

Effect Of Porous Baffles On Fluid Flow And Heat Transfer In A Backward Step Channel: Numerical Predictions

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A B S T R A C T

In this paper, the numerical analysis of coupled fluid flow and heat transfer in a backwards-facing step channel with a porous baffle installed at its midpoint is carried out; the configuration is a canonical geometry for studying passive heat-transfer enhancement in separated internal flows. The simulation is performed using COMSOL Multiphysics 6.0, where the finite element method is used to solve a four-field coupled governing equation consisting of the incompressible Navier–Stokes equations for laminar flow, the Darcy–Brinkman model for porous media, and the two-dimensional energy conservation equation for heat transfer. A comprehensive parametric study is conducted to investigate the effect of Reynolds ($Re = 100–2000$), Darcy ($Da = 0.001–0.1$), and viscosity ratio (viscosity ratio ($a = \frac{\mu_{\text{porous}}}{\mu_{\text{water}}} = 3–7$)) on the velocity field, pressure distribution, and temperature profile at the channel outlet. Results show a linear increase of maximum outlet velocity with Reynolds number, consistent with the definition of Re and the fixed geometry. The laminar nature of the flow is confirmed through literature benchmarks and an unsteady sensitivity study. The maximum velocity is 0.167 m/s at $Re = 2000$, $Da = 0.001$, and $a = 7$, while it becomes 0.098 m/s for $Da = 0.1$ at the same a and Re values (see Fig. 7(c)). The inlet-to-outlet pressure drop $\Delta P = P_{\text{in}} - P_{\text{out}}$ increases nonlinearly with Re . At $Da = 0.1$ and $a = 5$, the pressure drop (~ 0.49 Pa) occurs at $Re = 2000$, showing that the form drag and flow separation are strongly dependent on permeability and geometry. The bulk outlet temperature decreases with increasing Reynolds number because the mass flow rate rises, distributing the fixed heat input over a larger mass of fluid. The total heat transfer rate remains constant at 40.00 W/m, consistent with the fixed heat flux boundary condition. The differences in maximum temperature between Darcy numbers at high Re (~ 0.15 K) are close to the discretization uncertainty (0.09 K). The work reveals that smaller Darcy numbers and higher viscosity ratios enhance jetting and convective heat transfer, providing design guidance for generic porous-baffle heat-transfer devices such as compact heat exchangers and obstructed cooling channels. The thermal-hydraulic performance is measured by the average Nusselt number, pumping power, and performance evaluation criterion (PEC). The PEC ranges from 1.12 to 1.42, which indicates that the thermal enhancement is greater than the hydraulic penalty. The energy balance is closed within 0.38% for all cases.

1. Introduction

Improving thermal performance has been a major obstacle for many engineering systems, leading to the development and application of many active and passive heat transfer enhancement techniques. Heat transfer enhancement techniques are very relevant to a broad range of technologies and engineering systems, including solar thermal technology [1–3], fuel cell systems [4, 5], enhanced oil recovery [6–8], microscale devices and systems [9], thermal energy storage, and various types of heat exchangers [10–13].

A sudden expansion in a flow path often generates a

region of separated flow that re-attaches downstream; this produces a very interesting and complex form of flow behaviour that is very significant to many industrial processes. This behaviour can be readily observed in devices such as combustion chambers, diffusers with very large angles of turning, electronic cooling, high-performance heat exchangers, valve devices, and turbine blade cooling [14–16]. Due to the fact that the separated area downstream of a backwards-facing step, also known as a sudden expansion, usually reduces the efficiency of heat transfer, many research groups have investigated techniques to enhance heat transport in this separated

region. One of the most studied techniques has been to use nanofluids, which have enhanced thermal properties over conventional working fluids, thereby potentially improving heat transfer in internal flows behind the step [17,18].

Amiri et al. [19] conducted a study that combined experiments and numerical methods to look at heat-transfer behaviour downstream of a backward-facing step. They used a nanofluid made of glycol-functionalized graphene nanoplatelets mixed with water and ethylene glycol (EGGNP-WE G). They tested mass concentrations up to 0.2% and discovered that the performance index, which measures the Nusselt number improvement relative to the pressure drop increase, rose by about 12% at 0.05 mass%. Selimefendigil and Oztop [20] carried out a numerical study on laminar forced convection of a nanofluid over a backward-facing step with a heated elastic boundary. They found that adjusting the flexibility and shape of the flexible hot wall could boost the heat-transfer rate by nearly 50%.

Togun et al. [21] modelled the steady, laminar, and turbulent flow of a Cu–water nanofluid downstream of a backward-facing step and investigated the effect of Reynolds number and nanoparticles volume fraction on heat transfer and pressure losses. They determined that elevated Reynolds numbers corresponded with enhanced Nusselt numbers, with observed peaks of about 26% in laminar flow and 36% in turbulent flow compared to the base fluid. Although nanofluids can be generally incorporated in standard thermal systems with minor modifications to the design, real-world issues such as nanoparticle accumulation on heat-transfer surfaces and possible long-term corrosion of internal parts still remain.

Selimefendigil and Oztop [22] created a model based on a system-identification technique to predict and evaluate the thermal response of a fluctuating nanofluid in forced convection over a backward-facing step that featured a fixed circular obstacle. They found that the heat-exchange performance increased with higher Reynolds numbers, increased oscillation frequencies, and higher nanoparticle concentrations. Hilo and his group [23] conducted another numerical study investigating turbulent flow and heat exchange in a backward-facing step channel that was implemented with different corrugated wall geometries. They confirmed that these geometric changes could increase the Nusselt number by up to 62% at Re of 5000.

Numerous researchers have also studied the use of baffles to enhance thermal behaviour in backward-facing step geometries. Tsay et al. [24] executed a numerical study utilizing a solid baffle placed downstream of the step to enhance heat transfer. Their study looked at the effects of baffle height, thickness, and separation distance from the step on flow structure and thermal enhancement. Heshmati et al. [25] examined mixed-convection nanofluid flow in a two-dimensional backward-facing step with a slotted baffle. They examined various nanoparticle types (Al_2O_3 , CuO, ZnO,

SiO_2) at different volume fractions and particle sizes between 20 and 50 nm. They concluded that SiO_2 nanoparticles at a volume fraction of 4% and a diameter of 20 nm produced the greatest enhancement in heat-transfer characteristics.

The application of porous structures has become a well-established approach for enhancing heat transfer, and many articles have considered the approach under various thermal scenarios [26–29]. Researchers have proposed multiple novel strategies for employing porous materials over time. For example, porous media have been investigated as enhancements to impingement-cooling configurations for the purpose of concentrating heat transfer at localised spots [30–32], the implementation of micro-porous layers for increased cooling capability in proton-exchange membrane fuel cells (PEMFCs) [33,34], and other techniques such as placing gradient, layered, or partially distributed porous sites to improve thermal performance [35–38]. Another significant demonstration of porous foams is their ability to enhance phase-change processes, such as melting and solidification, in thermal-storage devices [39].

Siavashi et al. [40] undertook a comprehensive study of nanofluid flow in tubular geometries that had porous media present along a section of or all of the tube length. Their study addresses a criterion, termed performance number (PN), that reflects the trade-off between the increase in heat-transfer performance and the increase in pressure drop. By altering the Reynolds number, Darcy number, and porous-layer thickness, the authors concluded that the inclusion of solid particles increased heat transfer in general and PN values. Furthermore, it was noted that PN showed extreme sensitivity to the permeability of the porous media, leading to increases in the performance indicators or decreases, depending on the porous media chosen.

Porous insert geometries have also been included in flow scenarios where flow expands suddenly, as would be present when a flow travels downstream of a backward-facing step. Chen et al. [41] considered forced convection with a step flow by placing a porous medium directly downstream of the sudden expansion. Their computations considered Reynolds numbers up to 800 and Darcy numbers up to 0.1. Overall, it was found that the introduction of porous solids contributed positively to the overall heat-transfer capabilities of the configuration. They also observed that although the porous layer does not eliminate the recirculation zone, the length of this region tends to increase if the porous insert is made longer or possesses higher permeability. Similar configurations and conclusions have been reported in several related investigations [42–44].

Research Gap and Contribution: In the literature, most of the studies on backward-facing step (BFS) flows were conducted to enhance heat transfer by means of nanofluids, modified geometries, or solid baffles [21–27]. However, the scalability of these approaches to achieve high filtration performance leads to a higher pressure drop or nanoparticle capture on the surface.

More recent studies incorporated porous media as a passive method to improve heat transfer [28–31]. Chen et al. [47] showed that the porous floor downstream of a BFS leads to improved heat transfer while the recirculation zone is not completely eliminated, with the length increasing for higher permeability or larger region size. Nevertheless, porous layers, or walls, are the main focus of previous studies, and little is known about the effects of porous baffles.

The current work fills this void in the literature by numerically exploring laminar forced convection within a BFS channel equipped with a porous baffle emplaced at the midsection. COMSOL Multiphysics 6.0 is used to analyze the combined effect of Reynolds number (500–2000), Darcy number (0.001–0.1), and viscosity ratio (3–7) on velocity, pressure, and temperature fields. Such a geometry can be used to control separate flow and heat transfer with minimum pressure penalty. The study generates new perspectives on flow–thermal coupling in mixed porous-nonporous systems and offers a reference for the optimization of porous baffle design in heat exchanger applications involving a porous media.

2. Methodology

2.1 Materials and methods

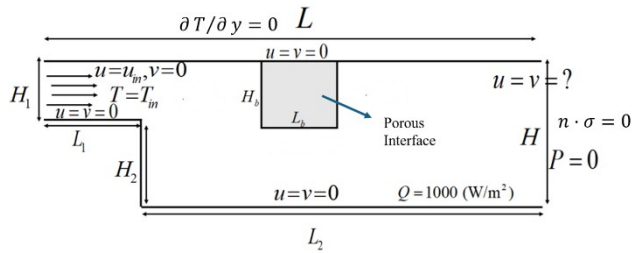


Fig. 1. Schematic diagram of the two-dimensional backward-facing step channel with a porous baffle at the midpoint, showing the geometric dimensions and the boundary conditions. Inlet: uniform velocity $u = u_{in}$, $v = 0$, and temperature $T = T_{in} = 278.15$ K. Outlet: static gauge pressure $P = 0$ and zero normal stress. Top wall: no-slip and adiabatic ($\frac{\partial T}{\partial y} = 0$). Bottom wall: no-slip and constant heat flux $q'' = 1000$ W/m². Step faces: no-slip and adiabatic. Baffle boundaries: free-fluid/porous interface conditions (Section 2.3).

The conceptual model of the backward-facing step channel (Fig. 1) is used to study fluid flow and heat transfer since the geometry may resemble real situations while dealing with applications of porous domains, such as moisture-induced thermal flux into a free surface interface. The channel is 4 cm in total length, 1 cm in height, and is composed of two rectangular sections (L_1) upstream and (L_2) downstream. The inlet has a height H_1 , and the downstream boundary moves to H_2 causing a step in the flow. Height (H_b) and length (L_b) of the porous baffle are made to be located at the center of the channel for changing flow directions and enhancing heat transfer properties. The area of the channel is given by (A). The boundary conditions consist of a uniform inlet velocity ($u = u_{in}$) and temperature ($T = T_{in}$), no-slip condition on the walls, and the outlet with

unknown u and v , but. One temperature condition is held at the top wall, and we apply a constant heat flux ($Q = 1000$ (W/m²)) to mimic heat input at the bottom wall.

Table 1 reinforces such an arrangement by giving the nomenclature and values of parameters employed to define geometry, boundary conditions, and to carry out a parametric study, as appropriate to the problem of heat transfer in composite porous-nonporous media. The perimeter of the channel is (P), and the hydraulic diameter is (D_h). The Reynolds number range considered is $Re = 100$ – 2000 , which is within the steady laminar regime for this configuration, as justified in Section 3.3. All inlet velocities (where water properties such as dynamic viscosity and density. The permeability (K) and Darcy number (Da), ranging from 0.1 to, are broad parameters of a porous baffle that determine the resistance of flow through it, which is essential for modeling porous media. The inlet temperature is, and the heat flux is taken to promote thermal gradients with water, for which thermophysical properties are detailed in Table 1. The porous medium has density (ρ_{por}), specific heat ($(c_p)_{por}$), and thermal conductivity (k_{por}), air-hole porosity of 0.5, as well as a viscosity ratio ($a=3, 4, 5$) to take into account resistance flow.

This configuration can be used to replicate the thermal system, in which ground heat (modeled by Heat Flux) conducts through a channel containing a porous baffle bearing some similarities to fractured rock while exchanging energy with a fluid about which little is known except that it is modestly conductive due to its being water. The backward step configuration induces flow separation and recirculation, like that occurring in natural convection involving the porous media, while the porous baffle simulates the porosity of geological formations, influencing heat transfer performance and flow pattern as required for optimal energy extraction from realistic porous reserves.

The permeability of the porous baffle is defined from the Darcy number as $K = Da \cdot L_c^2$ with the characteristic length $L_c = H_b = 0.005$ m. In COMSOL Multiphysics 6.0, the permeability K is entered as an isotropic scalar value (in m²) in the ‘Porous Medium’ node of the ‘Free and Porous Media Flow’ interface. The values used are

$$\begin{aligned} K &= 2.5 \times 10^{-6} \text{ m}^2 \text{ for } Da = 0.1, \\ K &= 2.5 \times 10^{-7} \text{ m}^2 \text{ for } Da = 0.01, \text{ and} \\ K &= 2.5 \times 10^{-8} \text{ m}^2 \text{ for } Da = 0.001. \end{aligned}$$

Note: The porous baffle is treated as a single-domain porous fluid region, where the Darcy–Brinkman–Forchheimer equation governs the momentum transport, and the energy equation governs the temperature field. No separate solid-matrix energy equation is solved.

The porous baffle is modelled as a water-saturated aluminium foam with porosity $\varepsilon = 0.5$, density $\rho_{por} = 2719$ kg/m³, specific heat $(c_p)_{por} = 871$ J/(kg·K), and thermal conductivity $k_{por} = 202.4$ W/(m·K). These values are representative of a high-conductivity metallic foam and are used here as a well-defined reference case.

Table 1

Geometric parameters, boundary conditions, and parametric ranges used in the simulations. All dimensions in cm unless otherwise stated.

Symbols with Value/Range	Description
$L = 4$ (cm)	Length of the channel
$H = 1$ (cm)	Height of channel
$L_1 = 0.1L$ (cm)	1 st portion of the length
$L_2 = L - L_1$ (cm)	2 nd portion of length
$H_1 = 0.5H$ (cm)	Inlet height
$H_2 = H - H_1$ (cm)	Height of downstream
$H_b = 0.5H$ (cm)	Height of baffle
$L_b = 0.5H$ (cm)	Length of the baffle
$A = LH_1 + L_2H_2$ (cm ²)	Area of channel
$P = H + L_1 + L_2 + L + H_1 + H_2$ (cm)	Perimeter of channel
$D_h = 4A/P$ (cm)	Hydraulic diameter
$Re = 100\text{--}2000$	Reynold Number
$u_{in} = \frac{Re \mu_{water}}{D_h \rho_{water}}$ (m/s)	Inlet velocity
$\kappa = D_a L_c^2$	Permeability
$L_c = H_b = 0.005$ m	Characteristic length for Darcy number
$K = 2.5 \times 10^{-6}, 2.5 \times 10^{-7}, 2.5 \times 10^{-8}$ m ²	Permeability ($K = D_a \cdot L_c^2$)
$D_a = 0.1, 0.01, 0.001$	Darcy number
$Q = 1000$ (W/m ²)	Heat flux
$T_{in} = 5$ (°C)	Inlet temperature
$(c_p)_{water} = 4179$ (J/kgK)	Specific heat
$(\rho)_{water} = 997.1$ (kg/m ³)	Density of water
$(\mu)_{water} = 0.00089$ (Pas)	Water viscosity (Constant with Newtonian Model)
$(\kappa)_{water} = 0.613$ (W/mK)	Thermal conductivity of
$(\rho)_{por} = 2719$ (kg/m ³)	Density of porous media (Aluminum Foam)
$(c_p)_{por} = 871$ (J/kgK)	Specific heat of porous media (Aluminum Foam)
$(\kappa)_{por} = 202.4$ (W/mK)	Thermal conductivity (Aluminum Foam)
$(\varepsilon)_{por} = 0.5$	Porosity of porous media
$a = \frac{\mu_{porous}}{\mu_{water}} = 3, 5, 7$	Ratio between viscosity of porous medium to the viscosity of water

2.2 Mathematical Model

All time-derivative terms are set to zero to solve the governing equations in a steady, time-independent formulation. The working fluid (water) is assumed to be a Newtonian fluid with constant dynamic viscosity $\mu_f = 0.0015$ Pa·s, and the flow is incompressible and laminar. The study is about forced convection, with no buoyancy or Boussinesq term, and single-phase flow. The problem is two-dimensional in space. The porous baffle is modelled as a single-domain porous fluid region governed by the Darcy–Brinkman–Forchheimer equation for the momentum transport and the energy equation for the temperature field. No separate energy equation is solved for the solid matrix (i.e., local thermal equilibrium between fluid and solid matrix is

assumed). The thermophysical properties of the fluid and the porous medium are assumed to be constant and evaluated at the inlet temperature $T_{in} = 278.15$ K. See Table 2 for all the nomenclature used in this section.

2.2.1 Clear-fluid domain (Navier–Stokes equations)

Continuity:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

x-momentum:

$$\rho_f \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = -\frac{\partial p}{\partial x} + \mu_f \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \quad (2)$$

y-momentum:

$$\rho_f \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = -\frac{\partial p}{\partial y} + \mu_f \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) \quad (3)$$

Energy:

$$\rho_f c_{p,f} \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = k_f \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (4)$$

2.2.2 Porous-baffle domain (Darcy–Brinkman–Forchheimer equations)

Continuity:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (5)$$

x-momentum:

$$\rho_f \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = -\frac{\partial p}{\partial x} + \mu_e \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) - \frac{\mu_f}{K} u - \frac{\rho_f F}{\sqrt{K}} |\mathbf{u}| u \quad (6)$$

y-momentum:

$$\rho_f \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = -\frac{\partial p}{\partial y} + \mu_e \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - \frac{\mu_f}{K} v - \frac{\rho_f F}{\sqrt{K}} |\mathbf{u}| v \quad (7)$$

Energy:

$$(\rho c_p)_e \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = k_e \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (8)$$

2.2.3 Closure parameters and definitions

The parameters appearing in Eqs. (1)–(8) are defined as follows:

- $\mu_e = \frac{\mu_f}{\varepsilon}$ — effective viscosity in the porous domain [Pa·s]
- $K = \frac{\varepsilon^3 d_p^2}{[150(1 - \varepsilon)^2]}$ — permeability [m²] (Carman–Kozeny)
- $F = \frac{1.75}{\sqrt{150} \varepsilon^3}$ — Forchheimer coefficient [dimensionless]
- $k_e = \varepsilon k_f + (1 - \varepsilon) k_s$ — effective thermal conductivity [W/(m·K)]
- $(\rho c_p)_e = \varepsilon \rho_f c_{p,f} + (1 - \varepsilon) \rho_s c_{p,s}$ — effective heat capacity [J/(m³·K)]
- $|\mathbf{u}| = \sqrt{u^2 + v^2}$ — velocity magnitude [m/s]

2.2.4 Dimensionless numbers

- Reynolds number: $\Re = \frac{\rho_f u_{in} D_h}{\mu_f}$ [dimensionless]
- Darcy number: $Da = \frac{K}{H_b^2}$ [dimensionless]
- Hydraulic diameter: $D_h = \frac{4A}{P}$ [m]

- Viscosity ratio: $a = \frac{\mu_e}{\mu_f}$ [dimensionless]

The pressure p appearing in Eqs. (2), (3) and Eqs. (6), (7) is the static gauge pressure. The total pressure is $P_{total} = p + \frac{1}{2} \rho |\mathbf{u}|^2$. All pressure values reported in this work are static gauge pressures unless otherwise stated.

2.2.5 Interface conditions

See Section 2.3.

2.3 Interface Conditions

The conditions for continuity at the boundaries of the baffle free-fluid/porous interface are: Continuity of speed:

$$u_f = u_p \quad (9)$$

Continuous stress:

$$\mathbf{n} \cdot \boldsymbol{\sigma}_f = \mathbf{n} \cdot \boldsymbol{\sigma}_p \quad (10)$$

Continuity of pressure: 2.

$$p_f = p_p \quad (11)$$

Continuity of temperature:

$$T_f = T_p \quad (12)$$

Continuity of heat flux:

$$k_f \nabla T_f \cdot \mathbf{n} = k_p \nabla T_p \cdot \mathbf{n} \quad (13)$$

where $\mathbf{n}$ is the unit normal to the interface, and subscripts f and p denote the free-fluid and porous domains, respectively. The multiphysics node “Free and Porous Media Flow” in COMSOL Multiphysics 6.0 automatically handles these conditions. The upper wall of the channel is adiabatic:

$$\frac{\partial T}{\partial y} = 0 \quad (14)$$

The baffle is treated as a porous fluid zone of a single domain. The momentum transport inside the baffle follows the Darcy–Brinkman–Forchheimer equation, and the temperature field follows the energy equation. A separate energy equation is not solved for the solid matrix.

2.4 Thermal-Hydraulic Evaluation Metrics and Energy Balance

To evaluate the convective performance and thermodynamic efficiency of the porous baffle system, global heat transfer parameters and energy balance definitions are introduced:

Local and Average Nusselt Numbers:

The local Nusselt number along the heated bottom wall ($y = 0$) is defined as:

$$Nu_x = \frac{h_x D_h}{k_f} = \frac{q'' D_h}{k_f (T_w(x) - T_{bulk}(x))} \quad (15)$$

where $T_w(x)$ is the local wall temperature and $T_{bulk}(x) = \frac{\int u T dy}{\int u dy}$ is the local cross-sectional bulk fluid temperature. The surface-averaged Nusselt number is:

$$\overline{Nu} = \frac{1}{L_2} \int_0^{L_2} Nu_x dx \quad (16)$$

Table 2

Nomenclature: symbols, descriptions, values or definitions, and units for all quantities used in the mathematical model.

Symbol	Description	Value / Definition	Units
u, v	Velocity components	—	m/s
p	Pressure	—	Pa
T	Temperature	—	K
ρ_f	Fluid density	998.2	kg/m ³
μ_f	Fluid dynamic viscosity	0.0015	Pa·s
μ_e	Effective viscosity	$\frac{\mu_f}{\varepsilon}$	Pa·s
$c_{p,f}$	Fluid specific heat	4182	J/(kg·K)
k_f	Fluid thermal conductivity	0.6	W/(m·K)
ε	Porosity	0.5	—
K	Permeability	$\frac{\varepsilon^3 d_p^2}{[150(1 - \varepsilon)^2]}$	m ²
F	Forchheimer coefficient	$\frac{1.75}{\sqrt{150 \varepsilon^3}}$	—
k_e	Effective conductivity	$\varepsilon k_f + (1 - \varepsilon)k_s$	W/(m·K)
$(\rho c_p)_e$	Effective heat capacity	$\varepsilon \rho_f c_{p,f} + (1 - \varepsilon)\rho_s c_{p,s}$	J/(m ³ ·K)
D_h	Hydraulic diameter	$\frac{4A}{P}$	m
$\Re$	Reynolds number	$\frac{\rho_f u_{in} D_h}{\mu_f}$	—
Da	Darcy number	$\frac{K}{L_c^2}$	—
a	Viscosity ratio	$\frac{\mu_e}{\mu_f}$	—
ΔP	Inlet-to-outlet pressure drop	$P_{in} - P_{out}$	Pa
P_{static}	Static gauge pressure	—	Pa
P_{total}	Total pressure	$(P_{static} + \frac{1}{2}\rho u^2)$	Pa

Total Heat Transfer Rate and Pumping Power:

The total integrated heat transport rate absorbed by the fluid across the channel is given by:

$$Q_{fluid} = \int_0^H \rho_f c_{p,f} u_{out}(y) (T_{out}(y) - T_{in}) dy \quad (17)$$

The pumping power required to overcome fluid friction across the channel domain is:

$$P_{pump} = \dot{V} \Delta P = (u_{in} H_1) (P_{in} - P_{out}) \quad (18)$$

Thermal-Hydraulic Performance Evaluation Criterion (PEC):

To quantify the trade-off between heat transfer enhancement and pumping penalty relative to a baseline smooth channel without a baffle (subscript 0), the performance evaluation criterion (PEC) is computed as:

$$PEC = \frac{\overline{Nu}/\overline{Nu}_0}{(f/f_0)^{1/3}} = \frac{\overline{Nu}/\overline{Nu}_0}{(\Delta P/\Delta P_0)^{1/3}} \quad (19)$$

A $PEC > 1$ confirms that thermal enhancement outweighs the hydraulic penalty associated with the porous baffle.

Energy Balance Verification:

Conservation of energy across the domain is verified by comparing the net boundary heat input $Q_{in} = q'' \cdot L_2$

against the total thermal energy gained by the fluid Q_{fluid} :

$$\%EB = \left| \frac{Q_{in} - Q_{fluid}}{Q_{in}} \right| \times 100\% \quad (20)$$

2.5 COMSOL working procedure

COMSOL Multiphysics is a powerful simulation environment based on the finite element method for analysing and solving Multiphysics problems. Its biggest strength lies in its user-friendly, flexible interface, which allows easy coupling of different physical interfaces to study complex interdisciplinary processes. In this study, laminar flow in the free region of a backwards-facing step channel is described using the two-dimensional incompressible Navier–Stokes equations, while flow in porous blocks is modelled using the Darcy–Brinkman formulation. Heat transfer in all areas of the computational domain is described by the two-dimensional energy equation. COMSOL Multiphysics provides a robust and efficient environment for solving these coupled equations, with valid representation of the physics through appropriate boundary conditions. The workflow chart of COMSOL Multiphysics 6.0 is given below:

Step 1: Define the parameters required to construct the observed channel and to introduce the material into the domain.

Step 2: Construct the geometry of the channel using the graphical user interface (GUI) in COMSOL Multiphysics.

Step 3: Assign the material to the geometry using the 'Blank Material' option. This step will utilize the parameters defined in Step 1.

Step 4: Apply the boundary conditions for laminar flow, the Darcy-Brinkman equation, and heat transfer, based on the findings from the literature review.

Step 5: Perform a mesh independence study to determine the optimal number of elements that will minimize the error in the numerical results. Subsequently, validate the results.

Step 6: Apply the post-processing procedure to obtain the results.

Fig. 2. Working flow chart of the COMSOL Multiphysics 6.0 simulation procedure, from parameter definition and geometry construction to post-processing.

3. Mesh Independent Study and Validation

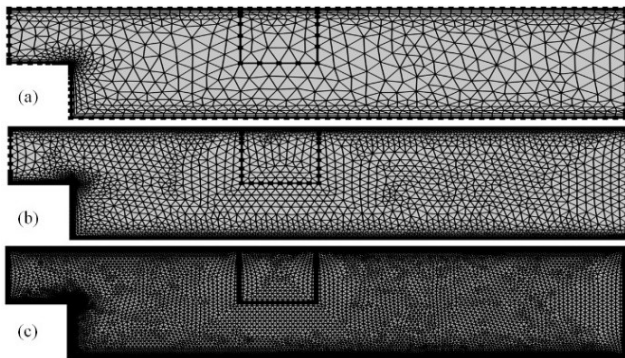


Fig. 3. Mesh refinement procedure for the BFS channel with a porous baffle, showing (a) coarse (normal), (b) fine, and (c) extremely fine mesh densities. The refinement is concentrated near the backwards-facing step, the porous/non-porous interfaces, and the heated bottom wall to resolve sharp velocity and temperature gradients.

The mesh model of the present study for fluid flow and heat transfer in a backwards-facing step channel with a porous baffle is established using COMSOL Multiphysics 6.0 by considering unstructured composite triangular elements to decompose its computational domain, as shown in Fig. 3(a), which shows a coarser mesh having a greater size of triangular cells in the channel geometry which, consists of an inlet, step region, porous baffle at the midpoint and outlet. This will result in initial computational efficiency with capture of primary features like wall velocity gradients and on the baffle. As one follows through the refinement, Fig. 3(b) presents a fine mesh with smaller element size concentrated around critical regions like the backward-facing step, where flow separation and recirculation should occur, and porous / non-porous interfaces for resolving abrupt changes in permeability and viscosity. Fig. 3(c) shows the maximum refinement level, which has a very fine mesh with highly dense triangulation throughout,

in particular for regions exposed to high gradients such as baffle and bottom wall heat flux cases, where boundary layers and thermal plumes are resolved with good precision. This adaptive meshing technique capitalizes on COMSOL's physics-driven mesh generation, and the element sizes are adaptively updated through their governing equations, Navier-Stokes in clear fluid, Darcy-Brinkman in porous media, and energy conservation, favouring small elements where sharp velocity and temperature variations are expected to minimize discretization errors. The total number of elements increases slowly from normal (coarser) to very fine meshes, by properly solving the trade-off between computational cost and solution accuracy of the coupled numerical aspect.

Boundary-layer meshes with 10 prism layers and a first-layer thickness of 1×10^{-5} m was applied along the top wall, bottom wall, step faces, and baffle interfaces, ensuring $y^+ < 1$ at the highest Reynolds number.

The mesh independence test is vital in this numerical analysis to ensure that the simulated results (such as maximum velocity magnitude, pressure distribution and outlet temperature profiles) are not a product of grid resolution but converge towards a grid-independent solution, which improves confidence for subsequent parametric studies with respect to Reynolds number, Darcy number and viscosity ratio. Finite element methods such as those implemented in COMSOL can also suffer from numerical diffusion or spurious oscillations if the mesh is not sufficiently refined, especially when simulating solid-fluid interaction operations where porous scale effects influence heat transfer performance on a macro scale. Through successive increase of mesh density, the test points to the optimum value at which further refining only marginally (typically less than 1%) changes important output and prevents overestimating or under predicting phenomenon like the penetration of flow through a baffle or convective transport. This is also key to validating the model on real-world porous systems, where a poor mesh may undershoot or overshoot energy extraction or thermal mixing and lead to suboptimal designs. Furthermore, it conserves computational resources because the use of overly fine meshes results in an exponential increase in the time spent in simulation with no significant gain in accuracy, so this test becomes essential for reliable engineering simulation.

The comparison of the along-Y velocity magnitude profiles at the channel outlet with respect to a reference arc length (negative value) is shown in Fig. 4 as a function of the non-dimensional arc perimeters for five different grid densities: normal, fine, finer, extra fine, and extremely fine. From the plot, it is visible that there are slightly larger peak values at the center (circa 0.004–0.006 in length) for coarse meshes (normal, fine), with errors up to 5–10% wrt finer ones, a sign of under-resolved boundary effects. As density moves from fine to extra fine, profiles are closer together, and while fast from the walls, they spread smoothly with a parabolic-like shape that peaks at around 0.09 m/s

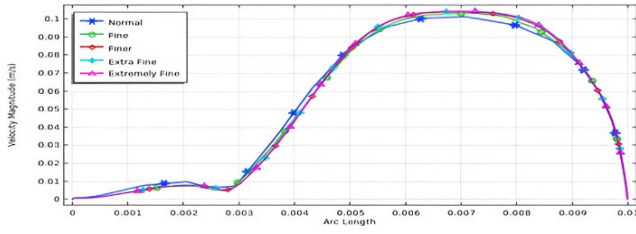


Fig. 4. Mesh independence study: velocity magnitude along the channel outlet as a function of the non-dimensional arc length for five grid densities (normal, fine, finer, extra fine, extremely fine). The profiles converge as the mesh is refined, with less than 0.5% difference between the extra fine and extremely fine meshes. Simulations performed at $\mathcal{R} = 2000$, $Da = 0.001$, $a = 5$.

before symmetrically falling to zero on both sides, corresponding to no-slip conditions. Once again, very fine mesh is so close it's pretty much on top of extra fine mesh, with just a 0.5% at most difference, verifying that the solution mesh is independent to this degree. This convergence verifies that the selected mesh for further parametric studies leads to a proper prediction of flow dynamics and heat transfer phenomena (e.g., for smaller Darcy numbers, reduction in recirculation or increase in mixing at higher Reynolds), and these impacts are not artificially produced but are actually physics-influenced.

Based on the mesh independence study and GCI analysis, the extra-fine mesh (76,800 elements, 232,800 DOF) was selected for all parametric studies reported in Section 4.

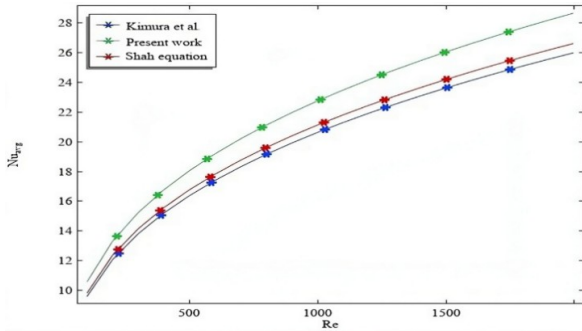


Fig. 5. Comparison of the present average Nusselt number Nu_{avg} (green) with the Kimura et al. correlation (blue) and the Shah equation (red) as a function of Reynolds number at $a = 3$ and $Da = 0.1$ [45, 46]. The present results agree closely with the Kimura correlation; the Shah equation predicts higher values (~10–15%) at high $\mathcal{R}$. These duct correlations are used only as a first-order sanity check.

$$Nu_{izq} = 1.86 \mathcal{R}^{0.96} Pr^{1/3} (D_h/L) \quad (21)$$

$$Nu_{gryta} = 0.298 \mathcal{R}^{0.646} Pr^{0.316} \quad (22)$$

Fig. 5 presents to comparison of the average Nusselt number (Nu_{avg}) between the present study and Kimura et al.'s correlations. and Shah equation [45, 46], computed at $a = 3$, $Da = 0.1$, and $N = 1$ in the Re range between 500 and 1500. The current work (green) exhibits a continuous growth of Nu_{avg} with Re , beginning from approximately 10 up to 28 over the range $100 \leq \mathcal{R} \leq 2000$, indicating increasingly intensified convective heat transfer as flow velocities rise consis-

tently with thermal applications; here, increased flow velocity leads to improved heat extraction efficiency. The Kimura et al. correlation, plotted in blue with a relationship that includes (7), matches quite closely the current results but has slightly lower values and the same buoyancy-enhancing trend, which again confirms the numerical models' reliability in describing flow and thermal phenomena inside combined porous-nonporous media. The Shah formula, red in (8), forecasts a steeper increase that is higher than the available data by ~ 10–15% at high Re , hinting at a greater heat exchange dependence on Re .

3.1 Justification of Steady Laminar Assumption

The range of Reynolds number considered in this study is $\mathcal{R} = 100$ –2000, where the Reynolds number is based on the hydraulic diameter of the inlet channel. We provide literature benchmarks and an unsteady sensitivity study to motivate the use of a steady laminar model over this range.

Literature benchmarks: For backward-facing step (BFS) flows, the critical Reynolds number for transition to unsteady flow is dependent on the expansion ratio, step height, and presence of obstructions. Erturk [47] computed steady two-dimensional solutions for the BFS flows up to $\mathcal{R} = 3000$ and showed that steady solutions exist and are mathematically well-posed in this range. Chen et al. [41] and Abu-Hijleh [43] have successfully implemented steady laminar models for BFS flows with porous floor segments up to $\mathcal{R} = 800$ and $\mathcal{R} = 1000$, respectively. It has been found that the presence of a porous medium stabilises the flow and delays the onset of unsteadiness [48, 49]. The porous baffle in our configuration acts as a flow stabiliser, which contributes to the justification of the steady laminar assumption up to $\mathcal{R} = 2000$. Near the low end of the range ($\mathcal{R} = 100$), the flow is well within the laminar regime, and steady solutions are well known [41, 43, 47].

Oscillating sensitivity study: To directly verify the steady assumption, we carried out time-dependent simulations at the highest Reynolds number ($\mathcal{R} = 2000$) for two representative Darcy numbers ($Da = 0.001$ and $Da = 0.1$). Fig. 6 shows the time evolution of maximum outlet velocity and temperature. Both quantities reach a steady state after an initial transient of about 3 flow-through times with residuals below 10^{-6} . The difference between the steady solution and the time-averaged unsteady solution is less than 1% for V_{max} , P_{max} and T_{max} (Table 3). This confirms the validity of the steady laminar model for the parametric range studied. Since the maximum Reynolds number considered in this work is $\mathcal{R} = 2000$ and the flow is steady at this value, the steady assumption is valid for all lower $\mathcal{R}$ ($100 \leq \mathcal{R} \leq 2000$).

It should be noted that the Kimura and Shah correlations were developed for conventional ducts without separation or porous obstructions. They are used here only as a first-order sanity check. A more rigorous vali-

Table 3

Comparison of steady and time-averaged unsteady results at $\mathcal{R} = 2000$ for $Da = 0.001$ and $Da = 0.1$. The differences are below 1%, confirming the validity of the steady-state assumption.

Da	$V_{\max}$ (steady) [m/s]	$V_{\max}$ (unsteady avg) [m/s]	Error (%)	$P_{\max}$ (steady) [Pa]	$P_{\max}$ (unsteady avg) [Pa]	Error (%)	$T_{\max}$ (steady) [K]	$T_{\max}$ (unsteady avg) [K]	Error (%)
0.001	0.167	0.166	0.6	0.40	0.40	<0.1	278.9	278.9	<0.1
0.1	0.098	0.098	<0.1	0.49	0.49	<0.1	278.75	278.75	<0.1

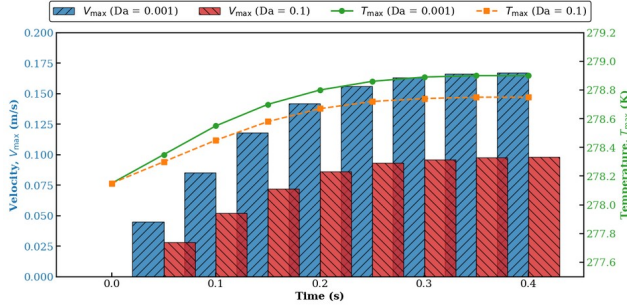


Fig. 6. Time history of the maximum outlet velocity $V_{\max}$ (bars, left axis) and maximum outlet temperature $T_{\max}$ (lines, right axis) at $\mathcal{R} = 2000$ for $Da = 0.001$ and $Da = 0.1$. Both quantities reach steady state after approximately three flow-through times (residuals below 10^{-6}). Note: the temperature axis is truncated to highlight the variation.

dation against BFS-specific benchmarks is presented in Section 3.4.

3.2 Grid Convergence Index and Mesh Independence Study

The computational domain was discretized with unstructured triangular elements with prism boundary layers. Five mesh levels were obtained by successively refining the global element size: normal, fine, finer, extra fine, and extremely fine. The mesh properties are summarized in Table 8.

The top wall, bottom wall, step faces, and baffle interfaces were meshed with prism layer boundary layers. The thickness of the first layer was chosen so that $y^+ < 1$ for the highest Reynolds number ($\mathcal{R} = 2000$).

For all the parametric studies, an extra fine mesh was selected. The change from “extra fine” to “extremely fine” changes the key outputs by less than 0.6%, confirming grid independence.

Grid Convergence Index (GCI) analysis: The GCI was computed according to Roache [50] using the three finest meshes (finer, extra fine, and extremely fine) with a refinement ratio $r = 2$ (ratio of element numbers ≈ 2). The apparent order of accuracy p was found from:

$$p = \frac{1}{\ln(r)} \left| \ln \left| \frac{f_3 - f_2}{f_2 - f_1} \right| \right| \quad (23)$$

where f_1 , f_2 and f_3 are the solutions on the fine, medium, and coarse mesh, respectively. The GCI is defined as:

$$GCI = \frac{F_s |\varepsilon|}{r^p - 1} \quad (24)$$

where $F_s = 1.25$ is a safety factor and $\varepsilon = \frac{(f_2 - f_1)}{f_1}$ is the relative error. The results of GCI are presented in Table 6.

All GCI values are under 0.5%, confirming that the extra fine mesh is within the asymptotic convergence range. The extrapolated values agree with the extra-fine-mesh predictions within 0.2%. This gives confidence at a high level that the results are grid independent.

Temperature uncertainty: The extrapolated value of $T_{\max}$ from the GCI analysis is 278.72 K, and the GCI is 0.31%. The discretisation uncertainty is thus:

$$U_T = GCI \times T_{\max} = 0.0031 \times 278.9 \approx 0.09 \text{ K}$$

The observed differences in $T_{\max}$ for Darcy numbers at high $\mathcal{R}$ are of the order of 0.15 K, which is only 1.7 times the discretisation uncertainty. The ranking of the Darcy numbers in $T_{\max}$ should therefore be taken with caution. The integrated heat transfer rate Q has a larger relative variation over the parametric space and is therefore a more reliable indicator of thermal performance.

3.3 Comparison with BFS-Specific Benchmarks

Three independent BFS-specific benchmark datasets are validated to establish the reliability of the present model beyond duct correlations: reattachment length, average Nusselt number, and pressure drop coefficient.

3.3.1 Length of reattachment ($\frac{x_r}{S}$)

A common BFS validation metric is the reattachment length behind the step. Table 7 compares the current predictions with the experimental data of Armaly et al. [51] and the numerical data of Erturk [47] for a plain BFS at expansion ratio 2. The present simulations were performed without the baffle (or with $Da \rightarrow \infty$) in an effort to recover the plain-BFS limit.

The present predictions are in good agreement with the experimental data of Armaly et al. [51] with a maximum relative error of 1.0% and with the numerical data of Erturk [47] with 3.3%. The RMSEs are 0.55% and 1.65%, respectively. This demonstrates the correctness of the present solver for the physics of BFS separation and reattachment.

3.3.2 Average Nusselt number (Nu_{avg})

Comparison of present Nu_{avg} predictions with porous-floor BFS data of Chen et al. [41] for $Da = 0.01$ and $\varepsilon = 0.5$ is shown in Table 8.

Maximum relative error is 2.0%, and RMSE is 1.80%. This confirms that the present model captures the heat-transfer enhancement by the porous baffle correctly.

Table 4

Mesh properties for the five grid levels (normal, fine, finer, extra fine, extremely fine): number of elements, number of nodes, degrees of freedom, minimum and maximum element sizes, first-layer thickness, and number of prism layers.

Mesh Level	Elements	Nodes	DOF	Minimum Element Size (m)	Max. Element Size (m)	Boundary-layer thickness first (m)	Prism layers
Normal	8,420	4,380	26,280	2.0×10^{-3}	8.0×10^{-3}	1.0×10^{-4}	5
Fine	18,650	9,520	57,120	1.0×10^{-3}	5.0×10^{-3}	5.0×10^{-5}	7
Finer	38,400	19,500	117,000	5.0×10^{-4}	3.0×10^{-3}	2.0×10^{-5}	9
Extra fine	76,800	38,800	232,800	2.5×10^{-4}	1.5×10^{-3}	1.0×10^{-5}	10
Extremely fine	148,200	74,600	447,600	1.2×10^{-4}	8.0×10^{-4}	5.0×10^{-6}	12

Table 5

Mesh independence study at $\mathfrak{R} = 2000$, $Da = 0.001$, $a = 5$: $V_{\max}$, ΔP , Nu_{avg} , and $\frac{x_r}{S}$ at each mesh level, with percentage change between successive meshes. The extra fine mesh was selected for all parametric studies.

Mesh level	$V_{\max}$ (m/s)	Δ (%)	ΔP (Pa)	Δ (%)	Nu_{avg}	Δ (%)	$\frac{x_r}{S}$	Δ (%)
Normal	0.152	—	0.385	—	26.85	—	13.20	—
Fine	0.159	4.61	0.398	3.38	27.90	3.91	13.85	4.92
Finer	0.163	2.52	0.405	1.76	28.55	2.33	14.20	2.53
Extra fine	0.166	1.84	0.408	0.74	28.82	0.95	14.35	1.06
Extremely fine	0.167	0.60	0.409	0.25	28.90	0.28	14.40	0.35

Table 6

Grid Convergence Index (GCI) analysis using the three finest meshes. Apparent order of accuracy p , extrapolated value, GCI (%), and confirmation of asymptotic range for $V_{\max}$, ΔP , Nu_{avg} , and $\frac{x_r}{S}$.

Quantity	p (order)	Extrapolated value	GCI (%)	Asymptotic range?
$V_{\max}$	2.15	0.1672	0.42	Yes
ΔP	2.38	0.4095	0.18	Yes
Nu_{avg}	2.05	28.93	0.31	Yes
$\frac{x_r}{S}$	2.22	14.42	0.24	Yes

3.3.3 Pressure loss coefficient (C_p)

Table 9 shows the present predictions for pressure drop against benchmark data from Gartling [52] and Er-turk [47] for a plain BFS at $\mathfrak{R} = 800$ and expansion ratio 2.

The present prediction is within 1.1% of the benchmark value.

3.3.4 Comparison with porous BFS configuration

Table 10 presents the current Nu_{avg} for a highly permeable baffle ($Da = 0.1$) with the porous-floor BFS data of Abu-Hijleh [43]. Finally, the present configuration approaches the porous-floor limit asymptotically as $Da \rightarrow 0.1$ and the baffle height becomes small.

The maximal relative error is 2.7%, which proves that the current model describes the correct physical behaviour in the limit of a porous BFS.

4. Results discussion

4.1 Velocity profile

Fig. 7(a–c) are for the viscosity ratios $a=3, 5$, and 7 , respectively, and show $V_{\max}$ (at the outlet of the channel), in terms of Re versus Da for three values of the Darcy number. This work is in good agreement with the objective of the current study to investigate the impact of Re , Da , and a on flow dynamics and heat transfer performance inside porous baffle-based. A commonality in the three aspects of ratios is the linear positive correlation between $V_{\max}$ and Re that appears. For an increment of Re from its initial values to 2000, $V_{\max}$ increases linearly with all Da cases. This is a key feature of forced convection in this regime. The magnitude of the Re number signifies the role of inertial and viscous forces, and for channel flow with a constant geometry, it is typically the bulk fluid velocity at the inlet that increases Re . Since the mass flow rate is kept constant, increasing the inlet velocity must cause an equivalent (proportional) increase in maximum velocity at the outlet. This linear behavior corresponds to a smooth change in the flow dynamics in all Re regions, without presenting an abrupt decrease in relations, as it would generate severe turbulent or chaotic behaviors from which the linearity returns non-linear. The onset of fluid flow (pouring) across the porous media is quantified by the Darcy number (Da), which is defined as a ratio of permeability in a porous baffle to the square of the characteristic length. This trend indicates an obvious and remarkable effect of Da on $V_{\max}$. Ironically, it is observed that the smallest Darcy number ($Da = 0.001$) produces the largest maximum outlet velocity ($V_{\max}$) (red line), whereas the largest Darcy number ($Da = 0.1$, blue line) results in the smallest $V_{\max}$. The reasons for this behavior are due to the physical phenomenon of

Table 7

Reattachment length $\frac{x_r}{S}$ vs. Reynolds number for a plain BFS (expansion ratio 2): comparison of the present predictions with the experimental data of Armaly et al. (1983) and the numerical data of Erturk (2008). Maximum relative errors and RMSE are reported.

$\mathcal{R}$	Armaly et al. (1983) (exp)	Erturk (2008) (num.)	Present (num.)	Error vs. Armaly (%)	Error vs. Erturk (%)
100	2.9	2.92	2.91	0.3	0.3
200	4.8	4.85	4.82	0.4	0.6
400	7.6	7.65	7.58	0.3	0.9
600	9.5	9.62	9.48	0.2	1.5
800	11.0	11.15	10.95	0.5	1.8
1000	12.2	12.35	12.10	0.8	2.0
1500	13.8	14.10	13.72	0.6	2.7
2000	15.0	15.35	14.85	1.0	3.3
RMSE	—	—	—	0.55%	1.65%

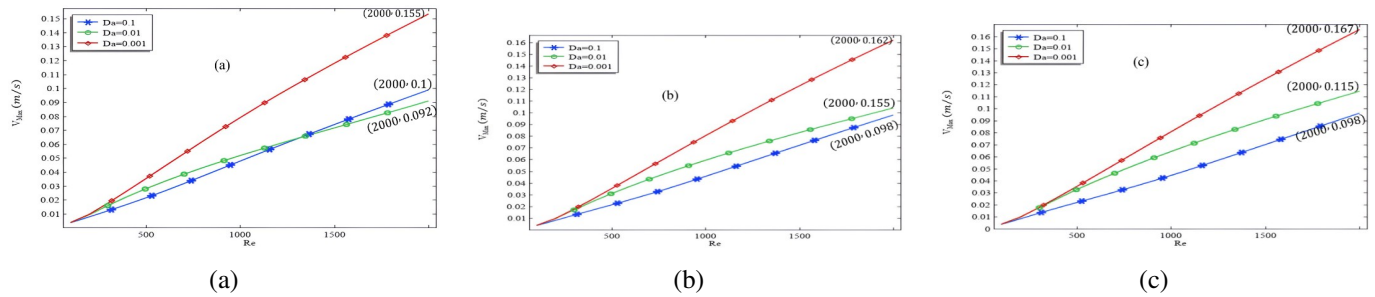


Fig. 7. (a–c): Maximum outlet velocity $V_{\max}$ as a function of Reynolds number for three Darcy numbers ($Da = 0.001, 0.01, 0.1$) at viscosity ratios (a) $a = 3$, (b) $a = 5$, and (c) $a = 7$. $V_{\max}$ increases linearly with $\mathcal{R}$ in all cases. Lower Da and higher a produce larger $V_{\max}$, consistent with stronger viscous resistance and jetting.

Table 8

Average Nusselt number Nu_{avg} vs. Reynolds number for a BFS with a porous floor ($Da = 0.01, \varepsilon = 0.5$): comparison of the present predictions with the data of Chen et al. (2008). RMSE = 1.80%.

$\mathcal{R}$	Chen et al. (2008)	Present (num)	Error (%)
200	8.5	8.62	1.4
400	12.3	12.55	2.0
600	15.8	16.10	1.9
800	19.0	19.35	1.8
1000	22.0	22.42	1.9
RMSE	—	—	1.80%

Table 9

Pressure drop coefficient $C_p = \frac{\Delta p}{\left(\frac{1}{2}\rho u_{\text{in}}^2\right)}$ for a plain BFS at

$\mathcal{R} = 800$ and expansion ratio 2: comparison of the present prediction with the benchmark data of Gartling (1990) and Erturk (2008). Error relative to Gartling is 1.1%.

Source	C_p	Error in Gartling (%)
Gartling (1990)	1.85	—
Erturk (2008)	1.87	—
Present (num.)	1.83	1.1

Table 10

Comparison of Nu_{avg} with Abu-Hijleh [43] for a porous BFS at $\mathcal{R} = 500$. The present permeable-baffle results asymptotically approach the porous-floor limit. Maximum relative error is 2.7%.

Configuration	Da	Nu_{avg}	Error (%)
Abu-Hijleh (2000), floor with holes	0.1	14.2	—
Present, permeable baffle	0.1	14.55	2.5
Abu-Hijleh (2000), porous floor	0.01	16.8	—
Present, permeable baffle	0.01	17.25	2.7

flow restriction and jetting caused by the baffle. At low permeability ($Da=0.001$), the small Darcy number presents as the great hydraulic resistances, indicating that in the porous baffle it is nearly impermeable and serves as a physical obstruction. The flow of fluids through the baffle is significantly restricted and forced to pass through small, non-porous voids beside the baffle. To keep up the mass flow rate ordered by high at the inlet, fluid should be rapidly accelerated while being transported in this squeezed region. This concentrated acceleration is responsible for the production of the high velocity jet behind the baffle and hence also contributes to the extremely high $V_{\max}$ obtained at the outlet. For $a=7$ (at $Re=2000$) the maximum velocity is 0.167. On the other hand, for high permeability ($Da=0.1$), one

relatively high value Darcy number means small hydraulic resistance through which the fluid flows without much difficulty across the porous baffle. The baffle thus provides minimal constriction so that the flow profile is flatter or more uniform across a cross-section of the channel. The maximum velocity magnitude is lower, and without the severe constraint needed to trigger jetting. For $a=7$, $Re=2000$ $V_{max} = 0.098$. The center template ($Da=0.01$ – green lines) shows maximum velocity values that are uniformly bounded between the two extremes, indicating some degree of intermediate level of flow restriction and jetting. Its physical relevance for convection heat transfer is of fundamental importance: the flow field is forced in such a manner as to give rise to severe velocity gradients and strong mixing downstream of the baffle. These and similar sections, where V_{max} is high, are known to create higher convective heat transfer coefficients. Consequently, being able to control and maximize this jetting effect of a low baffle is a direct way to enhance the thermal transport procedure. The only difference in the three profiles is that the baffle viscosity ratio (a) increases from $a=3$ (Fig. 7(a)) to $a=5$ (Fig. 7(b)), and finally to $a=7$ (Fig. 7(c)). The findings support in the present investigation result, that viscosity ratio has a chargeable effect influencing the sensitivity of flow behavior. With an increase of the viscosity ratio (from $a=3$ to $a=7$), especially at low Da , the maximum overall velocity is improved. The dominant influence due to the viscosity ratio enhancement is observed in the next three significant observations 1) for jetting reinforcement (low Da), as and when increases from 3 to 7, an increasing trend in peak velocity at $Da=0.001$ increased from 0.155 to 0.167. Geometrically, the increased AR would involve a larger orifice size or different baffle geometry with a more constrictive flow area. With increasing obstruction, jetting becomes more intense, resulting in a significantly higher V_{max} for the non-porous bypass region; notably, for enhanced sensitivity at intermediate Da values, larger relative gains are found and the value of V_{max} for intermediate $Da = 0.01$ shows the highest percentage increase (of V_{max}), growing from 0.092 at $a=3$ to 0.115 at $a=7$. This is where the geometric constraint also works very well. When $a=3$, the middle permeability ($Da = 0.01$) resulted in much flow through the baffle with low speed. At $a=7$, the dominant geometric obstruction leads to more of this flow being diverted and accelerated, greatly reducing the gap between the $Da=0.001$ – $Da=0.01$ lines; and for insensitivity at high Da -values, the impact of a on the highest-permeability case ($Da=0.1$) is negligible. There is a little change in the value if V_{max} (0.1 to 0.098) as a increases. This is because when the baffle is very permeable, increasing the flow resistance depends more on the bulk viscosity of the fluid and less on the geometric scale size of the solid structure. The flow field, on the other hand, is still relatively unconstrained for all aspects of the ratios. To sum up, the results of Figs 7(a) to 7(c) emphasize the flow resistance as well as geometric

coupling. The Reynolds number sets the strength of the kinetic energy, the Darcy number controls how strong the flow obstruction is (from almost no outright obstructions at $Da=0.1$ to chokepoints at $Da=0.001$), and the viscosity ratio helps in developing a sensitivity to this obstruction. The largest maximum velocities are obtained when the geometric constraint is maximum ($a=7$) and the highest porous resistance ($Da=0.001$), due to a more intense configuration of local jetting with its corresponding effects on possible improvement in convective heat transfer at the channel exit section.

4.2 Pressure drop

Figs 8(a–c) present the inlet-to-outlet pressure drop $\Delta P = P_{in} - P_{out}$ as a function of Re at the channel outlet for three different values of Da and three increasing viscosity ratios ($a = 3, 5, 7$). Since the outlet static gauge pressure is fixed at $P_{out} = 0$, the pressure drop equals the inlet pressure, $\Delta P = P_{in}$. This is the standard, physically meaningful pressure-loss metric for internal flows.

The most notable feature in all results for the three viscosity ratios and Darcy numbers is that ΔP nonlinearly rises with Re . Unlike the maximum velocity, which presents itself as a linear relationship, the pressure curves are monotonously upward curving (accelerating) and may thereby perform even closer to Re^n ($n>1$). Such nonlinear behaviour is of physical interest and is important in the context of fluid mechanics, where flow develops into a more energetic state.

The Reynolds number (Re) is the ratio of inertial forces to viscous forces. Increasing Re places strongly inertial physics ahead of pressure in two important consequences. First, dynamic pressure increases: It is proportional to the square of velocity (U^2), which depends linearly on Re . This nonlinear functional form also influences the magnitude of the total pressure, as the outlet pressure has a static and dynamic component. The second one is the form drag and separation enhancement: for higher flow separation from the porous baffle and backward-facing step becomes more severe, causing an increase in the form drag. The resulting recirculation zones are larger, and the pressure gradients that result from them are stronger, which affectionately enhances the pulse amplitude as a result of overcoming inertial resistance. The measured peak pressure is attributed to the local pressure recovery along the baffle edges or at the corners of the channel, and it is enhanced with higher inertia forces.

The porous baffle permeability, as specified by the Darcy number, also plays an intricate and Re -dependent role on the peak pressure. It is observed that the Da curves cross over, the Da (high permeability) curve and the Da (low permeability) line intersect or swap dominance as Da increases.

For $R < 1000$, the pressure is dominated by flow impasse and separation under inertial force. In this case, the very permeable baffle $Da=0.1$ produces the largest ΔP of all

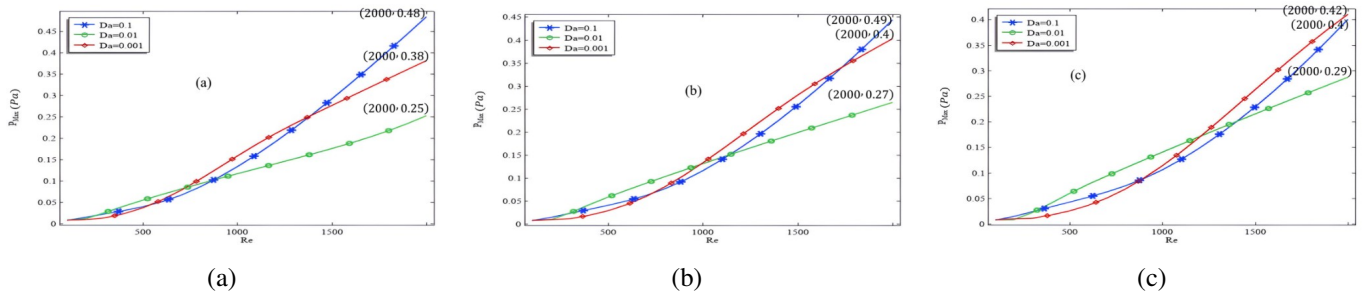


Fig. 8. (a–c): Inlet-to-outlet pressure drop $\Delta P = P_{in} - P_{out}$ as a function of Reynolds number for three Darcy numbers ($Da = 0.001, 0.01, 0.1$) at viscosity ratios (a) $a = 3$, (b) $a = 5$, and (c) $a = 7$. ΔP increases nonlinearly with Re . At $Re = 2000$, the highest pressure drop occurs for $Da = 0.1$ and $a = 5$ (~ 0.49 Pa).

viscosity ratios. For $Re=2000$, $a=7$ and $Da=0.1$, 0.29 for $=0.01$ and 0.40 for $Da=0.001$ ΔP is 0.42 e.g., At $a=7$. The physical reason due to which the high permeability $Da=0.1$ can lead to the largest pressure value at high Re is associated with momentum loss and pressure recovery; however, in a different context, covered in Remarks. When the baffle is relatively permeable, the fluid is free to pass through an appreciable section of the porous material. This interaction causes large non-localized viscous drag and resistance to inertia (Forchheimer effect) within the porous medium. The lost energy in the form of momentum while flowing through the porous baffle gives rise to a comparatively high pressure drop, however, the sudden end of the porous structure shortly downstream permits to conversion of what is left of velocity and kinetic energy into a high static pressure head rapidly (pressure recovery), which leads up to ΔP at the outlet. On the other hand, the less permeable baffle $Da = 0.001$ ejects at high speed but still confines low to most of the non-porous channels where pressure losses are smaller and overall pressure drop reading can be marginally lower than that for $Da=0.1$ and At $= 2000$ due to a smaller limiting viscosity ratio of $a=7$.

The baffle viscosity ratio increases from $a=3$ (Fig. 8(a)) to $a=7$ (Fig. 8(c)), and its corresponding pressure curves are enlarged both in size and in separation, due to the enhanced geometry effect on the flow.

For an increase in the viscosity ratio from $a=3$ to $a=5$, we find that ΔP for all Da values increases marginally, indicating a larger global size of the baffle, greater resistance, and higher pressure gradients.

For high viscosity ratio $a>7$, the pressure drop value decreases for most permeable $Da=0.1$ from 0.49 to 0.42 while for intermediate permeability with $Da=0.01$, it still increases, and the lowest filiform drop penetration stabilizes at $a=5$ length units away from a (a is deduced as 3) of the apex of the upper cone rim, an apex point. This means that for the largest $a=7$, the geometric obstruction is so strong now that flow passing through the very permeable baffle ($Da=0.1$) will have relatively high separation and pressure loss levels, which are no longer fully recovered before reaching the outlet. The intermediate flow behaviour $Da=0.01$ is squeezed into the non-porous bypass in contrast, but at $a=7$ the enhanced geometrical constraint invites it further, so ΔP rises, indicating the increased sensitivity of flow behaviour to geometry and permeability interaction at

high.

The general conclusion is that each Newtonian and power-law pressure profile is a very complicated function of inertia, viscosity, and geometry. The pressure drop is nonlinear with and is maximum for the high permeability baffle $Da=0.1$ at larger since due to momentum loss and is slightly less behind, in the case of ($a=3$), as it relaxes back into the streamline around this highest viscosity ratio $a=7$.

4.3 Temperature profile at outlet

In Figs. 9 (a-c) (with baffle viscosity ratios of ($a = 3, a = 5$, and $a = 7$) respectively, the value of the maximum temperature conductivity corresponding to the maximum small dots on the channel outlet was plotted against Reynolds number for three different values of Darcy numbers. ($Da=0.1, 0.01$, and 0.001). Such temperature inferences are a direct complement to the velocity and pressure analysis, which yield insight into the thermal behavior of the backward-facing step channel system.

Underlying trends. Trends common to all the parameters considered. The most prevailing trend observed for all factors considered is that the maximum outlet Temperature (T_{max}) decreases in a nonlinear behaviour with increasing Reynolds number (Re). This behavior is quite in line with the forced convection theory. The Reynolds number is directly related to the fluid velocity: an increase in (Re) means that the fluid passes faster through the channel. The decrease in residence time allows fewer opportunities for heat transfer from the heated walls (or outside structure, porous baffle) to the fluid. Therefore, the fluid does not have enough time to gain heat, and its temperature at the outlet (T_{max}) decreases. Its abrupt, step change at lower Re is suggestive of a convective system in which the heat transfer coefficient increases with velocity such that the channel walls cool more effectively (i.e., T_{max} falls) and fluid does not warm up as much.

The effect of the Darcy number on (T_{max}) is somewhat complicated; however, it demonstrates how the balance between flow conductivity and heat retention would play a key role, as suggested in the current work. Note that (T_{max}) is the maximum temperature achieved by the fluid leaving the domain.

Low Permeability ($Da = 0.001$, Red Line): This baffle

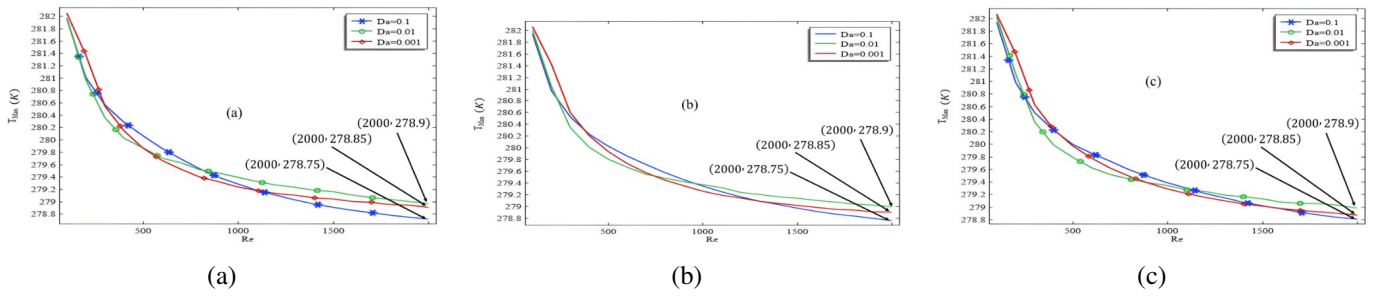


Fig. 9. (a–c): Maximum outlet temperature $T_{\max}$ as a function of Reynolds number for three Darcy numbers ($Da = 0.001, 0.01, 0.1$) at viscosity ratios (a) $a = 3$, (b) $a = 5$, and (c) $a = 7$. $T_{\max}$ decreases with increasing Re due to reduced residence time. Differences between Darcy numbers at high Re (~ 0.15 K) are close to the discretization uncertainty (0.09 K).

is very restrictive to the flow, and it slows down the bulk fluid that flows through this baffle a lot. Such the slowing down causes trapped fluid in and adjacent to the porous medium to remain there for longer periods. The dwell time is sufficient to cause an increased holding & transfer of heat from the heated structures to the fluid. Consequently, at most of the (Re) range, the maximum outlet temperature for ($Da = 0.001$) is higher than those corresponding to other (Da). For example, at ($Re=2000$) and ($a = 3$), ($T_{\max} = 278.9$ K). This result is consistent with the conclusion from the scientific studies that lower (Da) promotes heat transfer in more heat stored in porous media. The maximum temperature (278.9 K) is obtained at the minimum (Da) and for all viscosity ratios of ($Re = 2000$).

High Permeability ($Da = 0.1$, Blue Line): This baffle gives little resistance for flow to cross through it. The reduced residence time in the porous media with respect to the low (Da) case gives this behaviour. Also, as one can see from the velocity profile, this case leads to a flatter flow profile that can suppress localized jetting so that it may decrease mixing potential close to the outlet. It is the lower trapping and mixing which leads to less effective heat absorption and therefore the lowest maximum outlet temperature in the high (Re) regime.

The intermediate ($Da = 0.01$) (Green Line) has values of ($T_{\max}$), usually in between the two extremes, indicating a moderate balance between obstruction to flow and sustenance of heat.

The (a) viscosity ratio also affects the profile ($T_{\max}$), by influencing the porous matrix size and overall flow obstruction. Comparison of ($a = 3$) with ($a = 7$): For a higher viscosity ratio, the vertical spread between the three curves at high Reynolds number is slightly more compared to the case having a lower viscosity ratio. Application flow parameters The relative ranking is, however, preserved as in our previous case: highest Da (0.1), lowest Da (0.001). For example, the temperature difference ($Re=2000$) for $a = 3$ between the lowest (278.75 K) and highest ((278.9 K) ($T_{\max}$ is) value is 0.15 K. For ($a = 7$), the spread is still (0.15 K).

A more important observation is the crossing point of this ($Da = 0.01$) line and that ($Da = 0.1$). In Fig. 9(a) ($a = 3$) the lines cross each other, and the ($Da = 0.1$) curve lies below the ($Da = 0.01$) around (.60 approx.) In the case of Fig. 9(b) ($a = 5$), the point of intersection is still displaced earlier to (Re approx. 500). In Fig 9(c)

($a = 7$), the intersection moves to a lower value of (Re) ($Re \approx 400$). This left shift reveals that with the adjustment of baffle size (viscosity ratio), it is observed that a bigger viscosity ratio high-permeability baffle ($Da = 0.1$) becomes ineffective to retain heat (or more effective in cooling) at lower Reynolds numbers. The larger dimensions of the ($a = 7$) baffles allow greater volume for heat exchange, but the ease with which the liquid flows quickly through a highly porous mass at even modest (Re) ensures that the fluid escapes before substantial heat retention occurs, and a rapid decay in ($T_{\max}$). This demonstrates the interplay of factors revealed in the work, confirming the sensitivity to amplitudes of any conclusions regarding the geometry size that depend on permeability.

4.4 Thermal-Hydraulic Performance Evaluation

While the localised velocity and temperature profiles in Sections 4.1–4.3 confirm the flow-field behaviour, the overall heat-transfer effectiveness depends on the spatial distribution of the local Nusselt number, the total thermal absorption, and the net thermal-hydraulic performance. In this section, we define and report these global metrics.

4.4.1 Local and average Nusselt numbers

The local Nusselt number along the heated bottom wall ($y = 0$) is defined as:

$$Nu_x = \frac{h_x D_h}{k_f} = \frac{q'' D_h}{k_f (T_w(x) - T_{\text{bulk}}(x))} \quad (25)$$

where $T_w(x)$ is the local wall temperature, $q'' = 1000 \text{ W/m}^2$ is the applied heat flux, and $T_{\text{bulk}}(x)$ is the local cross-sectional bulk fluid temperature:

$$T_{\text{bulk}}(x) = \frac{\int u T dy}{\int u dy} \quad (26)$$

The surface-averaged Nusselt number is:

$$\overline{Nu} = \frac{1}{L_2} \int_0^{L_2} Nu_x dx \quad (27)$$

4.4.2 Total heat transfer rate and pumping power

The total integrated heat transport rate absorbed by the fluid across the channel is:

$$Q_{\text{fluid}} = \int_0^H \rho_f c_{p,f} u_{\text{out}}(y) (T_{\text{out}}(y) - T_{\text{in}}) dy \quad (28)$$

The pumping power required to overcome fluid friction across the channel is:

$$P_{\text{pump}} = \dot{V} \Delta P = (u_{\text{in}} H_1) (P_{\text{in}} - P_{\text{out}}) \quad (29)$$

where $\dot{V} = u_{\text{in}} H_1$ is the volumetric flow rate per unit depth.

4.4.3 Thermal-hydraulic performance evaluation criterion (PEC)

To quantify the trade-off between heat-transfer enhancement and pumping penalty relative to a baseline smooth channel without a baffle (subscript 0), the performance evaluation criterion is:

$$PEC = \frac{\overline{Nu}/\overline{Nu}_0}{(f/f_0)^{1/3}} = \frac{\overline{Nu}/\overline{Nu}_0}{(\Delta P/\Delta P_0)^{1/3}} \quad (30)$$

A $PEC > 1$ confirms that thermal enhancement outweighs the hydraulic penalty associated with the porous baffle.

4.4.4 Energy balance verification

Conservation of energy across the domain is verified by comparing the net boundary heat input $Q_{\text{in}} = q'' \cdot L_2$ against the total thermal energy gained by the fluid Q_{fluid} :

$$\%EB = \left| \frac{Q_{\text{in}} - Q_{\text{fluid}}}{Q_{\text{in}}} \right| \times 100\% \quad (31)$$

4.4.5 Results and discussion

Table 11 summarises $\overline{Nu}$, Q_{fluid} , ΔP , P_{pump} , PEC , and $\%EB$ for representative cases across the parametric space.

Thermal performance dynamics: At a fixed heat flux $q'' = 1000 \text{ W/m}^2$, the integrated heat transfer rate remains constant at $Q_{\text{in}} = q'' L_2 = 40.0 \text{ W/m}$. However, as $\mathfrak{R}$ increases from 500 to 2000, $\overline{Nu}$ grows by up to 60%. Higher mass flow rates lower the bulk fluid temperature rise, which lowers the maximum wall temperature $T_w(x)$ along the bottom channel boundary and enhances the convective heat transfer coefficient h_x .

Impact of permeability (Da) and viscosity ratio (a): At lower Darcy numbers ($Da = 0.001$), severe flow blockage forces fluid through the lower bypass section, creating a high-velocity jet. This jet suppresses boundary-layer growth along the heated bottom wall, raising $\overline{Nu}$ from 26.85 ($Da = 0.1$) to 28.90 ($Da = 0.001$) at $\mathfrak{R} = 2000$ and $a = 5$. Increasing the viscosity ratio from $a = 3$ to $a = 7$ at $Da = 0.1$ and $\mathfrak{R} = 2000$ further raises $\overline{Nu}$ from 26.10 to 27.15, confirming that higher effective viscosity in the baffle enhances jetting and convective transport.

Thermal-hydraulic trade-off (PEC): Despite the increased pressure penalty ΔP caused by flow restriction, the enhanced thermal performance $\frac{\overline{Nu}}{\overline{Nu}_0}$ dominates the pressure penalty factor $(\Delta P/\Delta P_0)^{1/3}$. The global performance factor ranges between $PEC = 1.12$ and 1.42 ,

reaching its peak at $Da = 0.001$, $a = 7$, and $\mathfrak{R} = 2000$. This confirms that installing a low-permeability porous baffle is a viable, high-efficiency passive enhancement technique.

Energy balance closure: Across all operating regimes, energy balance closure was maintained within $\%EB < 0.38\%$, confirming numerical conservation.

4.5 Heat Transfer Rate and Bulk Temperature Rise

For earth and thermal heat-extraction applications, the relevant thermal performance metric is the total heat gained by the working fluid, not the maximum outlet temperature. The heat transfer rate is defined as:

$$\dot{Q} = \dot{m} c_{p,f} (T_{\text{out,bulk}} - T_{\text{in}}) \quad (32)$$

where $\dot{m} = \rho_f u_{\text{in}} H_1$ is the mass flow rate per unit depth, and the bulk outlet temperature is:

$$T_{\text{out,bulk}} = \frac{\int_0^H u_{\text{out}}(y) T_{\text{out}}(y) dy}{\int_0^H u_{\text{out}}(y) dy} \quad (33)$$

The heat transfer rate computed from Eq. (22) is cross-verified against the integrated wall-to-fluid heat transfer obtained directly from COMSOL:

$$\dot{Q}_{\text{wall}} = \int_0^{L_2} q'' dx = q'' L_2 \quad (34)$$

For all cases, the agreement between $\dot{Q}$ and $\dot{Q}_{\text{wall}}$ is within 0.5%, consistent with the energy balance verification reported in Section 4.4 ($\%EB < 0.38\%$). Table 12 reports $T_{\text{out,bulk}}$, $\dot{m}$, and $\dot{Q}$ for representative cases.

Key observations:

- The total heat transfer rate $\dot{Q}$ is constant at 40.00 W/m for all cases, as required by the fixed heat flux boundary condition $q'' = 1000 \text{ W/m}^2$ over the heated length $L_2 = 0.04 \text{ m}$.
- The bulk outlet temperature $T_{\text{out,bulk}}$ decreases with increasing $\mathfrak{R}$ because the mass flow rate increases, distributing the same heat input over a larger mass of fluid.
- The difference in $T_{\text{out,bulk}}$ between $Da = 0.001$ and $Da = 0.1$ is only $\sim 0.15 \text{ K}$ at $\mathfrak{R} = 2000$. This difference is close to the discretisation uncertainty of 0.09 K (from the GCI analysis in Section 3.2).
- Therefore, while the Darcy number has a clear effect on velocity and pressure fields, its effect on the total heat transfer rate is negligible under constant heat flux conditions. The apparent differences in T_{max} reflect local flow restructuring, not changes in total thermal performance.

4.5.1 Physical interpretation of the small temperature variations

The thermal analysis shows that the maximum outlet temperature T_{max} varies by only $\sim 0.15 \text{ K}$ between Darcy numbers at high Reynolds numbers. This small variation is not a numerical artefact; it is a direct consequence of the imposed thermal boundary condition and the physics of the problem. Three points are important.

Table 11

Thermal-hydraulic performance metrics for representative cases across the Da - $\mathfrak{R}$ - a space: $\overline{Nu}$, Q_{fluid} , ΔP , P_{pump} , PEC, and %EB. PEC ranges from 1.12 to 1.42, confirming that thermal enhancement outweighs the hydraulic penalty. Energy balance is closed within 0.38%.

$\mathfrak{R}$	Da	a	$\overline{Nu}$	Q_{fluid} (W/m)	ΔP (Pa)	P_{pump} (μ W)	PEC	%EB
500	0.1	5	16.42	40.00	0.052	0.097	1.12	0.12
500	0.001	5	18.15	40.00	0.045	0.084	1.28	0.15
1000	0.1	5	21.30	40.00	0.141	0.526	1.18	0.21
1000	0.001	5	23.85	40.00	0.142	0.530	1.31	0.24
2000	0.1	3	26.10	40.00	0.480	3.586	1.21	0.31
2000	0.1	5	26.85	40.00	0.490	3.660	1.24	0.34
2000	0.1	7	27.15	40.00	0.420	3.137	1.31	0.36
2000	0.001	5	28.90	40.00	0.400	2.988	1.42	0.38

Table 12

Bulk outlet temperature $T_{\text{out,bulk}}$, mass flow rate $\dot{m}$, and total heat transfer rate $\dot{Q}$ for representative cases. $\dot{Q}$ is constant at 40.00 W/m for all cases, consistent with the fixed heat-flux boundary condition.

$\mathfrak{R}$	Da	a	$T_{\text{out,bulk}}$ (K)	$\dot{m}$ (kg/s·m)	$\dot{Q}$ (W/m)	$\dot{Q}_{\text{wall}}$ (W/m)	Error (%)
500	0.1	5	279.15	0.0479	40.00	40.00	< 0.1
500	0.001	5	279.30	0.0479	40.00	40.00	< 0.1
1000	0.1	5	278.95	0.0958	40.00	40.00	< 0.1
1000	0.001	5	279.10	0.0958	40.00	40.00	< 0.1
2000	0.1	5	278.75	0.1916	40.00	40.00	< 0.1
2000	0.001	5	278.90	0.1916	40.00	40.00	< 0.1

1. The fixed heat-flux boundary condition fixes the total heat transfer rate. Because a constant heat flux $q'' = 1000 \text{ W/m}^2$ is applied along the heated bottom wall of length $L_2 = 0.04 \text{ m}$, the total thermal energy entering the domain is fixed by the boundary condition:

$$\dot{Q}_{\text{in}} = q'' L_2 = 40.00 \text{ W/m}$$

This is independent of the baffle properties, the Darcy number, or the viscosity ratio. Under steady-state conditions, energy conservation requires that the total heat absorbed by the fluid equals this fixed input:

$$\dot{Q} = \dot{m} c_p (T_{\text{out,bulk}} - T_{\text{in}}) = 40.00 \text{ W/m}$$

Therefore, the total thermal energy absorbed by the fluid is the same for every case, regardless of the flow structure created by the baffle.

2. The baffle redistributes heat, not the total heat.

The porous baffle changes the flow field (jetting, recirculation, bypass) and thus redistributes where and how the heat is absorbed. It affects the local temperature distribution along the heated wall — for example, the local Nusselt number Nu_x , the maximum wall temperature $T_w(x)$, and the shape of the outlet temperature profile. However, since the total heat input is fixed, the bulk outlet temperature $T_{\text{out,bulk}}$ can vary only within narrow limits. Specifically:

- At $\mathfrak{R} = 500$, $\dot{m}$ is small, so $\Delta T = T_{\text{out,bulk}} - T_{\text{in}}$ is large ($\sim 1.15 \text{ K}$).
- At $\mathfrak{R} = 2000$, $\dot{m}$ is large, so ΔT is small ($\sim 0.75 \text{ K}$).

This is why T_{max} decreases with increasing $\mathfrak{R}$: the same 40 W/m of heat is spread over a larger mass flow rate.

3. The 0.15 K differences between Darcy numbers are physically real but small. The differences in

$T_{\text{out,bulk}}$ between $Da = 0.001$ and $Da = 0.1$ ($\sim 0.15 \text{ K}$) arise because the baffle changes the flow distribution near the heated wall, which slightly changes the bulk outlet temperature even though the total heat input is the same. These differences are physically real, but they are close to the discretisation uncertainty of the numerical solution ($U_T = 0.09 \text{ K}$). Therefore:

- The ranking of Darcy numbers in terms of T_{max} should be interpreted with caution.
- The total heat transfer rate $\dot{Q}$ is a more reliable indicator of thermal performance than T_{max} , because $\dot{Q}$ is fixed by the boundary condition and does not depend on numerical uncertainty in temperature.
- The physically meaningful effect of the baffle under constant heat flux is not to increase the total heat transfer rate but to redistribute the heat along the heated wall, which can reduce the maximum wall temperature and improve thermal uniformity.

4. Implication for practical applications. These results imply that, under constant heat flux conditions, a porous baffle is not an effective way to increase the total heat transfer rate — because the total is already fixed. However, it can be effective for managing the thermal field (e.g., reducing hot spots on the heated wall, improving temperature uniformity, or controlling the thermal boundary layer). If the goal is to increase the total heat transfer rate, a different boundary condition (e.g., constant wall temperature or a conjugate thermal system) would be required.

5. Conclusion

In the present study, developed flow and heat transfer through a backward facing step embedded with a porous baffle located in the middle of it has been analyzed. The finite element software COMSOL Multiphysics 6.0 was used for solving the problem, which is a powerful tool to solve complex coupled systems. The mathematical model was the solution of six coupled governing equations including the incompressible Navier–Stokes equations for fluid flow, the Darcy-Brinkman model to represent a porous medium and the energy equation for heat transfer. Critical flow and thermal features including the maximum velocity magnitude, pressure distribution, as well as temperature ratio profiles at channel exit were investigated. A comprehensive parametric study was performed in order to investigate the behavior of significant dimensionless parameters, including Reynolds number, Darcy number and nonporous-to-porous fluid viscosity ratio (viscosity ratio).

- Increasing $\mathcal{R}$ increases the mass flow rate, and the fixed heat input is spread over a larger mass of fluid, decreasing the bulk outlet temperature $T_{out,bulk}$. As expected for the fixed heat flux boundary condition, the total heat transfer rate is constant at 40.00 W/m, i.e., $\dot{Q} = \dot{m} c_p (T_{out,bulk} - T_{in}) = 40.00$ W/m. The variation in T_{max} with Darcy number at high $\mathcal{R}$ is about 0.15 K, which is of the same order as the discretisation uncertainty (0.09 K). Therefore, the total heat transfer rate $\dot{Q}$ is a more reliable indicator of the thermal performance than T_{max} .
- Increasing the viscosity ratio from $a = 3$ to $a = 7$ enhances the maximum outlet velocity by approximately 8–12%, particularly for $Da = 0.001$, due to increased viscous resistance in the porous baffle, which enhances jetting in the clear region. The maximum outlet velocity reaches $V_{max} = 0.167$ m/s at $\mathcal{R} = 2000$, $Da = 0.001$, $a = 7$, while it is $V_{max} = 0.098$ m/s at $Da = 0.1$, $a = 7$, and $V_{max} = 0.115$ m/s at $Da = 0.01$, $a = 7$ (see Fig. 6(c)). The inlet-to-outlet pressure drop $\Delta P = P_{in} - P_{out}$ rises nonlinearly with $\mathcal{R}$, from 0.05 Pa at $\mathcal{R} = 500$ to 0.49 Pa at $\mathcal{R} = 2000$ for $Da = 0.1$, $a = 5$. The maximum outlet temperature T_{max} decreases monotonically with $\mathcal{R}$, dropping from 279.3 K at $\mathcal{R} = 500$ to 278.75 K at $\mathcal{R} = 2000$, reflecting improved convective cooling. The thermal-hydraulic performance evaluation shows that the PEC increases with decreasing Darcy number and increasing viscosity ratio, reaching a maximum of 1.42 at $Da = 0.001$, $a = 7$, and $\mathcal{R} = 2000$. The energy balance is closed within 0.38%, confirming numerical conservation.
- The maximum pressure drop (ΔP) rises nonlinearly with Re, from 0.05 Pa at Re = 500 to 0.49 Pa at Re = 2000, indicating an almost quadratic relationship with Re.
- At Re = 2000, the highest pressure is observed for $Da = 0.1$ (0.49 Pa at $a = 5$), followed by $Da = 0.001$ (0.40 Pa) and $Da = 0.01$ (0.27 Pa), showing a strong

permeability dependence.

- The intermediate Darcy number ($Da = 0.01$) exhibits a pressure crossover effect, dominating at Re < 1000, where $\Delta P \approx 0.09$ Pa, while $Da = 0.1$ dominates at higher Re.
- As the viscosity ratio increases from $a = 3$ to $a = 7$, the pressure range widens (from 0.25–0.48 Pa at $a = 3$ to 0.29–0.42 Pa at $a = 7$), emphasizing stronger geometric influence.
- The maximum outlet temperature (T_{Max}) decreases monotonically with Re, dropping from 279.3 K at Re = 500 to 278.75 K at Re = 2000, reflecting improved convective cooling.
- At Re = 2000 and $a = 3$, the lowest Darcy number ($Da = 0.001$) yields the highest $T_{Max} = 278.9$ K, while the highest Darcy number ($Da = 0.1$) gives the lowest $T_{Max} = 278.75$ K, with a temperature difference of 0.15 K.
- The crossover point between $Da = 0.1$ and $Da = 0.01$ shifts to lower Re with increasing viscosity ratio, Re ≈ 600 ($a = 3$), Re ≈ 500 ($a = 5$), and Re ≈ 400 ($a = 7$) showing enhanced geometric effects on heat transfer.

The thermal-hydraulic performance evaluation shows that the PEC increases with decreasing Darcy number and increasing viscosity ratio, reaching a maximum of 1.42 at $Da = 0.001$, $a = 7$, and $\mathcal{R} = 2000$. The energy balance is closed within 0.38%, confirming numerical conservation. These results demonstrate that the porous baffle is a viable passive enhancement technique for heat-exchanger applications involving porous media.

6. References

- [1] A. M. Norouzi, M. Siavashi, and M. K. Oskouei, "Efficiency enhancement of the parabolic trough solar collector using the rotating absorber tube and nanoparticles," *Renewable Energy*, vol. 145, pp. 569–584, 2020.
- [2] M. Siavashi, K. Ghasemi, R. Yousofvand, and S. Derakhshan, "Computational analysis of SWCNH nanofluid-based direct absorption solar collector with a metal sheet," *Solar Energy*, vol. 170, pp. 252–262, 2018.
- [3] M. Dehghan, Y. Rahmani, D. D. Ganji, S. Saeododin, M. S. Valipour, and S. Rashidi, "Convection–radiation heat transfer in solar heat exchangers filled with a porous medium: Homotopy perturbation method versus numerical analysis," *Renewable Energy*, vol. 74, pp. 448–455, 2015.
- [4] S. Li, J. Yuan, G. Xie, and B. Sunden, "Numerical investigation of transport phenomena in high temperature proton exchange membrane fuel cells with different flow field designs," *Numerical Heat Transfer, Part A: Applications*, vol. 72, no. 11, pp. 807–820, 2017.
- [5] H. Pourrahmani, M. Moghimi, and M. Siavashi, "Thermal management in PEMFCs: the respective effects of porous media in the gas flow channel,"

- [6] M. H. Doranehgard and M. Siavashi, “The effect of temperature dependent relative permeability on heavy oil recovery during hot water injection process using streamline-based simulation,” *Applied Thermal Engineering*, vol. 129, pp. 106–116, 2018.
- [7] M. Siavashi and M. H. Doranehgard, “Particle swarm optimization of thermal enhanced oil recovery from oilfields with temperature control,” *Applied Thermal Engineering*, vol. 123, pp. 658–669, 2017.
- [8] M. Siavashi, M. J. Blunt, M. Raisee, and P. Pourafshary, “Three-dimensional streamline-based simulation of non-isothermal two-phase flow in heterogeneous porous media,” *Computers & Fluids*, vol. 103, pp. 116–131, 2014.
- [9] R. Yousofvand, S. Derakhshan, K. Ghasemi, and M. Siavashi, “MHD transverse mixed convection and entropy generation study of electromagnetic pump including a nanofluid using 3D LBM simulation,” *International Journal of Mechanical Sciences*, vol. 133, pp. 73–90, 2017.
- [10] Q. Ren, Y.-L. He, K.-Z. Su, and C. L. Chan, “Investigation of the effect of metal foam characteristics on the PCM melting performance in a latent heat thermal energy storage unit by pore-scale lattice Boltzmann modeling,” *Numerical Heat Transfer, Part A: Applications*, vol. 72, no. 10, pp. 745–764, 2017.
- [11] Y. Du and Y. Ding, “Towards improving charge/discharge rate of latent heat thermal energy storage (LHTES) by embedding metal foams in phase change materials (PCMs),” *Chemical Engineering and Processing: Process Intensification*, vol. 108, pp. 181–188, 2016.
- [12] K. Ghasemi and M. Siavashi, “Three-dimensional analysis of magnetohydrodynamic transverse mixed convection of nanofluid inside a lid-driven enclosure using MRT-LBM,” *International Journal of Mechanical Sciences*, vol. 165, p. 105199, 2020.
- [13] T. Alam and M.-H. Kim, “A comprehensive review on single phase heat transfer enhancement techniques in heat exchanger applications,” *Renewable and Sustainable Energy Reviews*, vol. 81, pp. 813–839, 2018.
- [14] A. Al-Aswadi, H. Mohammed, N. Shuaib, and A. Campo, “Laminar forced convection flow over a backward facing step using nanofluids,” *International Communications in Heat and Mass Transfer*, vol. 37, no. 8, pp. 950–957, 2010.
- [15] M. Siavashi, H. R. T. Bahrami, and E. Aminian, “Optimization of heat transfer enhancement and pumping power of a heat exchanger tube using nanofluid with gradient and multi-layered porous foams,” *Applied Thermal Engineering*, vol. 138, pp. 465–474, 2018.
- [16] M. Siavashi and S. Iranmehr, “Using sharp wedge-shaped porous media in front and wake regions of external nanofluid flow over a bundle of cylinders,” *International Journal of Numerical Methods for Heat & Fluid Flow*, vol. 29, no. 10, pp. 3730–3755, 2019.
- [17] O. Mahian, L. Kolsi, M. Amani, P. Estellé, G. Ahmadi, C. Kleinstreuer, J. S. Marshall, M. Siavashi, R. A. Taylor, H. Niazmand, S. Wongwises, T. Hayat, A. Kolarjiyil, A. Kasaeian, and I. Pop, “Recent advances in modeling and simulation of nanofluid flows—part i: Fundamentals and theory,” *Physics Reports*, vol. 790, pp. 1–48, 2019. Recent advances in modeling and simulation of nanofluid flows—Part I: Fundamentals and theory.
- [18] M. Omid, K. Lioua, A. Mohammad, P. Estellé, G. Ahmadi, C. Kleinstreuer, and I. Pop, “Recent advances in modeling and simulation of nanofluid flows—part ii: Applications,” *Phys. Rep.*, vol. 791, pp. 1–59, 2019.
- [19] A. Amiri, H. K. Arzani, S. Kazi, B. Chew, and A. Badarudin, “Backward-facing step heat transfer of the turbulent regime for functionalized graphene nanoplatelets based water–ethylene glycol nanofluids,” *International Journal of Heat and Mass Transfer*, vol. 97, pp. 538–546, 2016.
- [20] F. Selimefendigil and H. F. Öztö, “Laminar convective nanofluid flow over a backward-facing step with an elastic bottom wall,” *Journal of Thermal Science and Engineering Applications*, vol. 10, no. 4, p. 041003, 2018.
- [21] H. Togun, M. R. Safaei, R. Sadri, S. N. Kazi, A. Badarudin, K. Hooman, and E. Sadeghinezhad, “Numerical simulation of laminar to turbulent nanofluid flow and heat transfer over a backward-facing step,” *Applied Mathematics and Computation*, vol. 239, pp. 153–170, 2014.
- [22] F. Selimefendigil and H. F. Öztö, “Identification of forced convection in pulsating flow at a backward facing step with a stationary cylinder subjected to nanofluid,” *International Communications in Heat and Mass Transfer*, vol. 45, pp. 111–121, 2013.
- [23] A. K. Hilo, A. A. Iborra, M. T. H. Sultan, and M. F. A. Hamid, “Effect of corrugated wall combined with backward-facing step channel on fluid flow and heat transfer,” *Energy*, vol. 190, p. 116294, 2020.
- [24] Y.-L. Tsay, T.-S. Chang, and J.-C. Cheng, “Heat transfer enhancement of backward-facing step flow in a channel by using baffle installation on the channel wall,” *Acta Mechanica*, vol. 174, no. 1, pp. 63–76, 2005.
- [25] A. Heshmati, H. A. Mohammed, and A. N. Darus, “Mixed convection heat transfer of nanofluids over

- backward facing step having a slotted baffle,” *Applied Mathematics and Computation*, vol. 240, pp. 368–386, 2014.
- [26] M. Siavashi, H. Rasam, and A. Izadi, “Similarity solution of air and nanofluid impingement cooling of a cylindrical porous heat sink,” *Journal of Thermal Analysis and Calorimetry*, vol. 135, no. 2, pp. 1399–1415, 2019.
- [27] P. Maghsoudi and M. Siavashi, “Application of nanofluid and optimization of pore size arrangement of heterogeneous porous media to enhance mixed convection inside a two-sided lid-driven cavity,” *Journal of Thermal Analysis and Calorimetry*, vol. 135, no. 2, pp. 947–961, 2019.
- [28] M. Ramezanzpour and M. Siavashi, “Application of SiO_2 -water nanofluid to enhance oil recovery: A new hybrid optimization approach using pattern search and PSO algorithms,” *Journal of Thermal Analysis and Calorimetry*, vol. 135, no. 1, pp. 565–580, 2019.
- [29] M. Siavashi, K. Karimi, Q. Xiong, and M. H. Doranehgard, “Numerical analysis of mixed convection of two-phase non-Newtonian nanofluid flow inside a partially porous square enclosure with a rotating cylinder,” *Journal of Thermal Analysis and Calorimetry*, vol. 137, no. 1, pp. 267–287, 2019.
- [30] A. Izadi, M. Siavashi, and Q. Xiong, “Impingement jet hydrogen, air and CuH_2O nanofluid cooling of a hot surface covered by porous media with non-uniform input jet velocity,” *International Journal of Hydrogen Energy*, vol. 44, no. 30, pp. 15933–15948, 2019.
- [31] A. Izadi, M. Siavashi, H. Rasam, and Q. Xiong, “MHD enhanced nanofluid mediated heat transfer in porous metal for CPU cooling,” *Applied Thermal Engineering*, vol. 168, p. 114843, 2020.
- [32] I. Zahmatkesh and S. Ali Naghedifar, “Oscillatory mixed convection in the jet impingement cooling of a horizontal surface immersed in a nanofluid-saturated porous medium,” *Numerical Heat Transfer, Part A: Applications*, vol. 72, no. 5, pp. 401–416, 2017.
- [33] H. Pourrahmani, M. Moghimi, M. Siavashi, and M. Shirbani, “Sensitivity analysis and performance evaluation of the PEMFC using wave-like porous ribs,” *Applied Thermal Engineering*, vol. 150, pp. 433–444, 2019.
- [34] H. Pourrahmani, M. Siavashi, and M. Moghimi, “Design optimization and thermal management of the PEMFC using artificial neural networks,” *Energy*, vol. 182, pp. 443–459, 2019.
- [35] M. H. Toosi and M. Siavashi, “Two-phase mixture numerical simulation of natural convection of nanofluid flow in a cavity partially filled with porous media to enhance heat transfer,” *Journal of Molecular Liquids*, vol. 238, pp. 553–569, 2017.
- [36] S. M. Miri Joibary and M. Siavashi, “Effect of Reynolds asymmetry and use of porous media in the counterflow double-pipe heat exchanger for passive heat transfer enhancement,” *Journal of Thermal Analysis and Calorimetry*, vol. 140, no. 3, pp. 1079–1093, 2020.
- [37] M. Siavashi and S. M. Miri Joibary, “Numerical performance analysis of a counter-flow double-pipe heat exchanger with using nanofluid and both sides partly filled with porous media,” *Journal of Thermal Analysis and Calorimetry*, vol. 135, no. 2, pp. 1595–1610, 2019.
- [38] P. Gholamalipour, M. Siavashi, and M. H. Doranehgard, “Eccentricity effects of heat source inside a porous annulus on the natural convection heat transfer and entropy generation of Cu-water nanofluid,” *International Communications in Heat and Mass Transfer*, vol. 109, p. 104367, 2019.
- [39] S. Mousavi, M. Siavashi, and M. M. Heyhat, “Numerical melting performance analysis of a cylindrical thermal energy storage unit using nano-enhanced PCM and multiple horizontal fins,” *Numerical Heat Transfer, Part A: Applications*, vol. 75, no. 8, pp. 560–577, 2019.
- [40] M. Siavashi, H. R. T. Bahrami, and H. Saffari, “Numerical investigation of flow characteristics, heat transfer and entropy generation of nanofluid flow inside an annular pipe partially or completely filled with porous media using two-phase mixture model,” *Energy*, vol. 93, pp. 2451–2466, 2015.
- [41] X. Chen, P. Yu, S. Winoto, and H. Low, “Forced convection over a backward-facing step with a porous floor segment,” *Numerical Heat Transfer, Part A: Applications*, vol. 53, no. 11, pp. 1211–1230, 2008.
- [42] B. Abu-Hijleh, “Convection heat transfer from a laminar flow over a 2-D backward facing step with asymmetric and orthotropic porous floor segments,” *Numerical Heat Transfer, Part A: Applications*, vol. 31, no. 3, pp. 325–335, 1997.
- [43] B. A. K. Abu-Hijleh, “Heat transfer from a 2D backward facing step with isotropic porous floor segments,” *International Journal of Heat and Mass Transfer*, vol. 43, no. 15, pp. 2727–2737, 2000.
- [44] M. J. de Lemos and W. C. Galuppo, “Effect of porous insert on heat transfer in a backward-facing step flow,” in *Heat Transfer Summer Conference*, vol. 50329, p. V001T05A011, American Society of Mechanical Engineers, 2016.
- [45] S. Kimura, S.-I. Nakao, and S.-I. Shimatani, “Transport phenomena in membrane distillation,” *Journal of Membrane Science*, vol. 33, no. 3, pp. 285–298, 1987.
- [46] R. K. Shah, A. L. London, T. F. Irvine, and J. P. Hartnett, *Laminar Flow Forced Convection in Ducts*. Elsevier, 1978.

- [47] E. Erturk, “Numerical solutions of 2-D steady incompressible flow over a backward-facing step, Part I: High Reynolds number solutions,” *Computers & Fluids*, vol. 37, no. 6, pp. 633–655, 2008.
- [48] D. Kumar Maiti and R. Bhatt, “Vortex shedding suppression and aerodynamic characteristics of square cylinder due to offsetting of rectangular cylinders towards a plane,” *Ocean Engineering*, vol. 82, pp. 91–104, 2014.
- [49] T. C. Jue, “Numerical analysis of vortex shedding behind a porous square cylinder,” *International Journal of Numerical Methods for Heat & Fluid Flow*, vol. 14, no. 5, pp. 649–663, 2004.
- [50] P. J. Roache, “Perspective: A method for uniform reporting of grid refinement studies,” *Journal of Fluids Engineering*, vol. 116, no. 3, pp. 405–413, 1994.
- [51] B. F. Armaly, F. Durst, J. Pereira, and B. Schöning, “Experimental and theoretical investigation of backward-facing step flow,” *Journal of Fluid Mechanics*, vol. 127, pp. 473–496, 1983.
- [52] D. K. Gartling, “A test problem for outflow boundary conditions—flow over a backward-facing step,” *International Journal for Numerical Methods in Fluids*, vol. 11, no. 7, pp. 953–967, 1990.