to 1000. However, the influence of metal foams diminishes as the Reynolds number increases.

5.2.3 Comprehensive Performance Evaluation

The heat transfer performance and energy consumption are two important factors that should be considered when designing heat exchangers. For the evaluation of the heat transfer performance, the thermal load per unit length (Q/L) and the pumping power $P_p / L = \left(\Delta P / \mathrm{L}\right) \cdot V_f$, where V_f is the flow rate of the fluid, are utilized. The performance factor (I), given by Eq. (21), is introduced to assess the comprehensive performance.

$$I = \frac{\left(Q/L - P_p/\mathrm{L}\right)_{metal\ foam} - \left(Q/L - P_p/\mathrm{L}\right)_{plain\ tube}}{\left(Q/L - P_p/\mathrm{L}\right)_{plain\ tube}} \tag{21}$$

Table 2 shows the comprehensive performance characteristics of double-pipe heat exchangers with and without metal foams. As can be seen, inserting metal foams significantly increases the pressure drop. However, there is a remarkable improvement on the management of the thermal load (as much as 11 times), while the performance factor can increase by more than 700%.

Chapter 6

Conclusions

Forced convection in metal foam heat exchangers has been analyzed using the Brinkman-Forchheimer-extend-Darcy model and local thermal equilibrium energy model for a porous medium. Numerical results for the velocity and temperature profiles in metal foam pipes are first obtained for constant heat flux boundary conditions. The comparison between the numerical and analytical solutions displays the accuracy of our results. It can be seen that a lower Darcy number offers an enhancement in the heat transfer performance.

The effect of using metal foams in double-pipe heat exchangers as well as its considerable heat transfer enhancement over the plain-tube exchangers was investigated. Attributes related to the heat transfer enhancement at the expense of additional pressure drop were categorized. In this respect, a performance factor, was introduced and quantified to further help the evaluation and assessment of the compromise between heat transfer enhancement and pressure drop in designing a metal foam heat exchanger.

Table 1 Pertinent parameters for metal foam filled heat exchangers

Parameter	Value	Unit
c_{f1}	4.179×10^3	J/(kg*K)
k_{f1}	0.609	W/m/K
ρ_{f1}	997	kg/m^3
μ_{f1}	0.92×10^{-3}	Pa*s
c_{f2}	4.179×10^3	J/(kg*K)
k_{f2}	0.609	W/m/K
ρ_{f2}	997	kg/m^3
μ_{f2}	0.92×10^{-3}	Pa*s
R_1	0.01	m
R_2	0.02	m
T_{f1_in}	273.15	K
T_{f2_in}	373.15	K

Table 2 Comprehensive performance characteristics of metal foam heat exchangers compared to the plain tube heat exchanger

Case	Da	Cold water velocity (m/s)	Cold water flow rate (m^3/s)	Foam filled heat exchanger			Plain tube heat exchanger			$\frac{(\frac{\Delta P}{L})_{metal\ foam}}{(\frac{\Delta P}{L})_{plain\ tube}}$	$\frac{(\frac{Q}{L})_{metal\ foam}}{(\frac{Q}{L})_{plain\ tube}}$	I
				$\Delta P/L$ (Pa/m)	P_p/L (W/m)	Q/L (W/m)	$\Delta P/L$ (Pa/m)	P_p/L (W/m)	Q/L (W/m)			
1	10^{-4}	0.1	3.14×10^{-5}	28918	0.908	19986	15.94	0.0005	1783	1814	11.21	748%
2	10^{-4}	0.5	1.57×10^{-4}	544586	85.500	43268	193.84	0.0304	6518	2809	6.63	562%
3	10^{-2}	0.1	3.14×10^{-5}	2214	0.069	19629	15.94	0.0005	1783	139	11.00	732%
4	10^{-2}	0.5	1.57×10^{-4}	51645	8.108	42074	193.84	0.0304	6518	266	6.46	545%

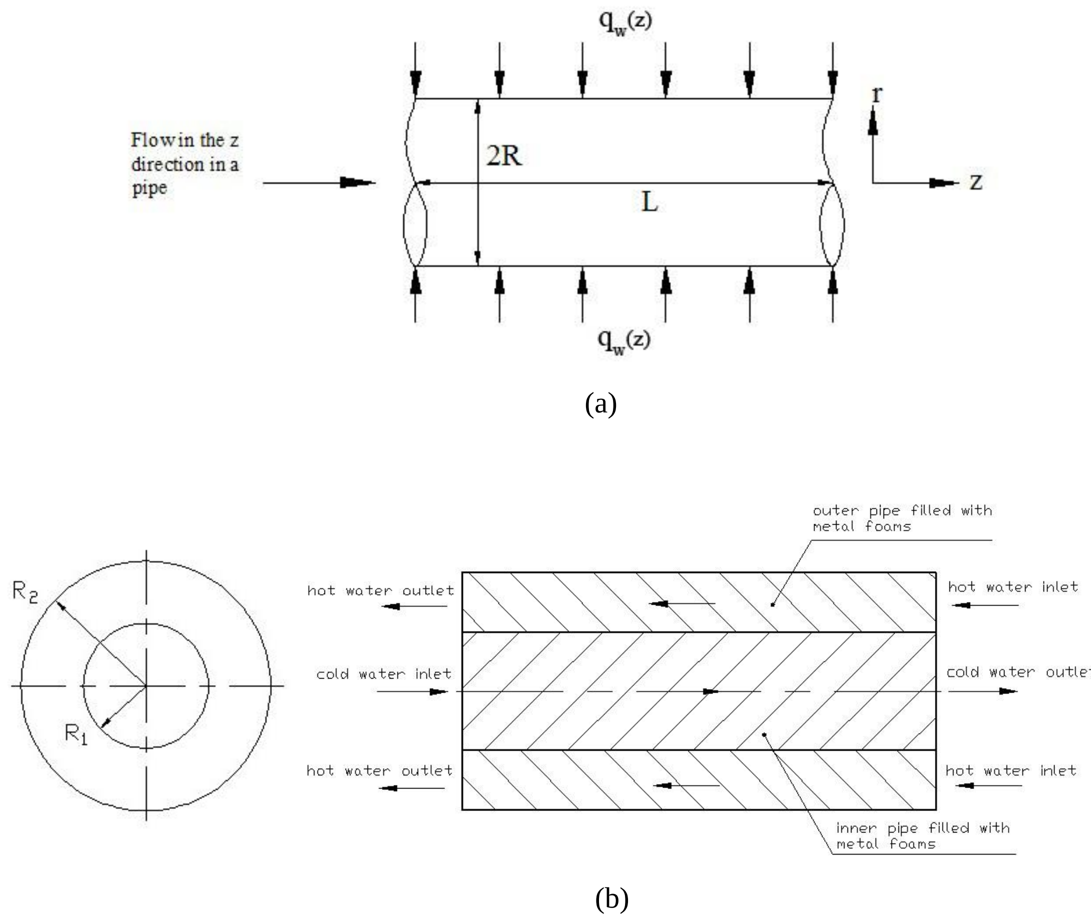
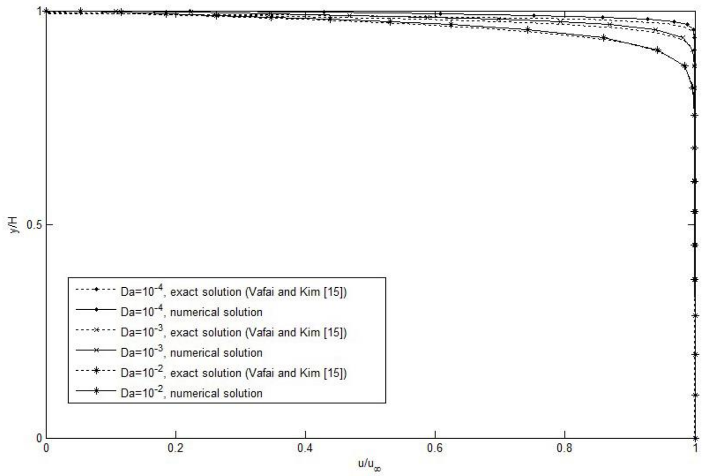
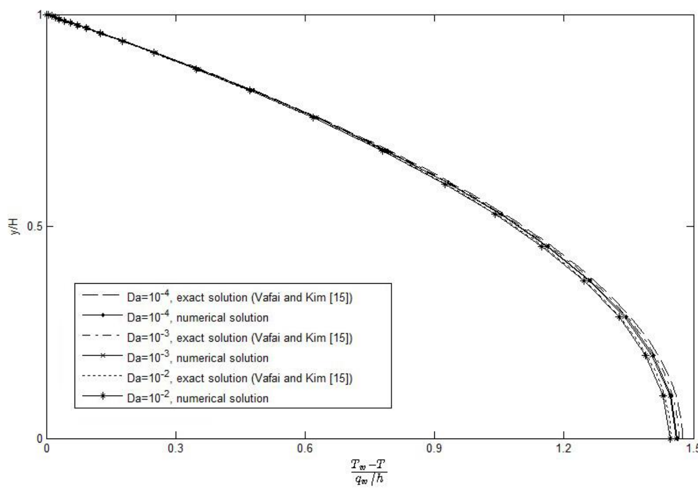


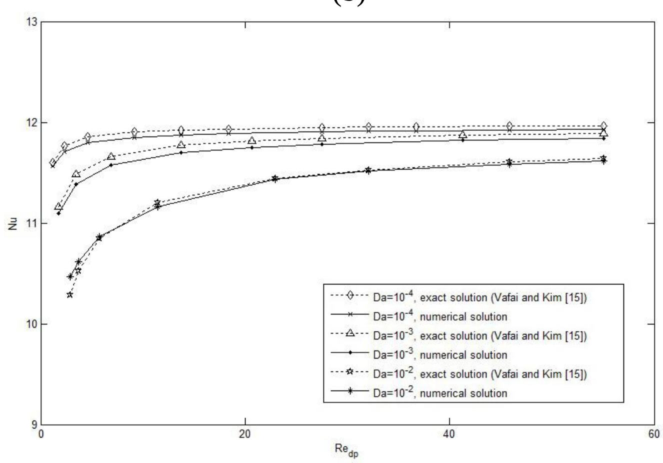
Fig. 1 Schematic of the double-pipe heat exchanger filled with metal foams: a) A single pipe filled with a metal foam, (b) Cross sectional view of the double-pipe heat exchanger.



(a)

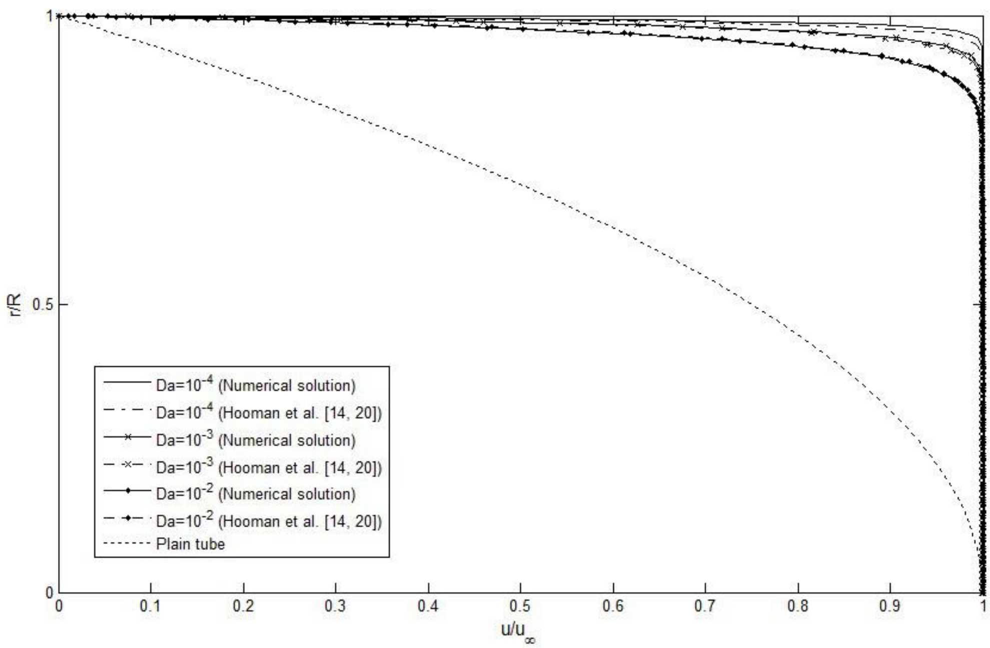


(b)

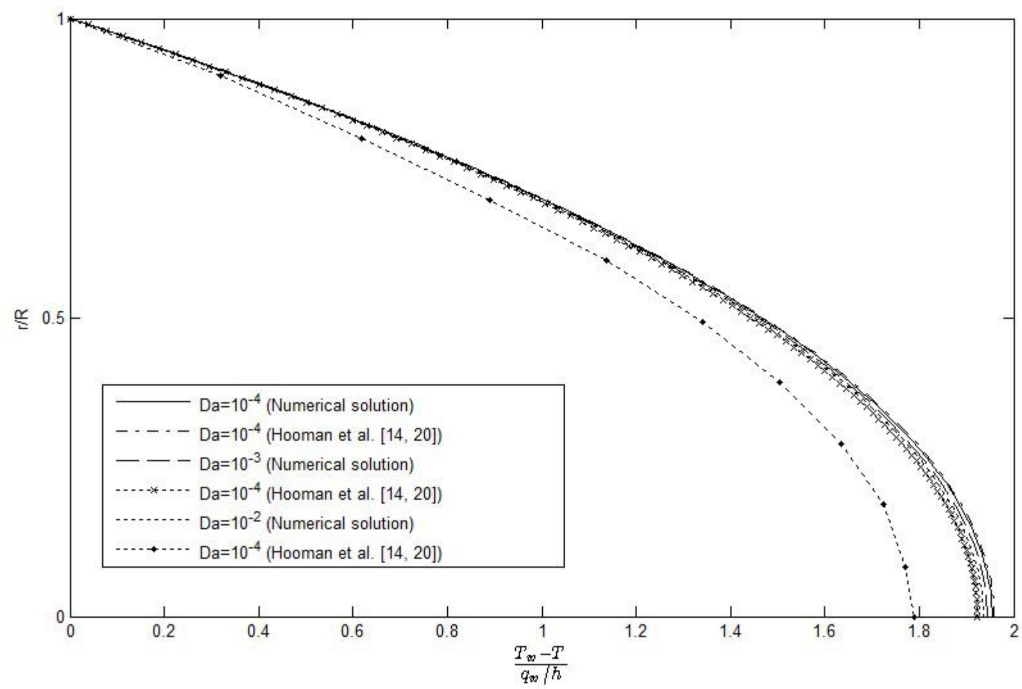


(c)

Fig. 2 Comparison of the fully developed Flow and Temperature fields in a rectangular duct: a) Velocity field, b) Temperature field, c) the Nusselt number.



(a)



(b)

Fig. 3 Comparison of the fully developed flow and temperature fields in a circular pipe: a) Velocity field, b) Temperature field.

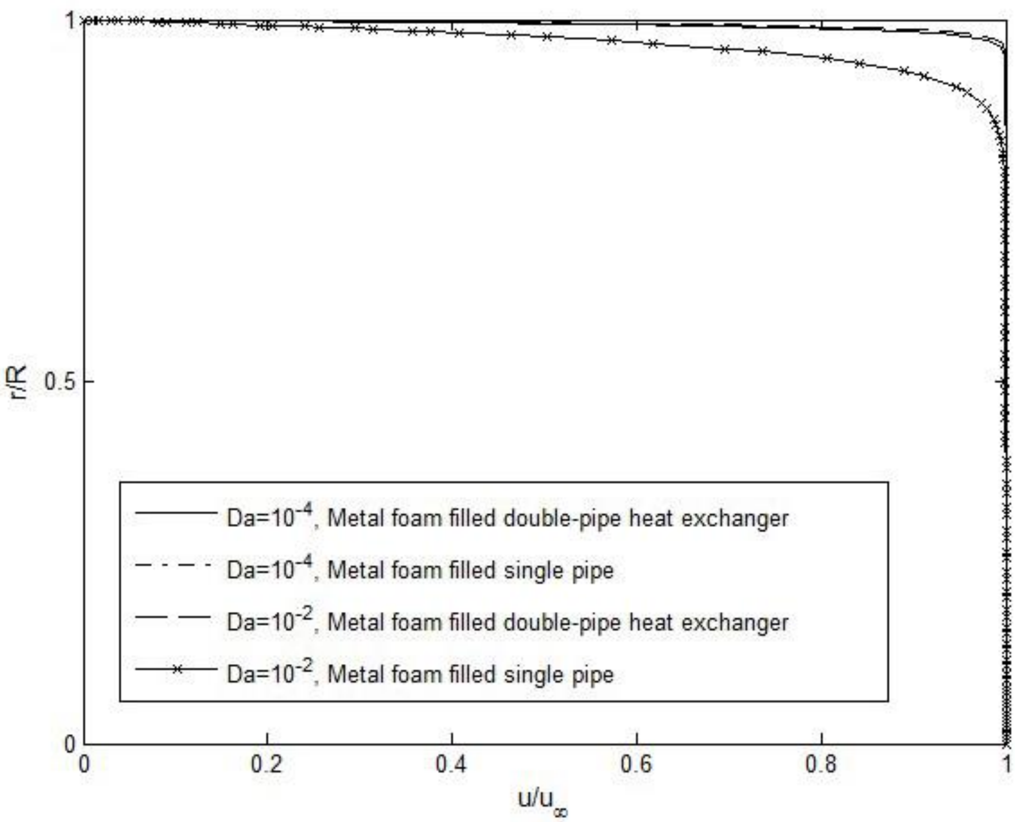
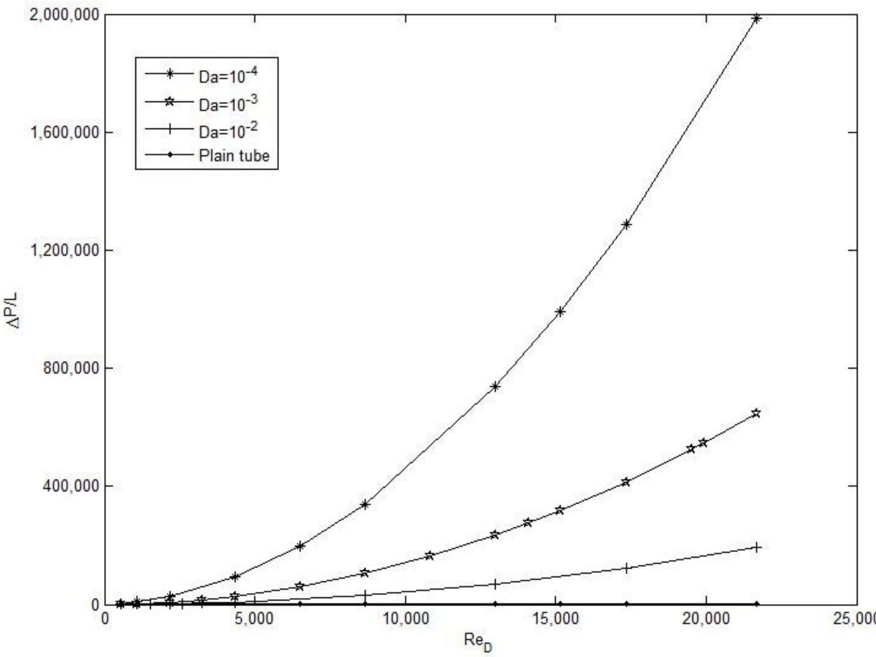
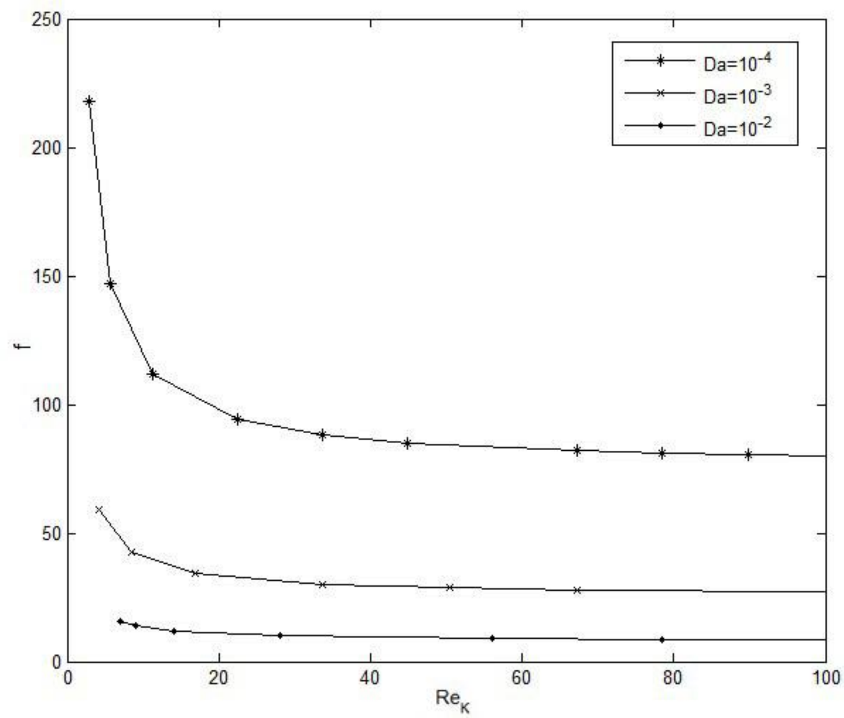


Fig. 4 Comparison of the velocity fields in a metal foam filled double-pipe heat exchanger.

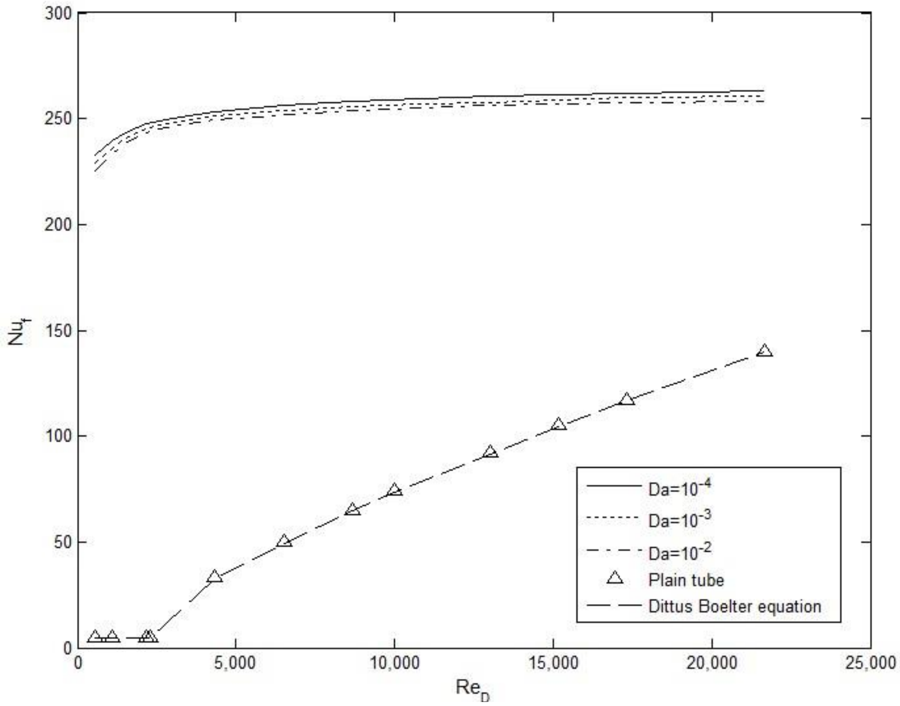


(a)

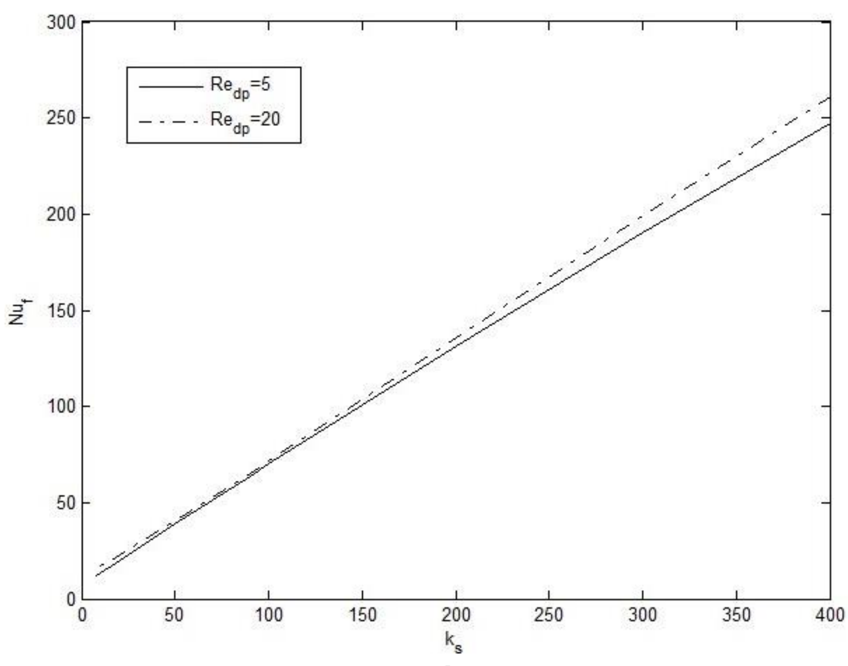


(b)

Fig. 5 Variation of the pressure drop with Darcy number in a metal foam filled pipe: a) Pressure drop, b) Friction factor.

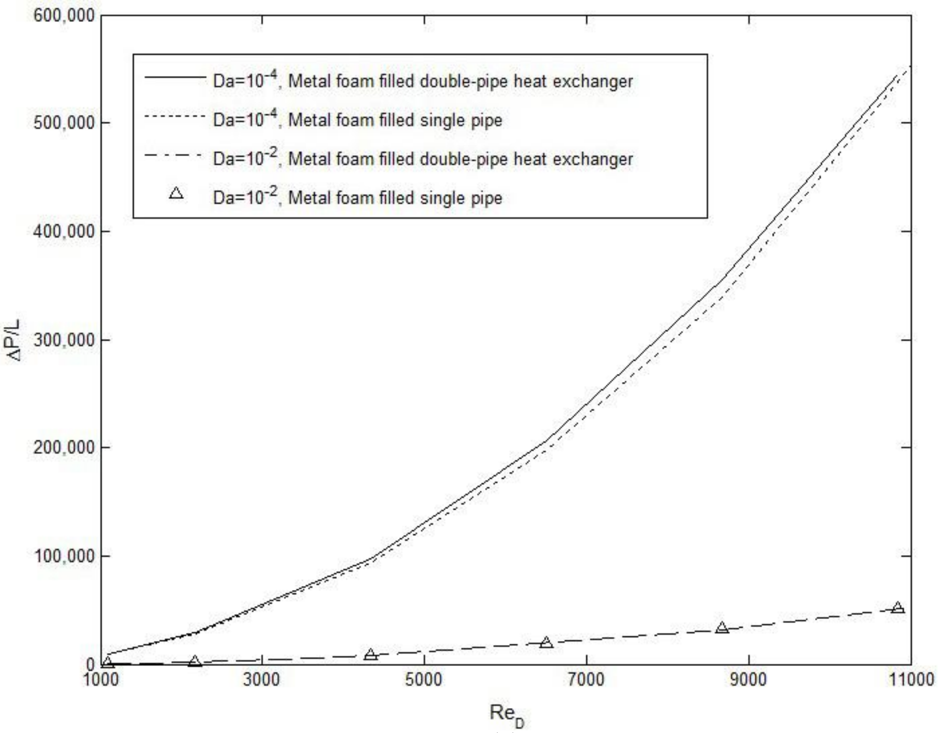


(a)

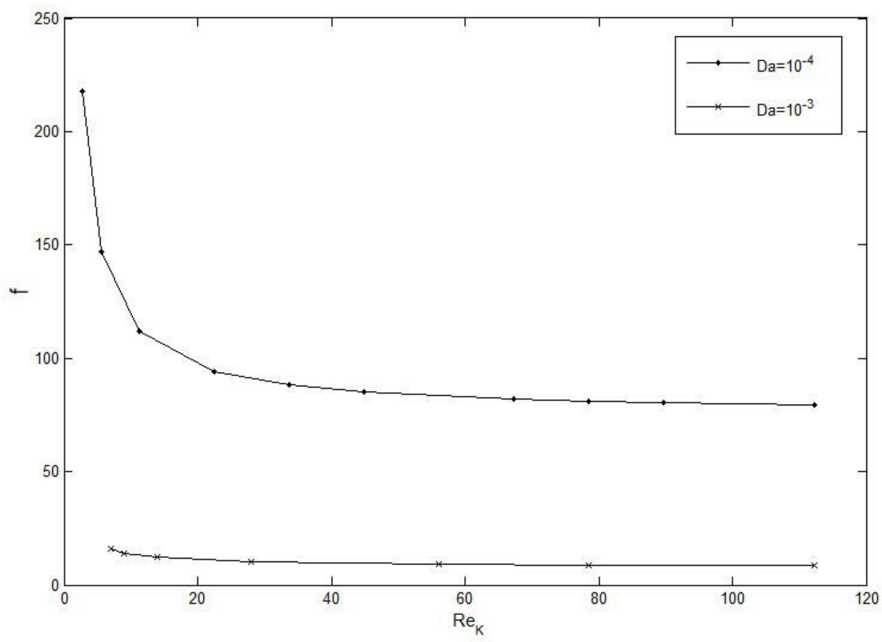


(b)

Fig. 6 Effect of Reynolds number, Darcy number and thermal conductivity variations on the Nusselt number for a metal foam filled pipe: a) Effect of Re_D and Da variations, b) Effect of k_s variations.

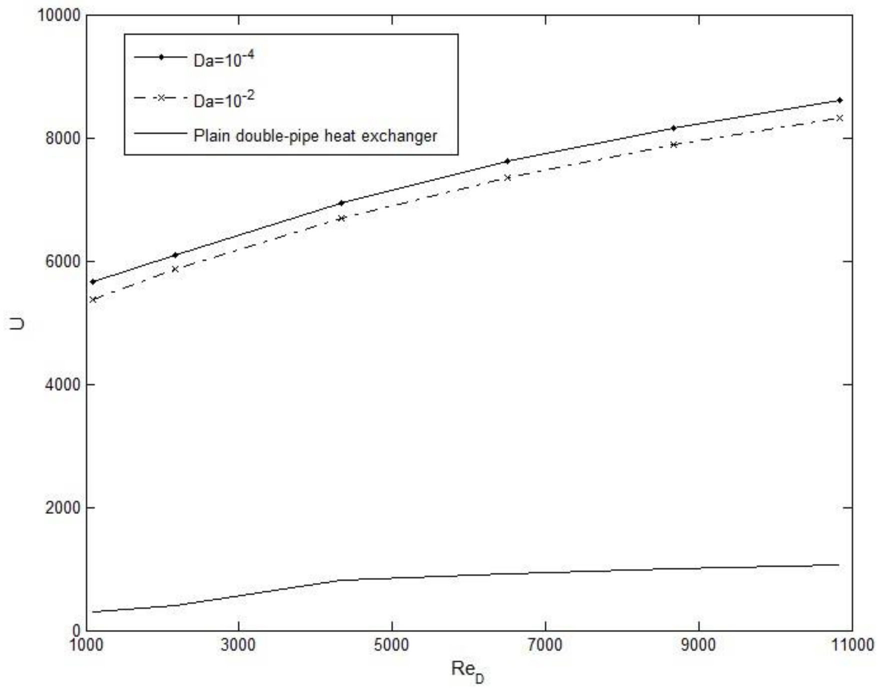


(a)

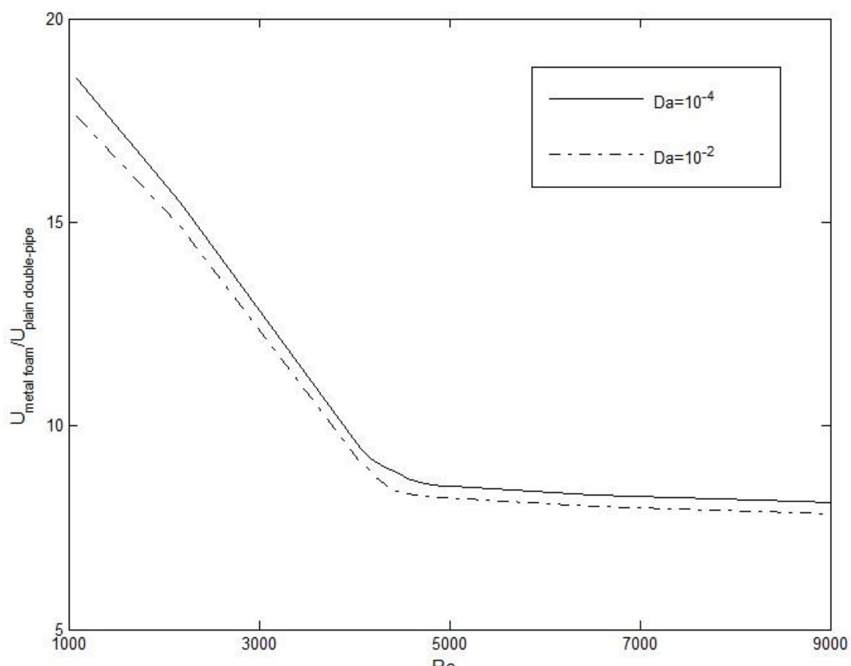


(b)

Fig. 7 Variation of pressure drop and friction factor variations with Darcy number in a metal foam filled double-pipe heat exchanger: a) Pressure drop, b) Friction factor.



(a)



(b)

Fig. 8 Comparison of the heat transfer performance in a metal foam filled double-pipe heat exchanger with a plain double-pipe heat exchanger: a) Total heat transfer coefficient, b) Ratio of total heat transfer coefficient.

References:

1. B. Alazmi and K. Vafai, Analysis of fluid flow and heat transfer interfacial conditions between a porous medium and a fluid layer, International Journal of Heat and Mass Transfer, vol. 44, pp. 1735-1749, 2001.

2. E. C. Ruiz, Modelling of heat transfer in open cell metal foams, University of Puerto Rico, Mayagüez Campus, 2004.

3. W. Lin, J. Yuan, and B. Sundén, Review on graphite foam as thermal material for heat exchangers, in World Renewable Energy Congress 2011, WREC 2011, 2011.

4. K. Boomsma, D. Poulikakos, and F. Zwick, Metal foams as compact high performance heat exchangers, Mechanics of materials, vol. 35, pp. 1161-1176, 2003.

5. S. Mahjoob and K. Vafai, A synthesis of fluid and thermal transport models for metal foam heat exchangers, International Journal of Heat and Mass Transfer, vol. 51, pp. 3701-3711, 2008.

6. V. Calmidi and R. Mahajan, The effective thermal conductivity of high porosity fibrous metal foams, Journal of Heat Transfer, vol. 121, pp. 466-471, 1999.

7. K. Boomsma and D. Poulikakos, On the effective thermal conductivity of a three-dimensionally structured fluid-saturated metal foam, International Journal of Heat and Mass Transfer, vol. 44, pp. 827-836, 2001.

8. A. Bhattacharya, V. Calmidi, and R. Mahajan, Thermophysical properties of high porosity metal foams, International Journal of Heat and Mass Transfer, vol. 45, pp. 1017-1031, 2002.

9. V. V. Calmidi, Transport phenomena in high porosity fibrous metal foams, Ph.D. Thesis, University of Colorado, 1998.

10. C. Zhao, T. Kim, T. Lu, and H. Hodson, Thermal transport in high porosity cellular metal foams, Journal of Thermophysics and Heat Transfer, vol. 18, pp. 309-317, 2004.

11. M. Kaviany, Principles of heat transfer in porous media: Springer-Verlag, 1991.

12. K. Vafai and C. Tien, Boundary and inertia effects on flow and heat transfer in porous media, International Journal of Heat and Mass Transfer, vol. 24, pp. 195-203, 1981.

13. R. Nazar, N. Amin, D. Filip, and I. Pop, The Brinkman model for the mixed convection boundary layer flow past a horizontal circular cylinder in a porous medium, International Journal of Heat and Mass Transfer, vol. 46, pp. 3167-3178, 2003.

14. K. Hooman and H. Gurgenci, A theoretical analysis of forced convection in a porous-saturated circular tube: Brinkman–Forchheimer model, Transport in porous media, vol. 69, pp. 289-300, 2007.

15. K. Vafai and S. J. Kim, Forced convection in a channel filled with a porous medium: an exact solution, Journal of heat transfer, vol. 111, pp. 1103-1106, 1989.

16. D. A. Nield, A. V. Kuznetsov, and M. Xiong, Thermally developing forced convection in a porous medium: parallel-plate channel or circular tube with isothermal walls, Journal of Porous Media, vol. 7, 2004.

17. K. Vafai and C. Tien, Boundary and inertia effects on convective mass transfer in porous media, International Journal of Heat and Mass Transfer, vol. 25, pp. 1183-1190, 1982.

18. D. A. Nield and A. Bejan, Convection in porous media: springer, 2006.

19. A. Amiri and K. Vafai, Analysis of dispersion effects and non-thermal equilibrium, non-Darcian, variable porosity incompressible flow through porous media, International Journal of Heat and Mass Transfer, vol. 37, pp. 939-954, 1994.

20. K. Hooman and A. Ranjbar-Kani, Forced convection in a fluid-saturated porous-medium tube with isoflux wall, International communications in heat and mass transfer, vol. 30, pp. 1015-1026, 2003.