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EXPERIMENTAL STUDY OF FLOW AND HEAT TRANSFER IN RECTANGULAR DUCTS WITH RIBBED SURFACES

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Abstract – The Pivotal focus on fluid and thermal equipment performance have remained to directly lower energy cost by utilizing varieties of surface structures including extended surfaces, treated surfaces, and rough surfaces. This has necessitated the resurgence of surfaces with dimples and protrusions for an enhanced system efficiency in electronic components, gas turbine blade cooling, vortex creation on air foil structures, combustion chambers, printed circuit boards, microfluidic passageways, and heat exchangers features. In this study, the performance characteristics, heat transfer enhancement (Nu/Nu_o), friction factor ratio (f/f_o), and overall thermal performance (OTP) of two test channels with distinct surface structures. The performance parameters were evaluated using experimental rigs, one with continuous spiral rib channel and the other one with discontinuous spiral rib channel. Thereafter, the results from both test channels were compared to a smooth surface channel. Comparing the performance characteristics, Nu/Nu_o, f/f_o, and OTP of the discontinuous spiral rib and continuous spiral rib were 31.5%, 91.3%, 4.4% and 81.9%, 113.6%, 38.4% higher than the smooth surface spiral rib channel. Finally, the study shows that the continuous spiral rib channel gave a lower pressure loss, and was established to possess higher heat transfer coefficient and overall thermal performance than the discontinuous spiral rib channel.

Keywords – Surface structures, dimples and protrusions, heat exchangers, rib channel, heat transfer coefficient

I. INTRODUCTION

This Heat exchangers are used in a variety of operations, including heat conversion, usage, and recovery in industrial, residential and commercial environments. Steam generation, cooling in thermal processing of agricultural and pharmaceuticals items, cogeneration in plants, water heating recovery and fluid heating in manufacturing amongst others are common examples where heat exchangers are used [1]. The increase of the performance of heat exchangers can lead to more cost-effective design, resulting in material, energy and cost savings in the heat transfer process. The necessity to improve the thermal efficiency of heat exchangers in order to save energy, material, and money has led to the creation and application of a variety of heat transfer augmentation techniques [2]. Heat transfer enhancement is another name for these approaches. The conventional ways of improving the thermal performance and heat transfer rate are the active methods which requires the use of external power sources, for example, fluid vibration, mechanical aids, fluid suction and injection, among others and passive methods such as the use of extended surfaces/rough surfaces/turbulators [3]. However, recent researches have been focused on the passive method of heat transfer due to ease of maintenance, low cost, and manufacture. In recent decades, the use of passive heat transfer enhancement methods characterized by the use of turbulators such as rib, groove/dimple, baffles, conical ring/nozzle, twisted tape, winglet, wing, among others have rapidly been developed [4].

In the work of [5] it is clear that grooves are very efficient in heat transfer, [6] also carried out an experiment concerning inclined ribs and it was discovered that slanted ribs creates a counter-turning stream in the duct which is responsible for the high Nusselt number, [7] research proved to be very useful in the design of highly thermal efficient heat exchangers, [8] carried out a numerical analysis of the heat transfer in a microchannel heat sink by using augmented rectangular ribs as turbulators, the result was that using hybrid methods significantly increased the efficiency of the heat sink. Heat transfer analysis was also carried out on arc shaped solar air heater by [9], and it was discovered that the thermal efficiency of the solar heater increases as the temperature difference decreases. [10] carried out a numerical study on the effect of friction factor on heat transfer and it was discovered that as the step and height ratio increases, friction factor decreases. An experimental research was carried out by [11] on the effect of rib size on heat transfer in a triangular duct and it was discovered that the Nusselt number tends to increase as the number of ribs and rib parts increases. [12] carried out an experiment involving heat transfer in rectangular ducts with ribs placed in rows and columns, it was discovered that the row-column arrangement of the ribs has a better heat transfer uniformity but the thermal uniformity and heat enhancement of the duct decreases. Recently, research and experiments have also been carried out on heat transfer using nanofluids, [13] critically reviewed experiments concerning nanofluids and concluded that the heat transfer performance increases as the heat flux increases but that the use of nanoparticles still needs further investigation.

Different shaped v-pattern baffles were also used as turbulators in a research carried out on heat transfer in a solar air channel [14], it was discovered that discretized broken v-shaped baffles has the highest thermal hydraulic performance when compared with other forms of baffles such as Rectangular cut baffle, U-shaped baffle, V-shaped baffle, Z-shaped baffle, Staggered diamond baffle and Inclined perforated baffle. Multi V-shaped perforated baffle were solely used in an experiment on the study of the heat transfer in rectangular ducts [15] and it was concluded that the perforation in the multi V-shaped baffles helped to enhance friction factor and heat transfer of a rectangular ducts. A comparative study was also carried out [16] on the effect of different turbulator arrangements on the hydraulic performance in an air passage and it was discovered that the use of different types of turbulators on a heated surface is a very good way to increase heat transfer and that V-shaped perforated blockages are better heat transfer compared to other blockage shapes. In the same year a review was also carried out on the effect of nanofluids on heat transfer [17] and it was concluded that metal nanoparticles are very high thermal conductor compared to ordinary fluids and that the use of copper-water as a nanofluid is more lucrative when compared to the use of alumina-water. [18] also compared the numerical and experimental results of using baffles on the effect of heat transfer in a rectangular duct, the result was an increase in thermal efficiency. A critical review was carried out on the effect of using triangular dimples on heat transfer with ducts and it was observed that triangular shaped dimples have a higher heat transfer efficiency, also triangular shaped dimples is better when used as turbulators as compared to using cylindrical dimples. [4] carried out an experiment on heat transfer in solar receiver combining chamfer-v-grooves (CVG) and punched-v-ribs (PVR), it was concluded that heat transfer was at its highest heat transfer when CVG and PVR were combined together as opposed to being used separately.

In the quest of trying to increase heat transfer and thermal efficiency, [19] used punctured winglets as a turbulator in a rectangular duct to measure the heat transferred, the result was that punctured winglets is a very good device to increase the thermal performance of an heat exchanger. [20] also studied the effect of triangular rib groove geometry on heat transfer and the result shows that the triangular rib-groove geometry has a better heat transfer efficiency as compared with heated plates without the triangular rib-groove arrangement. [21] studied the heat transfer in a two-pass channel with cylindrical dimples and v-shaped ribs, it was concluded that heat transfer was higher in the V-compound two-pass channel as compared with the v-rib and also thermal hydraulic performance was highest in the compound channel. In the following year, [22] also carried out both numerical and experimental investigation of fluid flow and heat transfer in a square duct, the result shows that the friction factor of the staggered arrangement was higher than that of the inline arrangement, the result also shows that the thermal performance of both configurations were similar to one another for Reynolds number between 30000 to 60000. Furthermore, [23] which concluded in their experiment that hybrid ribs has a better heat transfer efficiency as compared to rectangular and semicircular ribs, [10] also concluded in their experiment that the combination of ribs and grooves has a very high thermal efficiency as compared to any other combination

After careful and in-depth reviews, it can be inferred that heat transfer efficiency can further be improved by the combination of different configurations. Therefore, this study will focus on the effect of hybrid roughened ribs and grooves on the effect on heat transfer efficiency.

II. Materials and Method

This study employs different rib shape (continuous and discontinuous spiral arrangements). The creation and utilization of an experimental set-up to compare three (3) test channels with variable surface geometry. This rig's distinctive feature is that it can test as many test channels as possible. Figures 1, 2 and 3 depict the experimental setup's, its instrumentation and schematics as. This apparatus is flexible, allowing it to examine as many different types of duct surfaces as the test section may allow. A continuous spiral arrangement channel, a discontinuous spiral arrangement channel, and a plain-smooth surface channel are among the test samples. The thermal hydraulic performances of all of these are compared.

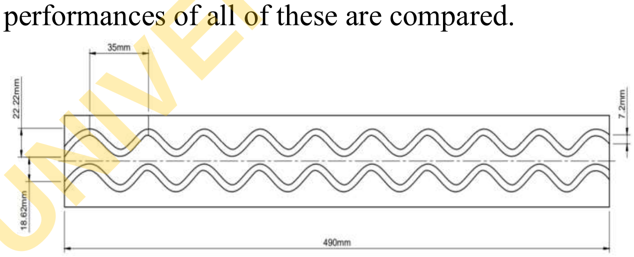

Figure 1: Continuous Spiral Arrangement

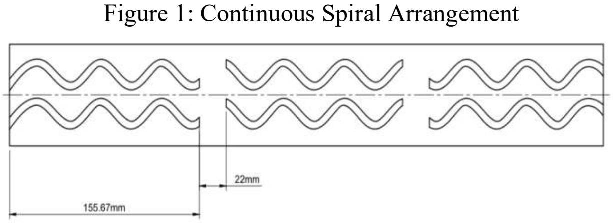

Figure 2: Discontinuous Spiral Arrangement

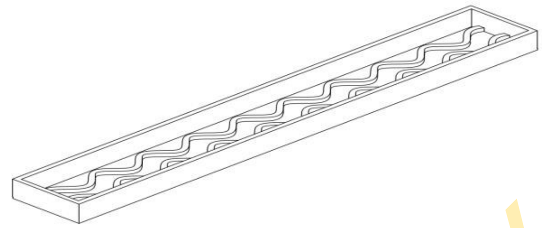

Figure 3: Spiral Duct Arrangement

A. *Setup, Procedure and Data Reduction*

A variable 10A blower, a variac, 1500W heating element, a digital temperature readout device, a pitot tube, and two digital manometers make up the entire arrangement. Figure 4 shows the experimental setup, which consists of a test section with a 50 mm inner square portion insulated by a 70 mm square section, both of which are 0.5m long. Fiberglass was employed as the insulating material, which was crushed into the pocket created by overlapping the two parts. Six ports with equal spacing make up the test portion. The first and last ports were utilized as differential pressure ports, while the middle four were employed to house thermocouples for sensing temperature across the channel's length. A flow straightener and a reducer make up the inlet construction. The flow straightener was made up of a honeycomb structure made up of patched straws.

The temperature readout device was linked to five thermocouple probes (0.18% uncertainty) in order to measure four temperature points in the test area as well as the exit temperature. In this study, the length of the test section was shortened to accommodate ducts, and the duct length was also cut to no more than ten times its hydraulic diameter in order to maintain the growing flow regime within each of the ducts.

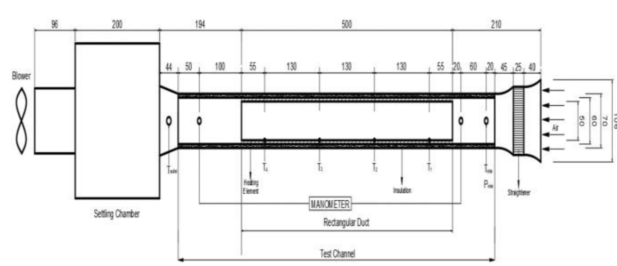

Fig. 4. The schematic arrangement of the experimental rig (dimensions in mm)

B. *Experimental Procedure*

A 1500 W spring type electric heating element is looped around the test channel (plain, continuous, and discontinuous spiral) and put into the fiberglass insulated test section to produce consistent heating. In the four ports of the test chamber, four Type-K thermocouples with an outer diameter of 0.5 mm were installed to measure the streamwise wall temperatures (T1, T2, T3, T4), as shown in Figure 3.16**.** Two tubes were inserted at the test section's intake and exit and connected to a digital manometer that measures the differential pressure (Pinlet, Poutlet). Another digital manometer was used to measure the pressure of the blower (Pblower) when it was engaged by the variable alternating current transformer at different speeds (VARIAC). The spring-type heating element was attached to the DC supply, while the terminals of the blower's coil were connected to the variac, which was subsequently connected to the DC power supply. As the blower's speed was changed, wall temperatures, pressure drop, and blower pressure were measured. The thermocouples are placed roughly 100 meters downstream of the test channel's exit to measure the mixed-mean temperature of the outflow.

Figure 5: Experimental setup showing points of measurements

C. Data Reduction

The gathered data was analyzed to identify the three channels' thermal-hydraulic properties. The amount of heat added to the fluid passing through the ducts is calculated.

$$Q = \dot{m} C_p (T_2 - T_1) \qquad 1.$$

The average heat transfer coefficient on each of the ducts' walls is derived using the following formula:

$$h = \frac{Q}{A_S \Delta T_{LM}} \qquad 2.$$

The averaged Nusselt number calculated from the heat transmission duct surfaces as;

$$Nu = \frac{hD_h}{k} \qquad 3.$$

where Dh is the duct's hydraulic channel diameter and k is is the fluid's thermal conductivity As a result, the log-mean temperature differential between the heating wall and the cool air flow, Tlm, is calculated [24].

$$\delta T_{lm} = \frac{(T_w - T_{in}) - (T_w - T_{out})}{\ln\left(\frac{T_w - T_{in}}{T_w - T_{out}}\right)} \qquad 4.$$

where Tw is the average temperature of the fluid-state surface of the test section. The friction factor is obtained from the pressure drop across the duct as:

$$f = \frac{\partial P}{[(\frac{L}{D_h})(\frac{1}{2}\rho u)u]} \qquad 5.$$

The dynamic pressure acquired as a result of the difference between the stagnation pressure, pstag, and static pressure, ps readings from the duct intake was used to calculate the duct inlet velocity.

$$V = \frac{\sqrt{2(P_{stag} - P)}}{\rho} \qquad 6.$$

The Reynolds number, on the other hand, was obtained from

$$\mathrm{Re} = \frac{\rho V L_p}{\mu} \qquad 7.$$

The heat transfer characteristics and friction factor for fully developed flow in a smooth duct serve as the baseline for comparing the performance of the test surfaces, allowing for a more efficient evaluation of the dimpled channels' thermal performance, which can be represented by the Performance Evaluation Criteria (PEC), a universal heat transfer unit evaluation parameter [25].

$$PEC = \frac{Nu / Nu_o}{(f / f_o)} \quad 8.$$

where Nu/Nu0 is the normalized Nusselt number, which is the ratio of the spiral channel's Nusselt number to that of the smooth channel, and f/f0 is the friction factor ratio of the spiral channel to that of the smooth channel.

D. Uncertainties

[26] provided the conventional uncertainty analysis used to quantify the uncertainties from the experiments. The uncertainty of a desired variable, R = f(x_i), is estimated from uncertainties, x_i, derived from a number of independent variables as follows:

$$\partial R = [\sum_{i=1}^{n} (\frac{\partial R}{\partial x_i} \partial x_i)^2] \quad 9.$$

While the uncertainty of a single measured value is estimated from bias error, Band precision error, and P of measurement, the uncertainty of a single measured value is computed from

$$\partial x_i = \sqrt{B_i^2 + P_i^2} \quad 10.$$

III. RESULTS

Nusselt number, Heat transfer enhancement,

Friction factors, Friction factor ratio, and overall

thermal performance of the test channel are

presented in table 6, 7, 8, 9 and 10 thus:

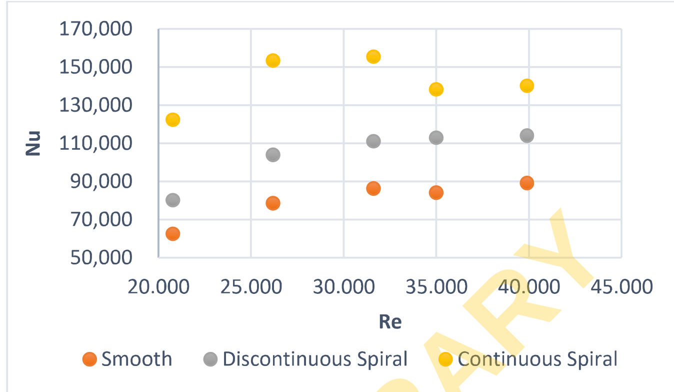

Figure 6: Average Nusselt numbers of the test channels

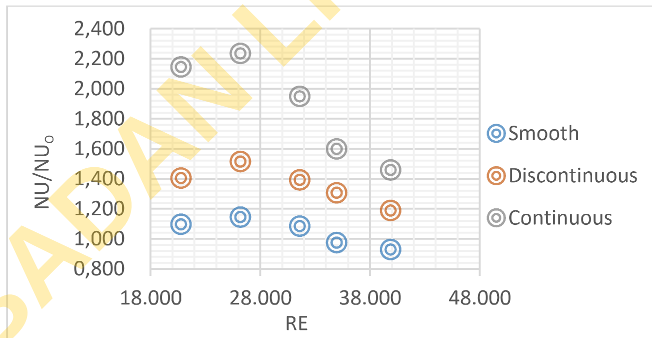

Figure 7: Heat transfer enhancement of test channels

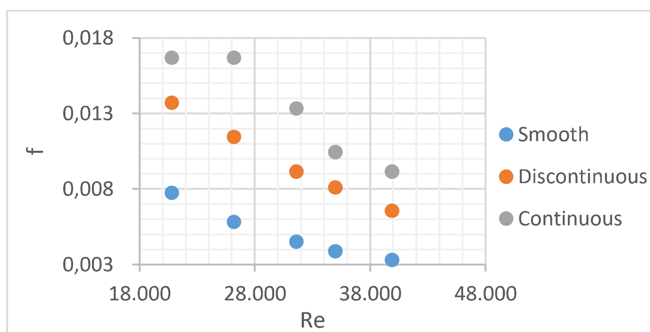

Figure 8: Friction factors of the test channels

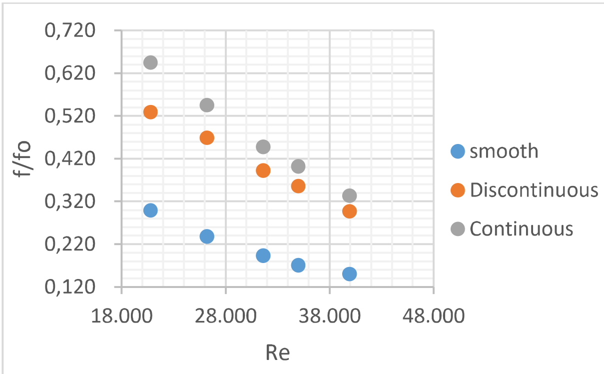

Figure 9: Friction factor ratio of the test channels

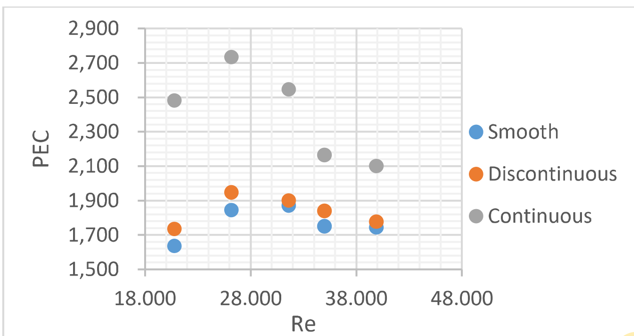

Figure 10: Overall thermal performance of test channel

IV. DISCUSSION

Figure 6 illustrated that the Reynolds number variation resulted in 29.2% and 74.4% higher Nusselt numbers for the discontinuous and continuous than the smooth channels. As shown in Figure 6, the Nusselt values for the continuous spiral channel range from 120 to 160, which is greater than the fully developed turbulent flow in a smooth channel. When compared to the channel with the continuous spiral arrangement, the channel with the discontinuous spiral arrangement has lower Nusselt values averaging 34.9%. As a result, of the three cases analyzed, the channel with the continuous spiral arrangement has the greatest Nusselt numbers. In Figure 7, the heat transfer enhancement (Nu/Nuo) of both the continuous spiral and discontinuous spiral appears to be 81.9% and 31.5% higher than the smooth test channel as Reynolds number is varied. The continuous spiral channel has higher Nu/Nuo of 18.3% than the discontinuous spiral channel. In general, the utilization of a new rib, as well as the comparison to a smooth surface, distinguishes this study. In Figure 8, the flow friction behavior for turbulent flow in test channels with Reynolds numbers ranging from 20,000 to 40,000 was determined. The friction factors of the smooth channel were also evaluated for the same Reynolds number in order to confirm the pressure loss within the equipment during experimentation. Figures 8 and 9 show the friction factors (f/f) and friction factor ratio (f/fo) of the test channels and the smooth channel. The friction factor drops as the Reynolds number grows, whereas f/fo, as shown in Figure 9, increases gradually but maintains nearly constant values at Re= 30,000. Figures 8 and 9 shows reasonable agreement with the literature, giving 91.7% and 91.3% as well as 160. 5% and 113.6% higher f/f as well as f/fo in the discontinuous and the continuous channels, respectively. Furthermore, in Figure 10, the heat transfer performance of the test channels could only be analyzed after incorporating the penalty effects related to friction losses in order to fully evaluate the thermal performance of the channels. The Figure reveals that the continuous spiral arrangement has the greatest thermal Performance value, which is greater than the discontinuous spiral by 32.6%. While the discontinuous spiral has a little better thermal performance than the Smooth channel by 4.4%.

V. CONCLUSION

Heat transmission, friction factor, and thermal enhancement factor of a smooth surface and ribbed surface (continuous and discontinuous spiral arrangement) system were investigated experimentally and compared at Reynolds number range of 20,000 to 40,000.

The following conclusions are drawn.

i. The continuous spiral rib arrangement was most effective in terms of heat transfer augmentation (81.8% and 18.33% more than the smooth and discontinuous configurations).
ii. For the smooth rib arrangement, the increase in Nusselt number is substantially more than the increase in friction factor.
iii. The continuous spiral rib arrangement channel was most effective in terms of the hydraulic performance i.e., relatively lower pressure loss compared to the smooth surface.
iv. The continuous spiral rib arrangement channel has the highest overall thermal

performance factor than the case of the embossed-teardrop protruded channel.

v. Results shows that the machined plate can compete with the special-made dimple if properly used in parts associated with essential need for swift heat transfer.

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