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Natural convection heat transfer in a rectangular channel with inclined longitudinal V-grooves

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Abstract
The principle of heat transfer enhancement has been widely applied in various fields, including electronic cooling—particularly in central processing units (CPUs)—heat exchanger units, gas turbine blade cooling, biomedical devices, and nuclear power plants. In the present study, a series of experiments was conducted to investigate free convection heat transfer in a horizontal rectangular duct. Three flat plates were tested: the first was a smooth plate without grooves, while the second and third had V-grooves inclined at 30° and 60°, respectively, relative to the horizontal. The objective was to examine the effect of varying heat flux rates, ranging from 148.6 to 1248 W/m², on the heat transfer coefficient and Nusselt number for both smooth and V-grooved plates. The results indicated that the V-grooved plate inclined at 30° significantly enhanced the heat transfer rate with increasing heat flux. Empirical correlations for the mean Nusselt number as a function of the Rayleigh number were derived for each of the plates tested.

Keywords
Heat transfer enhancement, Free convection, V-grooved plates, Nusselt number, Rectangular duct.

1.Introduction
One of the key mechanisms for thermal energy transfer is free convection, which plays a crucial role in addressing many practical thermal management challenges, both solvable and unsolvable under conventional conditions [1]. Heat transfer enhancement through free convection has been widely applied in various fields, including electronic cooling—particularly in central processing units (CPUs)—heat exchanger units, gas turbine blade cooling, biomedical devices, and nuclear power plants [2]. These applications are intrinsically linked to energy utilization technologies, where performance improvements are a primary concern [3–6]. Heat transfer occurs through three fundamental mechanisms: conduction, convection (both natural and forced), and thermal radiation [7]. Among these, natural convection is particularly significant in applications such as electronic devices, thermal management systems, and solar energy systems, where it is often a critical parameter for enhancing overall heat transfer efficiency [8–9].

Numerous numerical and experimental investigations have focused on how geometric modifications within enclosures influence energy loss or gain, as these alterations play a pivotal role in controlling thermal performance [10]. One effective approach to improving heat transfer from a heat-dissipating surface to the working fluid is the incorporation of grooves. These grooves introduce controlled surface roughness, disrupt the thermal boundary layer, and subsequently enhance convective heat transfer [11, 12].

In the pursuit of alternative and renewable energy sources, solar energy has emerged as a promising candidate. One practical application is the solar air heater. However, the rate of energy transfer from the absorber plate to the flowing fluid in such systems is often limited due to variable solar radiation intensity and unstable atmospheric conditions [13].

Various strategies have been employed to improve convective heat transfer rates, particularly in high-performance systems like gas and steam turbines, where efficient cooling of turbine blades is essential. Techniques such as the use of baffles, fins, jet

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impingement cooling, overhanging surfaces, roughened or serrated surfaces, and other turbulence-inducing methods have been explored. These enhancements aim to increase the heat exchange between the working fluid and heated surfaces while minimizing pressure drop, thereby optimizing the system's thermal performance [7]. The selection of an appropriate enhancement technique is typically based on application-specific performance criteria.

In this context, the implementation of grooved absorber plates with varying inclination angles in a rectangular duct is investigated as a means to enhance heat and mass transfer characteristics. Longitudinal corrugated grooves on the absorber plate surface disrupt the thermal boundary layer, reduce thermal resistance, and increase turbulence within the duct. This not only enhances the heat transfer from the absorber plate to the fluid but also benefits from the increased surface area provided by the grooves. Furthermore, the heat transfer rate is observed to increase with higher heat flux values, particularly when corrugated grooves are introduced at inclined angles.

This study aims to experimentally examine the effects of inclined longitudinal V-grooves on natural convection heat transfer in a horizontal rectangular duct. It evaluates the impact of varying heat flux on the heat transfer coefficient and Nusselt number, and proposes empirical correlations validated against established theoretical models.

The remainder of the paper is organized as follows: Section 2 presents the literature review; Section 3 describes the methodology; Section 4 discusses the results and analysis; and Section 5 concludes the study.

2.Literature review

Literature studies based on the characteristics of heat transfer and fluid flow aim to improve the energy transfer coefficient in such engineering applications. Therefore, the presented research reviews a set of studies that examined the basic concepts in performance improvement that are directly relevant to the current paper.

Ali [14] numerically investigated the effect of heat transfer by free convection from a plate placed inside a square-shaped cavity. The plate wall was heated at three different temperatures along its corrugated left side, while the right side was maintained at a cold temperature; the upper and lower sides were thermally insulated. Al-ali and Hamza [15] examined the impact of incorporating ribs within a rectangular channel on forced convection heat transfer and fluid flow characteristics. Their results were compared with those from smooth channels and channels using nanofluids. Zemani et al. [16] studied the properties and enhancement of convective heat transfer in an air-filled square duct, comparing it with an evacuated duct using various geometries, where the underside of the duct was partially heated.

Jehhef [17] investigated the effect of inclination angle, cavitation ratio, and surface temperature on free convective heat transfer from various flat smooth plate models heated from the upper surface. Kumar and Mahapatra [18] carried out experimental and numerical studies on free convection heat transfer inside a partially open square duct at different heat flux rates. Zhang et al. [19] examined the heat transfer coefficient of natural convection within a closed cavity, analyzing the influence of the electrical heater's height and radius, as well as the aspect ratio of the cavity. 20. Ameur [20] conducted a numerical investigation into the effect of corrugated fenders at different angles inside a rectangular channel. Zeng et al. [21] simulated energy transfer characteristics of a heat source using the Boltzmann method and calculated interior temperatures within a closed square duct, noting a decrease in Nusselt values with increased fluid volume under similar phase angles.

Tong et al. [22] explored free convective heat transfer induced by decay heat sources in complex flow channels using a triaxial model to analyze both convective and non-convective heat transfer phenomena. Karami et al. [23] numerically studied the influence of groove location and depth, along with the presence of nanoparticles, on turbulent heat transfer in a rectangular channel. Chordiya et al. [24] examined the effect of trapezoidal baffles in a saturated fluid-filled tank, both experimentally and numerically, concluding that obstructions enhance heat transfer performance in a partially open square cavity. Karabulut [25] investigated heat transfer and fluid flow enhancement in circular grooved channels by introducing baffles at varying heights and angles, showing improvements in the heat transfer coefficient.

Miansari et al. [26] numerically optimized free convection in vertical grooved cylinders by varying groove number and thickness, finding that greater surface area led to improved heat transfer. Said et al. [27] analyzed convective heat transfer in a container heated from both top and bottom surfaces, insulated at the sides, and filled with a nanomaterial-water mixture passing through a porous medium. Jobby et al. [28] presented a comprehensive study of convection heat transfer in horizontal and inclined rectangular cavities, using various internal shapes such as cylindrical, square, and rectangular baffles to assess their impact on heat and mass transfer.

Bilawane et al. [29] experimentally evaluated convective heat transfer from vertically grooved plates, comparing results with smooth plates by calculating Nusselt numbers. Baradi et al. [30] studied the effect of triangular grooves and CuO–water nanofluid flow in a square channel on turbulence and heat transfer performance. Kamal and Saihood [31] numerically assessed natural convection inside a rectangular chamber open on one side and exposed to room temperature, demonstrating that the use of porous materials enhanced heat transfer by increasing surface area. Mashud et al. [32] compared convective heat transfer and pressure drop in finned and circularly grooved cylinders, concluding that grooved configurations outperform smooth ones.

Mahmood and Raheem [33] investigated free convection of airflow in a rectangular enclosure with and without internal barriers of different geometries, analyzing barrier height, location, and inclination relative to Rayleigh number variation. Rashid et al. [34] examined the influence of semicircular grooves aligned with nanofluid flow on thermal performance in backward-facing step geometries. Ben and Ben [35] explored the convective heat transfer behavior on hot grooved cylinders with triangular and circular shapes, showing that higher Rayleigh numbers correlated with increased heat transfer. Ortiz and Koloszar [36] demonstrated that lower Prandtl numbers lead to boundary layer breakdown due to turbulence under intermediate heat flux conditions. Al-suhaibani et al. [37] conducted experiments on natural convection from square cylinder arrays with varying tilt angles and derived empirical correlations essential for thermal design applications.

The present study experimentally investigates free convective heat transfer within a horizontal rectangular duct using three types of plates: a smooth flat plate and two V-grooved plates inclined at 30° and 60° to the horizontal, respectively. The effect of varying heat flux rates on the air as the working fluid and the corresponding convective heat transfer characteristics was analyzed.

Therefore, this study plays a significant role in guiding the selection of suitable designs for thermal engineering applications. The key contributions of this work are as follows:

- The incorporation of V-grooved plates inclined at 30° and 60° enhances heat transfer to the fluid by disrupting thermal boundary layers along the rectangular duct, particularly at higher heat flux rates.
- Effective heat extraction from the aluminum plate is critical for maintaining adequate cooling, which directly contributes to improved operational efficiency, especially in turbine-related systems.

This paper presents a detailed description of the experimental setup and governing equations, includes an error analysis related to measurement and design uncertainties, and offers a comprehensive discussion of the experimental results, culminating in key conclusions.

3.Materials and methods

The present work involved the design and fabrication of the experimental test section, as illustrated in *Figure 1*. A series of experimental tests was conducted to study the effect of varying heat flux levels—specifically 148.6, 324, 568.6, 872.6, and 1248 W/m²—on the heat transfer coefficient and Nusselt number, using plates with and without V-shaped grooves. The test setup consisted of a wooden rectangular box with dimensions of 50 × 30 × 30 cm, open at both ends and thermally insulated at the top. An electric heater was mounted on the bottom surface of the box, fabricated as a metal panel with dimensions of 28 × 50 × 1 cm and a power capacity of 500 watts. Heat input was regulated using a variable voltage controller ranging from 0 to 240 volts, with ambient temperature maintained between 29 and 30 °C.

Three aluminum plates (thermal conductivity, k = 237 W/m·K) of size 28 × 50 × 0.8 cm were tested. The first plate was smooth, while the second and third plates were fabricated with 3 mm deep V-grooves inclined at 30° and 60° to the horizontal, respectively, as shown in *Figure 2*. These grooved plates were machined using a computer numerical control (CNC) milling machine (Vector 610) equipped with a high-speed steel tool (diameter = 10 mm), as depicted in *Figure 3*.

Temperature measurements were recorded using 20 type-K thermocouples strategically placed within the test setup. Five thermocouples were installed on each plate—one at the center and four at the corners. Three thermocouples were positioned at different heights inside the duct to measure the air temperature, and two additional thermocouples were mounted on the exterior to estimate heat losses. All temperature data were acquired using a BTM-4208SD 12-channel temperature data logger.

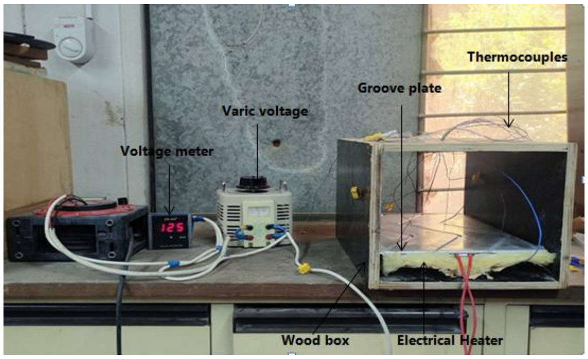


Figure 1 Photograph of the experimental setup showing the wooden test box, grooved plate, electrical heater, thermocouples, variac voltage controller, and voltage meter used for controlling and measuring heat input

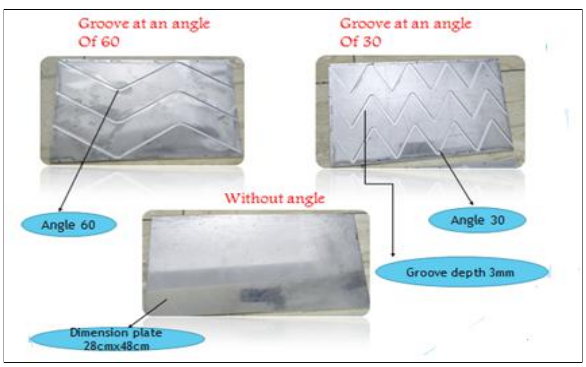


Figure 2 Dimensions of all test plates


Figure 3 CNC milling machine with high-speed steel tool

3.1Equations of data reduction

The Equations 1 to 7 were used to conduct the experimental analysis to calculate the rate of heat transfer by natural convection from a flat plate and two V-grooved plates to the air with changes in the heat flux rate.

The electrical power of the heater (Q) was calculated by Equation 1, Kwak and SONG [38];

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

Where, I=Electric current and V= Voltage

The heat flux (q'')can be calculated by Equation 2;

$$q'' = \frac{Q}{A} \quad (2)$$

A coefficient of heat transfer (h) can be determined by the following Equation 3;

$$h = \frac{q''}{(T_{s,ave.} - T_{a,ave.})} \quad (3)$$

Where,

$T_{s,ave.} = (T_1 + T_2 + T_3 + T_4 + T_5)/5$

$T_{a,ave.} = (T_5 + T_6 + T_7)/3$

The Rayleigh Number (Ra) was calculated from the following Equation 4;

$$Ra = \frac{g \times \beta \times \Delta T \times L^3 \times Pr}{v^3} \quad (4)$$

Where, $\beta = \frac{1}{T_f}$, and film temperature $T_f = \frac{(T_{s,ave.} + T_{air,ave.})}{2}$

All properties were estimated at (T_f)

The Nusselt number (Nu) can be written by Equation 5.

$$Nu = \frac{h \times L}{k} \quad (5)$$

To compare the empirical Nusselt numbers with the literature, Churchill and Chu's correlation was used as shown in Equation 6 [39];

$$Nu = 0.68 + \frac{0.67 \times Ra^{0.25}}{\left[1 + \left(\frac{0.492}{Pr}\right)^{9/16}\right]^{4/9}} \quad (6)$$

The mean Nusselt number (Nu_m) can be written from the following Equation 7:

$$Nu_m = \frac{1}{L}\int_0^L Nu\,dx \quad (7)$$

3.2Error data analysis

There are two primary factors that influence the accuracy of experimental results: the design of the test section and the precision of the measuring instruments. In this study, most of the observed errors are attributed to the measuring devices. To estimate the experimental uncertainty, Klein and McClintock's method was applied, as described by Holman [40] using Equation 8.

$$R = R(v_1, v_2, \ldots\ldots, v_n \ldots.) \quad (8)$$

Where R represents the error results and $(v_1, v_2, \ldots\ldots, v_n)$ is independent variables.

The uncertainty interval *w* in the results can be determined using Equation 9, as described in [41, 42].

$$w_R = \left[\left[\frac{\partial R}{\partial v_1} w_1\right]^2 + \left[\frac{\partial R}{\partial v_2} w_2\right]^2 + \cdots + \left[\frac{\partial R}{\partial v_n} w_n\right]^2\right]^{0.5} \quad (9)$$

The dimensions of the test section, along with the measuring devices and thermocouples used to record temperatures in all experiments, cannot be considered completely accurate [43]. Measurement errors are inevitable and are summarized in *Table 1*, as reported by Holman [40].

Table 1 Percentages of experimental data errors [44]

Independent variables (v)	Uncertainty Interval (w)
Temperature	±0.15 °C
Heater's voltage	±0.05 Volt
Heater's current	±0.002 Amp
Length of box	±0.003 m
Width of box	±0.003 m
High of box	±0.003 m

4.Result and discussion

The results are analyzed and discussed experimentally to investigate the effect of free convective heat transfer on the flat smooth plate and two V-grooved plates at angles of 30° and 60° and at different heat flux rates.

4.1Comparison of heat transfer coefficient

The effect of the heat flow rate against the transfer of heat coefficient for flat and grooved plates (30° and 60°) is shown in *Figure 4*. In general, the transfer of heat coefficient increases with increasing heat flow due to the increase of the difference between the plate surface and the fluid temperature. It is observed that the transfers of heat coefficient in presence of V-grooves are better than that of a flat plate. This is due to the presence of grooves working to break down the layers of thermal boundaries. Increasing the number of grooves causes a higher increase in the transfer of heat coefficient, as an increase of 28 % was recorded with a V-grooves plate at an angle of 30°, 23% with a grooves plate at an angle of 60°, and 18 % with smooth flat plate.

4.2Transfer of heat coefficient versus temperature difference

Figure 5 shows the relation between the transfer of heat coefficient and the temperature difference at each heat flow rate in the presence and absence of grooves on the plate surface. It is obvious from the figure that increasing the temperature difference causes to increase in the transfer of heat coefficient, especially with grooved plates at 30 and 60 angles. This is due to the increased breakdown of the layers of thermal boundary formed, which causes an increase in the process of heat exchange between the plate surface and the fluid.

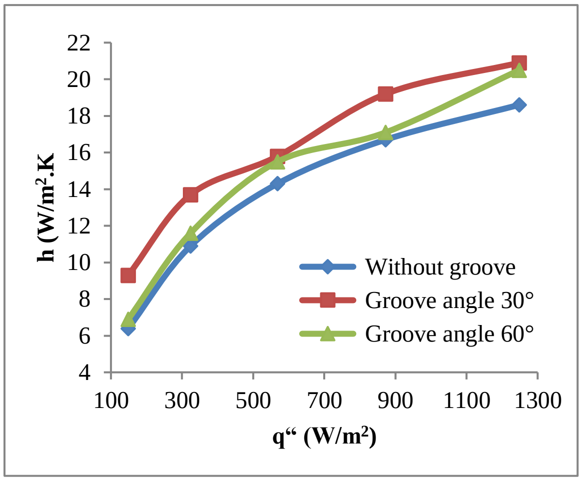


Figure 4 Heat transfer coefficient against heat supplied for all plates

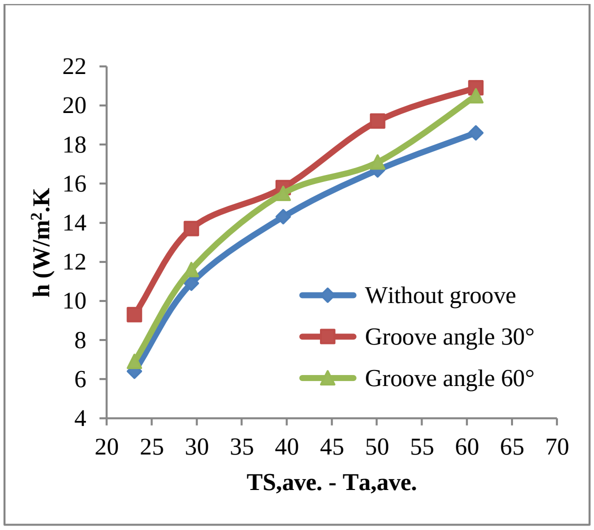


Figure 5 Heat transfer coefficient against average surface temperature difference for all plate

4.3 Average surface temperature versus Heat flux rate

The effect of varying heat flux rates on the average surface temperature is illustrated in *Figure 6*. In general, the average surface temperature of both flat and grooved plates increases with increasing heat flux. It is observed that the flat plate exhibits a higher average surface temperature compared to the grooved plates inclined at 30° and 60°. This is attributed to the smaller effective surface area of the flat plate, resulting in temperature increases of approximately 18% and 25% relative to the grooved plates at 60° and 30°, respectively.

4.4 Air temperature difference versus Heat flux rate

Figure 7 shows the air temperature difference versus the change in heat flux rates for the three plates (smooth plate and grooved at angles 30° and 60°). The figure shows that with increasing values of heat flow rates; the air temperature difference increases to reach the maximum value at q"=10. The highest values are obtained with the plates grooved at 30° because fracture of the thermal boundary layers is the greatest which caused increased heat transfer from the surface of the plate to the fluid.

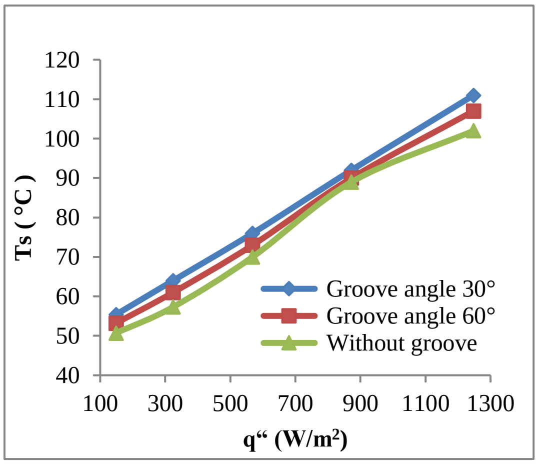


Figure 6 Heat flux rates against average surface temperature for all plate

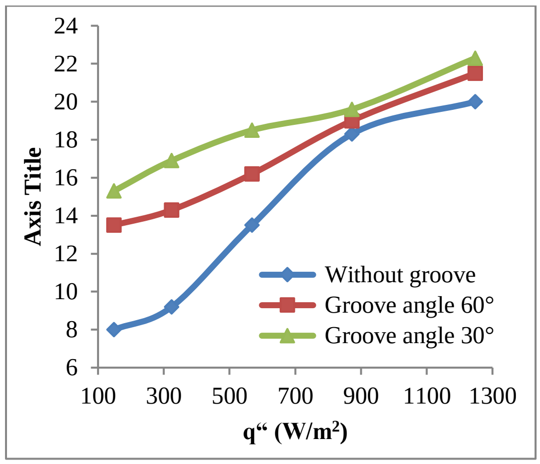


Figure 7 Heat flux rates against the air temperature difference

4.5 Nusselt numbers versus Rayleigh numbers for all plates

Figure 8 indicates a relationship of the Nusselt numbers against the Rayleigh number for flat and V-grooved plates at different heat flow rates. It is noted from the figure that Nusselt numbers increase with Rayleigh numbers because the transfer of heat coefficient is enhanced with an increase in temperature difference, and it has been explained in *Figure 4*. Maximum values of the Nusselt numbers were recorded at a plate grooved at 30° compared to a plate grooved at 60° and a smooth flat plate.

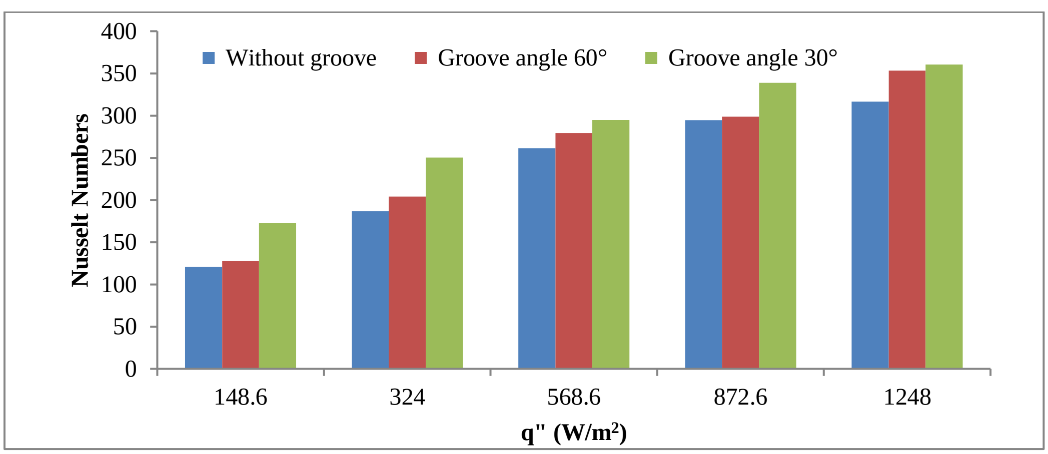


Figure 8 Variation of Nusselt number against Rayleigh number for all plates

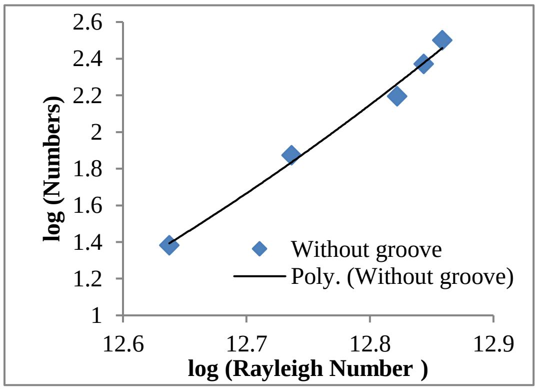
4.6Mean Nusselt number

The logarithmic of mean Nusselt numbers versus the logarithmic of Rayleigh number for smooth plate and V-grooved plates at angles 30° and 60° were shown in *Figure 9*. The empirical correlation was deduced as follows (Equations 10 to 12):

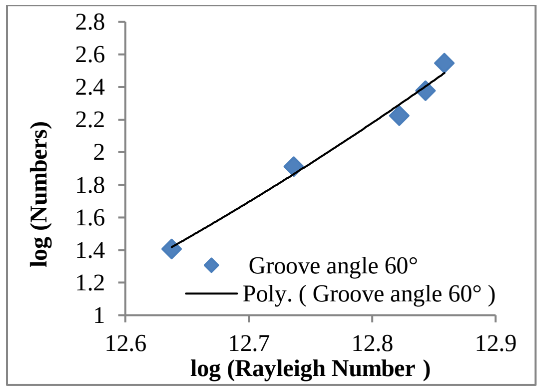
$$Nu_m = 2.8842\, Ra^2 - 68.72\, Ra + 409.21 \quad (10)$$

$$Nu_m = 2.4718\, Ra^2 - 58.188\, Ra + 4342.01 (11)$$

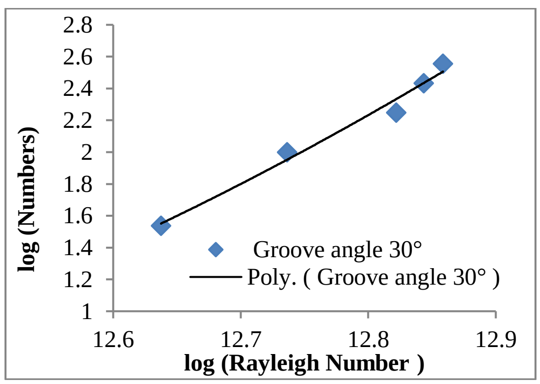
$$Nu_m = 2.1369\, Ra^2 - 50.163\, Ra + 294.21 \ (12)$$



(a) Without grooved plate



(b) with grooved plate (60°)



(c) with grooved plate (30°)

Figure 9 Correlation between mean Nusselt numbers and Rayleigh numbers

The uncertainty ratio within the inner and outer limits was estimated at a range of 24.1% to 28.8% for convective heat transfer inside a channel in the presence and absence of grooved plates. The Nusselt numbers were found to be within these limits. *Table 2* shows the variation of Nusselt numbers from their exact values. *Figure 10* shows the error bar of Nusselt numbers for different tests.

Table 2 Percent of experimental uncertainties

Test number	Internal uncertainty %	External uncertainty %	Total uncertainty %
1	25.7	0.5	26.2
2	28.3	0.5	28.8
3	27.0	0.5	28.2
4	24.4	0.5	24.9
5	23.6	0.5	24.1

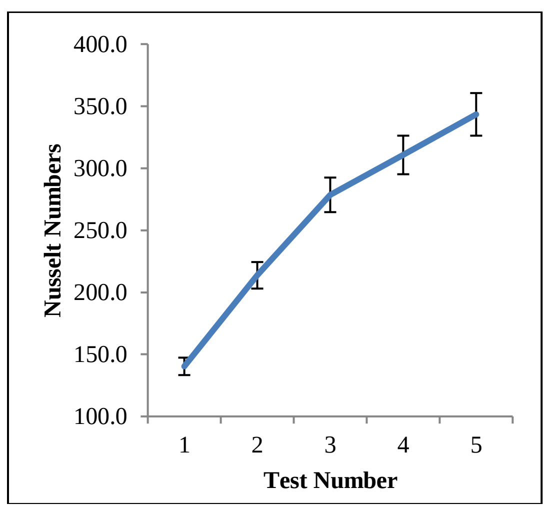


Figure 10 Error bars for the Nusselt numbers

4.7Comparison with the previous experimental works

Figure 11 provides a comparison between the current work and one of the previous literature Ben and Ben [35]. It is observed from the figure that there is a good convergence in the general behavior. It is noticed that the Nusselt numbers increases with increasing of the Raleigh numbers for all the plates.

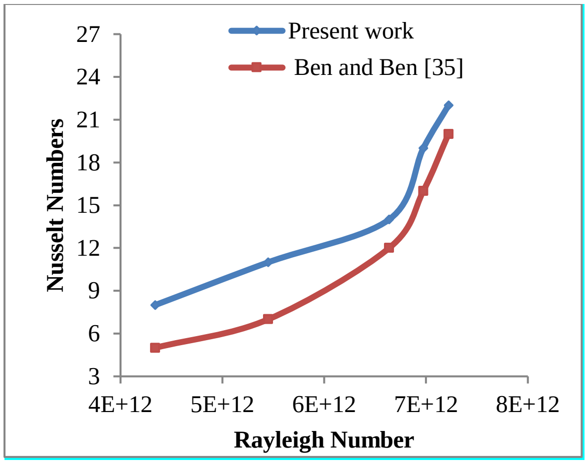


Figure 11 Comparison of the effect of Raleigh numbers on the Nusselt numbers between the present work and the previous experimental result

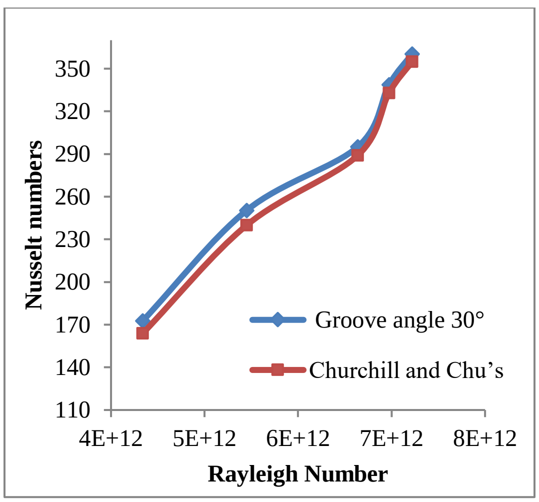
4.8Comparison between the theoretical and empirical Nusselt numbers

Figure 12 presents a comparison between Churchill and Chu's correlation (Equation 6) and the empirical Nusselt numbers obtained for V-grooved plates inclined at 30° and 60° under varying heat flux conditions. The results show a deviation of 5.2% for the plate inclined at 30° and 7.4% for the plate inclined at 60°, indicating good agreement between the theoretical and experimental data.

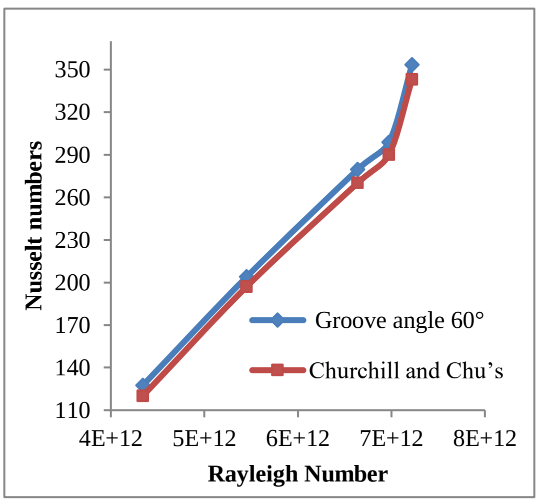
Limitations of this study are as under:

- Experimental tests carried out to investigate heat transfer by free convection in a horizontal rectangular duct.
- The three flat plates were used; The first plate was without V-grooves (smooth plate), and the second and third plates had V-grooves
- Tilt inclined at angles of 30° and 60° from the horizon respectively.

The study aimed to know the effects of varying the rates of heat flux from 148.6 to 1248 W/m^2 on the coefficient of heat transfer and the Nusselt numbers using smooth and V-grooved plates. A complete list of abbreviations is listed in *Appendix I.*



(a)



(b)

Figure 12 Comparison of the effect of Churchill and Chu's correlation and empirical nusselt numbers (a) Groove angle 30° (b) Groove angle 60°

5.Conclusion and future work

The present study experimentally investigates the impact of natural convection heat transfer from smooth flat plates and V-grooved plates inclined at 30° and 60° on the heat transfer coefficient and Nusselt number under varying heat flux rates. The key conclusions drawn from this work are as follows:

1. The highest enhancement in the heat transfer coefficient was observed with the V-grooved plate inclined at 30°, showing an increase of approximately 18% compared to the 60° grooved

plate and about 25% relative to the smooth flat plate.
2. The temperature difference between the average plate surface and the bulk air was greater for the 30° V-grooved plate than for the 60° grooved and smooth plates, indicating improved convective performance.
3. Surface temperature increased with rising heat flux across all plate configurations, with the 30° V-grooved plate consistently exhibiting superior thermal response.
4. The maximum Nusselt number values were recorded for the 30° V-grooved plate, with an increase of about 28% over the 60° grooved plate and approximately 33% over the smooth flat plate.

These results clearly demonstrate that a V-grooved surface inclined at 30° significantly improves natural convection heat transfer performance and is a promising design configuration for thermal management applications. For best performance in future applications a twisted fin and porous fin recommended be used as a fin in the rectangular channel.

Acknowledgment
None.

Conflicts of interest
The authors have no conflicts of interest to declare.

Data availability
Most of the datasets developed and used in this study are included in the published manuscript. The remaining datasets can be obtained from the corresponding author upon reasonable request.

Author's contribution statement
Sarmad A. Abdul Hussein: Study conception, design, data collection, supervision, investigation on challenges and draft manuscript preparation. **Suhaib J. Shbailat:** Data collection, conceptualization, writing – original draft, analysis and interpretation of results. **Mohammed A. Nima**: Analysis and interpretation of results, reviewing and editing. **Alyaa H Akhbala:** Making amendments and revisions to the manuscript and linguistic modifications.

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Appendix I

S. No.	Abbreviations	Description
1	CNC	Computer Numerical Control
2	CPUs	Central Processing Units
3	A	Cross-section duct area (m^2)
4	h	Transfer of heat coefficient ($W/m^2.K$)
5	I	Current (Amp.)
6	L	Duct length (m)
7	k	Thermal conductivity (W/m.K)
8	Nu	Nusselt numbers
9	q''	Heat flux (W/m^2)
10	Ra	Rayleigh Number
11	T	Temperature (°C)
12	V	Voltage (Volt)