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Numerical investigation of heat transfer in a partially porous-filled channel with core and wall configurations

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Abstract

The numerical simulations are carried out for analysis of heat transfer enhancement in the partially filled channel with the horizontal porous layer, considering two different configurations: at the core and at the walls of the channel. The Darcy-Brinkman-Forchheimer equation is used to model flow through the porous medium, and the governing equations are solved using a finite volume method. The study investigates the effect of the porous layer thickness, thermal conductivity and Darcy number on heat transfer performance and pressure drop. The detailed parametric analysis quantifies variations in the Nusselt number and pressure drop for both configurations. The study reveals that the pressure drop is consistently lower for porous layers located at the channel walls compared to the core for all Darcy numbers. The Nusselt number increases by up to 88% as the porous layer thickness ratio rises to 0.8, then decreases by approximately 19.5% at a lower thermal conductivity ratio of 2.5, but continues to rise at a higher ratio of 25. Increasing the thermal conductivity ratio to 25 improves the Nusselt number by up to 80.5% compared to a ratio of 2.5. Additionally, for lower Darcy numbers (up to 10^{-4}), the higher Nusselt number is achieved when the porous layer is located at the core. However, for higher Darcy numbers (above 10^{-3}), the Nusselt number is higher when the porous layer is at the walls. The results reveal that the position of the porous layer within the channel significantly affects its thermal and hydrodynamic performance.

Keywords: Channel flow; Darcy number; Heat transfer; Porous media; Pressure drop

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1. Introduction

Heat management in electronic components has become challenging due to technological advancement and miniaturisation. As conventional cooling methods are becoming obsolete, researchers are exploring alternative cooling technologies. The integration of porous materials into channels has shown great potential for enhancing heat transfer rates, with significant applications in electronic thermal management. The insertion of porous media in fluid channels modify the flow dynamics, resulting in reduced boundary layer thickness and enhanced heat transfer. Therefore, a detailed investigation of the heat transfer analysis is required for the optimal design of a channel containing porous inserts.

Heat transfer enhancement is crucial for improving the thermal performance of various industrial systems, including heat exchangers, cooling systems and electronic devices. The heat enhancement techniques are vital for sustainability and are categorised as either passive technique (e.g. porous inserts, nanofluids, turbulators), which require no extra energy, or active technique (e.g. magnetic fields, external pumps). Passive techniques are typically preferred for their low energy demands, offering a more sustainable alternative for improving heat transfer and flow without consuming large amounts of energy, which is increasingly important for energy-conscious industries.

Nomenclature

C – inertial coefficient
C_p – specific heat, J/(kg K)
Da – Darcy number
F – Forchheimer coefficient
g – gravitational acceleration, m/s^2
H – channel height, m
k – thermal conductivity, W/(m K)
k_r – thermal conductivity ratio
K – permeability, m^2
Nu – Nusselt number
p – pressure, Pa
Pr – Prandtl number
q'' – heat flux, W/m^2
r – porous layer thickness, m
r_p – porous layer thickness ratio
Re – Reynolds number
T – temperature, K
u – velocity in x-direction, m/s
$\vec{v}$ – velocity vector, m/s
x, y – Cartesian coordinates, m

Greek symbols

ε – porosity
θ – non-dimensional temperature
μ – dynamic viscosity, Pa·s
ρ – density, kg/m^3

Subscripts and Superscripts

c – core
e – effective
in – inlet
m – bulk mean
w – wall

The use of nanofluids, turbulators and porous inserts has been extensively studied in the literature as effective methods for enhancing heat transfer in various cooling systems. The details of these techniques, along with their performance metrics, working conditions and applications, are summarised in Table 1. Said et al. [1] presented a comprehensive review of passive heat transfer enhancement techniques employing nanofluids and turbulators. Alenezi et al. [2] experimentally investigated the effect of ultrasonic waves on heat transfer enhancement using the propylene glycol/water nanofluid in the minichannel. Their results showed that ultrasound at 27.8 kHz improved heat transfer, achieving a maximum Nusselt number increase of 11.3% for deionised water at the lowest flow rate (0.2 l/min). Although the study demonstrates the potential of active methods, the energy demand limits their applicability in low-power systems. The use of porous media is a well-established method for heat transfer enhancement, primarily due to its high surface area-to-volume ratio and ability to promote flow mixing. Hadim and Bethancourt [3] identified that full porous fills lead to prohibitive pressure drops, suggesting partial filling as a solution. Subsequent studies, such as Baig et al. [4], have continued this approach, often combining porous inserts with other methods like nanofluids. While the nanofluids and turbulators have been widely investigated, porous inserts have also shown more effective heat transfer enhancement while maintaining relatively low pressure drops and requiring no additional energy input. Owing to these advantages, porous materials represent an efficient and practical passive technique, motivating the present study to focus specifically on porous inserts for further investigation.

The study of heat transfer in porous channels or pipes, particularly under conditions such as constant wall temperature, constant heat flux and discrete heat sources, has received significant attention in recent research in the literature [3–9]. Several investigations have focused on understanding how these thermal boundary conditions influence flow behaviour and heat transfer in porous media. Hadim and Bethancourt [3] also investigated the parametric variation of heat source width in a channel and revealed that when the width of the heat source decreases, a significant increase in the heat transfer occurs and reduces the pressure drop. Hadim et al. [9] investigated the effect of discrete heat sources on the bottom wall of the porous channel using two different cases: full and partial porous channels. The results show the same heat transfer in both cases at low Darcy number and higher pressure drop in fully porous channels. Jen and Yan [10] proposed a 3D model to investigate the flow patterns in partially filled porous media. They revealed that strong counter rotating vortices are present in the channel because of the blowing effect of the porous layer. Hadim and North [11] investigated the effect of multiple inlet and outlet in a porous channel with a constant heat flux condition at the bottom wall. They observed that the pressure drop is minimised when the channel height equals the inlet width, while maintaining the same level of heat transfer enhancement. They also proposed the length-averaged correlation for the Nusselt number.

The use of porous materials in channels has been widely analysed with respect to its effects on flow behaviour, temperature change and pressure drop. Rachedi and Chikh [12] examined mixed convection heat transfer between a channel and an electronic component, inserting a porous material between the two. They demonstrated that at lower thermal conductivity ratios (up to 2), the effect on temperature drop is minimal. However, at higher thermal conductivity ratios (from 2 to 10), a temperature drop of 15% can be achieved. Yucel and Guven [13] studied forced convection heat transfer in parallel plates with multiple heat sources. They observed a significant rise in pressure drop with increasing Reynolds numbers in the presence of a porous layer. Huang et al. [14] investigated the effect of a porous cover on heat transfer enhancement in a channel using multiple heat sources. They revealed that the presence of a porous cover significantly enhances heat transfer compared to a setup without it. Da Silva Miranda and Anand [15] explored the role of porous baffles in enhancing convective heat transfer within channels. They found that a smaller spacing between the baffles results in better heat transfer performance, and that an increase in the

Table 1: Summary of important studies in heat transfer enhancement for channel/internal flows.

Authors	Geometry	Working fluid	Applications	Key findings
Alenezi et al. [2]	Minichannel	Propylene glycol/ water mixtures	Minichannel heat sinks, electronics cooling	1. Ultrasound (27.8 kHz) enhanced heat transfer, with a maximum Nusselt number increase of 11.3% for deionised water at the lowest flow rate (0.2 l/min).
Hadim and Bethancourt [3]	Porous channel	Dielectric fluid	Electronic cooling	1. A partially porous channel (porous layers only above heaters) achieves comparable heat transfer to a fully porous channel but with a much lower pressure drop. 2. Decreasing heat source width moderately improves heat transfer while significantly reducing pressure drop.
Baig et al. [4]	Microchannel	Al_2O_3-water nanofluid	Microchannel cooling, electronics thermal management	1. Finned cooking vessels significantly improve thermal performance and cooking rate compared to an unfinned vessel. 2. Hexagonal fins performed best, reducing boiling time by 13.63% and increasing maximum energy efficiency to 64.46%. 3. An economic analysis showed the hexagonal fin-based cooker could save up to 1495 MJ of fuel equivalent per year.
Promvonge [5]	Square duct with combined 30° V-fins and quadruple counter-twisted tapes	Air	Heat exchangers	1. The combined V-finned twisted tapes provide higher heat transfer and thermal performance than twisted tapes alone. 2. Heat transfer and friction factor increase with relative fin height but decrease with relative fin pitch.
Kalidasan et al. [6]	Solar box cooker with different fin configurations (pin, cross, hexagonal) on the cooking vessel	Water	Solar cooking	1. The fully wavy wall microchannel significantly enhances heat transfer, with a Nusselt number up to 3.87 times higher and thermal resistance only 0.41 times that of a smooth microchannel. 2. The flow becomes chaotic, intensifying fluid mixing and heat transfer with a relatively small pressure drop penalty.
Zhong et al. [7]	Square cross-section microchannel with fully wavy walls (all four sides)	Deionised water	Microchannel heat sinks for electronic cooling	1. The fully wavy wall microchannel significantly enhances heat trans-fer, with a Nusselt number up to 3.87 times higher and thermal resistance only 0.41 times that of a smooth microchannel. 2. The flow becomes chaotic, intensifying fluid mixing and heat transfer with a relatively small pressure drop penalty.
Mohammed et al. [8]	2D rib-groove channel (9 combinations of rectangular, triangular, trapezoidal ribs and grooves)	Nanofluids (Al_2O_3, CuO,SiO_2, ZnO with water, glycerin, engine oil)	Heat exchangers, turbine cooling, solar heaters	1. Rectangular rib-triangular groove provided the highest Nusselt number and best thermal enhancement factor. 2. SiO_2 nanofluid showed the highest heat transfer performance, despite its moderate conductivity, due to favourable thermophysical properties.

thermal conductivity ratio enhances heat transfer. Ko and Anand [16] also examined the effect of porous baffles on convective heat transfer in a channel, with the baffles installed alternately at the top and bottom walls. They revealed a 300% enhancement in heat transfer with baffles compared to the channel without them. Additionally, they developed a correlation between the ratio of heat transfer enhancement and the Reynolds number. Kim et al. [17] investigated the effect of porous material in a channel. They developed a correlation for the Nusselt number, specifically for Reynolds numbers in the range of 1000 to 3000.

The influence of temperature-dependent thermophysical properties on heat transfer enhancement in fully or partially porous channels has been examined in several studies. Dehghan et al. [18,19] examined the effect of thermal conductivity in porous channels. They proposed a correlation for the Nusselt number based on thermal conductivity and the shape of the channel. Mehata et al. [20] examined forced convection heat transfer in a wavy solar plant using porous blocks. They found that the average Nusselt number increased by 83.95% with the use of porous blocks compared to without them. Kaur et al. [21] investigated the effect of circular and semicircular heat sources in porous media and found that the lowest heat transfer occurred in the circular channel. Li et al. [22] examined the effect of multilayer porous inserts in a channel under non-equilibrium thermal conditions. The results showed that the pressure drop is directly proportional to the number of porous layers.

Due to their ability to enhance heat dissipation while maintaining a compact system design, channels partially filled with porous media have attracted considerable research attention. The temperature profile and Nusselt number used to quantify the heat transfer characteristics in such microchannels have been presented in previous studies [19,23]. Mahmoudi [24] developed analytical expressions for both the temperature field and Nusselt number in a microchannel. Dehghan [25] performed an

analytical study on the thermal behaviour of a channel partially filled with porous media. The addition of a porous material to a fluid undergoing convection results in a more uniform temperature distribution, which reduces thermal resistance and enhances heat transfer. The study also found that as the thickness of the porous material increases, the proportion of heat transferred by the porous medium relative to that transferred by the clear fluid flow increases. Dehghan et al. [26] developed analytical expressions for the local Nusselt number and the dimensionless temperature difference between the solid and fluid phases in a porous material placed inside a channel within the thermally developing region. Bibin et al. [27] numerically examined heat transfer in a partially filled porous channel using OpenFOAM. Their study reported that introducing a porous medium can enhance the heat transfer rate by 10.4 times compared to a channel without porous filling. They also found that this enhancement is most pronounced when the porous layer thickness ratio lies between 0.7 and 0.9. Kumar et al. [28] analysed heat transfer in sinusoidal, triangular, and trapezoidal channels containing porous media at their core. Their findings indicate that the trapezoidal channel achieves the highest thermal performance, with a maximum enhancement of 90%. Alkhamis [29] investigated heat transfer in a channel partially filled with porous material and equipped with solid turbulators. Their results indicate that the use of metal foam along with turbulator enhances both heat transfer and the friction factor. The combined effect of the metal foam and turbulators led to a fivefold improvement in thermal performance. Duan et al. [30] performed a numerical investigation of heat transfer in a wavy metal-foam sandwich configuration. Their analysis demonstrated that, under laminar flow conditions, the Nusselt number can increase by up to 118%.

Although many investigations have been conducted on heat transfer enhancement using porous layers in channels, a critical research gap remains. While the beneficial effects of partial porous layers are well recognised, a systematic numerical investigation into the influence of porous layer location and thickness ratio within a channel remains largely unexplored, particularly under constant heat flux boundary conditions. Addressing this gap, the present study focuses on analysing the effects of location of porous layer and thickness ratio on thermal and hydrodynamic performance to identify configurations that maximise heat transfer while minimising pressure losses. This study investigates convection heat transfer in a channel containing the porous layer with two different configurations: one near the walls and the other at the core of the channel. Partially filling the channel with the porous layer offers two primary benefits: it improves heat transfer performance and results in less pressure drop compared to a fully porous-filled channel. This work can be beneficial in industrial applications, such as heat exchangers, solar energy storage and electronic cooling. A uniform heat flux condition is applied at the channel wall. The effects of the thickness of the porous insert, thermal conductivity ratio and Darcy number on heat transfer enhancement are examined. Additionally, the velocity profile and pressure drop at different Darcy numbers are analysed to determine the optimal design of the channel.

2. Materials and methods

2.1 Governing equations

The governing equation for mass and momentum conservation for the fluid and porous region is explained in this section. The problem is simplified by assuming that the flow is 2D, steady and laminar. The Darcy-Brinkman-Forchheimer model [31] is employed to address the inertial effects in porous material.

The governing equations for mass and momentum conservation in the fluid region are as follows:

$$\nabla \cdot \vec{v} = 0, \tag{1}$$

$$\rho(\vec{v} \cdot \nabla)\vec{v} = -\nabla p + \mu \nabla^2 \vec{v}, \tag{2}$$

where $\vec{v}$ represents the velocity vector, μ denotes the dynamic viscosity, ρ is the density and p is the pressure.

The momentum equation, which incorporates the resistance of the porous medium using the Darcy-Forchheimer model [33], is given by [32]:

$$\frac{\rho}{\varepsilon}(\vec{v} \cdot \nabla)\vec{v} = -\nabla p + \frac{\mu}{\varepsilon}\nabla^2 \vec{v} - \frac{\mu}{K}\vec{v} - \frac{\rho F \varepsilon}{\sqrt{K}}|\vec{v}|\vec{v}, \tag{3}$$

where K represents the permeability, ε denotes the porosity, and F is the Forchheimer coefficient, which is a function of the porous geometry. Local thermal equilibrium is considered to be valid at the interface of the fluid and solid within the porous region.

The energy equation for fluid and porous region are written as [32]:

fluid region:

$$(\vec{v} \cdot \nabla)T = \frac{k}{\rho C_p}\nabla^2 T, \tag{4}$$

porous region:

$$(\vec{v} \cdot \nabla)T = \frac{k_e}{\rho C_p}\nabla^2 T, \tag{5}$$

where k is the thermal conductivity, C_p is the specific heat and T is temperature of the fluid, and k_e represents the effective thermal conductivity, which is influenced by the thermal conductivity of the solid phase.

To derive the non-dimensional form of the governing equations, scaling is applied to the velocity components, temperature and length, using the characteristic reference quantities: inlet velocity, heat flux and channel height, respectively. The dimensionless parameters are defined as follows:

$$X = \frac{x}{H}, \quad Y = \frac{y}{H}, \quad \vec{V} = \frac{\vec{v}}{u_{in}}, \quad P = \frac{p}{\rho u_{in}^2},$$
$$\mathrm{Re} = \frac{\rho u_0 H}{\mu}, \quad \mathrm{Da} = \frac{K}{H^2}, \quad \mathrm{Pr} = \frac{\mu C_p}{k}, \quad \theta = \frac{T - T_{in}}{q''(H/k)},$$
$$C = F\varepsilon, \quad r_p = \frac{r}{H}, \quad k_r = \frac{k_e}{k},$$

where H is the channel height, u_{in} is the inlet velocity in x-direction, u_0 is the average velocity in x-direction, q'' is the heat flux at the wall, θ is the non-dimensional temperature, r is the porous layer thickness, r_p is the porous layer thickness ratio, k_r is the thermal conductivity ratio, Pr is the Prandtl number and Re is the Reynolds number.

The non-dimensional form of the governing equations for mass and momentum conservation is written as:

$$\nabla \cdot \vec{V} = 0, \tag{6}$$

$$\frac{1}{\varepsilon}(\vec{V} \cdot \nabla)\vec{V} = -\nabla P + \frac{1}{\varepsilon\,\mathrm{Re}}\nabla^2\vec{V} - \frac{1}{\mathrm{Da\,Re}}\vec{V} - \frac{C}{\sqrt{\mathrm{Da}}}|\vec{V}|\vec{V}. \tag{7}$$

The energy equation in non-dimensional form is written as

$$(\vec{V} \cdot \nabla)\theta = \frac{k_r}{\mathrm{Re\,Pr}}\nabla^2\theta. \tag{8}$$

For flow in a channel, the Nusselt number for constant heat flux is calculated as [3]

$$\mathrm{Nu} = \frac{q''H}{k\,(T_w - T_m)} = \frac{1}{\theta_w - \theta_m}. \tag{9}$$

2.2 Problem statement and validation

Figure 1 illustrates the schematic of the computational domain. In one case, the porous layer is placed on the channel walls, and in the other case, the same porous material is located at the core of the channel. Geometric configurations partially or fully filled with a porous material are subjected to a constant and uniform heat flux. The flow enters the domain at a constant velocity and temperature. The thermophysical properties of the air are assumed to remain constant throughout the simulations. The model is based on assumptions that the porous region is uniform, isotropic, and filled with a single-phase fluid. The analysis spans a range of Darcy numbers (Da) from 10^{-6} to 1 and porous layer thickness ratio (r_p) from 0.2 to 1.

To mitigate the influence of grid-related effects on the simulation results, a grid independence study is conducted. The study examines the dimensionless axial velocity (u/u_{in}) profile in a partially filled channel with porous media. Three different mesh resolutions are tested, corresponding to 7500, 30 000 and 46 875 elements. The resulting axial velocity profiles are presented in Fig. 2. The variation in axial velocity for the 30 000 and 46 875 element cases is minimal, indicating grid independence. Based on these findings, a grid resolution of 30 000 elements is selected for further analysis to reduce computational cost.

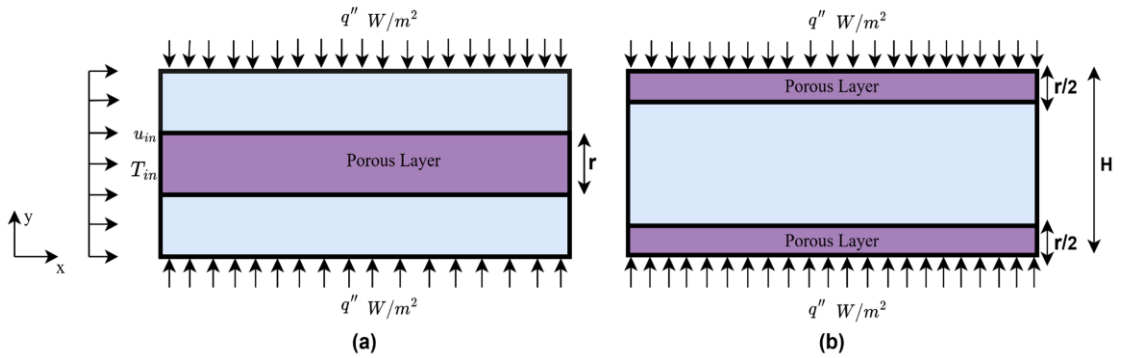


Fig. 1. The schematic representation of a partially filled channel used in the present study with two different configurations: (a) porous layer in the core of the channel, (b) porous layer at the walls of the channel.

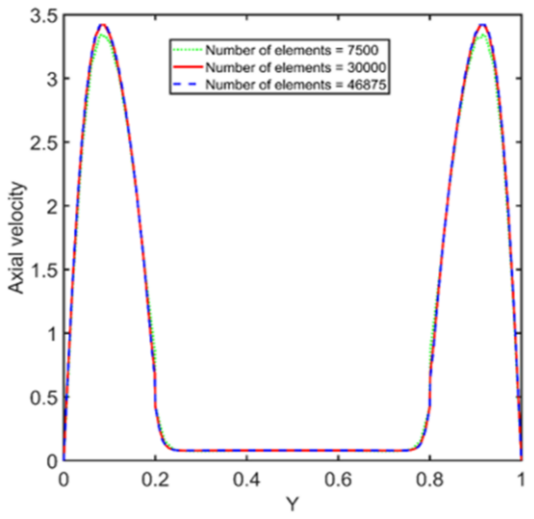


Fig. 2. Non-dimensional axial velocity variation for different grid resolutions at r_p = 0.6 and Da = 10^{-4}.

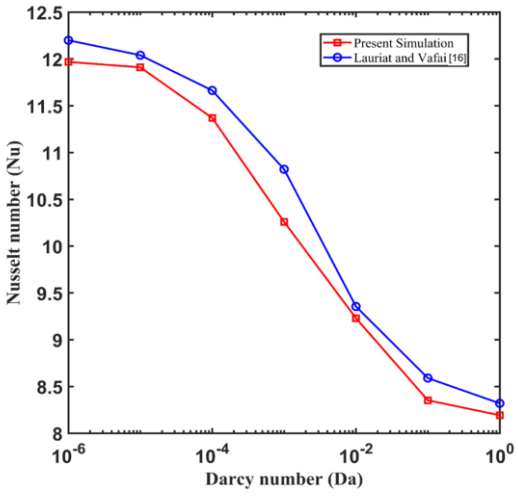


Fig. 3. Validation of the numerical model by comparing the Nusselt number variation with the Darcy number against benchmark results [17] for a fully porous channel.

To validate the numerical model, its results were compared with data reported in the literature. Specifically, the Nusselt number for flow through a fully porous channel with constant heat flux applied to the walls was compared to the findings of Lauriat and Vafai [34]. This comparison is presented in Fig. 3, where the agreement between the two sets of results demonstrates the accuracy of the developed numerical model in capturing the thermal performance of the porous medium.

To further establish the accuracy and reliability of the present numerical model, an additional validation is performed for a channel partially filled with porous media. The channel is partially porous, with porous layers placed directly above the heat sources, while the remaining regions of the channel are nonporous. The schematic representation of the validation geometry is shown in Fig. 4. The porous medium is modelled as homogeneous and isotropic and is saturated with a single-phase fluid that remains in thermal equilibrium with the solid matrix. The porosity and permeability are assumed to remain essentially uniform throughout the entire domain, including regions close to the channel walls.

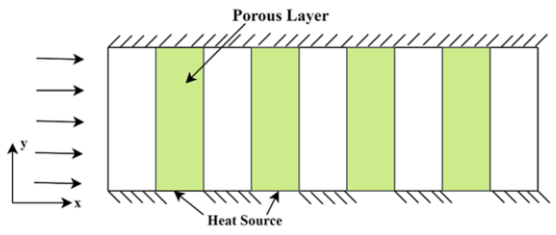


Fig. 4. Schematic diagram of a partially filled porous channel.

Additionally, the length-averaged Nusselt number is calculated at different Reynolds numbers and Darcy numbers for a partially filled porous channel. The obtained results (Fig. 5) exhibit good agreement with those reported in the previous study [9], further validating the numerical model used in the present work.

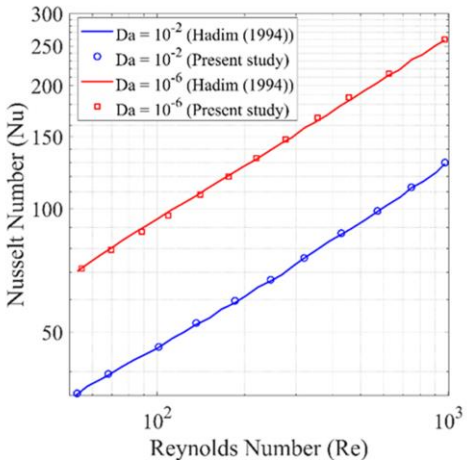


Fig. 5. Variation of the Nusselt number with the Reynolds number in a partially filled porous channel.

3. Results and discussion

The present study investigates the influence of a porous layer on heat transfer in a channel with two different configurations: one near the walls and the other in the core of the channel. The effects of the thickness ratio of the porous layer, the thermal conductivity ratio (k_r) and the Darcy number on heat transfer enhancement are examined. Additionally, the study examines the dimensionless axial velocity and pressure drop to optimise the balance between heat transfer enhancement and pumping power requirements.

3.1. Hydrodynamic characteristics

Figure 6 presents the axial velocity profiles for different values of porous layer thickness ratio (r_p). As the thickness ratio r_p increases, the fluid exhibits a channelling phenomenon, in which the flow is preferentially directed through the clear-fluid region between the porous layer and the channel wall. This results in

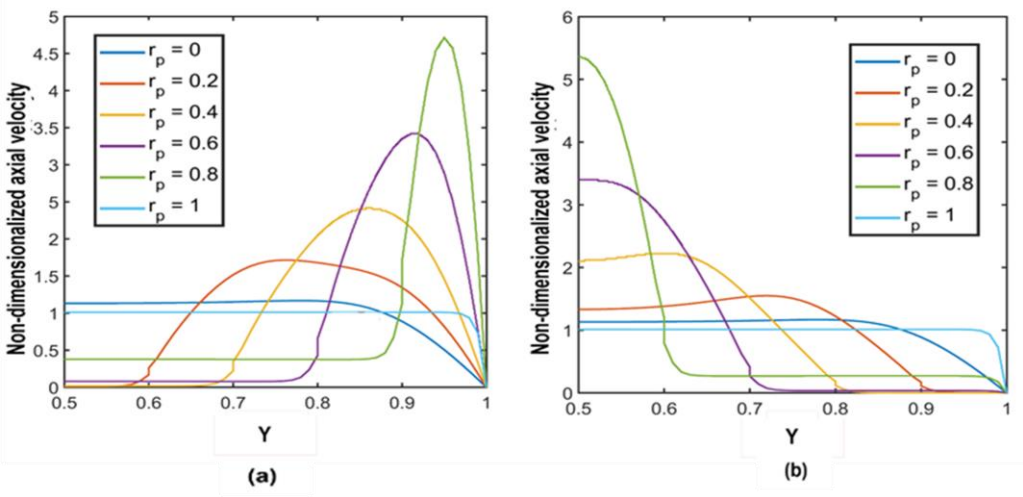


Fig. 6. The non-dimensional axial velocity profile for different porous layer thickness ratio (r_p) for both configurations: (a) porous layer embedded within the core of the channel, (b) porous layer at the walls of the channel.

a non-uniform distribution of flow, with fluid concentration occurring in localised regions around the porous layer, rather than maintaining a uniform flow across the entire cross-sectional area. For the case with a porous layer at the core (Fig. 6a), the maximum velocity of the fluid within the channel tends to increase and shift closer to the pipe surface as the thickness ratio increases, reaching a peak at r_p = 0.8. This increase in velocity is primarily attributed to the decrease in the cross-sectional area available for flow, which forces the fluid to accelerate to maintain the same flow rate. The peak velocity at r_p = 0.8 indicates an optimal configuration where the balance between the porous material's presence and the flow area is most favourable for fluid motion. For the configuration with the porous layer near the wall (Fig. 6b), the maximum fluid velocity increases and shifts toward the centre of the channel as the thickness ratio increases, reaching a peak at r_p = 0.8. This behaviour highlights the significant influence of the porous layer configuration on fluid dynamics, suggesting that careful design of these configurations can enhance thermal performance in applications, such as heat exchangers.

3.2. Pressure drop

The pressure drop in a channel is related to energy consumption. Reduced pressure losses allow for efficient fluid flow and enhance thermal performance. The pressure drop variation with different porous layer thickness ratios for the geometric configuration at different Darcy numbers is shown in Fig. 7. In this figure, subscript *c* represents the configuration when the porous layer is embedded within the core of the channel, and *w* represents the configuration when the porous layer is present at the wall of the channel. As the thickness of the porous layer increases, the resistance to fluid flow also rises, leading to higher pressure losses in both configurations of the channel, as depicted in Fig. 6. Similar pressure drop behaviour has been reported in previous studies [35,36]. This is primarily due to the increased surface area for frictional interactions and the longer path the fluid must traverse, which amplifies the energy required to maintain flow. Conversely, Darcy number is a measure of the porous medium's permeability relative to the fluid's viscosity, plays a crucial role in determining how easily fluid can flow through the material.

A higher Darcy number indicates greater permeability, which typically results in lower pressure losses for a given inlet velocity. It is interesting to note that there is a lower pressure drop when the porous layer is located near the wall than at the core of the channel. This trend becomes more pronounced as the thickness of the porous insert increases. This is because the porous layer on the walls allows for a more streamlined flow path, reducing turbulence and frictional losses. In contrast, a porous insert in the core can disrupt the flow more significantly, leading to increased resistance and higher pressure drop. For a fully porous channel, the pressure loss values are the same for both configurations.

3.3. Heat transfer characteristics

Figure 8 depicts the relationship between the Nusselt number (Nu) and the porous layer thickness ratio for a porous layer present in the channel core, under different Darcy numbers, at a lower thermal conductivity ratio. For a Darcy number of 10^{-2}, the thermal performance improves with increasing porous layer thickness, peaking at a porous layer thickness ratio of 0.6. However, as the porous layer thickness ratio is further increased to 0.8 and then to 1.0, the Nusselt number begins to decrease. In contrast, when the Darcy number is reduced to a value ranging between 10^{-3} and 10^{-6}, the trend changes. In these cases, the heat transfer performance continues to improve with increasing r_p up to 0.8, which results in significant enhancement of thermal performance compared to other cases. Similar to the previous case, a further increase in r_p from 0.8 to 1.0 results in a decrease in thermal performance. This is due to the channelling effect occurring for porous layer thickness ratio, r_p > 0.8. As the porous layer thickness increases beyond this point, the clear fluid gap becomes smaller, resulting in increased flow resistance. Consequently, the flow loses a distinct preferential path, leading to

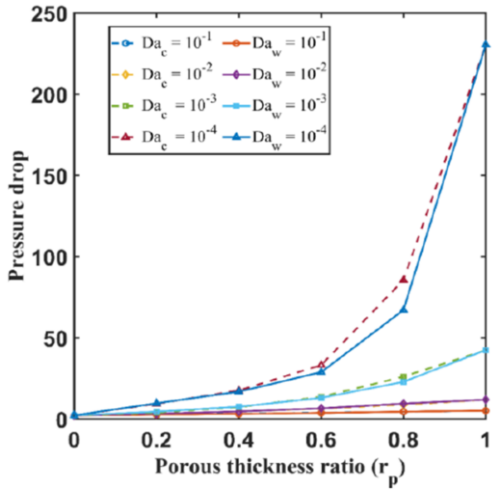


Fig. 7. Pressure drop with porous layer thickness ratio (r_p) at different Darcy numbers for both geometric configurations.

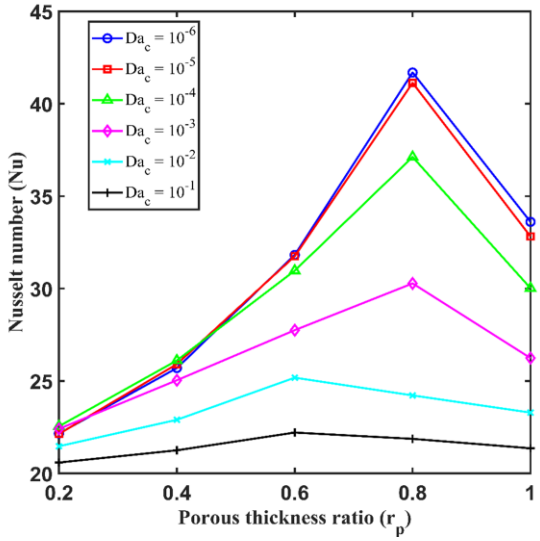


Fig. 8. Variation of the Nusselt number with the porous layer thickness ratio (r_p) at a thermal conductivity ratio (k_r) of 2.5 and different Darcy number (Da) values, when a porous layer is at the channel core.

a reduction in heat transfer performance, and resulting in a decrease in the Nusselt number. A similar variation in the Nusselt number has been reported in previous studies [36].

Figure 9 demonstrates the relationship between the Nusselt number (Nu), the porous layer thickness ratio (r_p) and the Darcy number (Da) for a higher thermal conductivity ratio. The results indicate that an increase in the thermal conductivity of the solid matrix improves the heat transfer rate. As the thermal conductivity ratio (k_r) increases, the porous material's ability to conduct heat also improves, resulting in higher Nusselt numbers. Notably, the impact of increasing thermal conductivity ratio becomes significant at higher porous layer thickness ratios. This is because as the porous layer thickness ratio (r_p) increases, the flow through the porous medium becomes less efficient, making the thermal conductivity of the solid matrix more important for heat transfer. Furthermore, when the porous material completely fills the channel (r_p = 1.0), the increase in thermal conductivity of the solid matrix results in the most substantial enhancement in heat transfer rates. In this fully filled configuration, the solid matrix can effectively conduct heat, maximising the thermal performance of the system. These findings emphasise that both the porous layer thickness ratio and the thermal conductivity of the solid matrix play a crucial role in heat transfer rates, with the extent of their impact being influenced by Darcy numbers.

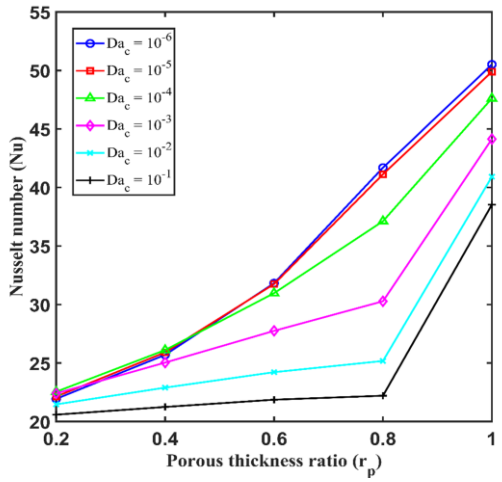


Fig. 9. Variation of the Nusselt number with the porous layer thickness ratio (r_p) at a thermal conductivity ratio (k_r) of 25 and for different Darcy number (Da) values, when a porous layer is at the channel core.

Figure 10 further demonstrates the influence of a higher thermal conductivity ratio on thermal performance across various configurations, comparing the porous layer placed at the core versus at the walls, for different Darcy numbers. Figures 10a and

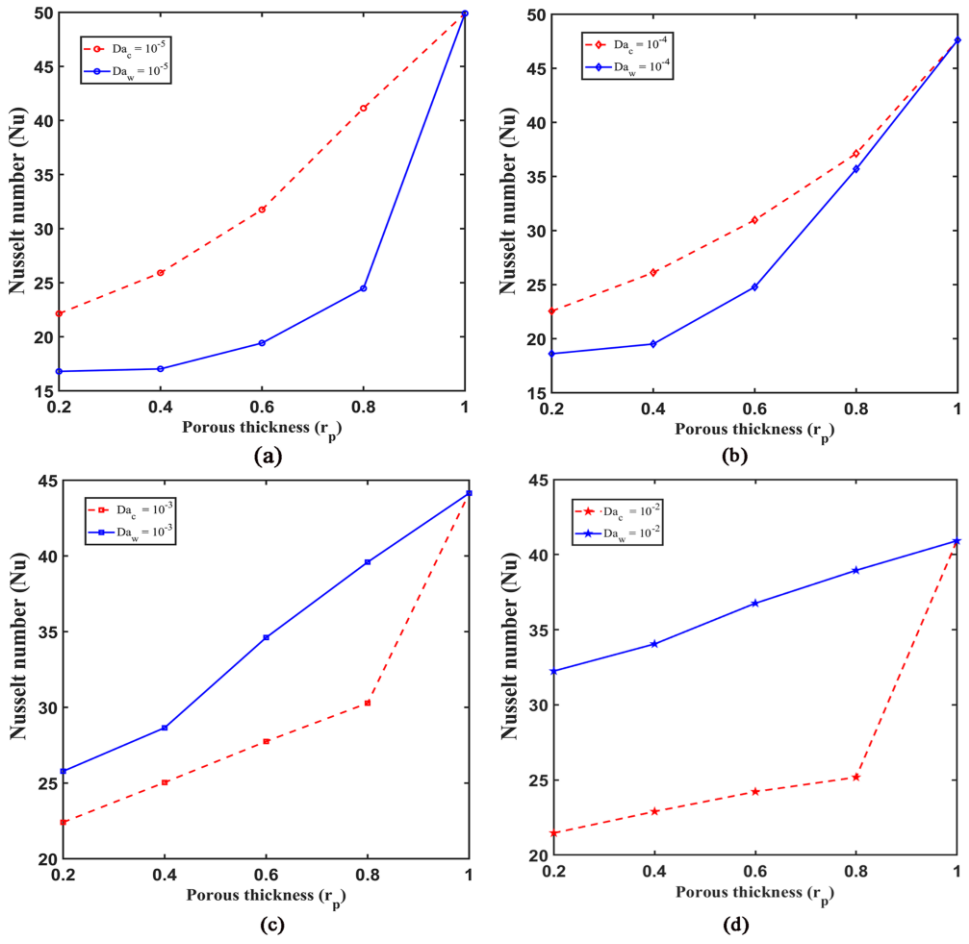


Fig. 10. Variation of the Nusselt number with respect to the porous layer thickness ratio (r_p) for two configurations: a porous layer in the core and a porous layer at the walls, for different Darcy numbers.

10b, corresponding to Darcy numbers of 10^{-5} and 10^{-4}, respectively, show that for less permeable porous media, positioning the porous layer in the channel core results in higher Nusselt numbers. However, this configuration also leads to higher pressure losses compared to the configuration with the porous layer adjacent to the walls, especially at lower Darcy numbers. In contrast, Figs. 10c and 10d, corresponding to Darcy numbers of 10^{-3} and 10^{-2}, respectively, reveal that higher permeability yields better thermal performance when the porous layer is located near the walls. This configuration achieves higher Nusselt numbers with lower pressure drops. The improved thermal performance in this case is due to higher permeability, which enhances fluid-wall interaction when the porous medium is adjacent to the walls, allowing the fluid to flow more freely and improving contact with the wall surface. This, in turn, increases convective heat transfer, resulting in a higher Nusselt number. Nusselt number as a function of the Darcy number for both cases for a porous layer thickness ratio of 0.8 is depicted in Fig. 11. The same value of the Nusselt number is observed for both configurations when the Darcy number is between 10^{-4} to 10^{-3}. The same transition of the Nusselt number with the Darcy number is seen for all porous layer thickness in this study. The thickness ratio of $r_p = 0.8$ is selected as it represents the optimal configuration that yields the maximum heat transfer enhancement among all thickness ratios investigated in the present study.

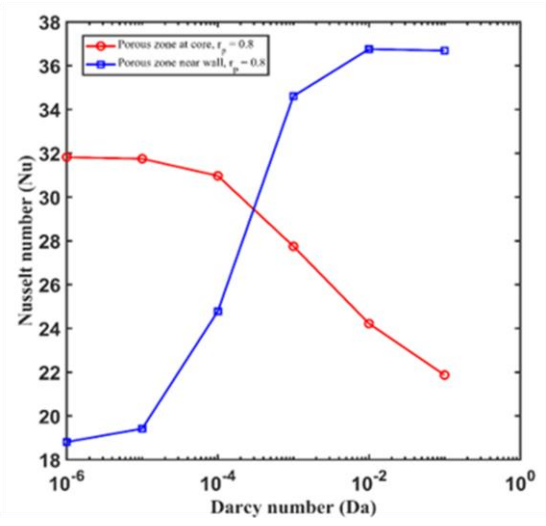


Fig. 11. Variation of the Nusselt number with the Darcy number at $r_p = 0.8$ for both configurations.

6. Conclusions

The present study numerically investigates the heat transfer enhancement in the channel using two different configurations: a porous layer in the channel core and a porous layer located near the channel wall. The effects of Darcy number ($0.000001 < Da < 0.01$), the ratio of porous layer thickness ($0.2 < r_p < 1$) and the thermal conductivity ratio are studied to understand how porous medium properties govern flow resistance and thermal performance. The main outcome of the study is summarised below:

1. The highest fluid velocity is observed when the porous layer thickness ratio is 0.8 in both geometric configurations. This indicates the optimal configuration for enhancing convective heat transfer between the fluid and the channel wall.
2. Pressure-drop analysis indicates that positioning the porous layer at the walls consistently results in lower pressure losses for all Darcy numbers, whereas placing the porous layer at the core introduces a stronger flow obstruction and therefore higher-pressure penalties. The pressure drop increases as the porous layer thickness ratio increases.
3. The results show that the porous layer thickness ratio plays a dominant role in flow-thermal behaviour. The maximum improvement in heat transfer is achieved at a thickness ratio of 0.8, where the Nusselt number increases by about 88% compared to the baseline. For the low thermal conductivity ratio of 2.5, increasing thickness beyond this point leads to a 19.5% reduction in the Nusselt number due to increased thermal resistance. In contrast, for a high conductivity ratio of 25, the heat transfer rate continues to rise with thickness, yielding as much as 80.5% improvement relative to the low-conductivity case.
4. At lower Darcy numbers, the Nusselt number is higher when the porous layer is located at the core of the channel, irrespective of the porous layer thickness ratio. In contrast, at higher Darcy numbers, the Nusselt number is higher when the porous layer is positioned near the walls, especially for high thermal conductivity ratios and for all porous layer thickness ratios.

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