Comparative Experimental Investigation of Convective Heat Transfer over Plain and Spherical Dimpled Surfaces under Heated and Cold Surface Conditions

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Abstract - Heat transfer enhancement using passive surface modification techniques offers an effective means of improving the thermal performance of heat exchangers and thermal management systems without additional energy consumption. This study experimentally investigates convective heat transfer over heated and cold staggered spherical dimpled surfaces under laminar forced convection. One plain surface and six dimpled surfaces with different linear and lateral pitch combinations were tested while maintaining constant dimple diameter and depth. The influence of dimple geometry on Reynolds number, Nusselt number, and convective heat transfer characteristics was evaluated under identical operating conditions. The results demonstrate that spherical dimples effectively disrupt the thermal boundary layer and generate secondary vortices, leading to enhanced heat transfer. For the heated surface, the optimum dimple configuration (30 mm × 50 mm pitch) achieved the highest Reynolds number of 71,907 and Nusselt number of 13,537, corresponding to a 24% increase in Reynolds number and a 61.85% increase in Nusselt number with only a 1.66% increase in effective surface area. For the cold surface, the same configuration produced the highest Reynolds number of 97,202 and Nusselt number of 4,193, representing a 22% increase in Reynolds number and a 208.96% increase in Nusselt number, while the increase in effective surface area remained only 1.66%. The greater enhancement observed for the cold surface is attributed to the lower dynamic viscosity of air, which promoted stronger flow development and intensified dimple-induced vortex formation. The findings demonstrate that heat transfer enhancement is governed primarily by boundary-layer disruption and secondary flow structures rather than the small increase in geometric surface area, confirming staggered spherical dimples as an effective passive heat transfer enhancement technique for both heating and cooling applications.

Keywords: Laminar forced convection; Staggered spherical dimples; Convective heat transfer enhancement; Reynolds number; Nusselt number; Boundary-layer disruption.

I. INTRODUCTION

Enhancing convective heat transfer is essential for improving the efficiency of heat exchangers, electronic cooling systems, and other thermal engineering applications. Among various passive enhancement techniques, dimpled surfaces have attracted considerable attention because they promote boundary-layer disruption and secondary vortex generation, thereby improving heat transfer with relatively low pressure penalties. Most previous studies have focused on heated dimpled surfaces under turbulent flow, while limited information is available on the comparative thermal performance of heated and cold dimpled surfaces under laminar forced convection. In addition, the influence of staggered dimple pitch on section-wise heat transfer characteristics has not been adequately investigated. Therefore, the present study experimentally evaluates six staggered spherical dimple array configurations under laminar forced convection. The thermal performance of heated and cold surfaces is compared in terms of heat transfer rate, average convective heat transfer coefficient, and section-wise heat transfer characteristics to identify the optimum dimple configuration for enhanced thermal performance.

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II. Literature Review

Several researchers have investigated passive heat transfer enhancement techniques using dimpled surfaces under various flow conditions. The major findings from previous studies are summarized below. Afanasyev et al. [1] conducted one of the earliest experimental investigations on spherical dimples fabricated on flat plates. Their study demonstrated that shallow dimples enhanced convective heat transfer by approximately 30–40% while causing only a marginal increase in pressure drop. The work established dimpled surfaces as an effective passive heat transfer enhancement technique. Chyu et al. [2] experimentally investigated concavity-based cooling passages containing staggered spherical dimples. They reported heat transfer enhancement approaching 2.5 times that of smooth channels and concluded that the generated vortex structures significantly improve thermal performance while maintaining acceptable hydraulic losses. Mahmood and Ligrani [3] experimentally studied flow structures and heat transfer characteristics in dimpled channels over a wide range of Reynolds numbers. Their work revealed that primary and secondary vortex pairs generated within the dimples promote boundary-layer disruption and produce high local Nusselt numbers downstream of each cavity. Ligrani et al. [4] investigated the influence of turbulence intensity on heat transfer over dimpled surfaces. They reported that inlet turbulence significantly modifies local heat transfer distribution and concluded that both flow conditions and dimple geometry govern the overall thermal performance. Ligrani et al. [5] further examined vortex development and local heat transfer characteristics over dimpled channels. Their results demonstrated that secondary vortices generated near the dimple rims contribute substantially to the enhancement of convective heat transfer. Vorayos et al. [6] experimentally compared staggered and inline dimple arrangements under forced convection. The staggered configuration consistently exhibited superior heat transfer performance owing to stronger vortex interaction and improved fluid mixing. Burgess and Ligrani [7] investigated the influence of dimple depth on thermal-hydraulic performance. They observed that increasing the depth-to-diameter ratio intensified vortex strength and increased Nusselt number, although accompanied by a moderate increase in friction factor. Leontiev et al. [8] experimentally evaluated various dimple spacing and density configurations. Their results indicated that optimum pitch and dimple density provide the best balance between heat transfer enhancement and pressure loss. Incropera et al. [9] presented the theoretical framework for convective heat transfer by introducing the governing relationships among Reynolds, Nusselt and Prandtl numbers. Their work continues to serve as the fundamental reference for analyzing forced convection heat transfer. Moon et al. [10] investigated the influence of channel height on heat transfer over dimpled surfaces. They concluded that channel geometry significantly affects vortex formation and heat transfer distribution, particularly around downstream dimple regions. Patel and Borse [11] experimentally analyzed forced convection heat transfer over dimpled plates and compared the results with those of a smooth surface. Their investigation confirmed that dimpled surfaces enhance heat transfer with comparatively lower pressure penalties than conventional roughened surfaces. Giram and Patil [12] studied six different dimpled plates having varying dimple densities. Their experiments showed that increasing dimple density and airflow rate improved the average Nusselt number and overall heat transfer coefficient. Ya-Ling He et al. [13] investigated heat transfer characteristics of ellipsoidal dimpled tubes. Their results indicated that ellipsoidal dimples provide better thermal performance than spherical dimples while maintaining relatively low pressure losses over a wide Reynolds number range. Amjad Khan et al. [14] experimentally studied natural convection over spherical dimpled surfaces under laminar conditions. Their investigation demonstrated that dimple orientation significantly influences heat transfer enhancement, particularly at lower Reynolds numbers. Verma et al. [15] analyzed different dimple geometries and reported that ellipsoidal dimples generate earlier vortex formation than circular dimples, resulting in improved thermal performance under laminar flow conditions. Sane et al. [16] experimentally investigated rectangular channels incorporating dimpled surfaces. They concluded that an optimum depth-to-diameter ratio exists beyond which the increase in pressure loss outweighs the improvement in heat transfer. NatVorayos et al. [17] investigated multiple dimple configurations under forced convection and demonstrated that staggered arrays consistently provide superior thermal-hydraulic performance compared with inline arrangements. Their work also emphasized the importance of optimizing pitch and dimple spacing to maximize heat transfer efficiency. The literature indicates that dimpled surfaces considerably enhance convective heat transfer through vortex generation and boundary-layer disruption while producing lower pressure losses than conventional protruded roughness elements. Previous investigations have primarily focused on turbulent flow over heated surfaces, whereas comparatively little attention has been devoted to laminar forced convection and direct comparison between heated and cold dimpled surfaces under identical operating conditions. Furthermore, the influence of staggered dimple pitch variation on section-wise heat transfer characteristics remains inadequately explored. Therefore, the present investigation experimentally evaluates six staggered spherical dimple array configurations under laminar forced convection on both heated and cold surfaces to identify the optimum pitch arrangement for enhanced thermal performance.

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III. EXPERIMENTAL SETUP

3.I Test Surfaces for Experimentation.

A total of seven test plates, comprising one plain (flat) plate and six staggered spherical dimpled plates, were fabricated for the present experimental investigation. The plain plate was used as the reference surface to evaluate the enhancement in convective heat transfer achieved by the dimpled configurations. All test plates were manufactured from 5 mm thick mild steel and were identical in overall dimensions, material properties, and heat transfer area. The only parameter varied among the dimpled plates was the arrangement of the dimple array, enabling the influence of dimple pitch on thermal performance to be investigated independently. The six dimpled test plates incorporated uniformly distributed spherical dimples arranged in a staggered configuration. The diameter and depth of the dimples were maintained constant for all specimens, whereas the linear pitch (X-direction) and lateral pitch (Y-direction) were varied according to the combinations listed in Table 1. The staggered arrangement was selected to promote boundary-layer disruption, secondary vortex formation, and enhanced fluid mixing, thereby improving convective heat transfer under laminar forced convection.

Table No.-1 X and Y axis pitch for the Pitch of Dimple arrays.

Surface No.	Array	
	X axis Pitch [Linear Pitch] mm.	Y axis Pitch [Lateral Pitch] mm
	Plain Surface.	
1.	25	50
2.	30	50
3.	25	60
4.	30	60
5.	30	55
6.	25	55

The geometrical configurations of the six dimple arrays are illustrated in Fig. 1. These configurations facilitated a systematic investigation of the influence of dimple pitch on the thermal and hydrodynamic behavior of the airflow. The overall dimensions of each plate were 700 mm × 500 mm, with each section having a length of 350 mm. Air supplied by the blower entered from the upstream end (Section 1) and flowed uniformly across the test surface before exiting through the downstream end (Section 2).

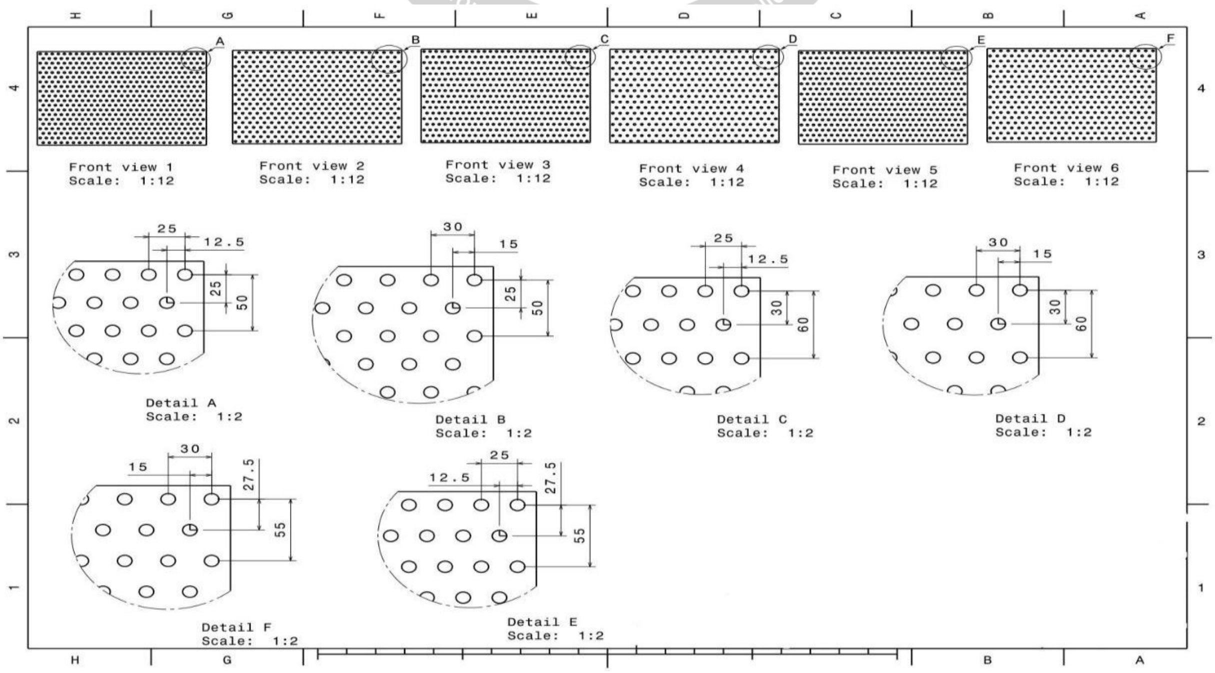

Fig. 1. Configuration of Dimple array.

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Figure 1 illustrates the detailed configuration of the six dimple arrays adopted in the present investigation During each experiment, the heat transfer rate, average convective heat transfer coefficient, and airflow rate were determined separately for both sections in addition to the overall surface performance. As the airflow progressed over the staggered dimple array, repeated interaction between the fluid and the dimples generated secondary vortices and intensified fluid mixing, resulting in variations in the local heat transfer characteristics between the upstream and downstream regions. The section-wise analysis therefore provided a comprehensive comparison of the thermal behavior of heated and cold dimpled surfaces under identical laminar forced convection conditions.

IV. EXPERIMENTAL PROCEDURE.

The experimental facility was designed to investigate the convective heat transfer characteristics of plain and staggered spherical dimpled surfaces under laminar forced convection. A centrifugal air blower driven by a 0.5 hp electric motor supplied a uniform airflow through a rectangular outlet having a width of 500 mm, corresponding to the width of the test plate, thereby ensuring nearly uniform air distribution over the entire heat transfer surface. A 3000 W strip plate heater was mounted beneath the test plate to provide a controlled and nearly uniform heat input during the heated surface experiments. Airflow velocity was measured using a calibrated digital anemometer positioned at the outlet of the test section. Surface temperatures of the test plate were monitored using a laser-based infrared temperature gun, whereas calibrated Iron–Constantan (Type J) thermocouples were installed at the inlet and outlet of the airflow passage to measure the corresponding air temperatures. Two independent dimmer stat units were incorporated into the experimental setup. One dimmer stat regulated the electrical power supplied to the strip plate heater, while the other controlled the blower speed to maintain the desired airflow velocity throughout the experiments. The arrangement provided independent control of the thermal and flow boundary conditions, thereby ensuring repeatable and reliable experimental measurements. The complete experimental setup employed for the heated surface investigation is shown in Figure 3.

4.1 Experimental Procedure for Heated Surface.

Prior to each experiment, the selected test plate was securely mounted on the experimental rig, and the strip plate heater was fixed beneath the test surface. The heater was energized through the dimmer stat, and the input voltage was maintained at 180 V to provide a constant heat input. The test surface was then allowed to attain thermal steady-state conditions. Surface temperature was monitored at 5-minute intervals using the infrared temperature gun until successive readings exhibited negligible variation, indicating that steady-state conditions had been achieved. Following thermal stabilization, the blower was switched on and the desired airflow velocity was established by regulating the blower speed using the second dimmer stat. The inlet air temperature was recorded using the thermocouple located upstream of the test section. After the airflow became stable, the outlet air temperature, airflow velocity, and surface temperatures at predetermined locations were measured and recorded. These experimental measurements were subsequently used to determine the heat transfer rate and the average convective heat transfer coefficient for the entire surfaces. The above procedure was repeated for the plain surface and each of the six staggered spherical dimpled test plates under identical operating conditions. Maintaining constant heat input, airflow conditions, and ambient surroundings throughout the investigation ensured that the observed differences in thermal performance were solely attributable to variations in the dimple array configuration.

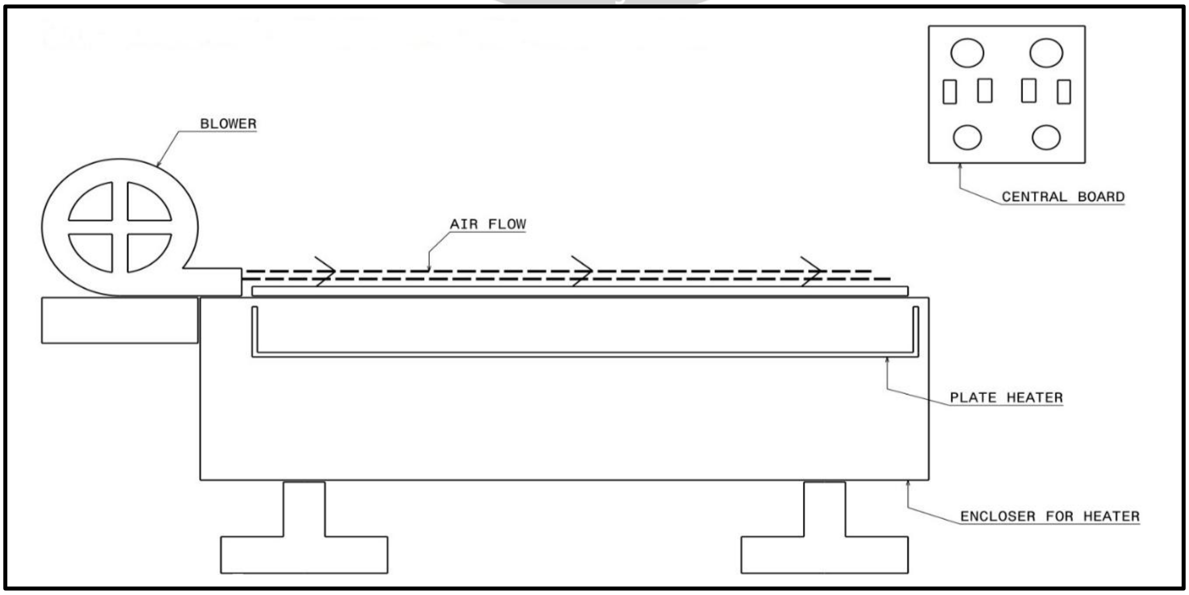

Fig. 2. Experimental set up for Heated Plate.

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4.2 Experimental Procedure for Cold Surface

To investigate the convective heat transfer characteristics of the cold dimpled surfaces, the experimental setup employed for the heated surface was suitably modified to provide a controlled cooling arrangement. A cold-water circulation system was incorporated into the experimental facility, consisting of a fabricated storage tank that served as the water sump. Cold water was continuously circulated from the sump to the test surface using a centrifugal water pump and returned to the tank after passing over the surface, thereby establishing a closed-loop cooling system. The temperature of the circulating water was maintained at approximately 12 °C throughout the experiments. The desired water temperature was achieved and sustained by periodically adding ice cubes to the storage tank. The schematic representation of the modified experimental arrangement is shown in Fig. 3.

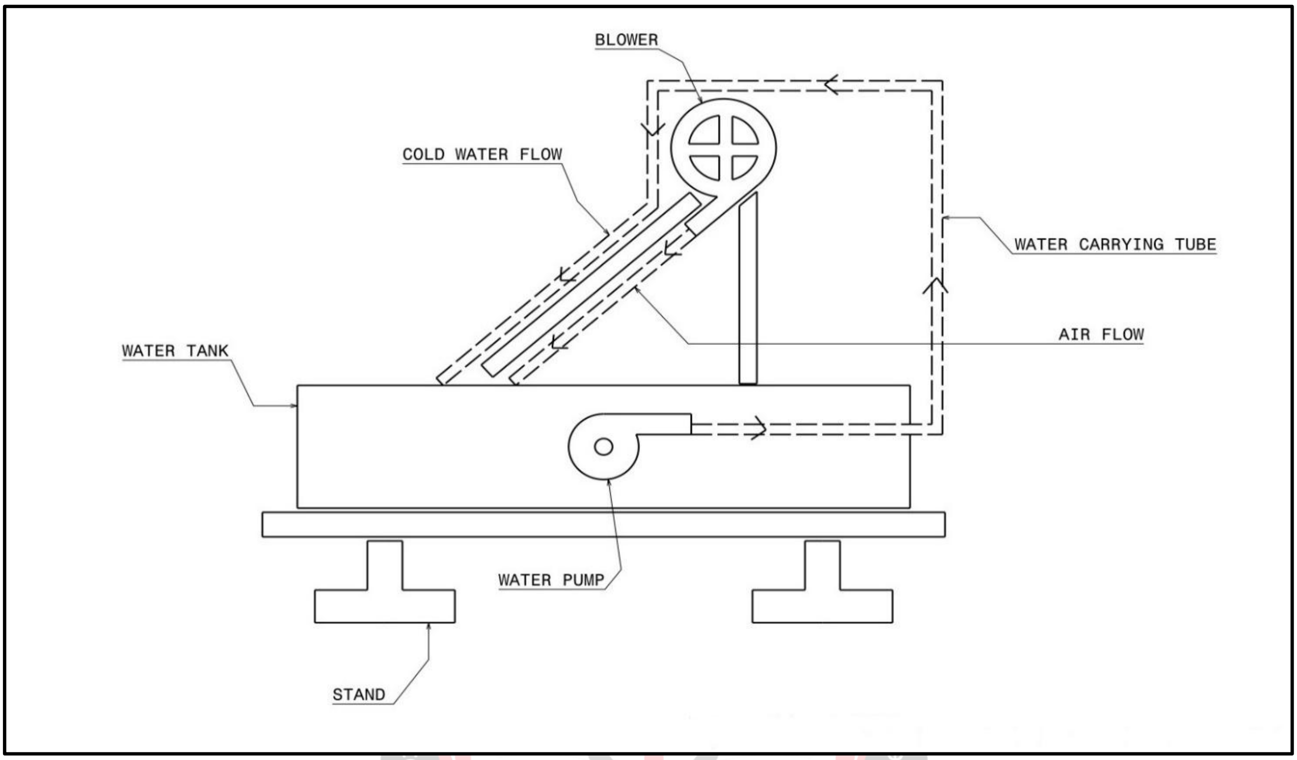

Fig. 3 Schematic Arrangement of Experimental Setup for Cold Surface

A constant water flow rate of 6 LPM was maintained during all experimental trials to ensure uniform cooling conditions over the test surface. Unlike the heated surface arrangement, the test plate was mounted at an inclination of 60° with respect to the horizontal. This orientation enabled a continuous and nearly uniform flow of cold water over the plain surface of the test plate under the influence of gravity. During the experiments, the plain surface was positioned upward to facilitate water flow, while the dimpled surface faced downward and remained exposed to the airflow. The blower was positioned accordingly to direct the air stream uniformly over the dimpled surface. After establishing the required water temperature and flow rate, the blower was switched on and adjusted to the desired airflow velocity using the dimmer stat. The system was allowed to attain steady-state conditions before measurements were recorded. The inlet and outlet air temperatures were measured using calibrated thermocouples, while the surface temperatures were monitored using a laser-based infrared temperature gun. Airflow velocity was measured using a calibrated digital anemometer at the outlet of the test section. Similar to the heated surface investigation, each test plate was divided into two equal sections along the direction of airflow. Experimental measurements were obtained separately for each section as well as for the entire surface. The recorded data were subsequently used to determine the heat transferred from the flowing air to the cold surface, the average convective heat transfer coefficient, and the section-wise thermal performance. Except for the modifications introduced in the cooling arrangement, all other experimental procedures, measurement techniques, instrumentation, and data reduction methods were identical to those adopted for the heated surface investigation.

V. Results and Discussion

5.1 Heated Surface

The data obtained after experimentation is used to plot different thermal characteristics. The details are discussed below.
Fig. 4 presents the relationship between Reynolds number and Nusselt number for the plain and dimpled surfaces. As expected, all dimpled surfaces exhibited higher Nusselt numbers than the plain surface, indicating improved convective heat transfer. The plain surface recorded the

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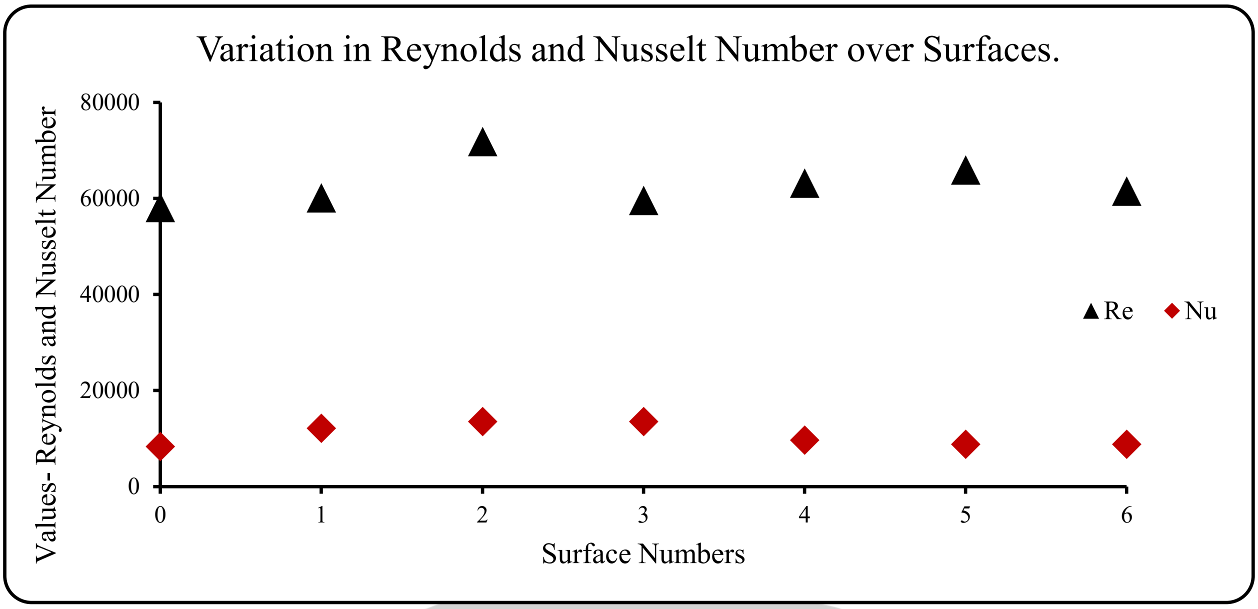
lowest Nusselt number of 8364 at a Reynolds number of 58,155, primarily due to the formation of a stable thermal boundary layer over the smooth surface. Among the dimpled configurations, Surface 2 achieved the highest Nusselt number (13,537) at Re = 71,907, while Surface 3 produced a nearly identical value

Fig. 4 Variation in Reynolds and Nusselt Number over Surfaces.



(13,528) despite operating at a much lower Reynolds number (59,624). This observation suggests that the dimple arrangement plays a significant role in enhancing heat transfer, in addition to the effect of flow velocity. In contrast, Surfaces 4, 5, and 6 exhibited comparatively lower Nusselt numbers, indicating that their dimple configurations were less effective in promoting fluid mixing. Fig. 5 shows the relationship between the percentage increase in surface area and the corresponding increase in Nusselt number for the dimpled surfaces. Although all dimpled plates had only a marginal increase in surface area (1.4–1.97%), the improvement in heat transfer was considerably higher. Surface 2 exhibited the highest enhancement in Nusselt number (61.85%) with only a 1.66% increase in surface area, followed closely by Surface 3 (61.74%). In contrast, Surfaces 5 and 6 showed only about 5.5–5.8% improvement despite having comparable increases in surface area.

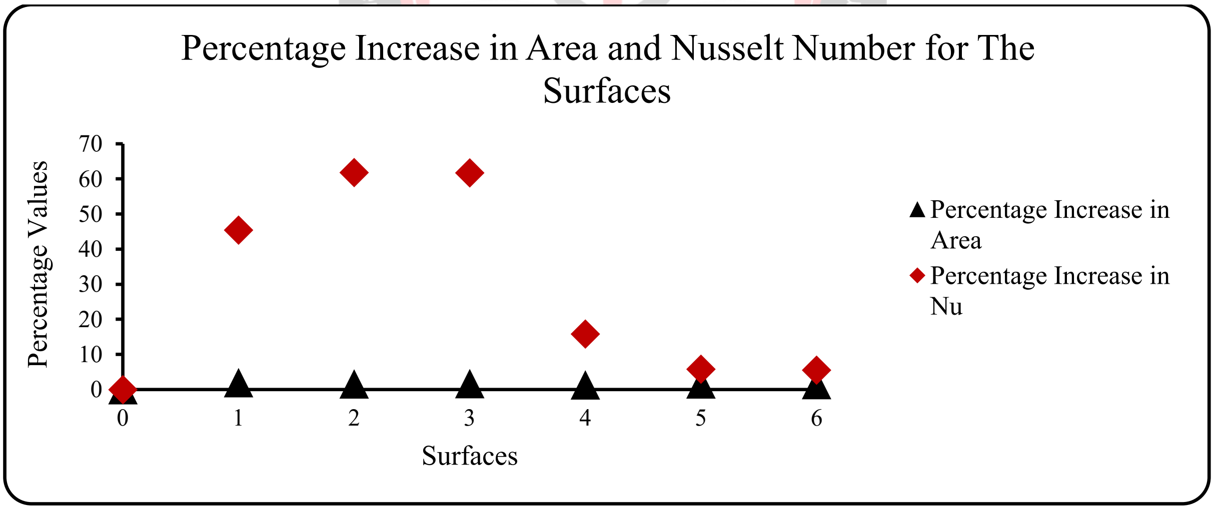


Fig. 5 Percentage Increase in Area and Nusselt Number for The Surfaces.

The percentage increase in characteristic length with the corresponding increase in Reynolds number for the dimpled surfaces is shown in Fig. 6. The incorporation of dimples resulted in only a marginal increase in the characteristic length, ranging from 1.4% to 1.97%, while the Reynolds number increased by 3–24%.

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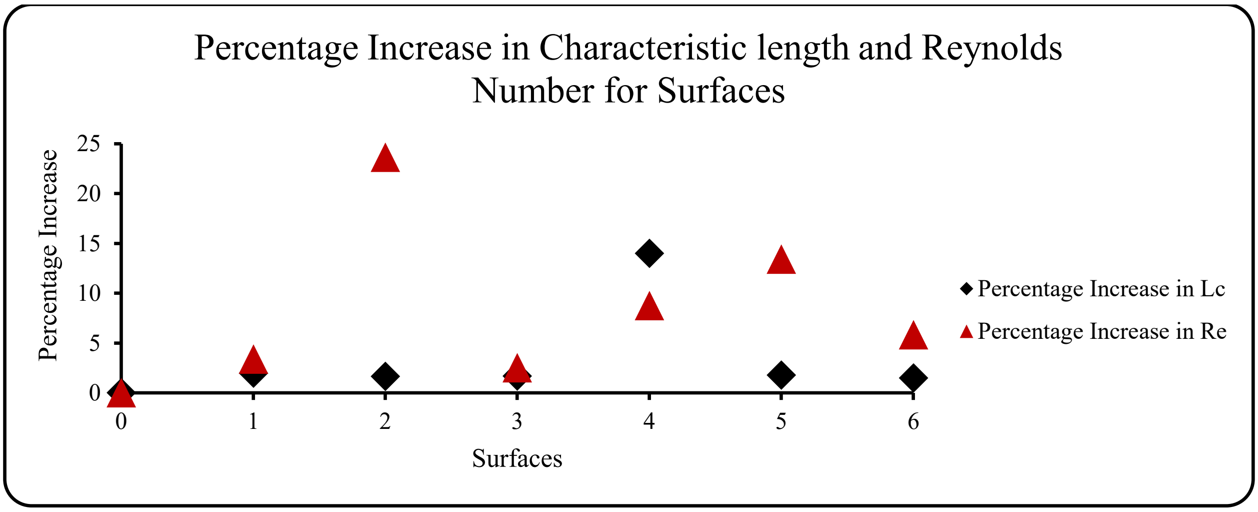

Fig. 6 Percentage Increase in Characteristic length and Reynolds Number for Surfaces.

Among the tested configurations, Surface 2 showed the highest increase in Reynolds number (24%) with only a 1.66% increase in characteristic length, indicating a more favorable flow behavior. In contrast, Surfaces 1 and 3 exhibited only a **3%** increase in Reynolds number despite having similar increases in characteristic length.

The relationship between the percentage increase in characteristic length and the corresponding enhancement in Nusselt number for the dimpled surfaces is illustrated in Fig. 7. Although the increase in characteristic length was relatively small, the improvement in heat transfer was significant for several configurations. Surfaces 2 and 3 achieved the highest increase in Nusselt number (62%) with only about 1.66–1.69% increase in characteristic length, followed by Surface 1 with a 45% enhancement.

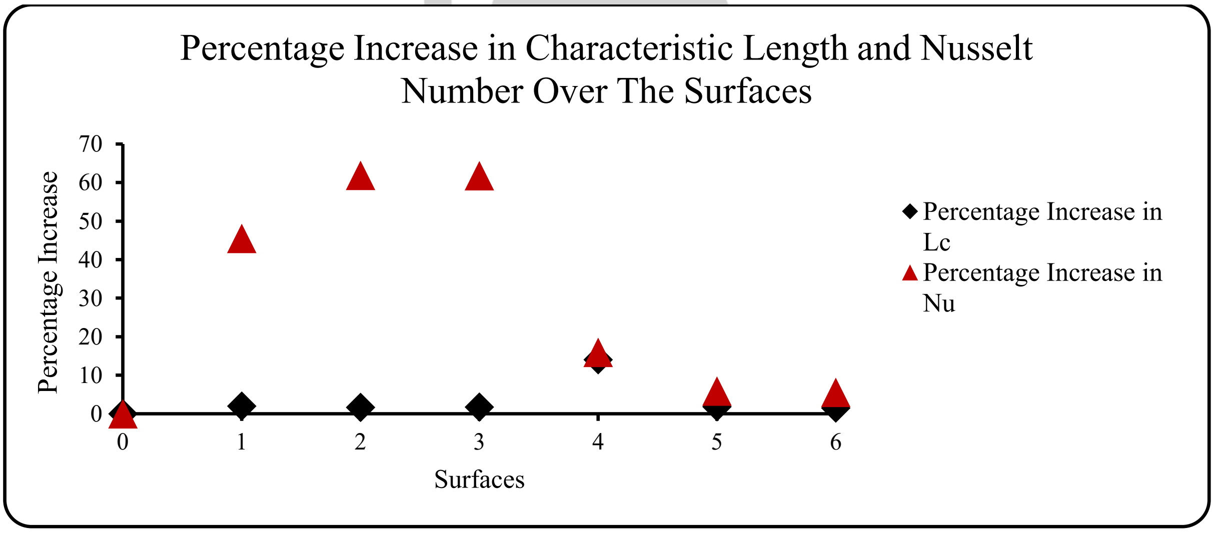

Fig. 7 Percentage Increase in Characteristic Length and Nusselt Number over The Surfaces.

4.2 Cold Surface

In the prior section, the results of heated surfaces have been discussed, now in the preceding sections the outcomes for the values of Enhancement in Heat transferred from air to the surface, increments in the average convective heat transfer coefficient and air flow .

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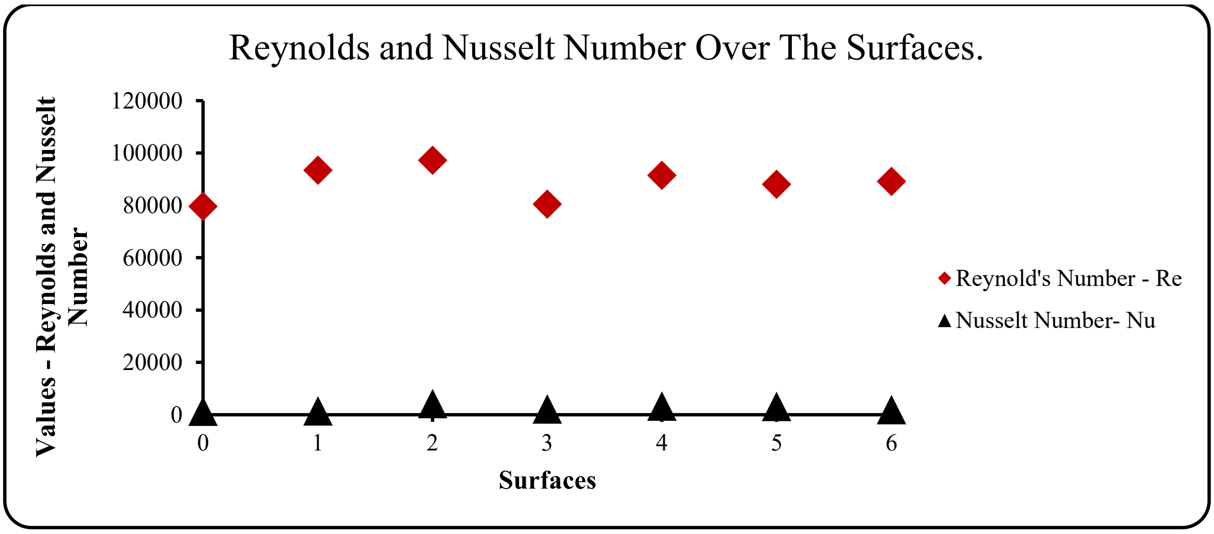

Fig. 8 illustrates the variation of Nusselt number with Reynolds number for the plain and dimpled cold surfaces. The plain surface recorded the lowest Nusselt number (1357) at a Reynolds number of 79,598, indicating the least effective convective cooling.

Fig. 8- Reynolds and Nusselt Number over the Surfaces.

All dimpled surfaces exhibited higher Nusselt numbers, confirming the beneficial effect of spherical dimples on heat transfer. Among the tested configurations, Surface 2 achieved the highest Nusselt number (4193) at Re = 97,202, followed by Surface 4 (3291) and Surface 5 (3094). However, Surface 1 produced only a marginal improvement despite operating at a relatively high Reynolds number, indicating that Reynolds number alone does not determine the cooling performance.

Fig. 9 illustrates the relationship between the percentage increase in effective surface area and the corresponding increase in Nusselt number for the cold surface. Although the effective surface area increased by only 1.4–1.97%, the Nusselt number increased significantly, with Surface 2 showing the highest enhancement (208.96%), followed by Surface 4 (142.52%) and Surface 5 (128.03%). In contrast, Surface 1 exhibited only a 3.64% improvement despite having the largest increase in effective surface area.

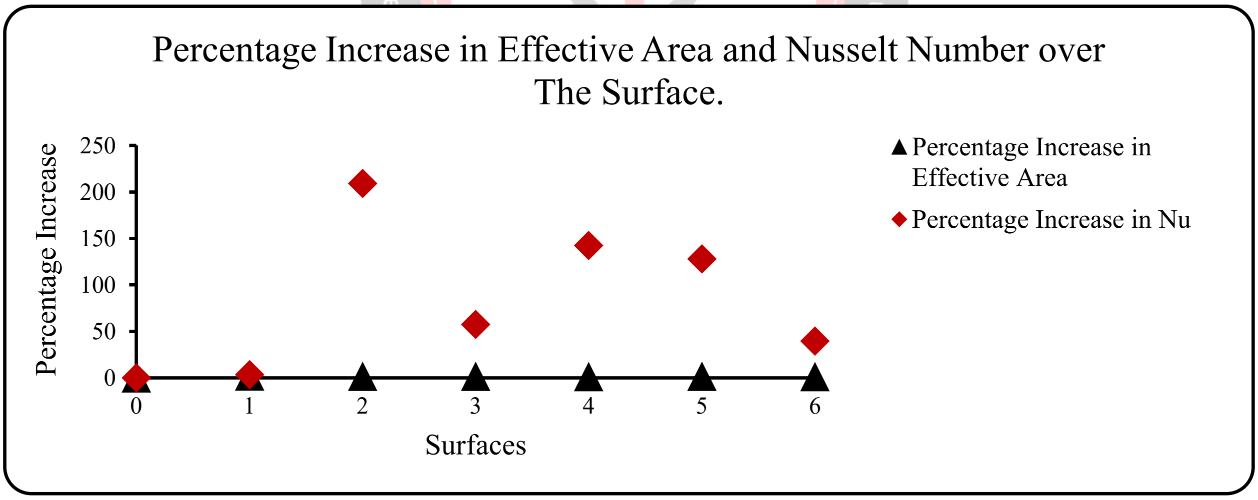

Fig. 9 - Percentage Increase in Effective Area and Nusselt Number over the Surfaces.

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Fig. 10 shows the percentage increase in characteristic length and variation in Reynolds number. Although the characteristic length increased by only 1.49–2.83%, the Reynolds number increased by up to 22%. Surface 2 exhibited the highest increase in Reynolds number (22%), followed by Surface 1 (17%) and Surface 4 (15%), while Surface 3 showed only a 1% increase.

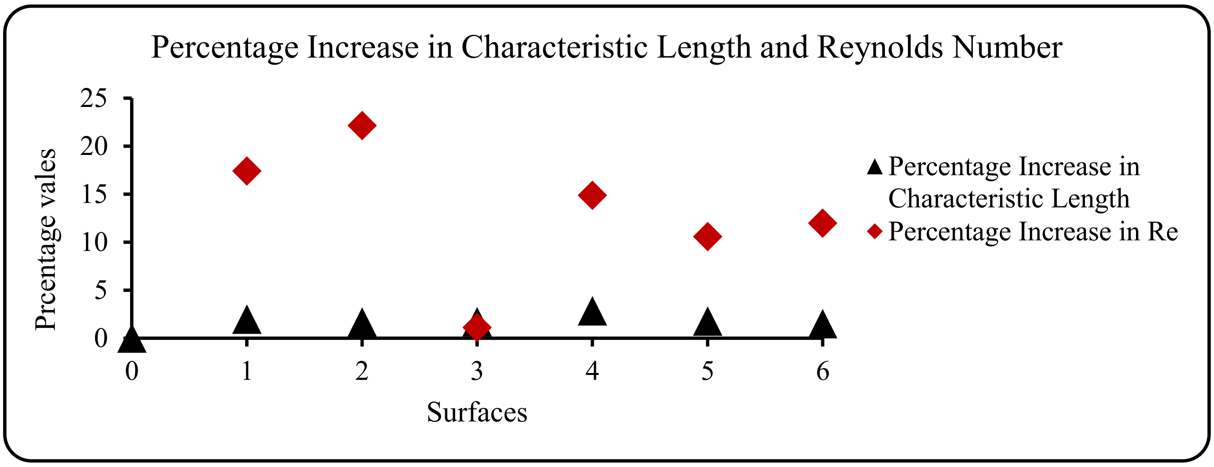

Figure 10 - Percentage Increase in Characteristic Length and Reynolds Number.

These results indicate that the increase in Reynolds number is influenced not only by the slight increase in characteristic length but also by the reduced air viscosity at lower temperatures and the flow modifications produced by the dimple geometry.

Fig. 11 illustrates the relationship between the percentage increase in characteristic length and the corresponding increase in Nusselt number for the cold surface. Although the characteristic length increased by only 1.49–2.83%, a substantial enhancement in Nusselt number was achieved. Surface 2 exhibited the maximum improvement (208.96%), followed by Surface 4 (142.52%) and Surface 5 (128.03%). In contrast, Surface 1 showed only a 3.64% increase despite having a relatively higher characteristic length. These results indicate that the enhanced convective cooling is primarily governed by the dimple-induced flow mixing and boundary layer disruption.

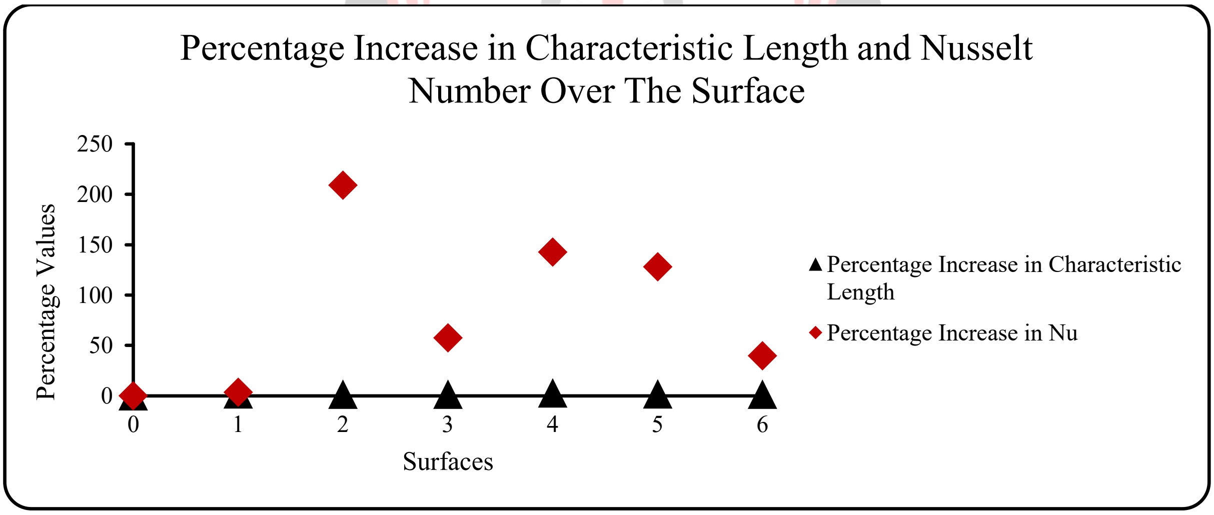

Fig. 11 - Percentage Increase in Characteristic Length and Nusselt Number over The Surface.

VI. CONCLUSIONS

The present work was towards experimental determination of effect of dimples on heat transfer over the heated and cold surface under forced convection condition with the varied dimpled array. The main conclusions are summarized as:

- The experimental investigation demonstrated that spherical dimples significantly enhance convective heat transfer under both heated and cooled surface conditions compared with a plain surface.

- The increase in effective surface area and characteristic length was relatively small (approximately 1.6–2.8% and 3.6–209%, respectively), indicating that the improvement in heat transfer was primarily due to the flow modifications induced by the dimple geometry rather than geometric changes alone.

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 For the heated surface, the higher air temperature increased the dynamic viscosity of air, which reduced the Reynolds number for a given flow condition and tended to suppress turbulence intensity.

 Despite the increase in viscosity, the spherical dimples effectively disrupted the thermal boundary layer, generated recirculation zones, and promoted fluid mixing, resulting in higher convective heat transfer coefficients and Nusselt numbers than those of the plain surface.

 For the cooled surface, the lower air temperature reduced the dynamic viscosity of air, leading to comparatively higher Reynolds numbers under similar flow conditions.

 The reduced viscosity facilitated stronger flow development around the dimples, while the dimple-induced vortices intensified boundary-layer disruption and fluid mixing, thereby enhancing convective cooling.

 The cooled surface exhibited greater heat transfer enhancement than the heated surface, with the optimum dimple configuration producing the highest Nusselt number among all the tested surfaces.

 The experimental results indicate that although the temperature-dependent viscosity of air influences the Reynolds number and convective transport characteristics, its effect is secondary to that of the surface geometry.

 The arrangement and spacing of spherical dimples govern the formation of vortical structures and the extent of boundary-layer disruption, making them the dominant parameters controlling convective heat transfer enhancement.

 The study confirms that an optimized spherical dimple configuration is an effective passive heat transfer enhancement technique for both heating and cooling applications.

Conflict of Interest.

The authors declare that there is no conflict of interest regarding the publication of this manuscript.

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