



Investigation of Bluff Body Shape Variation on Enhancing Heat Transfer Performance of Backward-Facing Step Flow

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Abstract: The control of flow separation phenomenon is a challenge that has attracted much attention from researchers in the context of heat and mass transfer. This phenomenon negatively affects heat transfer performance in thermal management applications. Flow control devices play a crucial role in minimizing the effects of flow separation. One of the fundamental geometries that supports understanding in flow separation control is the backward-facing step. Therefore, this study aims to investigate the utilization of bluff body shape variations, including cube, cylinder, and diamond shapes, as passive flow control devices on heat transfer performance in backward-facing step flow. The present study used a Computational Fluid Dynamics solver, followed by a variation of the Reynolds number, $50 \leq Re \leq 400$. Computational results show that the bluff body significantly reduces the primary recirculation zone and compresses the thermal boundary layer, strengthening the temperature gradient and improving the heat transfer rate. The cube variation demonstrates the optimal thermal performance, exhibiting an augmentation in the average Nusselt number of up to 28.15% at $Re = 400$, resulting the highest overall Performance Evaluation Criterion.

Keywords: backward-facing; cube; cylinder; diamond; heat-transfer



SDGs Problem Solving

Research and problem-solving in this study align with SDGs Chapter #9 by applying passive flow control to optimize thermal management systems. This approach improves heat transfer performance without requiring external energy sources, advancing technological innovation, and supporting the development of sustainable and energy-efficient industrial infrastructure.

Conflict of Interest Statement: The author(s) declares that the research was conducted in the absence of any commercial or financial relationship that could be construed as a potential conflict of interest.

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1. INTRODUCTION

Within the broader context of heat and mass transfer studies, the control of the flow separation phenomenon has garnered significant attention from researchers. One of the theoretical geometry configurations frequently employed to elucidate this phenomenon is the Backward-facing step (BFS) (Hu, Wang and Fu, 2015). This geometry configuration has been widely used in various industry heat transfer cases, such as combustion, gas turbine engines, cooling systems, aircraft, and buildings (Chen *et al.*, 2018). The BFS generally encompasses two fundamental phenomena: flow separation and reattachment point (Armaly *et al.*, 1983). In the specific context of thermal management applications, studies have indicated that flow separation can adversely affect the heat transfer performance of a system (Velazquez, Arias and Mendez, 2008). However, the following phenomenon, reattachment point, has a beneficial effect on improving the heat transfer process. Therefore, this mechanism is frequently employed as a local-based thermal management mechanism (Selimefendigil and Öztop, 2016).

The effect of flow separation on the heat transfer process has been a subject of considerable interest among researchers in recent decades. Research has used adiabatic circular cylinders as passive flow control devices in BFS geometry (Kumar and Dhiman, 2012). The findings indicated that implementing the circular cylinder at position $y_c = 1.5$ yielded the optimal local thermal performance, exhibiting a maximum Nusselt number (Nu) enhancement of up to 155%. However, the device's performance does not demonstrate maximum results because the average Nu output is still low. Furthermore, additional enhancements have been identified that integrate corrugated walls on the downstream side of the BFS. It has been determined that there is a substantial increase in Nu, reaching up to 62%, at a Reynolds number (Re) of 5000 (Hilo *et al.*, 2020). The development of research on BFS flow engineering in thermal management continues until studies utilize engineering on the bottom wall slope to control the growth of the flow separation zone (Bayraktar, 2020). The results demonstrated that increasing the slope of the bottom wall resulted in an increase in the local Nu behind the step. In addition, an alternative study proposed an alternative configuration by adjusting the tilt angle on the global BFS geometry (Issakhov *et al.*, 2021). The investigation's findings demonstrate that the effect of buoyancy force has a substantial impact on the performance of heat transfer and enhances the efficiency factor. Furthermore, the advent of research endeavors that employed active flow control, which utilized multiple jets modified in the BFS geometry, marked a significant turning point in the field (Coskun, Cadirci and Gunes, 2020). The computational results indicate that the temporal interactions formed due to the installation of the device have a substantial impact on the fluid's mixing process. This condition leads to an enhancement in Nu, increasing up to 78.5% in thermal performance.

A thorough review of the extant literature reveals a discernible progression in developing research to enhance control of flow separation phenomena. However, the development of thermal performance is hindered by several factors, including the distribution of heat transfer processes that tend to be concentrated in specific areas, the incorporation of ineffective configurations, and the high power consumption of control devices (Selimefendigil and Öztop, 2015). Consequently, further research is necessary to optimize the thermal performance of the BFS flow. While the implementation of a bluff body as a passive flow control device is recognized as an effective strategy for enhancing thermal performance without external energy sources, current literature

predominantly focuses on conventional circular cylinder configurations. Angular structures like cubes and diamonds generate entirely different wake dynamics and flow separation behaviors compared to standard circular cylinders. The specific influence of these geometric variations on the primary recirculation zone and thermal boundary layer compression has not been comprehensively explored. Therefore, this research aims to examine variations in the geometry and shape of the bluff body as a passive flow control device. The primary goals are to improve the thermal management performance of the BFS flow and to minimize the flow separation phenomenon. Therefore, it is anticipated that this research will meaningfully contribute to advancing knowledge in thermal management engineering in BFS flows.

2. METHODOLOGY

2.1. Backward-Facing Step (BFS)

In examining the flow separation phenomenon, the backward-facing step (BFS) has emerged as a prevalent geometry form within the research community. In the case of internal flow, the main feature of BFS is the sudden expansion of the cross-sectional area ([Julian, Anggara and Wahyuni, 2023](#)). The sudden expansion of the pipe's cross-sectional area is attributed to incorporating an additional step, resulting in a significant increase in the cross-sectional area. The expansion ratio in the BFS geometry is set in Equation 1. The case of internal flow through BFS includes several common phenomena, such as flow separation, recirculation zone, reattachment point, and pressure gradient ([Julian, Anggara and Wahyuni, 2024](#)). It has been demonstrated that these features play an important role in various engineering applications.

$$\frac{H}{h} = 1 + \frac{S}{h} \quad (1)$$

where H , h , and S are the total height of the downstream channel (mm), the height of the upstream channel, and the height of the steps, respectively.

2.2. Geometries

The present study focuses on engineering the flow separation phenomenon in BFS flow by applying bluff body geometry variations as a passive flow control method. Specifically, the variations in the bluff body geometry are divided into three different shapes: cube, cylinder, and diamond. As illustrated in [Figure 1](#), the bluff body has been installed in the downstream area of the BFS, with its placement location designated at coordinates $X_c = 0.6$ and $Y_c = 1.5$. The findings inform the configuration and dimensions of the flow control devices of research conducted by Ankit Kumar and Amit K. Dhiman, which aimed to optimize performance outcomes ([Kumar and Dhiman, 2012](#)). The dimensions and configuration of the devices in BFS are described in [Table 1](#).

2.3. Geometries

The present study employs a computational approach based on Computational Fluid Dynamics (CFD) as a numerical model to describe the BFS flow phenomenon. This approach was selected as an efficient and cost-effective research method compared to experimental ([Julian, Anggara and Wahyuni, 2023](#)). The flow dynamics are governed by the Reynolds Averaged-Navier-Stokes (RANS) equations. RANS comprises two primary governing equations: continuity, stipulated in Equation 2,

and momentum, delineated in Equation 3 (Julian, Anggara and Wahyuni, 2023). The energy equation is also incorporated into Equation 4 to regulate the heat transfer process during fluid interaction with solid (Julian et al., 2024).

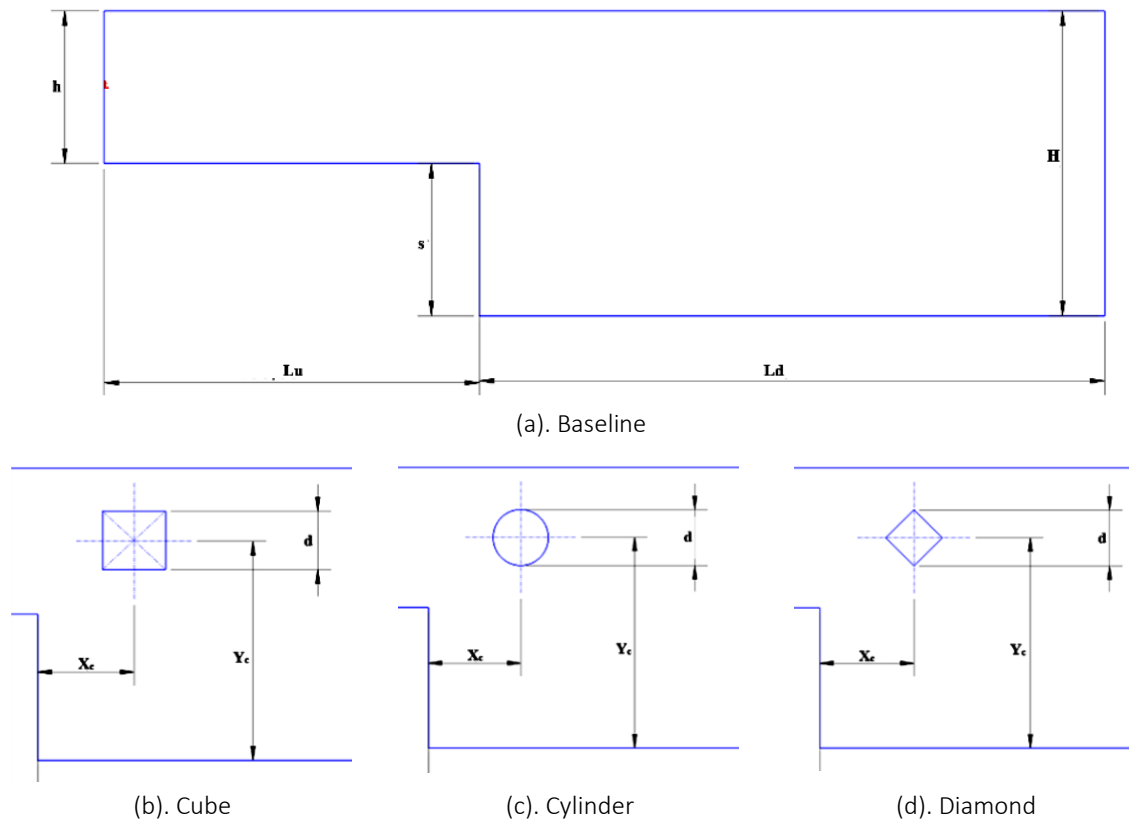


Figure 1. Geometries configuration

Table 1. Detail geometries configuration

Parameters	Dimensions (mm)
h	10
H	20
s	10
Lu	100
Ld	300
X _c	6
Y _c	15
d	4

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i) = 0 \quad (2)$$

$$\frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_i} (\rho u_i u_j) = \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right) \right] + \frac{\partial}{\partial x_i} (-\rho \overline{u_i' u_j'}) \quad (3)$$

$$\frac{\partial(\rho u_i T)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[\left(\frac{\mu}{Pr} + \frac{\mu_t}{\sigma_T} \right) \frac{\partial T}{\partial x_i} \right] \quad (4)$$

where ρ , u , P , μ , Pr , and T are fluid density, velocity, pressure, viscosity, Prandtl number, and temperature, respectively.

2.4. Mesh and Boundary Condition

The computational domain is discretized using a structured quadrilateral mesh to ensure numerical accuracy without generating excessively small elements. Boundary conditions are established to define the internal flow parameters of the backward facing step. The upstream inlet velocity varies based on the tested Reynolds numbers ranging from 50 to 400 according to Equation 5 (Julian *et al.*, 2024). A parabolic velocity profile is applied using Equation 6 to simulate a fully developed flow (Julian, Anggara and Wahyuni, 2023). The channel boundaries and bluff body surfaces are specified as adiabatic no slip walls at an ambient temperature of 299 K. The downstream lower wall is assigned a constant hot temperature of 300 K. The non dimensional temperature formulation is outlined in Equation 7. Heat transfer performance is evaluated using the Nusselt number expressed in Equation 8. The downstream extremity is set as a zero-pressure outlet. Furthermore, the Performance Evaluation Criterion (PEC) formulated in Equation 9 is utilized to assess the overall heat transfer efficiency. Figure 2 delineates the complete mesh details and boundary conditions.

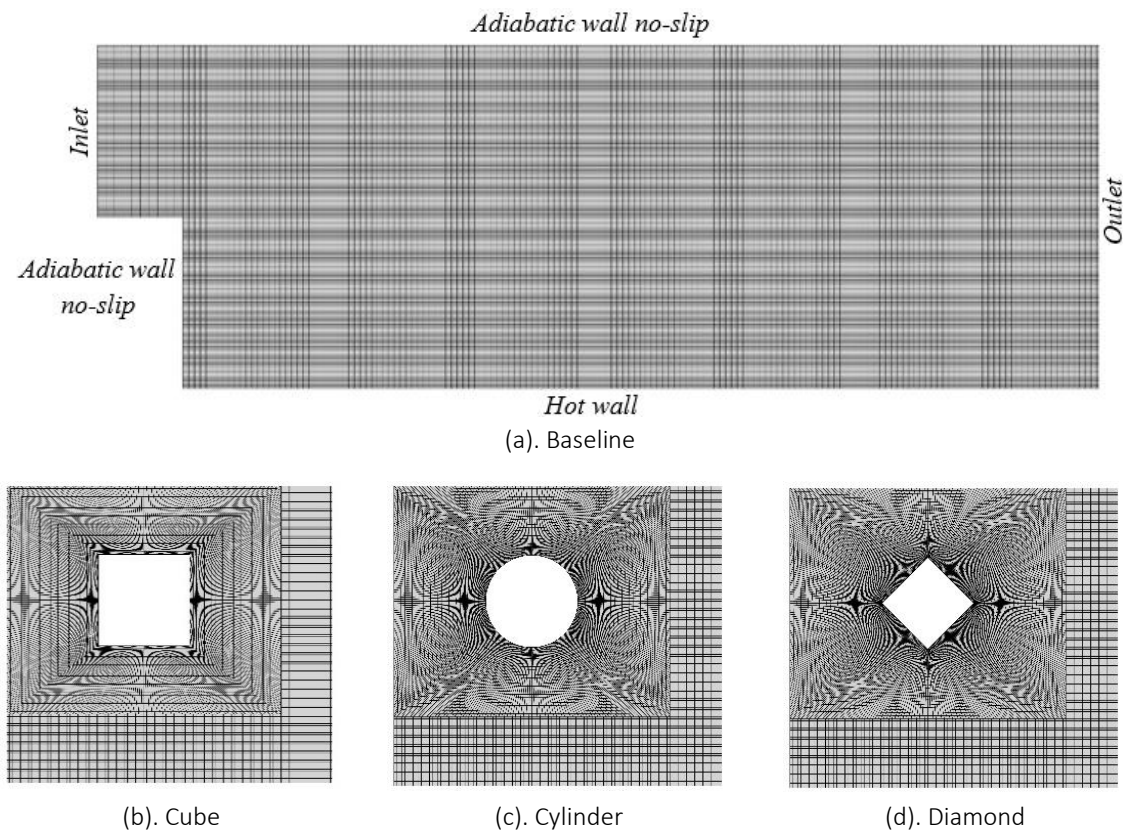


Figure 2. Detail mesh and boundary conditions

$$Re = \frac{\rho D U_b}{\mu} \quad (5)$$

$$U_{\max} = \left(\frac{3}{2}\right) U_b \quad (6)$$

$$\theta = \frac{(T - T_0)}{(T_w - T_0)} \quad (7)$$

$$Nu = \frac{h_s \times h}{K} \quad (8)$$

$$PEC = \frac{\left(\frac{Nu_{avg, modification}}{Nu_{avg, baseline}} \right)}{\left(\frac{f_{modification}}{f_{baseline}} \right)^{\frac{1}{3}}} \quad (9)$$

$$f = \Delta P \frac{D_h}{L} \frac{2}{\rho U_b^2} \quad (10)$$

Where Re , U_b , D_h , L , $U_{\max}$, θ , T_{in} , T_w , Nu , h , K , and f are Reynolds number, average velocity, hydraulic diameter, total length, maximum velocity, non-dimensional temperature, inlet fluid temperature, hot wall temperature, Nusselt number, convective heat transfer coefficient, thermal conductivity, and friction factor, respectively.

To execute the numerical simulations, a commercial Computational Fluid Dynamics solver was utilized. The numerical solution was achieved by implementing a coupled algorithm to resolve the pressure and velocity fields simultaneously. For the spatial discretization, the gradient was evaluated using the least squares cell based method (McQueen et al., 2022). Furthermore, a second order scheme was applied for the pressure spatial discretization alongside a second order upwind scheme for the momentum equations to ensure high accuracy in capturing the flow separation phenomena.

2.5. Mesh and Boundary Condition

A mesh independence test was conducted to evaluate the sensitivity of the computational model and ensure numerical accuracy. The spatial domain was discretized into three grid resolutions comprising fine (200,000 elements), medium (100,000 elements), and coarse (50,000 elements) meshes. The grid refinement ratio was established using Equation 11. The Grid Convergence Index methodology proposed by Roache was employed to quantify the numerical uncertainty based on velocity values extracted at coordinates $x = 0.12$ and $y = -0.005$ (Roache, 1994). The error deviations between the fine and medium grids alongside the medium and coarse grids were calculated using Equations 13 and 14. Equation 16 was utilized to verify that the numerical solutions successfully reached the asymptotic range of convergence. Following this verification, the true asymptotic value was determined using Equation 17.

The complete independence test results are summarized in Table 2. Based on these quantitative findings, the fine mesh was selected as the optimal domain model for all subsequent computational simulations.

Table 2. Mesh independence test

Mesh	Fine	Medium	Coarse
Velocity	0.11472000	0.11293200	0.11004300
$\bar{p}$		0.692223467	
r		2	
GCI_{fine}		3.214%	
GCI_{coarse}		5.1930%	
$f_{rh} = 0$		0.117623673	
$\frac{GCI_{coarse}}{GCI_{fine} r^{\bar{p}}}$		1	
Error	2.46861%	3.98871%	6.44485%

3. RESULTS AND DISCUSSION

3.1. Validation

Before the analysis stage, the numerical calculation data undergoes a validation process to ensure its level of data actualization against the experimental data. This research compares with the secondary data in the experimental studies of (Kumar & Dhiman, 2012) and (Armaly et al., 1983). As demonstrated in Figure 3, the Nusselt number (Nu) and reattachment point (XR) distribution data along the hot wall behind the step is compared. The numerical approach successfully yields graphical trends that are highly consistent with the findings from both experimental studies. Although minor deviations are present, the discrepancy remains completely negligible and confirms the overall validity of the computational model.

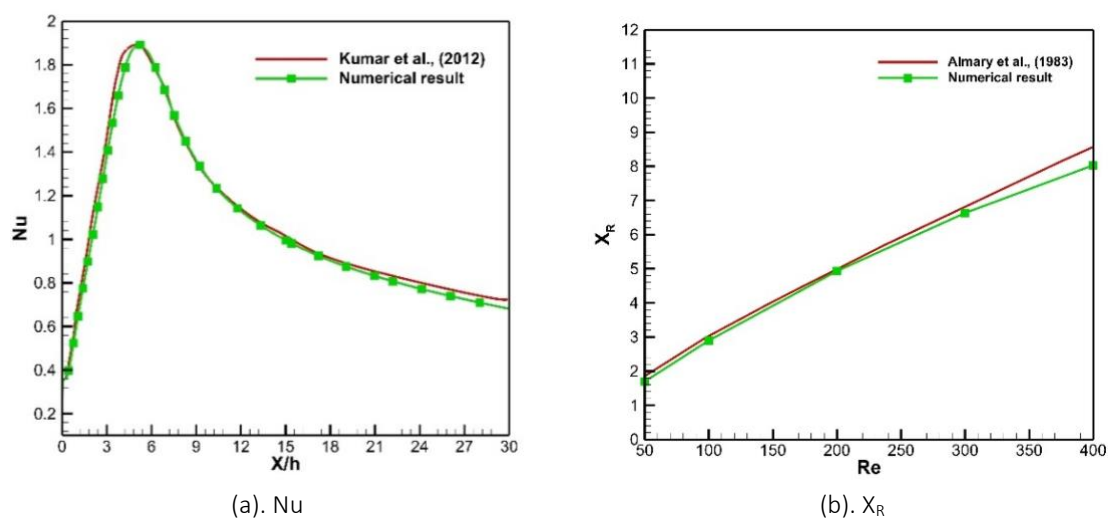


Figure 3. Validation

3.2. Analysis

Internal flow over a backward facing step is fundamentally characterized by flow separation and a subsequent reattachment point (XR). As depicted in Figure 4, the XR location expanded proportionally with the Reynolds number. This parameter critically indicates the overall size of the primary recirculation zone. For the baseline configuration, this linear expansion reached a maximum XR of 8.033 at $Re = 400$, which aligns precisely with the established experimental findings of Armaly *et al.* (Armaly *et al.*, 1983). The integration of passive flow control devices significantly suppressed this downstream shift. The cylinder and diamond geometries demonstrated comparable effectiveness by successfully restricting the XR location to 1.84 and 1.77 at $Re = 400$. Ultimately, the cube shaped bluff body exerted the most substantial suppression by minimizing the XR location to 1.52 under the identical flow conditions.

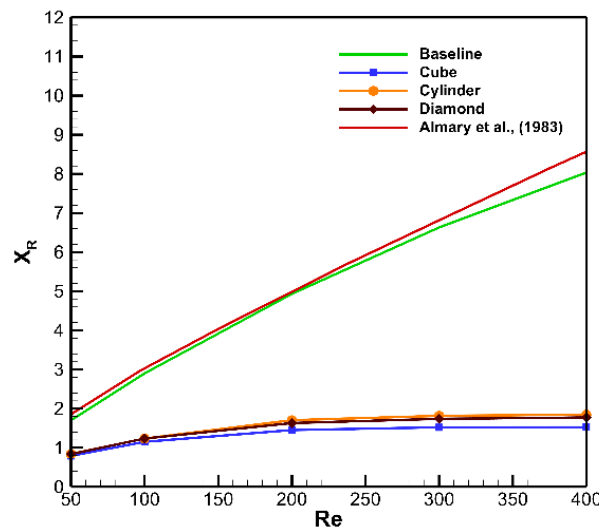


Figure 4. Location of reattachment point (X_R) against Reynolds number (Re)

From the perspective of heat transfer, the phenomenon of flow separation, occurring in the context of flow past the BFS, has been shown to exert a detrimental influence on the system's thermal performance. As demonstrated in Figure 5, the Nu distribution on the hot wall is presented at every Re condition. The suboptimal thermal process, attributable to the flow separation phenomenon, is typified by a decline in the Nu value within the region immediately adjacent to the step. The localized drop in Nu indicates the fluid's constrained capacity to absorb heat. This condition deteriorates in proportion to the increase in Re . However, a reattachment point has been shown to counterbalance the thermal performance compensation within the flow separation area. The peak value of Nu at the local position indicates an increase in the heat transfer process. As the parameter Re increases, the peak value of Nu obtained becomes more significant. The maximum value of Nu is attained when $Re = 400$, yielding a result of 2.43. Furthermore, a local shift in the peak value of Nu is observed, concomitant with the shift of XR.

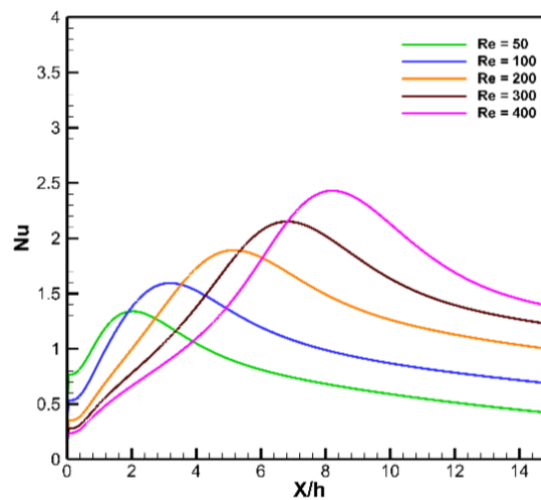


Figure 5. Distribution of Nusselt number (Nu) at each Reynolds number (Re)

The ability of the passive flow control devices to restrict the displacement of XR substantially improved the thermal performance of the backward facing step flow. As illustrated in Figure 6(a), all three bluff body variations enhanced the Nu distribution on the heated wall immediately behind the step. At lower Reynolds numbers between 50 and 100, the peak Nu values increased notably with no significant performance discrepancy among the different shapes. However, the performance gap between the geometries widened significantly as the Reynolds number increased. The cube geometry achieved the highest peak Nu of 7.08, followed by the diamond at 5.92 and the cylinder at 5.82. While the bluff bodies strategically enhanced heat transfer within the flow separation region, they concurrently caused a noticeable decline in the downstream \$Nu\$ distribution. The relationship between the local maximum Nu location and the normalized XR position was examined further in Figure 6(b). The data revealed that the reattachment point did not invariably correspond to the exact location of maximum heat transfer. Instead, the correlation between XR and the peak Nu location depended heavily on the flow regime. At higher Reynolds numbers, the spatial shift pattern of the XR location relative to the maximum Nu became almost comparable.

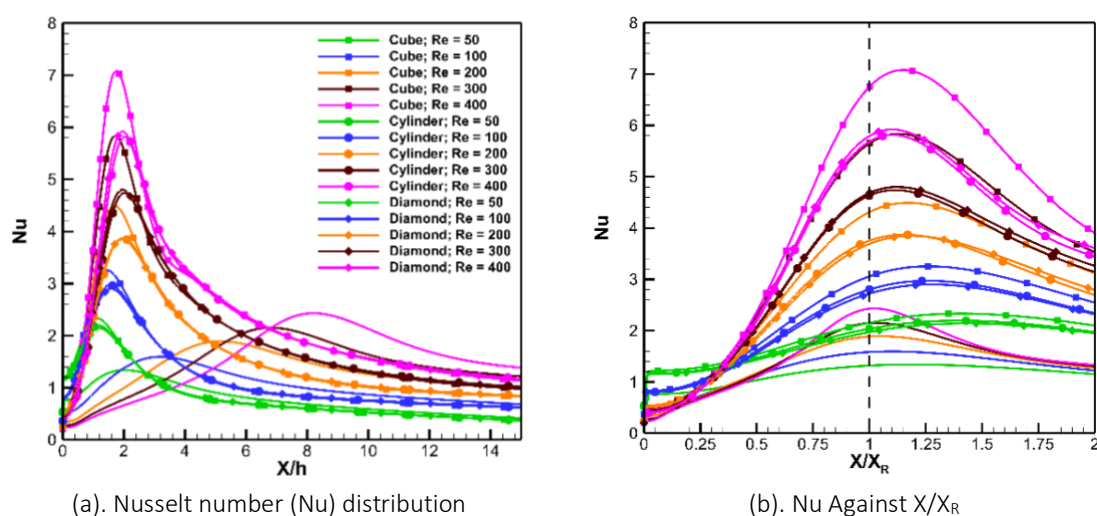


Figure 6. Effect of passive flow control on Nu distribution

The performance of the heat transfer process is demonstrated in Figure 7. The mean Nu distribution is exhibited in relation to increasing Re in each passive flow control form variation. Re plays a crucial role in the heat transfer capability of the BFS flow system. The resulting thermal performance is directly proportional to the increase in Re. Furthermore, the implementation of passive flow control, incorporating three variations of bluff body geometry, has been demonstrated to enhance the thermal performance of BFS. These findings directly align with the study conducted by Kumar & Dhiman (Kumar and Dhiman, 2012). The quantitative analysis of heat transfer performance data for the BFS flow is presented in Table 3. A comparison of the device's performance with cylinder and diamond shapes reveals no significant differences. The mean maximum Nu distribution is 1.607 and 1.623, respectively. The maximum heat transfer performance is achieved with the cube shape variation, where the average Nu distribution reaches 1.704, with a 28.15% performance improvement.

Table 3. Heat transfer performance of BFS flow with passive flow control installed

Baseline		Cube			Cylinder			Cube		
Re	Nu _{Avg}	Re	Nu _{Avg}	%	Re	Nu _{Avg}	%	Re	Nu _{Avg}	%
50	0.514	50	0.621	20.80%	50	0.604	17.55%	50	0.604	17.43%
100	0.748	100	0.899	20.12%	100	0.869	16.19%	100	0.867	15.84%
200	1.050	200	1.226	16.75%	200	1.173	11.72%	200	1.176	12.01%
300	1.212	300	1.483	22.36%	300	1.404	15.89%	300	1.414	16.72%
400	1.329	400	1.704	28.15%	400	1.607	20.89%	400	1.623	22.10%

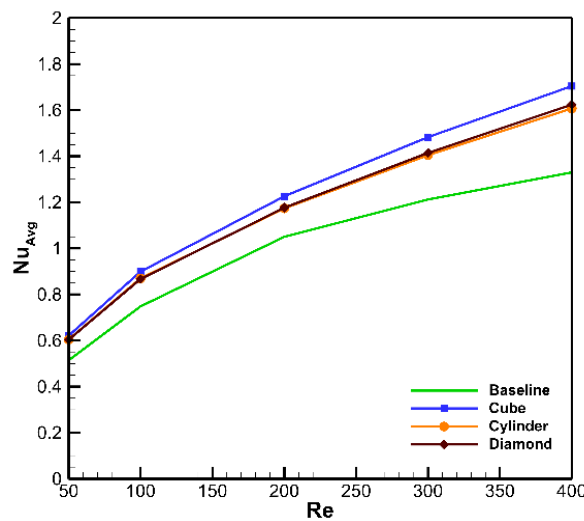


Figure 7. Average distribution of Nusselt number (Nu)

Subsequent analyses visualized the backward facing step flow phenomena following the implementation of the three passive flow control variations. As illustrated in Figure 8, Figure 9 and Figure 10, streamline, streamwise velocity, and non-dimensional temperature contours meticulously examined the flow characteristics. This visual data investigated the fluid structures formed against the velocity gradient and the fundamental heat transfer processes at a Reynolds number of 400. The contours clearly demonstrated that flow separation acted as the foundational element driving the entire flow dynamics. Flow separation manifested immediately after the fluid passed the step and induced a primary recirculation zone.

This recirculation zone persisted until the fluid ultimately reattached to the downstream lower wall. A low velocity vortex defined this region due to a localized vacuum area. Thermally, the fluid interaction with the heated wall generated a thermal boundary layer that expanded along the characteristic length. The primary recirculation zone induced a substantially thick thermal boundary layer immediately near the step.

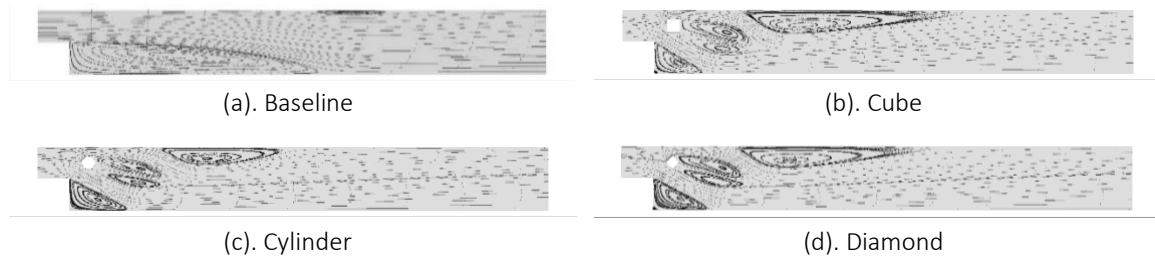


Figure 8. Streamline contour at $Re = 400$

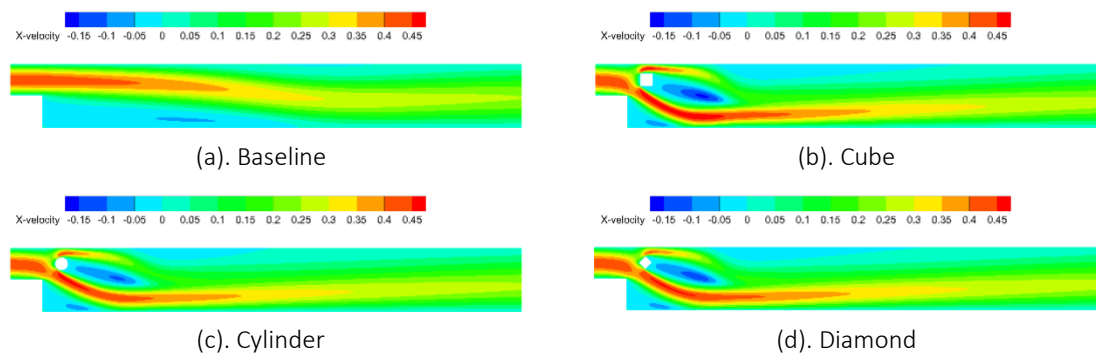


Figure 9. X-velocity contour at $Re = 400$

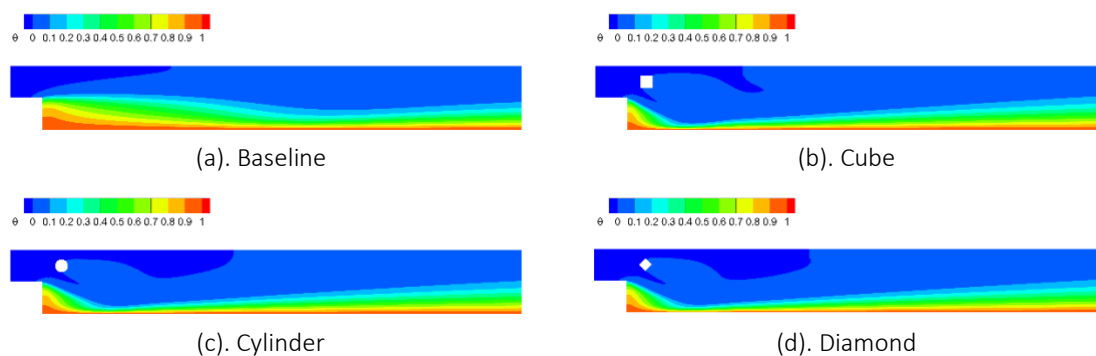
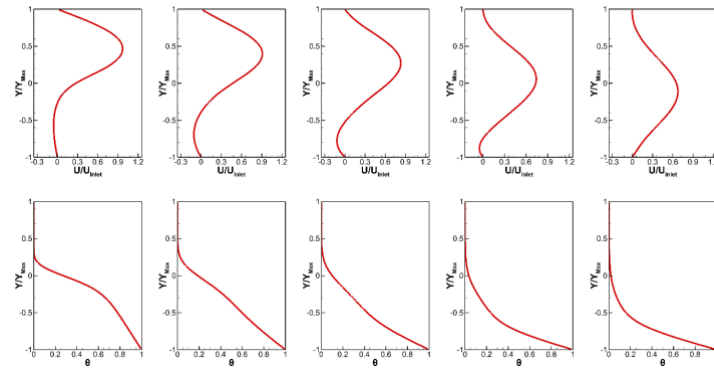
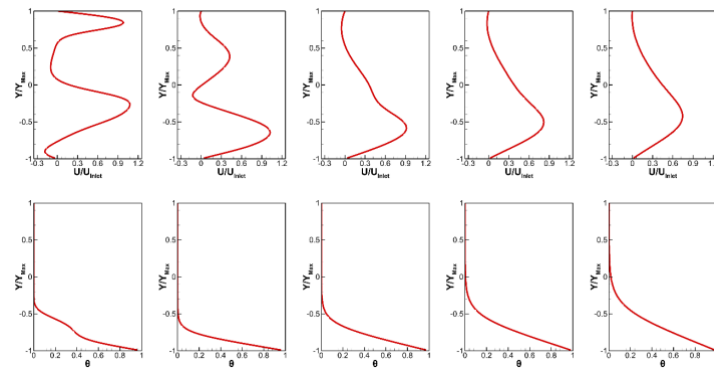


Figure 10. Non-dimensional temperature contour at $Re = 400$

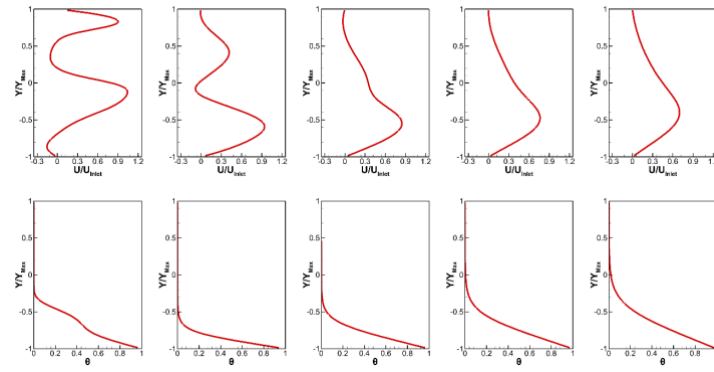
However, this boundary layer was effectively suppressed near the reattachment point. This suppression served as a critical factor in enhancing the temperature gradient and facilitating an optimal heat transfer rate. The installation of the bluff bodies introduced high flow complexity. Advanced phenomena emerged including a wake zone immediately behind the obstacle and a secondary recirculation zone on the upper wall, which significantly diminished the primary recirculation zone. The narrow gap between the step corner and the bluff body augmented the fluid velocity and altered the momentum dictating the fluid return to the lower wall.



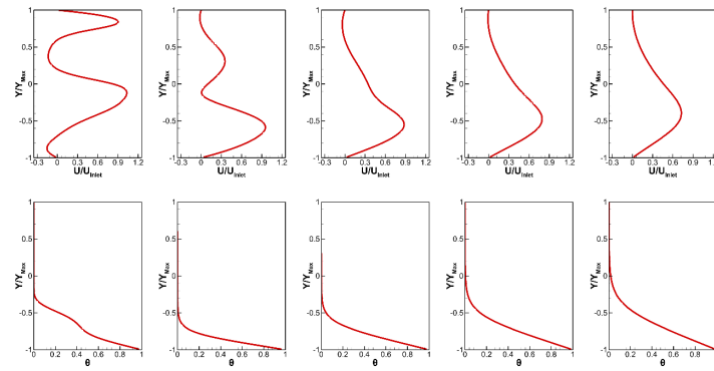
(a). Baseline



(b). Cube



(c). Cylinder



(d). Diamond

Figure 11. X-velocity and non-dimensional temperature profile at $Re = 400$

The cube configuration proved most effective in reducing the primary recirculation zone and thinning the thermal boundary layer to maximize the temperature gradient. Nevertheless, the cube shape concurrently formed the most substantial wake zone and secondary recirculation flow. This adverse condition generated significantly greater pressure drops compared to the cylinder and diamond geometries.

Specifically, the normalized velocity and non-dimensional temperature profiles are displayed in [Figure 11](#). In the flow past the BFS, the primary recirculation zone formed due to flow separation is characterized by a velocity profile with a negative vector direction. In the context of laminar flow, the magnitude of the fluid's velocity with a negative vector initially exhibits a minor tendency to fluctuate until it ultimately diminishes and ceases to be redeveloped. The implementation of body bluffs in BFS exerts a substantial influence on the velocity profile that is established. The primary recirculation zone is minimized, and two zones with high-velocity gradients are evident due to the formation of narrow gaps. Nevertheless, the zone characterized by a velocity profile with a negative vector direction is subdivided into two distinct regions: the wake zone formation and the secondary recirculation zone. Examining the temperature profile reveals a substantial increase in fluid temperature above the hot wall in the primary recirculation zone. This condition indicates a thick thermal boundary layer, which is known to result in suboptimal heat transfer performance. As the local position XR is approached, the temperature gradient increases, suppressing the thermal boundary layer that forms. Implementing passive flow control has been demonstrated to suppress the thermal boundary layer substantially. The resultant interaction between the fluid and the solid forms a high-temperature gradient, particularly in the region adjacent to the step. This phenomenon engenders an optimal heat transfer process, leading to the development of the thermal boundary layer and the subsequent regeneration of fluid flow.

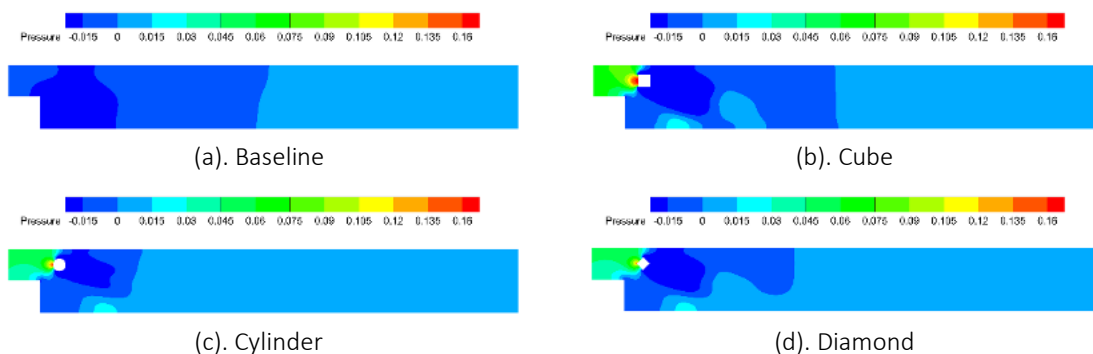


Figure 12. Pressure contour at $Re = 400$

Furthermore, [Figure 12](#) illustrates the pressure contour employed to assess the pressure distribution in the BFS flow. It has been observed that the primary recirculation zone formed in the baseline configuration results in a substantial pressure drop behind the step. The pressure drops resulting from a vacuum zone within the separated flow has been demonstrated to have a detrimental effect on efficiency. The implementation of passive flow control, employing a range of bluff body shapes, has markedly reduced the pressure drop behind the step. However, utilizing a bluff body during the process gives rise to an additional pressure drop in the device's immediate vicinity. The bluff body, characterized by its diamond shape, was found to have the smallest wake zone, thereby maximizing fluid flow efficiency. Despite its superior performance in terms of heat transfer, the bluff body with

a cube shape is accompanied by a resulting wake zone that becomes dominant, thereby generating a larger low-pressure zone. This condition is attributable to the larger frontal area, which engenders substantial stagnation pressure on the frontal area of the bluff body.

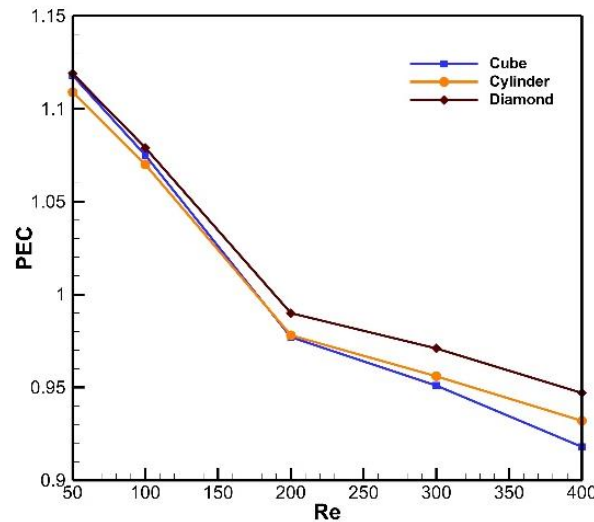


Figure 13. Performance Evaluation Criterion (PEC)

To quantify the balance between heat transfer enhancement and the penalty caused by pressure drop, the system is evaluated using the Performance Evaluation Criterion (PEC). Figure 13 visualizes the PEC values for the three variations of bluff body geometry across different Reynolds number levels. At lower flow regimes with Re values of 50 and 100, all three device configurations generate PEC values exceeding 1. This condition indicates that the benefits gained from thermal augmentation are significantly greater than the compensation for additional pumping power losses. However, the PEC value demonstrates a consistent downward trend as Re increases for all shape variations. When Re reaches the range of 200 to 400, the effect of pressure drop losses becomes much more dominant and causes the PEC value to fall below 1. It is essential to highlight that although the cube shape previously proved to yield the highest Nu, the diamond shape provides the most superior thermohydraulic efficiency due to lower pressure drop. The diamond configuration consistently maintains the highest PEC value throughout the testing range. This fact confirms that the diamond geometric shape offers the most optimal compromise between the increased heat transfer rate and flow resistance in BFS flow applications.

4. CONCLUSION

This study computationally investigated the physical mechanisms governing heat transfer augmentation in a backward facing step flow utilizing various bluff body geometries as passive flow control devices. The introduction of angular and circular obstacles fundamentally altered the flow separation behavior by modifying momentum transport near the step. The restricted spatial gap between the step corner and the bluff body accelerated the fluid stream and enhanced local momentum. This accelerated momentum forcibly disrupted the primary recirculation zone and caused the fluid to reattach earlier on the heated bottom wall. Physically, this early fluid reattachment actively suppressed the development of the thermal boundary layer. The compressed thermal boundary layer established a significantly steeper temperature gradient at the interface

between the fluid and the solid wall. This steep temperature gradient was the primary physical mechanism that elevated the local Nusselt number and the overall convective heat transfer rate.

While all tested geometries successfully improved the thermal distribution, their unique profiles produced vastly different vortex formations and wake dynamics. The flat frontal area of the cube geometry generated immense stagnation pressure and a massive downstream wake. This extensive vortex region maximized fluid mixing and achieved the highest peak thermal performance with a 28.15 percent augmentation in the average Nusselt number at a Reynolds number of 400. However, the massive wake of the cube concurrently extracted a severe toll on the flow momentum and caused a substantial pressure drop penalty. Upon evaluating the strict compromise between heat transfer enhancements and pressure penalties using the Performance Evaluation Criterion, the diamond geometry emerged as the most efficient design. The streamlined angular profile of the diamond mitigated excessive vortex expansion and preserved fluid momentum significantly better than the cube. Ultimately, the diamond configuration delivered the most superior overall thermohydraulic performance and proved to be the highly optimal passive flow control geometry for backward facing step thermal management systems.

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