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Abstract: *The present work predicts the thermo-hydraulic performance of baffled metal foams (MF) filled in an asymmetrically heated channel. The province taken up for the investigation consists of a horizontal channel in which an aluminum solid plate is placed on the top of the channel. A constant heat flux is provided on the heater, which is attached to the solid plate. Heat transfer (HT) from the plate in the channel is carried away by the water fluid flowing at different inlet velocities (0.02 to 0.3 m/s). The novelty of the examination is to use baffle MF to increase the HT rate and reduce the pressure loss, which reduces the fluid's pumping power. The flow and HT along the baffle MF are predicted using Darcy Extended Forchheimer (DEF) and Local thermal equilibrium models, respectively. The present adopted methodology in the study is authenticated with the help of comparing pressure loss and heat transfer coefficient results of experiments and present numerical results and found a fairly good agreement between them. The results of both baffle MF and completely filled MF scenarios are presented and discussed. From the results of pressure loss and HT, it is noted that the baffle MF gives a lesser heat transfer rate compared to completely filled MF, but the pressure loss is approximately reduced by 50% for baffle MF compared to completely filled MF.*

Keywords: **Baffle, Metal Foam, Porous Media, Channel, Nusselt number**

1. Introduction

Efficient thermal management continues to provide a significant challenge in modern technical applications, which include electronic cooling, automotive heat exchangers, and energy storage systems. The growing need for compact yet practical thermal solutions has prompted research into efficient heat transfer enhancement methodologies. MF establish themselves as desirable materials owing to their substantial porosity, large surface area, and increased convective properties. Their open-cell structure enhances the fluid contact area, thus enhancing both conduction and convection transfer of heat [1]. However, MF relatively low turbulence restricts their convective heat transfer efficiency. To overcome this limitation, researchers have used flow-disrupting elements, such as baffles, to improve turbulence intensity and to promote secondary flow patterns [2]. Baffles cause localized flow separation and recirculation, enhancing fluid mixing and heat transfer rates. Combining metal foams and baffles provides an advanced yet highly efficient approach to increasing channel heat transfer performance, specifically under asymmetric heating conditions [3]. The foam porosity, baffle positioning, and flow dynamics influence the enhancement of heat transfer and pressure drop. Boomsma [4] noted that aluminum foams significantly enhance heat transfer rates due to increased surface area and better mixing. Calmidi and Mahajan [5] developed a theoretical model for predicting the effective thermal conductivity of metal foams, illustrating the crucial role of porosity and pore dimensions in enhancing heat transmission. Bhattacharya *et al.* [3] experimentally confirmed these findings, showing improved thermal performance with higher porosity and pressure drop. Dukhan *et al.* [6] emphasized the relationship between foam microstructure and thermal performance, demonstrating the relationship between improved heat transfer and higher flow resistance. Arbak and Dukhan [1] experimentally studied the effect of open-celled MF inserted in an asymmetrically heated channel by considering water as the working fluid. In recent years, substantial research has been conducted on metal foams and baffles' thermal and hydraulic performance in heat exchanger applications. Chen *et al.* [17] investigated the

efficiency of a metal-foam baffle exhaust heat exchanger for waste heat recovery. Adding metal foams significantly improves heat transfer due to augmented surface area and turbulence generation. Wu et al. [22] investigated a partially filled porous foam cylindrical receiver. They demonstrated that decreasing the foam material may improve thermal performance while maintaining the pressure drop within acceptable limits. Said et al. [21] investigated using multi-walled carbon nanotubes (MWCNT) and water-based nanofluids in a shell-and-tube heat exchanger, showing significant heat transfer and pressure drop improvements.

Liu et al. [8] conducted a design analysis on double-layer asymmetric flower baffles, emphasizing the critical role of baffle geometry in improving heat transfer and flow resistance. Batule et al. [14] increased heat transfer and pressure drop efficiency by adding curved circular spines inside a circular pipe, indicating that geometric modifications may improve thermal performance. Aldawi [7] investigated the effect of spiral twisted tape inserts in flat coiled tubes, showing that specific designs significantly improve heat transfer rates and reduce pressure loss. Promvonge and Skullong [16] examined louvered winglet tapes inside heat exchanger tubes, showing that the configuration of the tapes significantly influenced entropy generation and heat transfer rates. Ji et al. [9] performed a numerical study on vibration and thermal transfer properties of conical spiral elastic bundle heat exchangers with baffles, suggesting that the baffle design improves heat exchange efficiency. Prajapati et al. [15] evaluated the thermodynamic performance of shell-and-tube heat exchangers through advanced exergy analysis, emphasizing the significance of minimizing exergy destruction while improving system efficiency. Maghsoudali et al. [20] used 3D modeling to assess the effect of finned tubes on shell-and-tube heat exchangers, indicating that fin shape significantly influences thermal and hydraulic performance. Rao and Sai (2023) [2] conducted a numerical analysis of a shell-and-tube heat exchanger (STHX), such as continuous helical baffles, showing higher thermal efficiency due to improved mixing as well as lower pressure drop. Gu et al. [18] investigated the fluid flow dynamics and thermal transfer of shell sides of heat exchangers, showing that optimized flow channels decrease pressure loss while improving heat transfer efficiency. Lei et al. [19] developed and assessed novel shell-and-tube heat exchangers using louvre baffles, indicating that baffle spacing and angle are critical variables for improving performance. Bahiraei et al. [11] investigated the thermal-hydraulic performance of nanofluids in heat exchangers using trapezoidal inclined baffles, concluding that the baffle configuration influences both heat transfer and pressure drop. Li et al. [13] combined experimental and numerical techniques to evaluate the thermal efficiency of shell-and-tube heat exchangers with longitudinal flow, highlighting the vital role of flow direction in heat transfer efficiency.

With respect to the above discussion, the MF is used in many applications to improve heat transfer and found limited studies on MF filled in asymmetrically heated channels. It is also observed from the literature that the baffle used in the channels plays a vital role in the enhancement of heat transfer due to the proper mixing of the fluid. Hence, the present study investigates the benefit of baffle MF on pressure loss and heat transfer improvement. The study's novelty is to investigate numerically the use of MF as a baffle inserted at different locations inside an asymmetrically heated channel. The present province examined resembles the applications in electronic cooling. Essential variables, including Reynolds number, Nusselt number, and pressure drop, are assessed to understand the thermal performance.

2. Methodology

Figure 1 illustrates the physical domain selected for the numerical examination to evaluate the effect of the metal foam baffle in the present study. The domain considered for the investigation consists of a horizontal channel with dimensions of 101.6 × 20.3 × 50.8 mm, and an aluminum solid plate measuring 101.6 × 12.6 × 50.8 mm is positioned at the top of

the channel. A heater is embedded with a solid aluminum plate at the top to supply heat to the plate, which is subjected to a constant heat flux. Water is the working fluid and flows through the channel at various velocities (0.02–0.3 m/s), carrying heat away from the heated plate. An MF baffle with dimensions of $10 \times 20.3 \times 50.8$ mm is placed directly below the aluminum plate at different locations inside the channel to enhance the heat transfer rate. A total of 5 MF baffles are considered to determine the channel's thermal-hydraulic performance. The schematic of the domain selected for the present investigation resembles the experimental study referenced in Ref. [5]. Figures 1 (a) and (b) show the completely filled MF channel and MF filled as a baffle in the channel, respectively.
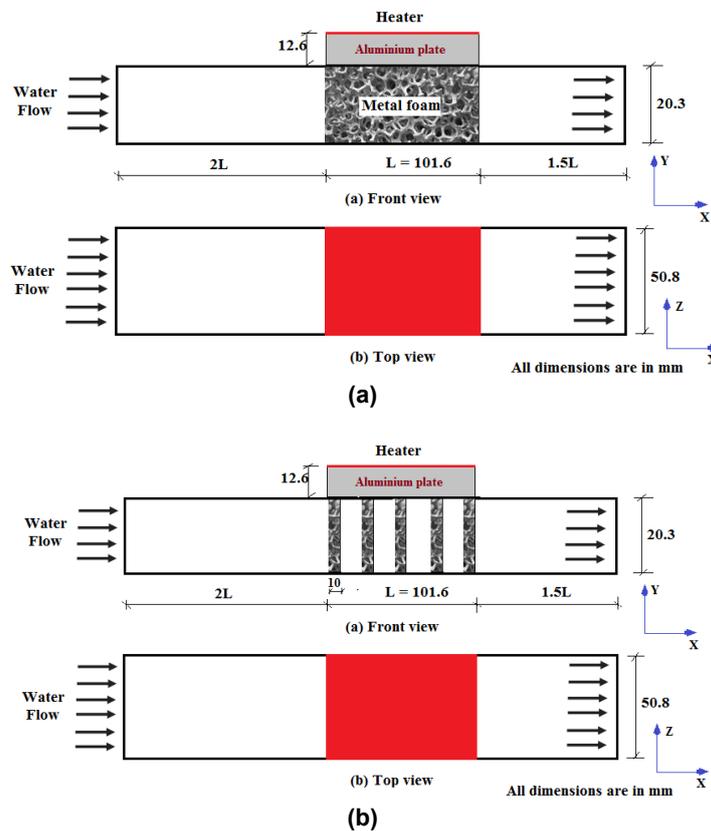

Figure 1. Geometry of the study (a) completely filled MF (b) Baffled MF

Because of the symmetrical configuration of the physical province shown in Figure 1 along Z – the axis, a two-dimensional computational province is chosen, which consists of a test unit (MF filled channel) upstream (2 times the test unit length) and downstream of the channel (1.5 times length of the test unit). The inlet of the channel is specified with constant velocity inflow, while the outlet of the channel is given with zero pressure. The channel's bottom and top side walls are taken as adiabatic walls with no-slip condition. The heater placed at the aluminum solid metal plate is defined with constant heat flux (11625 W/m^2).

The current calculations are obtained using ANSYS Fluent 15.0. The fluid velocity flowing through the horizontal channel varies between 0.02 to 0.3 m/s. The hydraulic diameter-based Reynolds number (Re) calculated varies from Re = 577 to Re = 8660. Contingent on the value of 'Re,' the flow in the channel is presumed as laminar (Re < 2300), transition flow (2300 < Re < 6000), which is predicted with the help of k-kl-ω model and turbulent flow (Re > 6000) predicted using k-ε model. The governing equations for flow through the unfilled portion (open region) of the channel are given in Eq. (1 − 3).
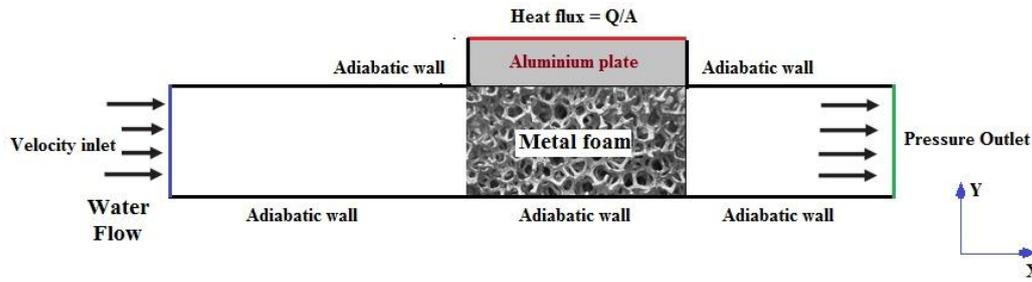
Figure. 2. Computational domain and Boundary conditions

Continuity equation	$\dfrac{\partial(\rho_f u_i)}{\partial x_i} = 0$	(1)
Momentum equation	$\dfrac{\partial(\rho_f u_i u_j)}{\partial x_j} = \dfrac{\partial p}{\partial x_i} + \dfrac{\partial}{\partial x_j}\left((\mu_f + \mu_t)\left(\dfrac{\partial u_i}{\partial x_j} + \dfrac{\partial u_j}{\partial x_i}\right)\right)$	(2)
Energy equation	$\dfrac{\partial(\rho_f u_j T)}{\partial x_j} = \dfrac{\partial}{\partial x_j}\left(\left(\dfrac{\mu_f}{Pr_f} + \dfrac{\mu_t}{Pr_t}\right)\dfrac{\partial T}{\partial x_j}\right)$	(3)

The flow inside the metal foam region is foretold by adding the DEF model as a source to the momentum equation, which takes care of porous medium properties such as permeability, porosity and drag coefficient. The heat transport across the metal foam space is foretold by means of the LTE model, which is similar to the studies of Mohamad [12]. Flow transport inside the porous foam is considered as laminar as permeability-based 'Re' < 150 as per Kotresha B [10]. The governing equations (4) – (8), as per [10], are used in the present investigation with respect to porous metal foam province.

Continuity equation:	$\dfrac{\partial(\rho_{nf}\varepsilon u_i)}{\partial x_i} = 0$	(4)
Momentum equations:	$\dfrac{\partial(\rho_{nf} u_i u_j)}{\partial x_j} = -\varepsilon\dfrac{\partial p}{\partial x_i} + \dfrac{\partial}{\partial x_j}\left(\mu_{nf}\left(\dfrac{\partial u_i}{\partial x_j} + \dfrac{\partial u_j}{\partial x_i}\right)\right)$ $- \varepsilon\left(\dfrac{\mu_{eff}}{K}u_i + \rho_{nf}C\,\lvert u\rvert u_i\right)$	(5)
Energy Equation: LTE	$(\rho c)_{nf} u_i \nabla\text{T} = \nabla.\left(k_{eff}\nabla\text{T}\right)$	(6)
where	$(\rho c)_{eff} = (1-\varepsilon)(\rho c)_s + \varepsilon\,(\rho c)_{nf}$	(7)
	$k_{eff} = (1-\varepsilon)\,k_s + \varepsilon\,k_{nf}$	(8)

The suffix *'eff'* in Eq. (4 - 8) denotes the effective parameters determined based on the solid and fluid properties of the metal foam. In contrast, the effective viscosity (μeff) is taken to be the same as that of fluid viscosity in the current study. The pressure and velocity are integrated with the help of the SIMPLE scheme, and the gradient is calculated with the help of least square cell based along with second-order discretization for all the variables. A boundary layer grid generation is adopted at the channel wall such that the Y+ is limited to less than 2.0. The convergence standard set for flow variables is 1e-5; for turbulence factors, it is 1e-3, and 1e-8 is for energy.

3. Results and Discussion
The following variables are evaluated to examine the consequences of fluid flow and HT physiognomies.

3.1 Flow characteristics

Reynolds number (Re)

$$Re = \frac{UD_h}{\nu_{nf}} \tag{9}$$

Friction factor (FF)

$$FF = \frac{\Delta P}{\left(\frac{L}{D_h}\right)\left(\frac{\rho U^2}{2}\right)} \tag{10}$$

where $'\Delta P'$ is the pressure drop arising due to metal foam porous medium

3.2 Thermal characteristics

Temperature difference

$$\Delta T = T_{avg,Plate} - T_{in} \tag{11}$$

$T_{avg,Plate}$ is the average temperature of the heater surface and T_{in} is inlet fluid temperature.

Nusselt number

$$Nu = \frac{QD_h}{Ak_{nf}\Delta T} \tag{12}$$

Performance Evaluation Criteria (PEC)

$$PEC = \frac{Nu_{cc}}{Nu}\left(\frac{f}{f_{cc}}\right)^{\frac{1}{3}} \tag{13}$$

Pumping power (PP)

$$PP = UA_{inlet}\Delta P \tag{14}$$

A_{inlet} is the area at the inlet of the channel.

3.3 Grid Sensitivity Analysis

The grid independence analysis is performed for three grid sizes such as 17100, 65000 and 130400 and with reference to grid size 130400, the deviation in terms of ΔP (pressure loss) and ΔT (temperature difference) is calculated for the other two grid sizes and is shown in Figure 3. It is found from the examination that the deviation between 65000 and 130400 grid sizes is very less; hence 65000 grid size is selected for further numerical investigation in the present work to save computational time and space.
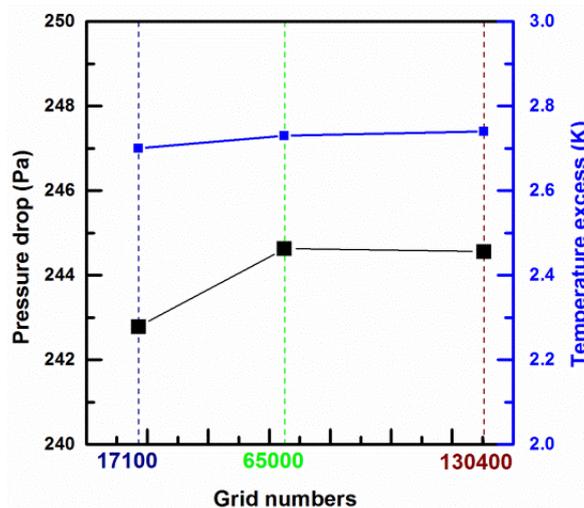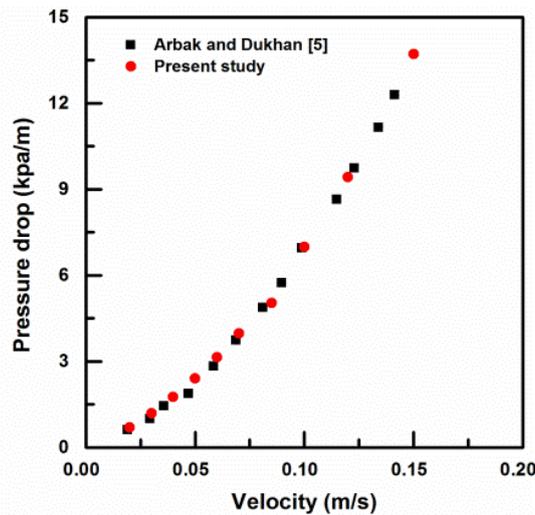

Figure 3. Grid independence study

Figure 4. Validation of flow solution

To authenticate the methodology adopted in the study, metal foam of 20 PPI (K = 3.88X10-8, F =0.085, ε = 0.918), which is considered by Arbak and Dukhan [5] in his experiments, is taken up for the simulations. The comparison of pressure drop obtained in the current investigation and experimental results of Arbak and Dukhan [5] is shown in Figure 4. It is found that both the numerical and experimental result matches very well with a deviation of ±3%. The Nusselt number is calculated along the length of the heater surface and is shown in Figure 5. It is noticed from the comparison that the heat transfer performance decreases along the length of the heater surface for both scenarios. The average deviation between numerical and experiments is ±12%, and it is also important to note that the uncertainty present in the experimental Nusselt number calculation is around ±8%.
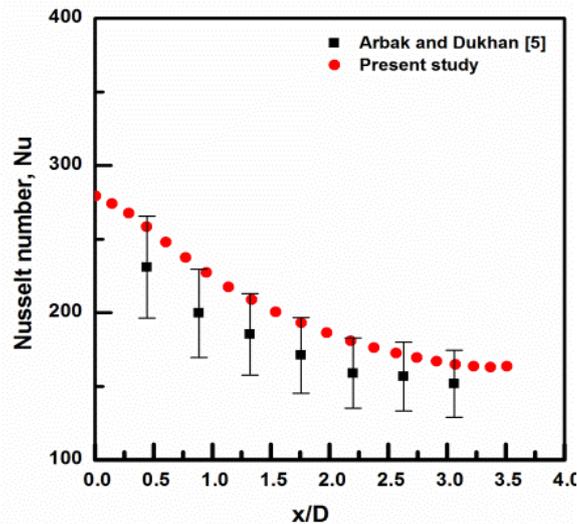

Figure 5. Validation of thermal solution

3.4 Flow results

Figure 6 illustrates the pressure drop variation with inlet velocity for two distinct configurations: completely filled MF and MF as baffles. The pressure drop significantly increases with increasing inlet velocity in both designs, indicating an inherent relationship between flow rate and resistance under porous and obstructed flow conditions. However, the completely filled MF design consistently shows a substantially higher pressure drop across the whole velocity range than the baffled MF. The significant increase in pressure drop results from the higher flow resistance provided by the full MF complex and dense internal structure. However, it also leads to increased frictional losses and higher resistance.

At lower inlet velocities (about 0.05 m/s), the pressure loss for the entire MF is higher than for the MF baffle design. This suggests that even at low flow rates, the higher friction and flow disruption within the full MF result in a significant resistance penalty. As the inlet velocity increases, the pressure drop increases nonlinearly, indicating that flow resistance within the foam structure becomes more critical at higher velocities due to increased turbulence and inertial effects. At a particular velocity of 0.2 m/s, the pressure loss achieved in baffle MF is approximately 55% less than in the completely filled MF case. This will certainly compensate for the pumping power of the fluid and reduce the operating cost by 50%. Even at an inlet velocity of 0.30 m/s, the pressure drop in the full MF design is around 45 kPa/m. In contrast, the MF baffle design shows a pressure drop of about 17 kPa/m, indicating reduced pressure loss using baffle MF.
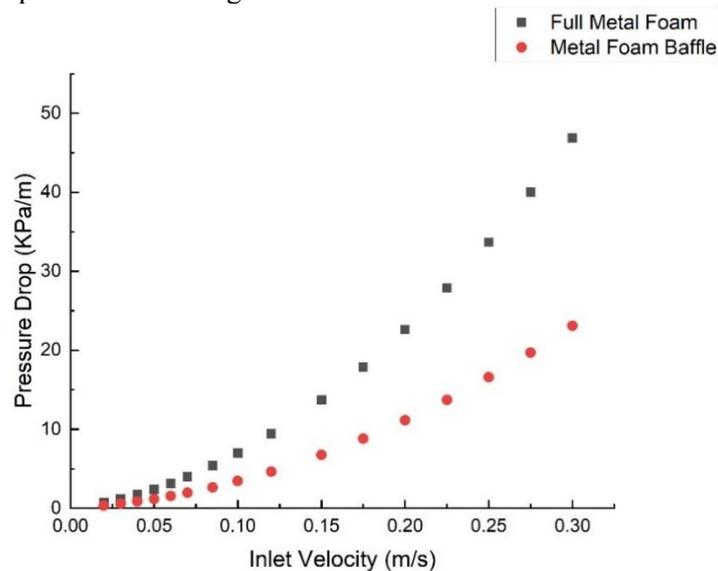

Figure 6. Pressure loss variation with velocity

3.5 Thermal Results

Figure 7 depicts the variations of Nusselt with flow Reynolds number of water for two distinct designs. The Nusselt number is the dimensionless heat transfer coefficient, which increases with an increasing Reynolds number in both designs, showing an improvement in convective heat transfer at higher flow rates. However, the full MF design consistently exhibits higher Nusselt numbers across the entire range of Reynolds numbers than the MF baffle design. This indicates that the full MF offers improved heat transfer efficiency due to its enhanced surface area and increased turbulence within the foam structure. At Re = 1000, the Nusselt number for the complete MF configuration is nearly 150, representing an increase of about 36% compared to approximately 110 obtained for the MF baffle structure. The initial improvement in heat transfer efficiency is due to the large surface contact area of the full MF, which enhances the heat absorption and dissipation rate, even at lower flow velocities. An average of 36% increase in heat transfer rate is observed for completely filled MF compared to baffle MF for the range of Reynolds numbers studied in the investigation. It is important to note the percentage increase in heat transfer and pressure loss enhancement. It is suggested from the analysis that the baffle MF configuration can be considered in applications where pressure loss is an essential key parameter in the design of thermal systems.
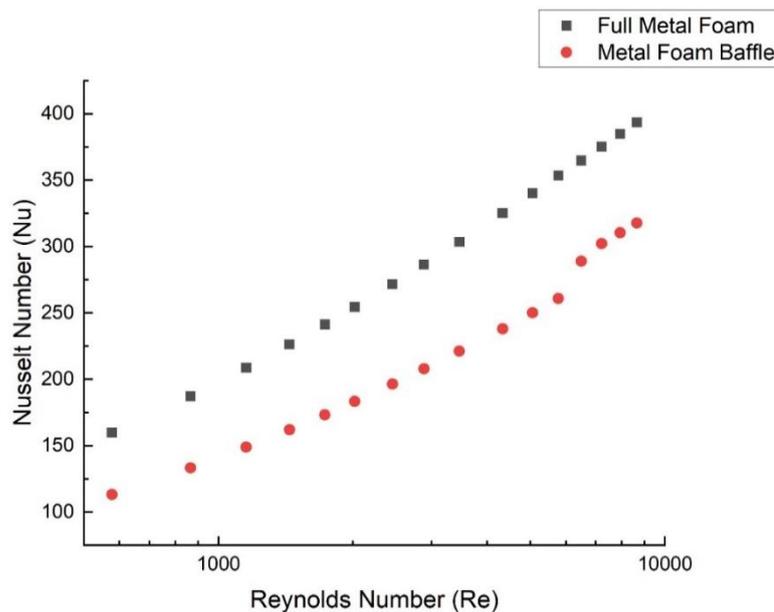
Figure 7. Nusselt number (Nu**) vs the Reynolds number (**Re**)**

4. Conclusions

The present work uses numerical simulation to perform the thermal and flow analysis of two designs: completely filled MF and baffle MF configurations in an asymmetrically heated channel. The province taken up for the investigation consists of a horizontal channel in which an aluminum solid plate is placed on the top of the channel. The heater attached to the solid plate provides a constant heat flux. The water fluid flowing at different inlet velocities (0.02 to 0.3 m/s) carries HT from the plate in the channel. The DEF flow model and LTE thermal model are used to forecast the flow physics and thermal properties through the MF in the channel. The present numerical results are validated with experimental results available in the literature. The following points are drawn based on the results of pressure loss and Nusselt number.

- The pressure loss increases with increased water flow velocity for both configurations. An average of 50% reduction in pressure loss is observed for baffle MF compared to completely filled MF in the range of the Reynolds number studied. With respect to the pressure loss baffle, the MF configuration is suitable as it reduces operating costs by 50%.

- The heat transfer rate increases with increased flow Reynolds number for both designs analyzed. The completely filled MF case produces an average of 36% higher heat transfer than the baffle MF design.

- Based on the outcome of the investigation, it is suggested that the baffle MF configuration can be considered in applications where pressure loss is an essential key parameter in the design of thermal systems.

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