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Research Article

Experiments on Turbulent Flow in a Duct Affected by Diamond Obstacles with Aligned and Staggered Arrangements

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ABSTRACT

The convection heat transfer flow of air in a rectangular duct with and without diamond obstacles arranged in two ways (aligned and staggered) was experimentally examined in this work to examine the performance of the duct with and without the presence of obstacles and the impact of these obstacles on the enhancement of heat transfer in the duct. The purpose of the diamond obstacles was to enhance heat transfer within the duct. A variety of turbulent conditions were utilized for the tests. (Reynolds number=10000–35000). Pitch to obstacle height ratios of 4, 5, and 6 were employed with aluminum barriers at height ratios of 0.16, 0.25, and 0.3. According to the experiments' findings, these barriers improved heat transfer by enhancing the mixing process between air and the heated walls. This is seen when the obstacles are applied with obstacles height to duct height (e/H) = 0.3 and pitch to obstacle height (p/e) = 6 for the two configurations. The staggered configuration is thought to provide the best enhancement of thermal performance, equal to 174 % at $Re = 10,000$, while the aligned arrangement is 156% under the same conditions. That means that the staggered arrangement is better than the aligned case. Regarding the pressure drop, the findings demonstrate that the staggered arrangement yields the highest friction factor values for all investigated velocities, as well as the highest values for the elevation and pitch ratios; it reaches 0.14 at a Reynolds number = 10,000 for staggered and 0.122 for aligned. These findings demonstrate that staggered diamond barriers dramatically enhance turbulence and heat-transfer rates with tolerable pressure penalties, giving essential experimental data for compact heat-exchanger design.

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1. Introduction

The first thing engineers look at when trying to optimize the performance of any thermal device is energy efficiency. It explains methods for enhancing heat transmission between heat exchanger fluids. The heat exchanger is one of the most widely used forms of thermal equipment, with uses in a variety of industries and fields, including power plants, refrigeration

and air conditioning, food and chemical, and automotive. To improve heat transfer in heat exchangers, a variety of techniques have been used. Using obstacles or turbulators to encourage fluid mixing is one low-effort method of improving heat transport [1-5]. Because of its huge specific surface area, tortuous fluid channels, and lightweight nature, porous metal has attracted more and more attention from

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scientists and engineers in recent decades [6]. Heat transfer devices like reactors, heat sinks, and heat exchangers have made extensive use of it [7,8].

It is essential to point out that many analysts continue to aim to improve the effectiveness of the heating and ventilation systems. In order to enhance thermal performance, the analysts carried out extensive exploratory and explanatory research, which demonstrated that obstructions in the stream section increase turbulence in the liquid stream. Balances, ribs, baffles, cables, and other components are typically inserted as part of the detachable method [9-11]. Utilizing obstacle turbulators represents one of the most important passive methods for boosting heat transfer. The two most significant effects of being close to the ribs are a reduction in volume and a toll. As much as possible, intermittent obstacles are utilized for enhancing the heat transmission setup in various cooling applications [12-14]. The effects of 90° broken ribs attached to a hot absorber plate were investigated by Sahu and Bhagoria [15]. The range of the Reynolds number was 4000–12000. The rectangular channel's angle proportion was fixed at 8, its harshness height (e) was 1.5 mm, and the size to hydraulic diameter ratio (e/D_h) was 0.034. The testing used a range of discomfort and pitch shifts from 10 to 30 in the expansion. They observed a significant increase in Nu value at Reynolds number as a result of their analyses. At a pitch of 2 cm, the heat transmission coefficient was at its highest value. Additionally, the test results showed that the Nu increased from 1.25 to 1.5 times the smooth channel's value. Additionally, a set of tests on turbulent heat exchange and interaction calculations was carried out by Wang and Sunden [16] in a square conduit that was roughened by dependable and intermittent obstacles on a single surface. The rib e/D_h percentage was 0.16 in both situations. The fraction of the obstacle pitch was 12.0. The experimental results of this study demonstrated that the obstacles' shapes had a clear effect on the heat transfer coefficient.

In order to test heat transmission and improvement, Keshmiri et al. [17] used a rectangular duct for various values of p/e , 6 to 12. The results showed that the most noticeable heat transfer value that was at $p/e = 9$. Feng et al. [18] investigated the cooling enhancement with internal ribs and external coolant film. Eren et al. [19] used pierced ribs to assess the increase in heat transmission in a rectangular channel. Estimates were made using the joined obstacle pitch percentage of $p/e = 12$ and a height proportion of 0.1. In comparison to the smooth channel, it was

discovered that the roughed conduit provided an estimated 34.1% higher heat transfer value. Smaism [20] conducted tests and a theoretical analysis for the use of rectangular channels with rectangular obstacles of p/e ranging from 6.66 to 15 and different heights. It was discovered that the channel's thermal performance and heat transfer improved as the ribs' height expanded.

To achieve the best overall performance and test the pressure drop, the current experiment applies three distinct diamond obstacle heights and pitches with aligned and staggered arrangements. Additionally, the temperature rise in the channel wall is evaluated and compared for different obstacle configurations. The obstacles' lengths were 30 mm. Tests were conducted at a range of Reynolds numbers, approximately 10,000 to 35,000. Considering the benefits of ease of manufacture and installation, the two configurations provide good improvements in overall performance with a slight reduction in pressure for $(e/H)=0.35$ and $(p/e)=5$.

2. Experimental Setup

The experiment rig is designed and constructed to satisfy the requirements of the ongoing research endeavor. It is composed of the elements seen in Figs. 1 and 2, which represent the schematic design and picture of the test rig. The rectangular channel was made of galvanized iron that was 2 mm thick. The channel is measured as follows: length ($L=1$ m), height ($H=60$ mm), and width ($W=150$ mm). Overall performance can be increased by installing obstacle turbulators in the flow channel to improve heat transmission between the fluid and the heated surface. The obstacles are made of 0.7 mm thick galvanized steel. Multiple pitch ratios and height ratios $(e/H) = 0.15, 0.25$, and 0.35 , $(p/e) = 3, 4$ and 5 are examined for diamond obstacles with a 30 mm length. This study experimentally investigated the convection heat transfer of air flowing in a rectangular duct with and without diamond obstructions set in two different configurations (aligned and staggered). They are arranged in arrays one after the other along the lower surface of the duct and at different pitches from the entrance to the exit of the duct. The obstacles' placement and form are depicted more clearly in Fig. 3.

To create a consistent heat flow ($q=1000$ W/m²), a heater is applied to the bottom surface. To stop the heat from dissipating to the outside of the channel, glass wool insulation has been employed to insulate the bottom hot surface. The work was carried out in a room at a temperature that ensures a constant air temperature of 27°C. Readings were taken after

the flow rate was stabilized for thirty minutes for each case tested to ensure stability.

The top and side walls were thermally insulated with 25 mm glass-wool sheets to guarantee one-directional heat flow from below, while the bottom wall received a consistent heat flux of 1000 W/m^2 .



Fig. 1. The experimental setup

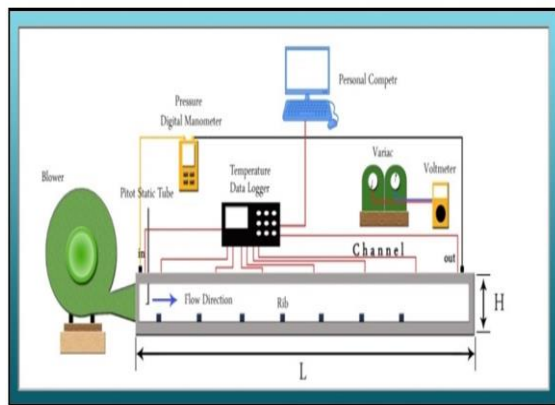


Fig. 2. The schematic diagram of test cases

Regarding the instruments for measuring, the temperature data logger (displayed in Fig. 4), additionally referred to as the temperature monitor, is a tool that can independently record the temperature on the channel's lower surface by employing T-type standard thermocouples that are positioned at various points along the bottom surface to measure temperatures ranging from -270°C to 400°C . The values of the static pressure difference along the duct are obtained using the Fluke 922 airflow model. Additionally, the same device is employed to measure the velocities by attaching it to the Pitot Tube. The Pitot-static tube is utilized to measure the velocity of the inlet airflow stream. The pitot-static tube has two concentric tubes with the ends turned through a right angle. It has a modified ellipsoid nose configuration to form a single forward-facing hole. The digital pressure manometer employed for this investigation is shown in Fig. 5.

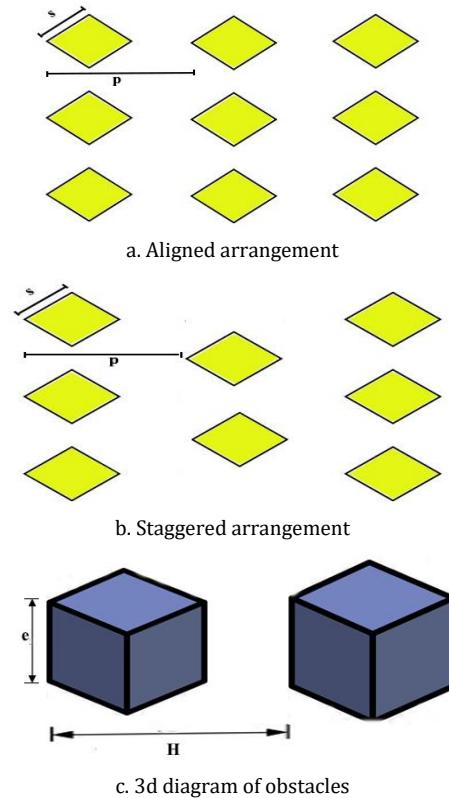


Fig. 3. The obstacles arrangements



Fig. 4. The temperature data logger

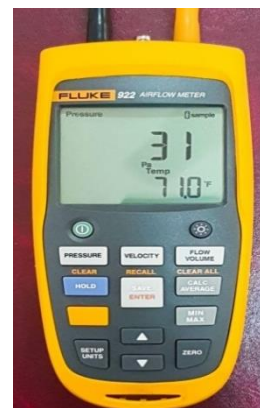


Fig. 5. The digital Airflow meter

Also, the digital multimeter is connected with an output line from the Variac in parallel to measure the voltage supply to the duct to get the required heat flux on the lower surface. The specifications of the multimeter used in the experiment are type VICTOR VC890D, with an accuracy of $\pm 0.8\%$.

Table 1. Geometrical parameters for the physical model

The tested channel			
Length (L)		1000 mm	
Height (H)		60 mm	
Width (W)		150 mm	
Thickness		2 mm	
The cases of the arrangement			
The case	e/H	p/e	s
Aligned	0.15, 0.25	4, 5,	30 mm
staggered	and 0.35	and 6	

2.1. Experimental Calculations

The airflow velocity, average wall temperature, inlet and outlet airflow temperature, and pressure drop along the rectangular duct—all of which have been measured in testing with and without obstacles for various cases—are the basis for the properties of the turbulent airflow in the current study, including Reynolds number, friction factor, and Nusselt numbers. Air served as the working fluid in this experiment, flowing turbulently through the duct with a uniform heat flux applied to the bottom surface. The equations required to compute the variables are validated in this section.

The following is a specification of the net heat input to the fluid derived from the electrical energy input to the system:

The air's heat transfer rate (Q) was calculated as follows:

$$Q = V.I \quad (1)$$

Now, the coefficient of heat transfer can be calculated as in the equation below [21]:

$$\bar{h} = \frac{q}{T_w - T_b} \quad (2)$$

Fluent by volume calculates the fluid's total bulk temperature. Integral utilizing a weighted average of mass [22].

$$T_b = \frac{\int_0^L \int_0^H \int_0^W \rho c_p u T \, dx \, dy \, dz}{\int_0^L \int_0^H \int_0^W \rho u \, dx \, dy \, dz} \quad (3)$$

Fluent by volume calculates the fluid's total bulk temperature. Integral utilizing a weighted average of mass [23]:

$$T_w = \frac{1}{A} \int T_i dA_i = \frac{1}{A} \sum_{i=1}^n T_i |A_i| \quad (4)$$

The friction factor is defined using the Darcy-Weisbach equation [24].

$$f = \frac{\Delta P \cdot D_h}{\frac{1}{2} \rho u_{avg}^2 \cdot L} \quad (5)$$

P_{in} was calculated by surface integral by means of area weighted average [25].

$$P = \frac{1}{A} \int P_i dA_i = \frac{1}{A} \sum_{i=1}^n P_i |A_i| \quad (6)$$

P_{out} was specified as a boundary condition and was selected as zero Pascal.

The mean Nusselt number (Nu) [26] is estimated as follows:

$$Nu = \frac{\bar{h} \cdot D_h}{k} \quad (7)$$

Overall performance [27] is considered here as a measure of improving the overall performance of the channel:

$$\eta = (Nu_w / Nu_0) / (f_w / f_0)^{1/3} \quad (8)$$

The thermal enhancement ratio is (Nu_w / Nu_0), the friction factor ratio is (f_w / f_0), and the friction factor and Nusselt number of the smooth duct are denoted by f_0 and Nu_0 , respectively.

2.2. Experiment Procedure

The steps listed below provide specifics of the experimental process used to accomplish the goals of the current investigation.

1. Assembling the experimental components by placing the channel inlet and the adaptor in the blower's outlet. The thermocouples are linked to the data recorder after being distributed around the channel's bottom surface.
2. Maintaining a virtually constant ambient temperature in the lab and ensuring that it is set up and appropriate for the experimental investigations.
3. To choose the preferred air flow speed, turn on the blower and adjust the speed regulator. The intended air speed was checked using a digital air flow meter.
4. To modify and calculate the necessary heat flux, connect the heating source in series with the variac and switch them on.
5. It takes 30 minutes to achieve steady-state reach after operation.
6. To determine the equipped energy, measure the equipped voltage with a multimeter and the current using a clamp meter.
7. Make a table to record the pressure drop reading, the input and outflow temperatures to determine the bulk temperature, and the wall temperatures on the channel's bottom surface to get the average wall temperature.
8. The preceding steps are repeated at five distinct speeds after the findings have been recorded.

9. To cover every example needed for the present study, repeat the previous stages.

2.3. Uncertainty

The three-time continuous staggered obstacles test was conducted again under identical conditions to determine the friction factor and Nusselt number for all Reynolds number values in order to guarantee the accuracy of the data derived from the experimental work. The three tests' results demonstrated a high degree of agreement; the differences between them did not surpass 1% and 2% for the friction factor and Nusselt number, respectively. The measurement errors for the friction factor and Nusselt number are displayed in Tables 2 and 3, respectively. The specifics of the uncertainty analysis are presented in the following equation:

$$\bar{x}_{mean} = x_{average} = \frac{\sum_{i=1}^n x_i}{n} \quad (9)$$

where

x_i : is the values of each measurement.

n : is the total number of measurements of x .

The standard deviation of (x) denoted by (σ), given by [28]:

$$\sigma = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n-1}} \quad (10)$$

The standard deviation of the mean. It is also called the standard error and is denoted by (σ_m), given by [28]:

$$\sigma_m = \frac{\sigma}{\sqrt{n}} \quad (11)$$

The final measured value of (x) is given by [28]:

$$x = \bar{x} \pm \sigma_m = \bar{x} \pm \frac{\sigma}{\sqrt{n}} \quad (12)$$

Table 2. The measurement uncertainties of friction factor

Re	x_1	x_2	x_3	$\bar{x}_{mean}$	σ_m	Percentage error (%)
10000	0.05	0.053	0.052	0.053	0.00088	2.464
15000	0.05	0.05	0.05	0.05	0.00066	2.279
20000	0.05	0.051	0.0488	0.049	0.00064	2.206
25000	0.04	0.046	0.045	0.045	0.0006	2.222
35000	0.03	0.034	0.034	0.034	0.00034	1.682

Table 3. The measurement uncertainties of Nusselt number

Re	x_1	x_2	x_3	$\bar{x}_{mean}$	σ_m	Percentage error (%)
10000	62.3	62.7	63.7	62.9	0.442	1.217
15000	77.3	78.64	76.8	77.6	0.577	1.179
20000	84.5	84.1	85.5	84.7	0.435	0.891
25000	93.3	94.79	92.8	93.63	0.6	1.097
35000	110.85	111.3	109	110.5	0.58	0.904

3. Results and Discussion

3.1 Validation

To ensure the credibility of the experimental work, the mean Nusselt number and friction factor values for the smooth rectangular channel were determined prior to the experiments on the rectangular channel. Empirical correlations for the Nusselt number and friction factor are

used to validate the experimental results. The Dittus-Boelter equation is used to validate the Nusselt numbers for the smooth channel [29].

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (13)$$

The heat transfer findings obtained from Eq. (13) and the current work agree correctly, with deviations of less than 4%, as illustrated in Fig. 4.

As seen in Fig. 5, the measured friction factor was also compared to the Blasius equation [30]. The average deviation of the results does not go above 5.8%.

$$f = 0.316 \text{ Re}^{-0.25} \quad (14)$$

A comparison between the current findings for aligned obstacles and the published findings of Gupta et al. [30] is shown in Figs. 6, 7, and 8. The thermal-hydrodynamics performance, friction factor, and Nusselt values are among the validations.

They tested with and without a gap in a rectangle-shaped duct with transverse obstacles organized with $(p/e) = 10$ and $e/Dh = 0.06$. The case with a 90-degree angle of attack and no gap was chosen for testing. As can be seen, the results' variations do not go above 14%, 15%, and 8%, respectively. This discrepancy results from variations in the channel configuration and pitch ratio values.

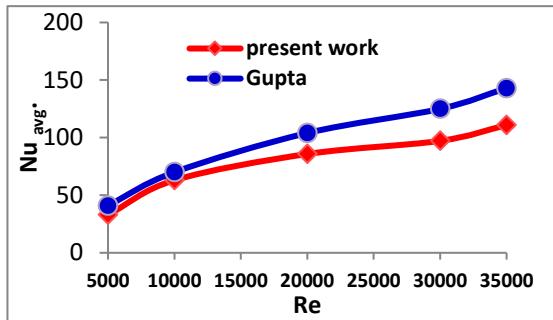


Fig. 6. Comparison between the current study and Gupta et al. [30] for Nusselt number results

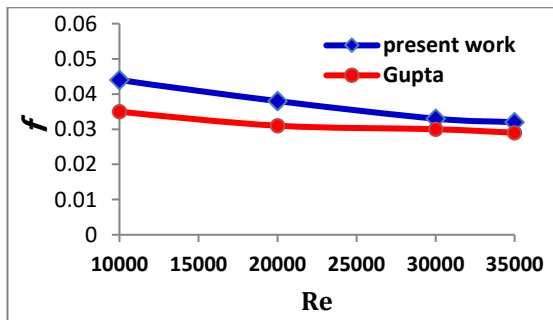


Fig. 7. Comparison between the current study and Gupta et al. [30] for friction factor results

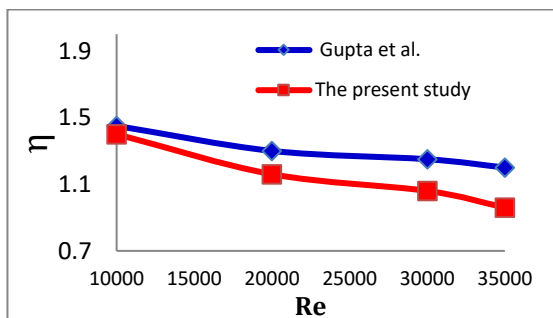


Fig. 8. Comparison between the current study and Gupta et al. [30] for overall performance results

3.2 Experimental Results

• Effect of hieght ratio (e/H)

Three obstacle heights were chosen in order to investigate and assess their impact on the channel's overall performance, friction factor, and Nusselt number. These figures were calculated as the obstruction height divided by the entire duct height. accordingly, these are $(e/H) = 0.15, 0.25$, and 0.35 .

The impact of height ratio (e/H) on the Nusselt number findings is displayed in Fig. 9. It is evident that the height ratio (e/H) significantly influences the heat transfer coefficient, which explains the Nusselt values. Tests have generally shown that greater Nusselt number values are associated with a bigger height ratio in the duct with obstacles. Therefore, with an unchanged value of $(p/e) = 6$, the Nusselt number corresponding to $(e/H) = 0.35$ is greater than those of $(e/H) = 0.15$ and 0.25 . The Nusselt number readings for $(e/H) = 0.35$ are shown to have risen by 16.35% when compared to $(e/H) = 0.15$ and 8.35% when compared to $(e/H) = 0.25$.

Nusselt numbers are increase with obstacles height because raising the obstacles can create alarger recirculation area beneath them, which increases turbulence intensity and vortex strength and improves heat transfer. For aligned arrangements, that is. According to the data, the (Nu) for staggered arrangements increases by 18.8% when compared to $(e/H) = 0.15$ and 11.5% when compared to $(e/H) = 0.25$.

Compared with the aligned arrangements, the staggered arrangements provided a 23% advantage in this test.

The main ways that e/H improves convective heat transfer are (i) expanding the separated wake and reattachment region, which causes the thermal boundary layer to thin repeatedly; (ii) strengthening the shear layer and producing turbulent kinetic energy at the tips of obstacles; and (iii) encouraging wake-wake interactions when obstacle spacing is small enough. These processes account for the increase in Nusselt number that has been observed experimentally with

As for the pressure drop, which is summarized here by the friction factor, Fig. 10 shows that the staggered arrangement gives the highest values for all heights, as the friction factor rises with increasing obstacle height.

Raising the ratio of obstacle to duct height together with e/H , raises forming drag and viscous dissipation by strengthening separation of flow at obstacle edges and increasing hydraulic obstruction. Longer wakes and stronger vortex formation are produced by larger obstacles; wake-wake interactions create

prolonged 3-dimensional turbulent structures that significantly raise the friction factor when wakes start to overlap (with suitably large (e/H) and/or small (p/e)).

Because of the more intricate wake interactions in the staggered geometry, the friction factor in the current trials was around 14.8% higher in the staggered arrangement (≈ 0.14) than in the aligned case (≈ 0.122) at $Re = 10,000$.

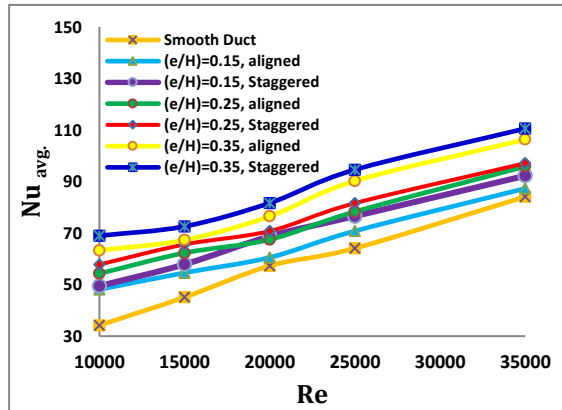


Fig. 9. Effect of height ratio on Nusselt number for $(p/e) = 6$

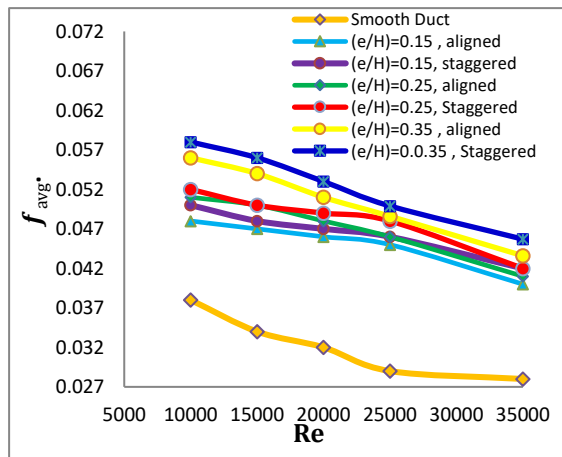


Fig. 10. Effect of height ratio on friction factor for $(p/e) = 6$

• Effect of pitch ratio (p/e)

One significant factor influencing the heat transfer area for obstacles is the pitch distance. By altering the ribs' pitch to height ratio (p/e) , it can be examined. $(P/e) = 4, 5$, and 6 are the three ratios that were examined. To ascertain how each ratio affects the Nusselt number, friction factor, and overall performance, these ratios are examined. For each ratio, these tests were carried out with Reynolds numbers ranging from $10,000$ to $35,000$.

The effects of the pitch ratio on the average Nusselt number and friction factor for various values of Reynolds number are described in Figs. 11 and 12. Experimental results reported that the presence of rib turbulators in the tested channel significantly increases average Nusselt

number values for all Reynolds number values compared to the smooth channel. It is clear that when the Reynolds number increases to $20,000$, the increase in Nusselt number becomes more regularly increasing and more stable.

The flow separation following the obstacles and the reattachment to the wall are the main justifications for increasing convective heat transfer. It is evident from the data that the Nusselt number rises with increasing (p/e) until it reaches $(p/e) = 6$. This is because the separation zone and the length of reattachment are influenced by the distance between the obstacles, which in turn affects the size of the recirculating zones behind the obstacles. Pitch ratio is thought to be a governing element in establishing the intended expansion of circulation zone size. As the pitch ratio rises to a tested limit value $(p/e) = 6$ for the arrangement of staggered obstacles, these zones' strength and size most likely grow in this case, and the findings reported that the staggered case gives more heat transfer; it is about 19% higher compared to the aligned case. As shown in Fig. 12, the friction factor increases with increasing the Reynolds number.

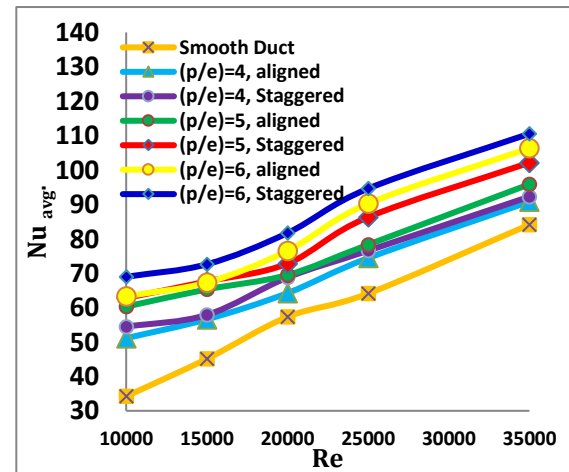


Fig. 11. Effect of pitch ratio on Nusselt number for $(e/H) = 0.35$

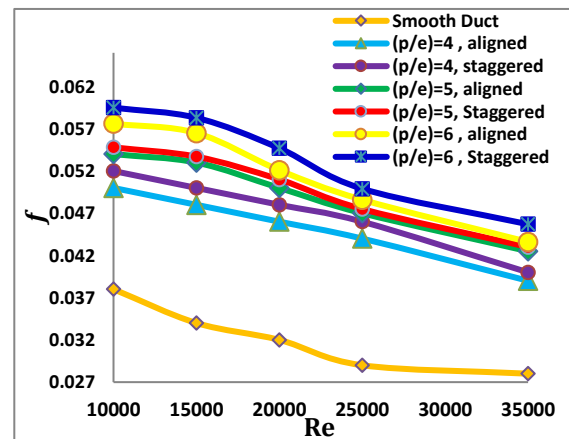


Fig. 12. Effect of pitch ratio on friction factor for $(e/H) = 0.35$

The improvement in thermal-hydrodynamics performance against the Reynolds number at $(e/H) = 0.35$ and $(p/e) = 6$ is shown in Fig. 13, because in these situations, testing produced the largest heat transfer. This figure illustrates how the variety of thermal-hydrodynamics performance decreases as the Reynolds number rises. Compared to aligned, the staggered arrangement resulted in the largest improvement in thermal-hydrodynamic performance. It was found to be 18.86% higher than the aligned ratios.

When impediments are employed, heat transfer improves more slowly as the Reynolds number rises because the flow does not have enough time to switch between barriers and heat transfer. As a result, overall performance improves less as Reynolds number rises.

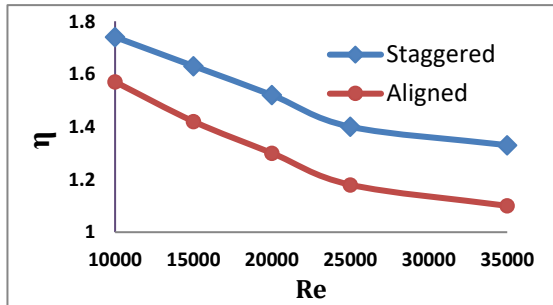


Fig. 13. The variation of the thermal-hydrodynamics performance for various Re for the arrangements of obstacles for $(p/e) = 6$ and $(e/H) = 0.35$

3.3 Empirical Correlations

For all of the study's examples, empirical correlations between the acquired experimental data and the friction factor, average Nusselt number, and thermal-hydraulic performance have been carried out. This analysis was performed using the XLSTAT software.

For these cases, the correlation is for varying pitch ratios and obstacle heights. For this obstacle configuration, the empirical equations for the Nusselt number, friction factor, and thermal-hydraulic performance are shown in Eqs. 21, 22, and 23, respectively.

$Re \leq 35000$ for 10,000

$$Nu = 1.33442 \left(\frac{H}{e}\right)^{0.05497} \left(\frac{p}{e}\right)^{-0.343} P_r^{0.4} Re^{0.46093} \quad (15)$$

$$f = 9.53644 \left(\frac{H}{e}\right)^{-0.21962} \left(\frac{p}{e}\right)^{-0.56202} P_r^{0.4} Re^{-0.41667} \quad (16)$$

$$\eta = 37.412 \left(\frac{H}{e}\right)^{-0.00365} \left(\frac{p}{e}\right)^{-0.33955} P_r^{0.4} Re^{-0.40299} \quad (17)$$

For the Nusselt number, friction factor, and thermal-hydraulic performance, the differences between the actual and projected values are within 2%, 4%, and 3%, respectively. The cases of $(p/e) = 6$ and $(e/H) = 0.35$ for the Reynolds

numbers under consideration have been utilized in Figs. 14, 15, and 16 to illustrate these variances.

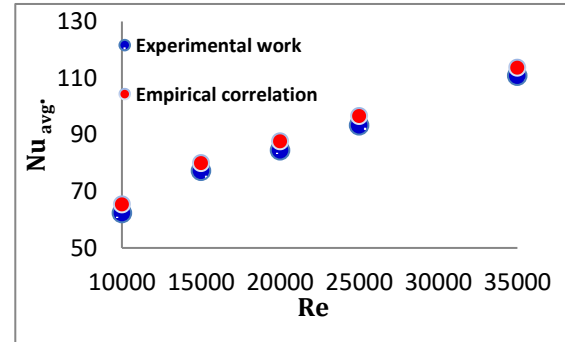


Fig. 14. Comparison between experimental and empirical correlations results of Nu for the case of $(p/e) = 6$ and $(e/H) = 0.3$

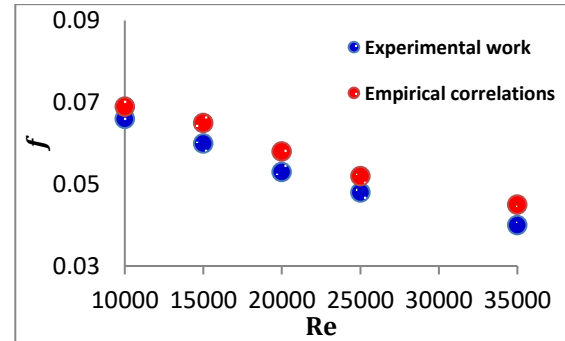


Fig. 15. Comparison between experimental and empirical correlations results of f for the case of $(p/e) = 6$ and $(e/H) = 0.3$

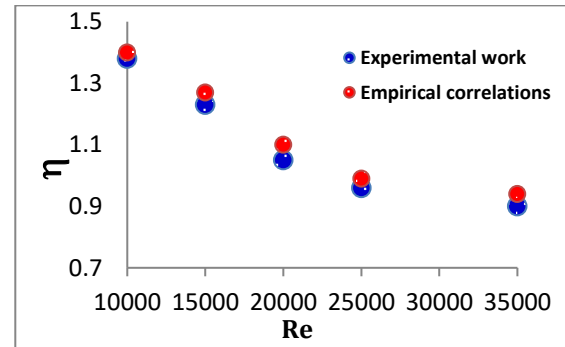


Fig. 16. Comparison between experimental and empirical correlations results of η for the case of $(p/e) = 6$ and $(e/H) = 0.35$

4. Conclusions

The following can be inferred from the present simulation's results:

- Overall, the results show that the (Nu) with velocity was evident in the staggered case and in all studied cases. This is because increased air mixing, created larger vortices and increased the recirculation area, enhancing the chance for dut. At $Re=35,000$,

the maximum Nu for the staggered case was found.

- When comparing the heights of the obstacles and the value of pitch between them, the highest height and the highest pitch had the greatest impact on improving the heat transfer. The value of Nu varied from 68.96 to 110.55 for the staggered obstacle cases, while it ranged between 63.3 and 105.34 for the aligned obstacle case
- In the staggered example at Re=10,000, the duct's maximum thermal-hydraulic performance is 1.74%. When this kind of barrier arrangement is added, the increase in heat transfer is more than the increase in the coefficient of friction values that the channel was subjected to in the same case.
- At staggered at Re=10,000, the highest friction factor value obtained is 0.12. As a result, it should be mentioned that the channel's overall performance increased along with the pressure drop (Δp), and as a result, (f)
 - The aligned case has the minimum friction factor and overall performance as compared to the staggered arrangements at $(e/H) = 0.15$ and $(p/e)=4$ for all Reynolds number values.

5. Recommendation

For better heat transfer, turbulence can be raised by:

- Hybrid Turbulence-Control Concepts: By combining diamond turbulators with acoustic excitation, controlled vibration, or spinning inserts, future research could integrate passive and active enhancement techniques. These hybrid systems can retain minimal pressure losses while greatly increasing mixing. Porous or Perforated Diamond Obstacles: It is anticipated that adding perforations or a porous metallic structure to the diamond obstacles may increase secondary vortex interactions and create multi-scale turbulence, improving local convective coefficients and lowering form drag.
- Nano-Enhanced Airflow: To increase efficient thermal conductivity and boundary-layer disruption, especially in low Reynolds-number applications, the use of hybrid nanofluids or nanofluid-seeded air (suspensions of nanoparticles in air) could be studied. Angle-Variable and Adaptive Geometries: Real-time thermal performance optimization under changing flow and

thermal loads would be possible with the employment of morphing or modifiable diamond obstacles with tunable inclination angles, moving the field closer to intelligent heat-exchanger systems.

- Integration with Wavy and Corrugated Surfaces: To achieve compound vortical structures and sustained reattachment zones for optimal heat transfer, future designs can investigate the complementary impacts of corrugated channel surfaces and diamond barriers.

Nomenclatur

A	Cross section area [m ²]
Dh	Hydraulic diameter [m]
e	Obstacle Height [m]
f	Friction factor
h	Convective heat transfer coefficient [W/m ² K]
H	Channel height [m]
K	Thermal conductivity [W/mK]
L	Length of channel [m]
Nu	Nusselt number
p	Pitch of obstacles [m]
P	Pressure [Pa]
Re	Reynolds number
s	Length of obstacles [m]
T	Temperature [K]
u	Velocity of air [m/s]
η	Overall performance

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Conflicts of Interest

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Authors Contribution Statement

Sarah Rabeea Nashee: Conceptualization; Formal Analysis; Investigation; Methodology; Resources; Software; Supervision; Validation; Writing – Original Draft; Writing – Review & Editing.

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