Experimental Analysis of Thermal - Hydraulic Enhancement in a Shell-and-U-Tube Heat Exchanger Using Copper Foam Baffles / Original Article

Researcher Rawaa Hussein Al-meammar

Mechanical Engineering Department,
Faculty of Engineering, Qom University, Iran
e-mail: rawaa.almeammar@yahoo.com

التحليل التجريبي لتعزيز الأداء الحراري والهيدروليكي في مبادل حراري من نوع الغلاف وأنابيب على شكل حرف U باستخدام حواجز من رغوة النحاس

الباحث رواء حسين المعمار

قسم الهندسة الميكانيكية، كلية الهندسة، جامعة قم، ايران

Abstract

Porous metal foams have emerged as a promising medium for thermal management due to their high surface area, low density, and excellent thermal conductivity. This study investigates the effect of copper porous foam baffles with different pore densities (5 and 10 PPI) on the thermal and hydraulic performance of a shell-and-U-tube water–water heat exchanger. An experimental setup was developed to evaluate the Nusselt number, heat transfer coefficient, pressure drop, and friction factor over a range of Reynolds numbers across the annular side. The results revealed that incorporating porous metal foam significantly enhanced heat transfer compared with the smooth-tube configuration. At Re ≈ 7600, the 5 PPI and 10 PPI foams increased the Nusselt number by approximately 21% and 50%, respectively. However, these enhancements were accompanied by notable increases in pressure drop and friction factor due to the reduced permeability and increased flow resistance of the porous structure. The 10 PPI foam provided the highest heat transfer improvement but at the expense of greater hydraulic losses. Overall, the 5 PPI configuration achieved a balanced trade-off between thermal efficiency and pumping power, confirming that optimizing pore density is essential for practical heat-exchanger design using porous media.

Keywords: Porous metal foam, Copper baffles, Heat transfer enhancement, Thermal–hydraulic performance, Double-pipe heat exchanger, Pressure drop.

المستخلص

برزت الرغوات المعدنية المسامية كوسط واعد في التطبيقات الحرارية نظرًا لمساحتها السطحية العالية، وكثافتها المنخفضة، وموصليتها الحرارية الممتازة. تبحث هذه الدراسة في تأثير حواجز مصنوعة من رغوة نحاس مسامية ذات كثافات مسام مختلفة (5 و10 مسام / البوصة PPI) على الأداء الحراري والهيدروليكي لمبادل حراري ماء–ماء من نوع الغلاف وأنابيب على شكل حرف U.تم تطوير منظومة تجريبية لقياس عدد Nusselt، ومعامل انتقال الحرارة، وفقدان الضغط، ومعامل الاحتكاك ضمن مدى من أعداد Reynolds على جانب الحيز الحلقي (الفراغ بين الغلاف والأنبوب). أظهرت النتائج أن إدماج الرغوة المعدنية المسامية أدى إلى تعزيز ملحوظ في انتقال الحرارة مقارنةً بحالة الأنبوب الأملس. فعند عدد Reynolds يقارب 7600، زادت قيمة عدد Nusselt بنحو 21% و50% عند استخدام رغوة بكثافة 5 و10 مسام/بوصة على التوالي. إلا أن هذا التحسن في الأداء الحراري ترافق مع زيادة واضحة في فقدان الضغط ومعامل الاحتكاك نتيجة انخفاض النفاذية وارتفاع مقاومة الجريان داخل الحواجزالمسامية. وقد حققت رغوة 10 PPI أعلى تحسن في انتقال الحرارة، ولكن مقابل خسائر هيدروليكية أكبر. بشكل عام، أظهرت رغوة 5 PPI توازناً أفضل بين الكفاءة الحرارية وقدرة الضخ المطلوبة، مما يؤكد أن اختيار الكثافة المثلى للمسام يُعد عاملاً أساسياً في تصميم المبادلات الحرارية المعتمدة على الأوساط المسامية لأغراض تطبيقية عملية.

الكلمات المفتاحية: الرغوة المعدنية المسامية, حواجز نحاسية, تعزيز انتقال الحرارة, الأداء الحراري–الهيدروليكي, مبادل حراري مزدوج الأنبوب, فقدان الضغط.

1. Introduction

Porous metal foam is an innovative material that has gained increasing attention in recent years due to its unique properties, such as low density, high permeability, good thermal conductivity, and a large surface-to-volume ratio. These properties make it an ideal medium for enhancing heat transfer by increasing the heat exchange area and stimulating flow turbulence, resulting in higher Nusselt values and cooling efficiency. Metal foams have been widely used in thermal applications, ranging from electronic refrigeration and solar energy systems to thermal storage and waste heat recovery. These applications respond to the challenges of rising global energy demand and the need for efficient and sustainable solutions[1] [2]. In the field of heat exchangers specifically, studies have shown that incorporating metal foam into tubes or tube bundles can double heat transfer rates compared to conventional exchangers, despite the associated increase in pressure loss. Hybrid designs that combine traditional fins and metal foam have also proven to significantly improve thermal performance while reducing energy consumption, making them a promising alternative to conventional heat exchanger technologies. Therefore, porous metal foam represents a promising direction towards developing more efficient and reliable heat exchangers to meet the requirements of modern industries in reducing energy consumption and carbon emissions[3] .Several studies consistently highlight that porous metal foams enhance heat transfer but at the cost of increased pressure loss, and that careful optimization of foam distribution and pore density is essential to achieve balanced performance. For example, Pavel and Mohamad[4] showed that porosity, pore size, and Reynolds number strongly affect heat transfer rates and pressure drops, concluding that porous inserts significantly improve heat transfer with only a moderate increase in pressure loss if properly designed. Similarly, Hamzah, *et al.*,[5] confirmed experimentally that the best thermal performance was achieved at lower PPI values, where heat transfer was enhanced while pressure losses remained manageable. This trade-off between improved heat transfer and higher pressure drop was also evident in double-tube heat exchangers. Both Marri, *et al.*,[6] and Faal, *et al.*,[7] reported that fully filling the exchanger with foam maximized heat transfer but led to substantial pressure penalties, whereas partial filling using foam baffles provided a more favorable balance between thermal efficiency and hydraulic resistance. These findings align with Alhusseny, *et al.*,[8], who proposed a composite design employing foam coatings and axial rotation to reduce pumping power requirements, further demonstrating that strategic foam distribution can mitigate pressure losses while maintaining strong thermal performance. On the other hand, studies focusing on shell-and-tube and exhaust gas heat exchangers highlighted additional benefits of porous baffles. Abbasi, *et al.*,[9], Chen, *et al.*,[10], and Tian, *et al.*,[11] all showed that incorporating porous baffles significantly improved thermal-hydraulic performance by reducing stagnant flow zones and enhancing mixing. Optimized designs were able to increase heat transfer rates by up to 523.81 kW9]], boost the area goodness factor by more than 500% [10], and simultaneously improve heat transfer by 92% while reducing pressure drop by 65% [11]. However, these works were mostly numerical and did not fully account for long-term operational issues such as fouling.

2. Experimental Setup

A complete water–water heat exchanger rig was designed and constructed, as shown in Figure 1. The system mainly consists of three sections: the test section (Heat exchanger), the hot-water circuit, and the cold-water circuit. In the test section, two copper tubes with U-shaped were fixed inside a transparent Perspex shell and they were concentered together with two thermal Teflon flanges. Hot water flowed through the copper tubes, while cold water was circulated through the annular shell side. Both hot and cold water loops were operated using two independent pumps to maintain the desired flow rates. The hot water was heated in a tank by an electric heater and then circulated through the copper tubes, while cold water was supplied to the shell side. To measure the temperature distribution, six thermocouples were employed: four to monitor the inlet and outlet temperatures of both fluids, and two others placed on the surfaces of the copper tubes. To enhance the thermal performance of the heat exchanger, circular copper porous metal inserts were used as baffles inside the shell side. The porous medium was tested at two pore densities: 5 PPI and 10 PPI. A differential mercury manometer was connected across the shell side to measure the pressure drop. Flow rates of hot and cold water were controlled using two flow meters and valves. Table 1 illustrate the rig specifications.

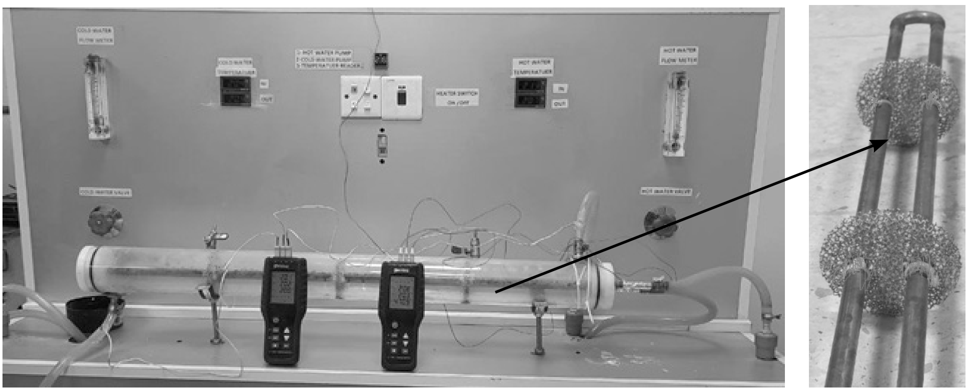

Figure 1: The test rig

Table 1: The specifications of the test rig's parts

Test section (1) Length (2) Inner diameter of copper pipe, di (3) Outer diameter of copper pipe, do (4) Inner diameter of annular pipe, Di	1 m 1.07 cm 1.27 cm 7.4 cm
Porous metal foam (1) Materials (2) Thickness (3) Shape (4) Diameters (5) Number (6) Pore density	Copper 1 mm circular 7.4 cm 3 5 PPI, 10 PPI
Water Pumps (1) Type (2) Max. flow (3) Power	End suction pump 35 L/min 370 W
Temperature recorder -Channel -Sensor type -Operation temperature	4 channels type (K) thermocouple -20 to 150 ºC
Flow meters -water flow meter, Range	0 to 18 L/min

3. Testing Procedure

A shell and U tube heat exchanger was employed where hot water was heated to 60 °C and circulated through the inner copper U tube, while cold water was supplied in counter flow through the annulus. The hot and cold water volumetric flow rates were precisely adjusted using flow meters, and all measurements were taken after achieving steady-state conditions. For each test, the inlet and outlet temperatures of both streams were recorded to evaluate the heat transfer coefficient. To ensure reliability, the experiments were repeated for five different cold-water flow rates (9 to 13 LPM), resulting in multiple runs for comparative analysis but the hot water flowrate was adjusted to 13 LPM. Three configurations of the inner tube were investigated: (i) a smooth pipe without any modification, (ii) the pipe fitted with porous copper foam of 5 PPI, and (iii) the pipe fitted with porous copper foam of 10 PPI. This experimental methodology allowed for a systematic comparison of thermal performance under different operating and structural conditions.

4. Mathematical Formulation

The heat transfer for hot and cold water sides are calculated using equations the following equations [12][13]:

$$Q_h = \dot{m}_h Cp_h \left(T_{hi} - T_{ho}\right) \qquad \textbf{(1)}$$

$$Q_c = \dot{m}_c Cp_h \left(T_{ci} - T_{co}\right) \qquad \textbf{(2)}$$

$$Q_{avg.} = \frac{Q_h + Q_c}{2} \qquad \textbf{(3)}$$

The convective heat transfer coefficient of the inner pipe can be estimated from equation 4:

$$h_c = \frac{Q_c}{A_s\left(T_{s.inner\ pipe} - T_{an}\right)} \qquad (4)$$

Where:

$$A_s = 2\left(\pi\, d_o\, L\right) \qquad \textbf{(5)}$$

And the Nu number of the annular side :

$$Nu_c = \frac{h_c \, D_H}{K} \quad (6)$$

Where D_H is the hydraulic diameter and calculated as follows:

$$D_H = \frac{4 \, A_c}{P_{wet}} \quad (7)$$

$$A_c = A_{c.an} - 2 \, A_{c.inner} = \frac{\pi}{4} D_i^2 - 2 \times \frac{\pi}{4} \, d_o^{\,2} \quad (8)$$

$$P_{wet} = \pi d_{i.an} + 2 \, \pi d_{o.inner} \quad (9)$$

The Re number for both sides :

$$Re_h = \frac{\left(\rho \, v \, D_H\right)_h}{\mu_h} \quad (10)$$

$$Re_c = \frac{\left(\rho \, v \, d_i\right)_c}{\mu_c} \quad (11)$$

Equation (12) is used to calculate the annular friction factor.

$$f_c = \frac{\Delta P \left(\frac{D_H}{L} \right)}{\frac{\rho \, v^2}{2}} \quad (12)$$

And to calculate the theoretical Nu number, Dittus-Boelter correlation is used:

$$Nu_c = 0.023 \left(Re \right)^{0.8} \left(\mathrm{Pr} \right)^{0.33} \quad (13)$$

$$Pr = \frac{Cp. \, \mu}{K} \quad (14)$$

5. Uncertainty Analysis

The accuracy of the experimental results depends on the calibration of the measuring devices and the accuracy of the readings. Calibrated devices such as thermocouples and flow meters were used, and the experiments were repeated twice to ensure reliability. The uncertainty was ±0.5 °C for the thermocouple and ±0.05% for the flow meter. Uncertainty analysis for the calculated quantities, Reynolds and Nusselt numbers, thermal load, and annular friction factor was carried out according to Equations (13) to (17) [14]. The obtained values are presented in Table (2). Since the uncertainties in thermo-physical properties like, specific heat, viscosity, density, and thermal conductivity were insignificant وthey were neglected. The manufacturer indicates that the possible deviation in the dimensions (diameters and length) does not exceed ±0.5 mm.

$$\frac{Un_{Re_{an}}}{Re_{an}} = \pm\sqrt{\left(\frac{Un_{\dot{m}}}{\dot{m}}\right)_{an}^{2} + \left(\frac{Un_{\dot{d}_{an}}}{d_{an}}\right)^{2} + \left(\frac{Un_{\mu_{an}}}{\mu_{an}}\right)^{2}} \qquad \textbf{(15)}$$

$$\frac{Un_{Q_{an}}}{Q_{an}} = \pm\sqrt{\left(\frac{Un_{\dot{m}}}{\dot{m}}\right)_{an}^{2} + \left(\frac{Un_{\dot{T}_i}}{T_i - T_o}\right)^{2} + \left(\frac{Un_{\dot{T}_o}}{T_i - T_o}\right)^{2}} \qquad \textbf{(16)}$$

$$\frac{Un_{h_{an}}}{h_{an}} = \pm\sqrt{\left(\frac{UQ_{\dot{m}}}{Q}\right)_{an}^{2} + \left(\frac{Un_{\dot{d}_{an}}}{d_{an}}\right)^{2} + \left(\frac{Un_{\dot{L}_{an}}}{L_{an}}\right)^{2}\left(\frac{Un_{\dot{T}_i}}{T_i - T_o}\right)^{2} + \left(\frac{Un_{\dot{T}_o}}{T_i - T_o}\right)^{2}} \qquad \textbf{(17)}$$

$$\frac{Un_{Nu_{an}}}{Nu_{an}} = \pm\sqrt{\left(\frac{Un_{h}}{h}\right)_{an}^{2} + \left(\frac{Un_{\dot{d}_{an}}}{d_{an}}\right)^{2} + \left(\frac{Un_{k_{an}}}{k_{an}}\right)^{2}} \qquad \textbf{(18)}$$

$$\frac{Un_{f_{an}}}{f_{an}} = \pm\sqrt{\left(\frac{Un_{\Delta P}}{\Delta P}\right)_{an}^{2} + \left(\frac{Un_{\dot{d}_{an}}}{d_{an}}\right)^{2} + \left(\frac{Un_{L_{an}}}{L_{an}}\right)^{2} + \left(\frac{Un_{\rho_{an}}}{\rho_{an}}\right)^{2} + \left(\frac{Un_{v_{an}}}{v_{an}}\right)^{2}} \qquad \textbf{(19)}$$

Table (2): Uncertainty for experiments parameters

Parameters	Uncertainty %		
	Plain Pipe	Porous metal foam (5 PPI)	Porous metal foam (10 PPI)
Reynold's Number	2.94	2.81	1.005
Heat Transfer	4.46	2.96	1.66
Heat Transfer Coefficient	2.58	3.66	5.27
Nusslte's Number	0.9652	0.9652	0.9666
Friction Factor	3.25	3.08	1.24

6. Results and Discussion

A water-to-water in a shell and U- tube heat exchanger model was employed to investigate the thermal enhancement using copper porous foam as baffles in annular side.15 experiments were done in order to illustrate the porous metal foam and its porosity on the thermal and hydraulic performance of the heat exchanger. The results obtained from these experiments are summarized in the following sections.

6.1 The Model Validation

A comparison was conducted between the Nusselt number obtained for the plain pipe and the value calculated from the established Dittus-Boelter correlation [15]. Figure 2 presents this comparison, contrasting the experimentally determined Nusselt numbers across a range of Reynolds numbers with the corresponding theoretical predictions. The results demonstrating good agreement with an error margin equal 8.32%.

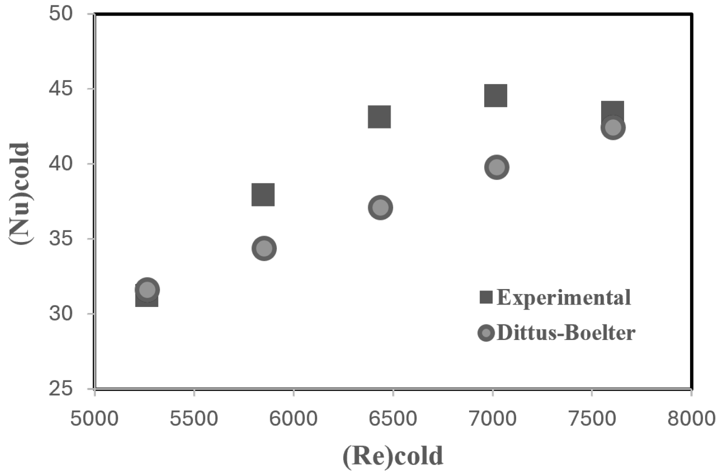

Figure 2: Comparison between experimental and correlated Nusselt number values for the smooth pipe

6.2 Thermal and hydraulic Performance

The thermal and hydraulic performance of the shell and U-tube heat exchanger was evaluated for the three configurations. The results are presented as a function of the Reynolds number (Re) to illustrate the effect of pore density on both heat transfer and flow resistance. The thermal performance was assessed using the Nusselt number (Nu), heat transfer coefficient (h), and heat transfer rate (Q) across the annular side.

As shown in Figure 3, Nu number increases with Reynolds number for all cases, indicating enhanced convective heat transfer at higher flow rates. At Re ≈ 7600, the smooth tube achieved a Nu of about 40, while the 5 PPI and 10 PPI configurations reached approximately 62 and 83, respectively, representing average enhancements of 21% and 50% over the smooth pipe. This improvement is attributed to the increased turbulence intensity and larger specific surface area created by the open-cell copper foam, which promotes stronger fluid mixing and disrupts the thermal boundary layer. Figure 4 shows the average heat transfer coefficient followed a similar trend because the smaller pores in the 10 PPI foam provide greater solid–fluid contact area and enhance conductive heat spreading through the copper matrix, thus improving the overall convective performance.

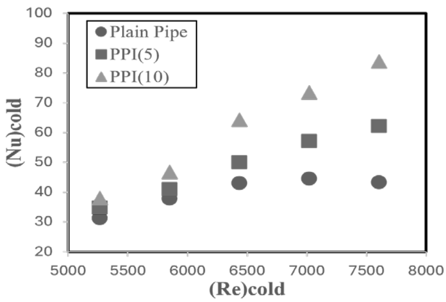

Figure. 3: Nu number vs. Re number of three different configuration

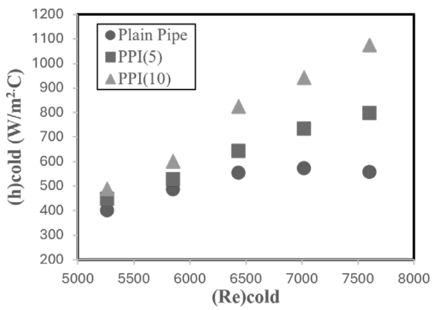

Figure. 4: Heat transfer coefficient vs. Re number of three different configuration

On the other hand, the hydraulic behavior was evaluated through pressure drop (ΔP) and friction factor (f) to quantify the effect of the porous structure on flow resistance.

Based on the results represented in the Figures 5 and 6 for pressure drop and friction factor across varying Reynolds numbers, the results clearly demonstrate the significant hydraulic impact of incorporating porous metal foam baffles in the annular side of the shell and U-tube heat exchanger. Compared to the plain pipe baseline, both the PPI(5) and PPI(10) configurations exhibit a substantial increase in both the friction factor and the measured pressure drop for any given Reynolds number. This is a direct consequence of the complex, tortuous flow path created by the porous matrix, which intensifies flow disruption and mixing. Furthermore, the finer-pored PPI(10) foam consistently generates a higher flow resistance than the PPI(5) foam across the operational range, highlighting the direct relationship between pore density and hydraulic penalty. These figures underscore the classic trade-off in heat exchanger enhancement: the dramatic improvement in heat transfer performance offered by the metal foam is achieved at the cost of a greatly increased pressure drop, which necessitates higher pumping power for the system.

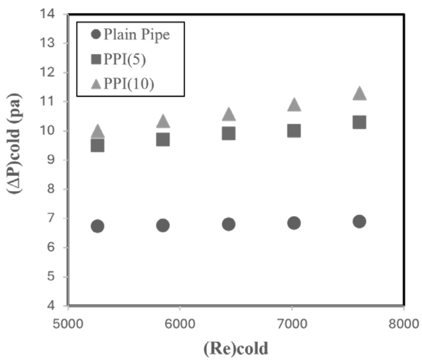

Figure 5: Pressure drop vs. Re number of three different configuration

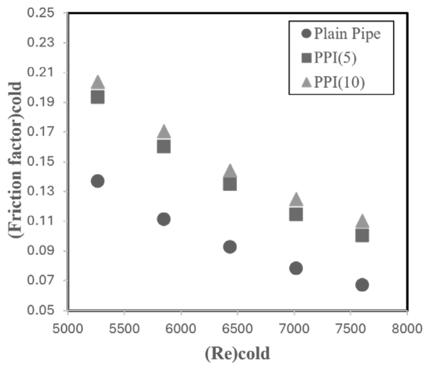

Figure 6: Friction factor vs. Re number of three different configuration

6.3 Thermal Performance Factor

An effective thermal enhancement in a heat exchanger is achieved when the rise in the heat transfer coefficient is greater than the corresponding increase in pressure drop, as represented in Eq. (20) [16]. The relationship between the Thermal Performance Factor (TPF) and Reynolds number is illustrated in Figure 7. The results reveal that TPF exceeds unity for every enhanced configuration, confirming the improvement in thermal performance among the tested designs.

$$TPF = \frac{\dfrac{Nu_{.porous\ metal}}{Nu_{.plain\ pipe}}}{\left(\dfrac{f_{.porous\ metal}}{f_{.plain\ pipe}}\right)^{\frac{1}{3}}} \quad \text{..........(20)}$$

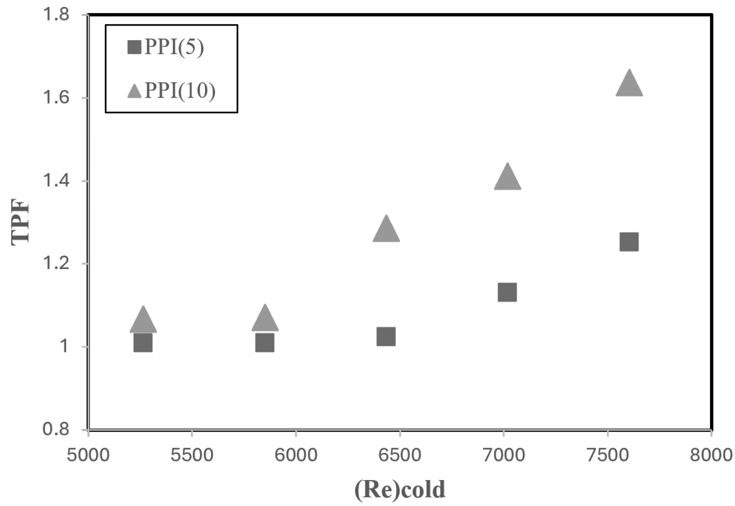

Figure 7: Thermal performance factor vs. Re number of the heat exchanger

7. Conclusion

The experimental investigation demonstrated that the integration of copper porous metal foam baffles markedly enhances the heat-transfer performance of a shell-and-U-tube heat exchanger. The open-cell structure of the foam increases surface area and induces turbulent mixing, leading to higher Nusselt numbers and heat-transfer coefficients compared with a smooth pipe. Specifically, the 5 PPI and 10 PPI foams improved heat transfer by up to 21% and 50%, respectively, confirming the strong influence of pore density on thermal performance. Nevertheless, these enhancements come with increased pressure losses and friction factors, particularly for the 10 PPI configuration, where the smaller pore size intensifies flow obstruction. Hence, while high-density foams offer superior heat-transfer capability, their hydraulic penalty limits overall system efficiency. The results suggest that moderate pore densities (around 5 PPI) provide the most effective balance between improved thermal characteristics and acceptable pressure drops.

NOMENCLATURE

A_s	Surface area, m^2
A_c	Cross sectional area, m^2
Cp	Specific heat, kJ/kg. °C
d_i	Inside diameter of the inner pipe, m
d_o	Outside diameter of the inner pipe, m
D_i	Inside diameter of the annular pipe, m
D_o	Outside diameter of the annular pipe, m
D_H	Hydraulic diameter, m
f	Friction factor
h	heat transfer coefficient, W/m^2. °C
k	Thermal conductivity, W/m. °C
L	Length of the tube, m
m	Mass flow rate, kg/s
Nu	Nusselt number
Pr	Prandtl number
Q	Heat transfer rate, W
Re	Reynolds number
T	Temperature, °C
Un	Uncertainty
ΔP	Pressure drop, pa
v	Velocity, m/s

Greek symbols

μ	Dynamic viscosity, kg/m.s
ρ	Density, kg/m^3

Subscript

h	hot
c	cold
i	Inlet
o	Outlet

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