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CFD Analysis to Analyze Thermal Characteristics of a Heat Exchanger Handling Combination of Dimple Ribs

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ABSTRACT

It takes a lot of energy to have a good life. For living and working, humans nowadays rely on an abundant and constant source of power. Because energy is depleting at such a rapid rate, it has become important to use heat more efficiently, which necessitates that we preserve. As a result of the global energy crisis, many researchers have worked to improve the efficiency of thermal systems and reduce the size and thus energy consumption rates, which is one of the most critical problems due to the large and continuing increase in consumption, the increasing scarcity of energy resources, and the high cost. A 3-dimensional numerical (3-D) simulation was used to evaluate the thermal properties of a heat exchanger managing a combination of dimple ribs. Handling air flow velocity through the channel was varied from 3.97 to 5.80 m/s. is the subject of this study. The heat transfer physiognomies of a heat exchanger managing a combination of dimple ribs were studied using the simulation tool ANSYS 19.2. The numerical results showed that the combination of dimple ribs enhanced heat transfer significantly more than the dimple. In comparison, the average Nusselt number of a combination of dimple ribs is 4.28 percent higher than that of a channel with dimpled Plate.

KEYWORDS: *Surface enhancement, dimples and ribs, heat transfer characteristic, friction factor, Nusselt number, thermal performance, CFD*

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I. INTRODUCTION

Many researchers have worked to improve the efficiency of thermal systems and reduce the size and thus energy consumption rates as a result of the global energy crisis, which is one of the most critical problems due to the large and continuous increase in consumption and the increasing shortage of energy resources as well as the high cost.

Heat transfer enhancement is the process of applying various strategies to improve a system's heat transfer rate and thermo hydraulic performance. Heat transfer enhancement methods are used to improve heat transfer without significantly affecting the overall realisation of systems, and they cover a wide range of applications where heat exchangers are used for functions such as air conditioning, refrigeration, central heating systems, cooling automotive components, and many other applications in the chemical industry.

Heat transfer improvement techniques are divided into three categories: passive, active, and compound procedures. Passive methods, on the other hand, do not require any additional energy to improve the thermohydraulic performance of the system. Active methods require external power to input the process; however, active methods do not require any additional energy to improve the thermohydraulic performance of the system.

External power is not required in passive approaches; rather, the geometry or surface of the flow channel is altered to improve the thermohydraulic performance of the systems. Inserts, ribs, and rough surfaces are used to increase fluid mixing and turbulence in the flow, resulting in an increase in total heat transfer rate. Passive heat transfer enhancement approaches also offer several advantages over other heat transfer enhancement techniques, such as cheap cost, ease of manufacture, and installation.

Rib turbulators can effectively improve heat transmission, but they usually come at a high cost in terms of pressure loss. Dimple methods have recently become popular because they improve heat transmission while posing a minimal pressure penalty. Several researchers have seen heat transfer amplification when dimple surfaces are combined with nanofluids, but the primary worry is an increase in the system's friction factor, which is also managed within acceptable limits.

II. LITERATURE REVIEW

Several studies have reported increased heat transfer when dimple surfaces are combined with nanofluids, however the primary worry is an increase in the system's friction factor, which is managed within acceptable ranges. The following is a summary of research on dimpled surfaces, nanofluids, and dimple surfaces combined with nanofluids.

2.1. Previous Work

Several studies have reported on the formulation of nanofluids using various types of nanoparticles, as well as their capacity to conduct convective heat transfer.

Maxwell (1873) was the first to report the thermal conductivity enrichment of conventional fluids with the challenges of sedimentation, clogging and erosion in flow tracks [1]. Afterward, **Masuda et al. (1993)** examined the thermal conductivity enhancement with the addition of micro-sized solid particles into the base fluid (single phase), but also encountered the same problems of sedimentation, enhanced pumping power, erosion and clogging [2]. **Hamilton-Crosses (1962)** also contributed by extending the work of Maxwell and provided the more accurate model to predict the thermophysical properties of the particles suspended fluids [3].

In 1995, the work of **Choi** revolutionized the field of heat carrying fluids when first time fabricated the nanofluids that exhibited enhanced thermal transport properties with better stability in comparison of fluids containing the milli and micro sized solid particles [4].

With this invention, researchers started to investigate the nanofluids with great interest.

Pak and Cho (1998) conducted heat transfer and friction factor experiments for Al_2O_3/water and TiO_2/water nanofluids in the Reynolds number range from 104 to 105 and the particle concentration ranging from 0% to 3% and observed heat transfer enhancement compared to the base fluid (water); they also propose newly-developed Nusselt number correlation [5].

Later on, **Xuan and Li (2001)** used Cu/water and Cu/transformer oil nanofluids and observed heat transfer enhancements as compared to the base fluids. In another study, **Xuan and Li(2002)** observed heat transfer enhancement of 60% for 2.0% volume concentration of Cu/water nanofluid flowing in a tube at a Reynolds number of 25000 and they report separated Nusselt number correlations for laminar and turbulent flow, respectively [6,7].

Wen and Din (2004) conducted heat transfer experiments for Al_2O_3/water nanofluid in a tube under laminar flow and they observed heat transfer enhancement of 47% at 1.6% volume fraction as compared to the base fluid (water) [8].

Heris et al. (2007) also used Al_2O_3/water nanofluids in a tube under laminar flow and observed heat transfer enhancement using constant wall temperature boundary conditions [9].

Williams et al. (2008) reported convective heat transfer enhancement with alumina/water and zirconia/water nanofluids flow in a horizontal tube under turbulent flow [10].

Duangthongsuk and Wongwises (2010) found heat transfer enhancement of 20% and 32% for 1.0% vol of TiO_2/water nanofluid flowing in a tube at Reynolds numbers of 3000-18000, respectively [11].

Moraveji et al. (2011) simulated water-Al_2O_3 nanofluid through a tube under a constant heat flux. They found that the heat transfer coefficient rises by increasing the nanoparticle concentration and Reynolds number. Furthermore, the heat transfer coefficient increases by particle diameter reduction [12].

Ghozatloo et al. (2014) obtained heat transfer enhancement of 35.6% at a temperature of 38 °C for 0.1 wt% of graphene/water nanofluids flow in a tube under laminar flow [13].

Sundar et al. (2012) found heat transfer enhancement of 30.96% with a pumping penalty of 10.01% for 0.6% vol of Fe_3O_4/water nanofluid flow in a tube at a Reynolds number of 22000 [14].

Sundar et al. (2014) observed heat transfer enhancement of 39.18% with a pumping penalty of 19.12% for 0.6% vol of Ni/water nanofluid flow in a tube at a Reynolds number of 22000 [15].

Delavari et al. (2014) numerically simulated the heat transfer in a flat tube of a car radiator at laminar and turbulent regimes. They showed the ability of CFD to simulate the flow field and temperature distribution profile well and reported an increment of Nusselt number with increasing the nanoparticle concentration [16].

Chandrasekhar et al. (2017) experimentally investigated and theoretically validated the behavior of Al_2O_3/water nanofluid that was prepared by chemical precipitation method. For their investigation, Al_2O_3/water at different volume concentrations was studied. They concluded that the increase in viscosity of the nanofluid is higher than that of the effective thermal conductivity. Although both viscosity and thermal conductivity increases as the volume concentration is increased, increase in viscosity predominate the increase in thermal conductivity. Also various other theoretical models were also proposed in their paper [17].

Hady et al. (2017) experimentally investigated the performance on the effect of alumina water (Al_2O_3/H_2O) nanofluid in a chilled water air conditioning unit. They made use of various concentrations ranging from 0.1-1 wt % and the nanofluid was supplied at different flow rates. Their results showed that less time was required to achieve desired chilled fluid temperature as compared to pure water. Also reported was a lesser consumption of power which showed an increase in the cooling capacity of the unit. Moreover the COP of the unit was enhanced by 5 % at a volume concentration of 0.1 %, and an increase of 17 % at a volume concentration of 1 % respectively [18].

Rohit S. Khedkar et al. (2017) experimental study on concentric tube heat exchanger for water to nanofluids heat transfer with various concentrations of nanoparticles in to base fluids and application of nanofluids as working fluid. Overall heat transfer coefficient was experimentally determined for a fixed heat transfer surface area with different volume fraction of Al_2O_3 nanoparticles in to base fluids and results were compared with pure water. It observed that, 3 % nanofluids shown optimum performance with overall heat transfer coefficient 16% higher than water [19].

Han et al. (2017) in double tube heat exchanger flow turbulent and counter examine the enhancement of heat transfer by using nanoparticles aluminum oxide in water. They examine the rate of heat transfer with 0.25% and 0.5% by volume concentration at various inlet temperatures. In pipe flow the Reynold's number should be greater than 4000 for turbulent flow, as we know with higher turbulence the rate of heat transfer is high so with the various concentration and inlet temperature they also examine the rate of heat transfer at various Reynold's number i.e., 20000, 30000, 40000, 50000, and 60000. After the experiment they analyze that by using different concentration of Al_2O_3 that include 0.25% and 0.5% by volume concentration with Reynold's number varies from 20000 to 60000 maximum increase in heat transfer coefficient is about 9.7% and 19.6% respectively at 40^0C, and with same volumetric concentration and also Reynold's number varies from 20000 to 60000 the maximum increase in heat transfer coefficient is about 15% and 29% for volume concentration 0.25% and 0.5% respectively at 50^0C. Comparing the result at 40^0C and 50^0 with the volume concentration of 0.25% and 0.50%, the increase in heat transfer coefficient is about 5.3% and 9.4% respectively. The main conclusion is that at same nanoparticles concentration we can increase the rate of heat transfer with the increase in inlet temperature of nanofluid, which shows that nanofluid dependency on temperature. Nusselt number is also deal with heat transfer, by using nanofluid the Nusselt number also increases about 8.5% and 17% at the volumetric concentration of 0.25% and 0.50% respectively [20].

Akyürek et al. (2018) experimentally investigated the effects of Al_2O_3/Water nanofluids at various concentrations in a concentric tube heat exchanger having a turbulator inside the inner tube. Comparisons were done with and without nanofluid in the system as well as with and without turbulators in the system. Results were drawn and a number of heat transfer parameters were calculated on the basis of observed results. Various heat characteristics such as change in Nusselt number and viscosity with respect to Reynolds number, behaviours of nanofluid at various volume concentrations, changes in heat transfer coefficient, effect of the difference of pitch of turbulators on the heat transfer of nanofluid etc. were studied. They concluded that there exists a relationship between the varying pitches and the turbulence in the flow caused i.e. when the pitch is less there is more turbulence and vice versa [21].

Dimpled surfaces are recommended as a typical heat transfer enhancement passive approach because to their light weight, low pressure drops values, simplicity of production, and cheap maintenance costs. The potential of dimple surface approach in diverse thermal systems has been investigated via several experiments and computer analyses.

Kanokjaruvijit et al. (2005) increased heat transmission in cooling tubes with dimpled (concavity imprinted) surface area can be achieved in comparison to sticking out ribs. Heat exchangers and other hot-area elements might benefit from it (nozzle, blade, combustor lining, etc.). The friction factor and heat transfer coefficients were established experimentally in concavities (dimples) on one wall of rectangle-shaped channels surface. The thermal efficiency of dimple surfaces was higher than that of flat surfaces. The kind with continuous ribs, demonstrating that improved heat transmission can be achieved. Concavities were used to do this while keeping a low pressure drop [22].

Moon et al. (2012) A transient big band liquid crystal approach was utilised to investigate an 8 X 8 jet array impingement on a staggered dimples array at Reynolds number 11,500. The distance between the perforated plate and the target plate was estimated for jet diameters of 2, 4, and 8. When hemispherical and cusped elliptical dimple geometries were examined, it was revealed that the outcomes of hemispherical and cusped elliptical dimples behaved similarly. However, in terms of cost, manufacture, and pressure loss, the hemispheric form seemed preferred [23].

Griffith et al. (2015) The rate of heat transfer in rotating rectangular cooling channels was explored, and it was observed that channel orientation had a different impact, with the trailing-edge channel growing in Nusselt at the same rate as the orthogonal channel. Furthermore, the dimpled channel performs similarly to a 45-degree angled rib channel in terms of spanwise heat transmission, but with less variation [24].

Lauffer et al. (2017) Heat transfer studies were carried out utilising heater foils and a steady-state apparatus with liquid crystals on a rectangular dimpled channel with a 6-aspect ratio. Localized rib designs were revealed to boost heat transmission in these important locations while decreasing strain dramatically [25].

Chang et al. (2019) For four sets of dimple fin channels with rectangular cross sections, a channel aspect ratio (AR) of 6, and three varied fin length (L) to channel hyd, heat transmission and friction factor were evaluated. Using Reynolds numbers ranging from 1500 to 11,000 and Re on heat transfer upon channel with dimpled fins, Da (d), ratios (L=d) 8.9, 6.2, and 3.5, respectively. For both Re and L=d, the convex–convex dimpled fin channel showed the maximum Heat Transfer Enhancement [26].

Suresh et al. (2011) The heat transfer enhancement using the combination of helically dimpled tube and CuO-water nanofluid was investigated, and it was discovered that there was no increase in friction factor, and the thermohydraulic efficiency was 10% greater than plain tube utilizing the combination [27].

Rajabi et al. (2019) has used the nanofluids to run numerical simulations on heated walls of microchannel with spherical dimples and found that at a certain dimple depth, every 2% increase in volume fraction improves heat transmission by 2% [28].

Li et al. (2014) has analyzed that heat transfer rises with rise in concentration in microchannel with dimple + protrusion using Al_2O_3-water nanofluid [29].

Firoozi et al. (2019) explored the effect of Al_2O_3-water nanofluid on variety of dimple configuration, In the case of water flowing through the tubes, the improved tubes provide better average output as pitch is decreased, tube height is increased, and the filling angle is increased [30].

Josephine et al. (2019) the experimental analysis of the influence of dimpled layouts on flow and heat transfer properties is presented in this paper. Three plate surfaces (smooth, equally distributed spherical dimples, and irregularly distributed spherical dimples) were created and put in a channel one after the other. The average Nusselt number increases with the Reynolds number as a result of heat interaction with the airflow. Over the smooth channel, the equally and unevenly dimpled plate channels experienced a 75.7 percent and 91.8 percent increase in Nusselt number, respectively. The flow friction coefficients of the uniformly and unevenly dimpled plate channels were only 0.59 percent and 0.67 percent higher than those of the smooth plate channel, respectively [31].

2.2. Problem Formulation

Several researchers have seen heat transfer amplification when dimple surfaces are used, but the primary worry is an increase in the system's friction factor, which is also managed within acceptable limits. Therefore this investigation deals with the thermal and flow behavior of heat exchanger handling combination of dimple ribs to amplifies the heat transfer rate with very less increase in friction factor.

Therefore, a 3-dimensional numerical (3-D) simulation was used to evaluate the thermal properties of a heat exchanger managing a combination of dimple ribs. Handling air flow velocity through the channel was varied from 3.97 to 5.80 m/s. is the subject of this study. The heat transfer physiognomies of a heat exchanger managing a combination of dimple ribs were studied using the simulation tool ANSYS 19.2.

III. GEOMETRY SETUP AND MODELLING

The geometry of the Plate Heat Exchanger used in the simulation analysis was taken from one of the research scholar's **Josephine et al. (2019)** with exact dimensions, Plate was made of mild steel of 220mm in length, 140mm width and 10mm thickness. Eleven rows of spherical dimples with print diameter of 15 mm and depth of 5 mm were created on the upper side of the plates in the stream-wise direction having a longitudinal pitch of 19.5 mm. In the proposed work, eleven rows of spherical dimples with print diameter of 15 mm and depth of 5 mm were created on the upper side of the plate and having rib on the lower side of the plate. The model was created using the ANSYS (fluent) workbench 19.2 program.

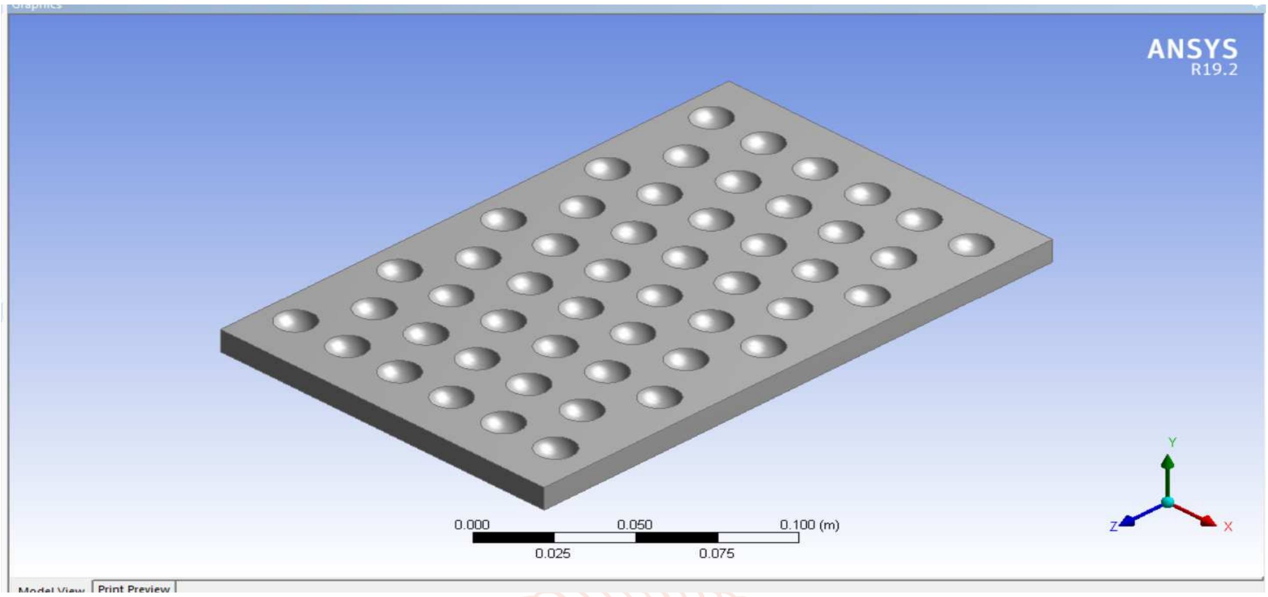

Figure 1. Geometry of the Plate Heat Exchanger with spherical dimples created on the upper side of the plate (Josephine et al. (2019).

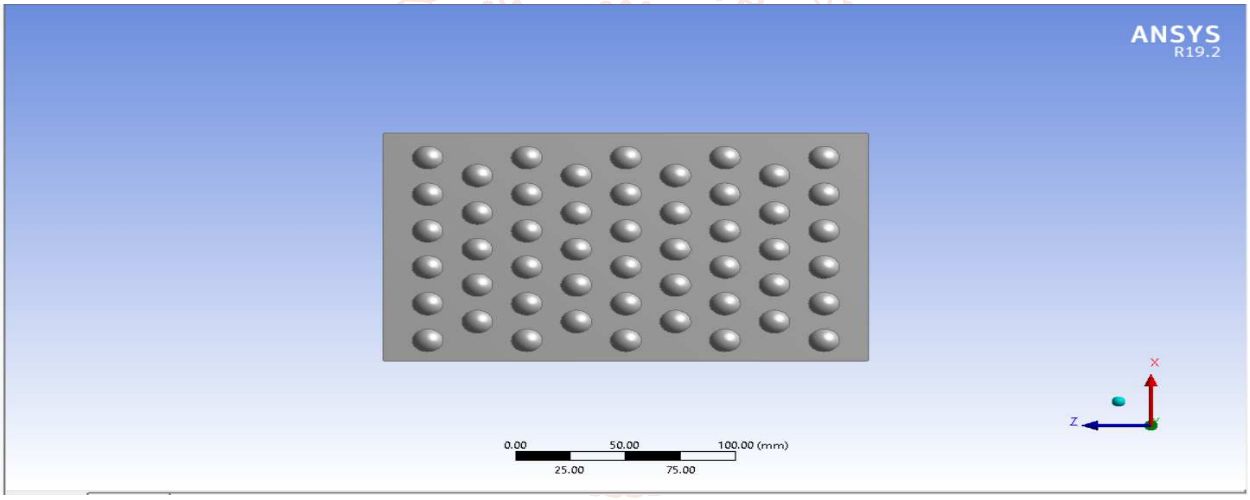

Figure 2. Geometry of the Plate Heat Exchanger with spherical dimples created on the upper and having rib on the lower side of the plate (Proposed work).

A three-dimensional discreet PFCHE model with fin was built in the pre-processor stage of ANSYS FLUENT R19.2. Although grid type is related to the output of simulation, the whole structure is discretized in the finite volume of the tetrahedral grids of Quad core such that thermal properties are reliably calculated with fine grids.

Table 1 Mesh details

The applied design	Number of nodes and elements
Josephine et al. (2019 Work	21157 and 99258
Proposed Work	30411 and 145212

To compute, the Fluent 19.2 program was used. A finite element method was used in experiments to distinguish the governing equations. The researchers used a simpler algorithm for this convective term, and the second order upwind method was used to relate pressure and velocity calculations.

Turbulence was solved using a regular k-epsilon equation in conjunction with flow and energy equations.

A 3-Dimensional steady state turbulent flow method was used in the computational simulation. To solve the dilemma, the governing equations for heat flow and conjugate transfer were tuned to the simulation setup's conditions. The following are the governing equations for mass, momentum, energy, turbulent kinetic energy, and turbulent energy dissipation:

Mass:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0$$

Momentum:

$$\frac{\partial(\rho u_i u_k)}{\partial x_i} = \frac{\partial\left(\mu \frac{\partial u_k}{\partial x_i}\right)}{\partial x_i} - \frac{\partial p}{\partial x_k}$$

Energy Equation:

$$\frac{\partial(\rho u_i t)}{\partial x_i} = \frac{\partial\left(\frac{K}{C_p}\frac{\partial t}{\partial x_i}\right)}{\partial x_i}$$

The renormalization-group (RNG) k-ε model was used in this work because it can provide better estimates of close wall flows and flows with high streamline curvature. Additionally, the thermal effect parameter was chosen in improved wall treatment panel transport equations for the RNG k-ε model in common form.

Turbulent kinetic energy:

$$\frac{\partial(\rho K)}{\partial t} + \frac{\partial(\rho u_i K)}{\partial x_i} = \frac{\partial\left(\alpha_k \mu_{eff} \frac{\partial k}{\partial x_j}\right)}{\partial x_j} + G_k + \rho\varepsilon$$

Turbulent energy dissipation:

$$\frac{\partial(\rho \epsilon)}{\partial t} + \frac{\partial(\rho u_i \varepsilon)}{\partial x_i} = \frac{\partial\left(\alpha_\varepsilon \mu_{eff} \frac{\partial \varepsilon}{\partial x_j}\right)}{\partial x_j} + C_{1\varepsilon}\frac{\varepsilon}{k}G_k + C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}$$

Where, G_k represents the generation of turbulent kinetic energy due to the mean velocity gradient and

$$\mu_{eff} = \mu + \mu_t$$

$$\mu_t = C_u \rho \frac{k^2}{\varepsilon}$$

The empirical constants for the RNG k-ε model are allotted as following:

$C_{1\varepsilon} = 1.42$

$C_{2\varepsilon} = 1.68$

$\alpha_\varepsilon = 1.39$

The velocity of the input air flow through the channel was varied between 3.97 and 5.80 m/s. The discretized flow domain was configured with suitable boundary conditions. Inlets were assigned input air flow boundary conditions, while outlets were assigned pressure outlet boundary conditions. Plate is heated using 400 W heating element.

IV. RESULTS AND DISCUSSIONS

4.1. CFD Validation

Computational models offer detailed and well-founded outcomes. However, numerical representations of physical measurements must be validated. The plate model with dimple is done with the aim of validating the numerical model, and the results are correlated with data from Josephine et al. (2019) who studied the investigation of flow and heat transfer in a channel with dimpled plate.

4.1.1. Contours for channel with dimpled plate at air flow velocity = 3.97 m/s

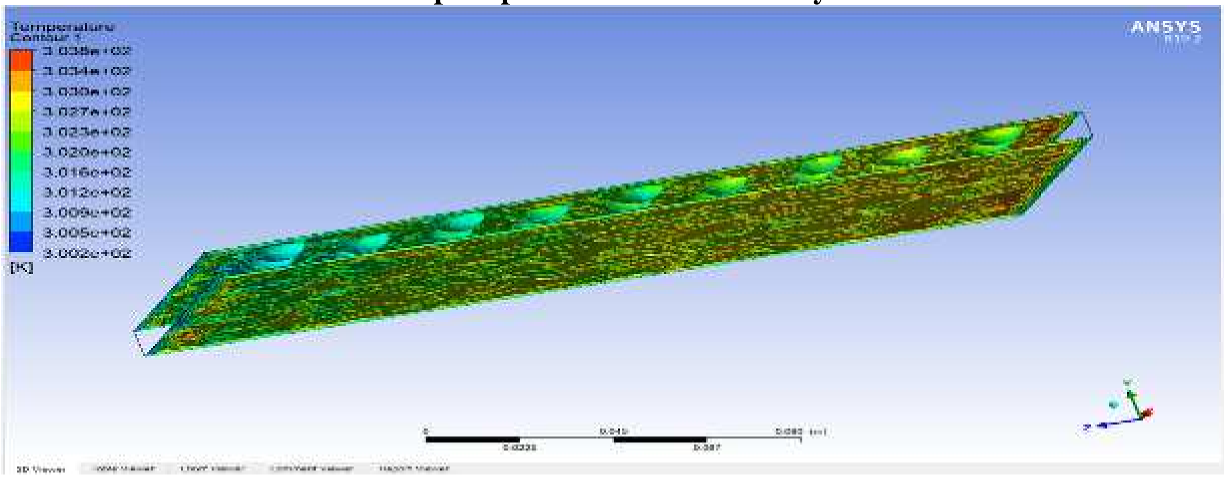

Figure 3. Temperature contour view 2 at air flow velocity = 3.97 m/s for channel with dimpled Plate.

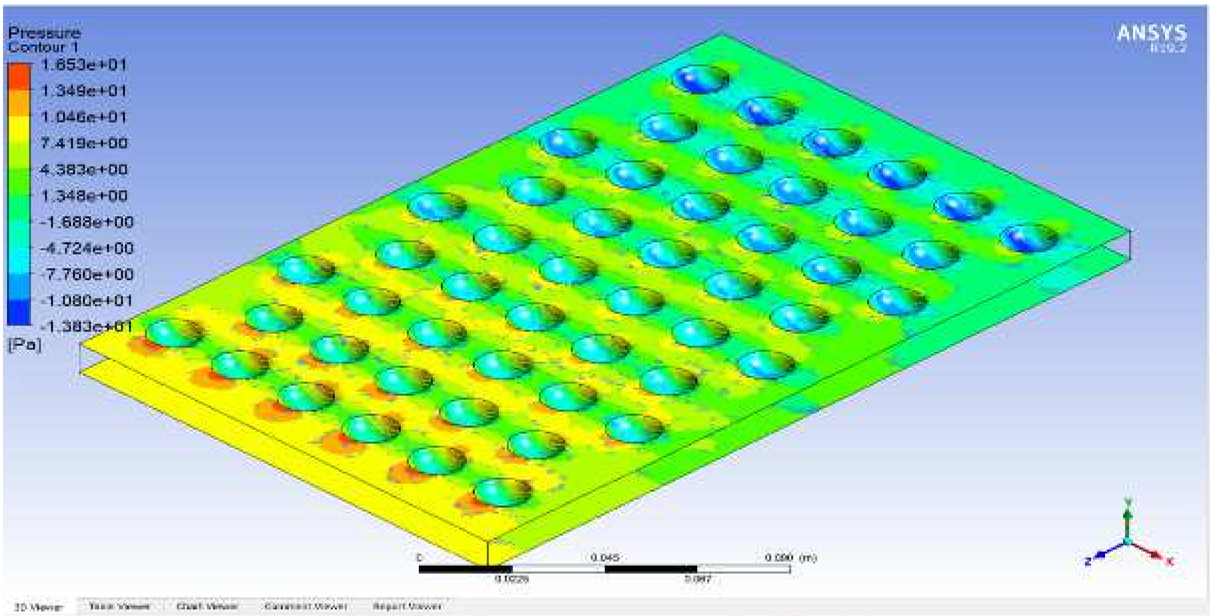

Figure 4. Pressure contour at air flow velocity = 3.97 m/s for channel with dimpled Plate.

To compute the Nusselt number value at different flow rate of inlet, CFD measurements were used. The Nusselt number of the CFD modeling have been compare to the values of the Josephine et al. (2019) measurements.

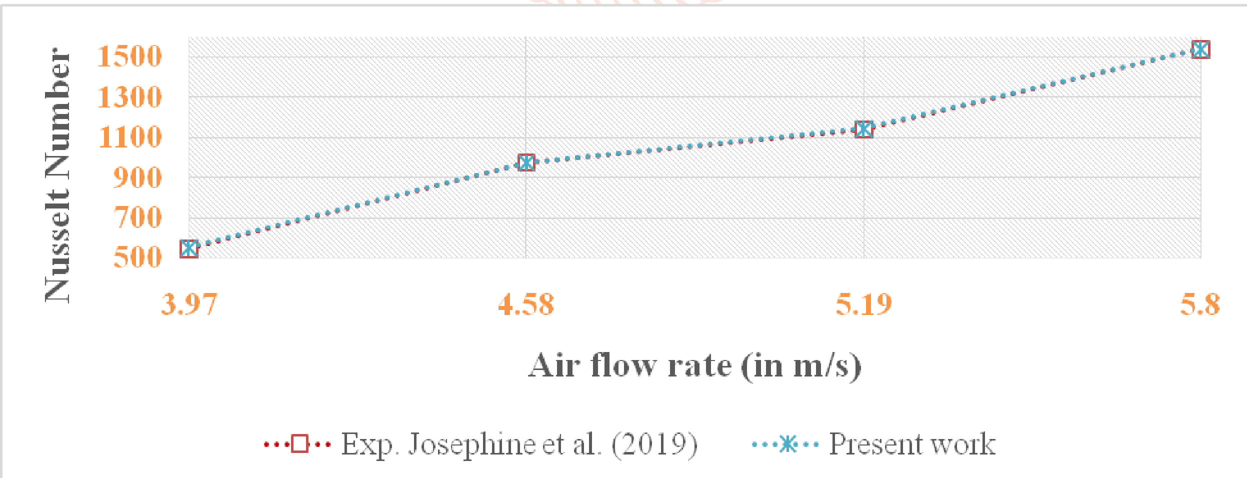

Figure 5. CFD Nusselt number comparison with experimental result from Josephine et al. (2019).

The experiment and the numerical results of the Nusselt number values agree well.As a result, we should state that the corrugated channel CFD model is right here.

4.2. Effect of spherical dimples on the upper and having rib on the lower side of the plate

The velocity of the input air flow through the channel was varied between 3.97 and 5.80 m/s. Plate is heated using 400 W heating element.

4.2.1. Contours for channel with spherical dimples on the upper and having rib on the lower side at air flow velocity = 3.97 m/s

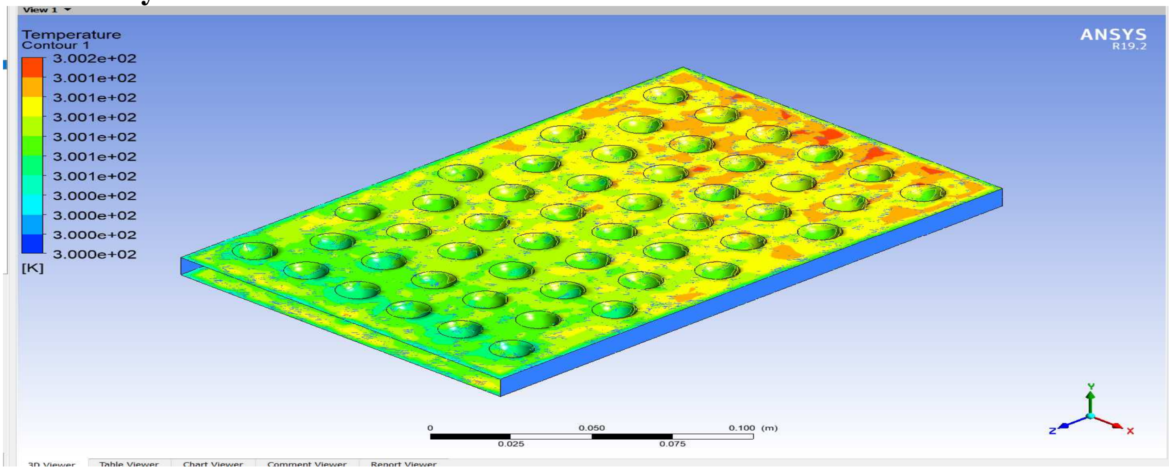

Figure 6. Temperature contour channel with spherical dimples on the upper and having rib on the lower side at air flow velocity = 3.97 m/s.

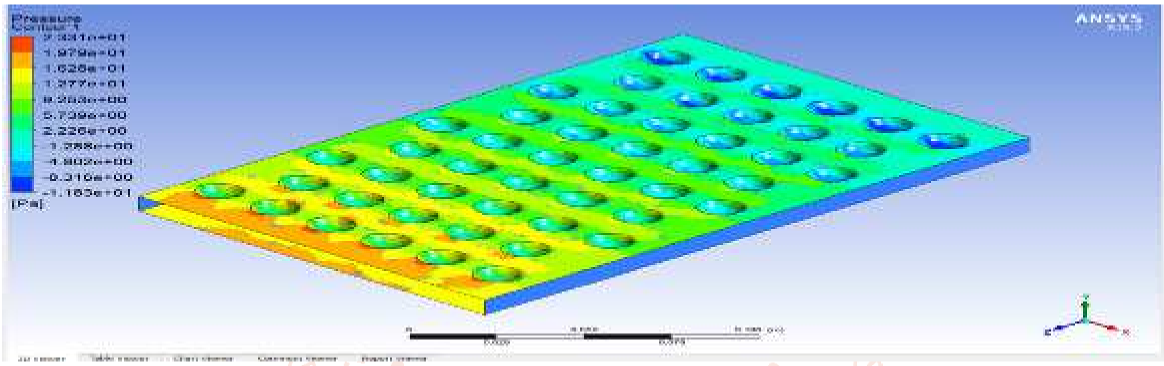

Figure 7. Pressure contour channel with spherical dimples on the upper and having rib on the lower side at air flow velocity = 3.97 m/s.

4.2.2. Contours for channel with spherical dimples on the upper and having rib on the lower side at air flow velocity = 5.80 m/s

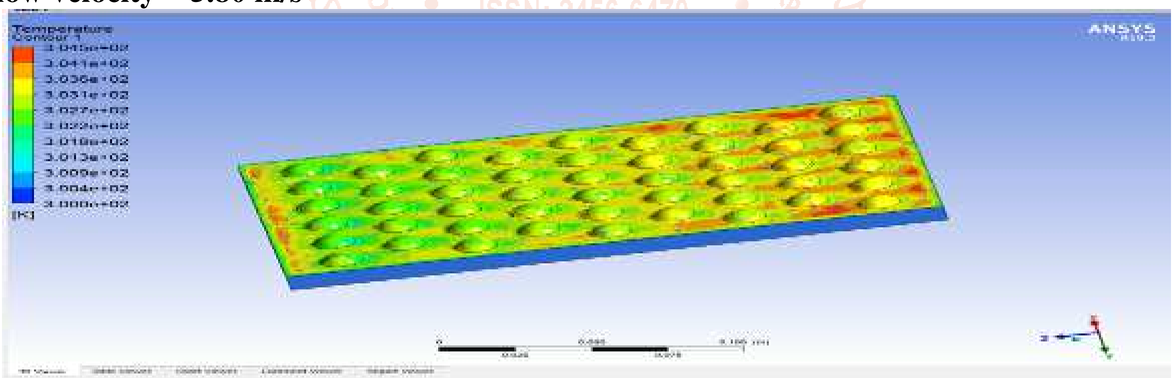

Figure 8. Temperature contour channel with spherical dimples on the upper and having rib on the lower side at air flow velocity = 5.80 m/s.

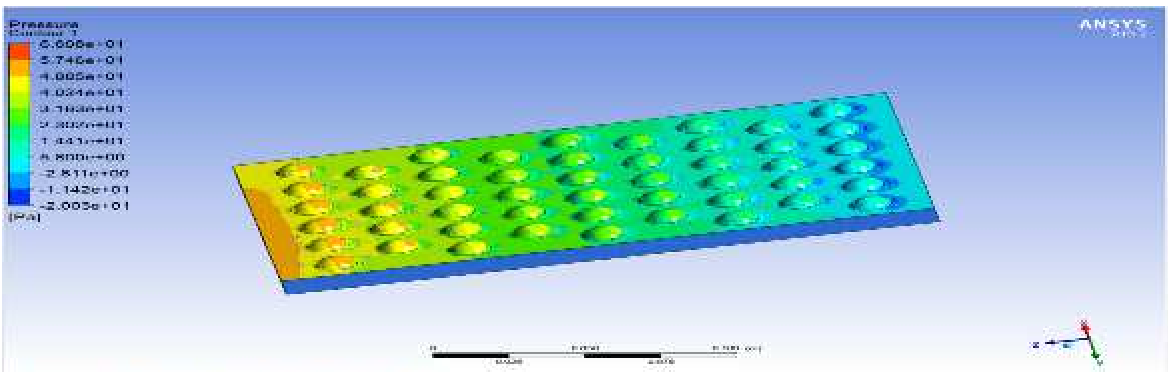

Figure 9. Pressure contour channel with spherical dimples on the upper and having rib on the lower side at air flow velocity = 5.80 m/s.

4.3. Comparison of Nusselt number and friction factor for previous work and proposed work

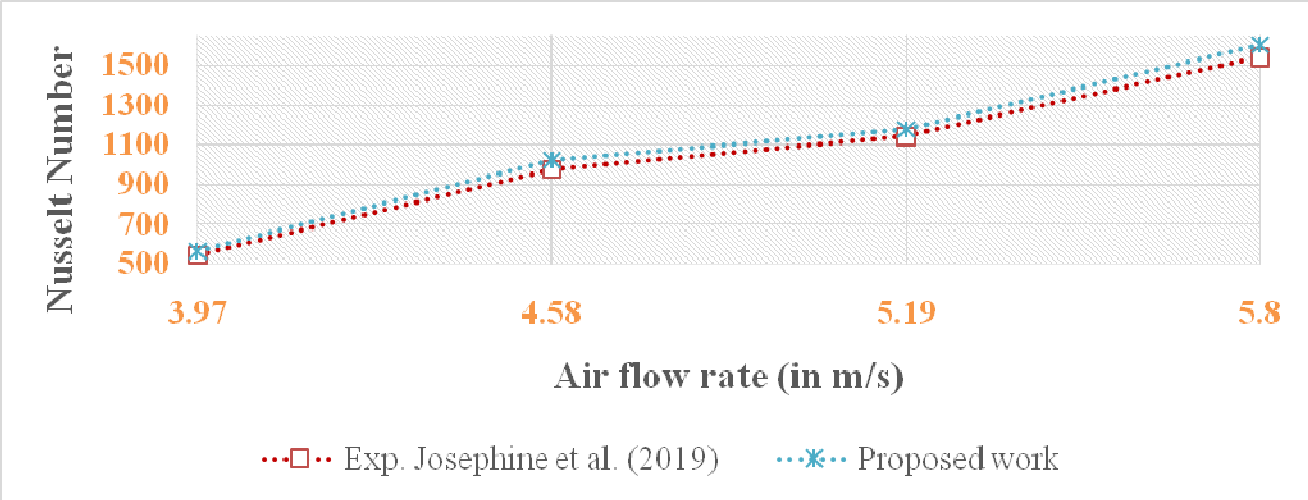

Figure 10. Comparison of Nusselt number for previous work and proposed work.

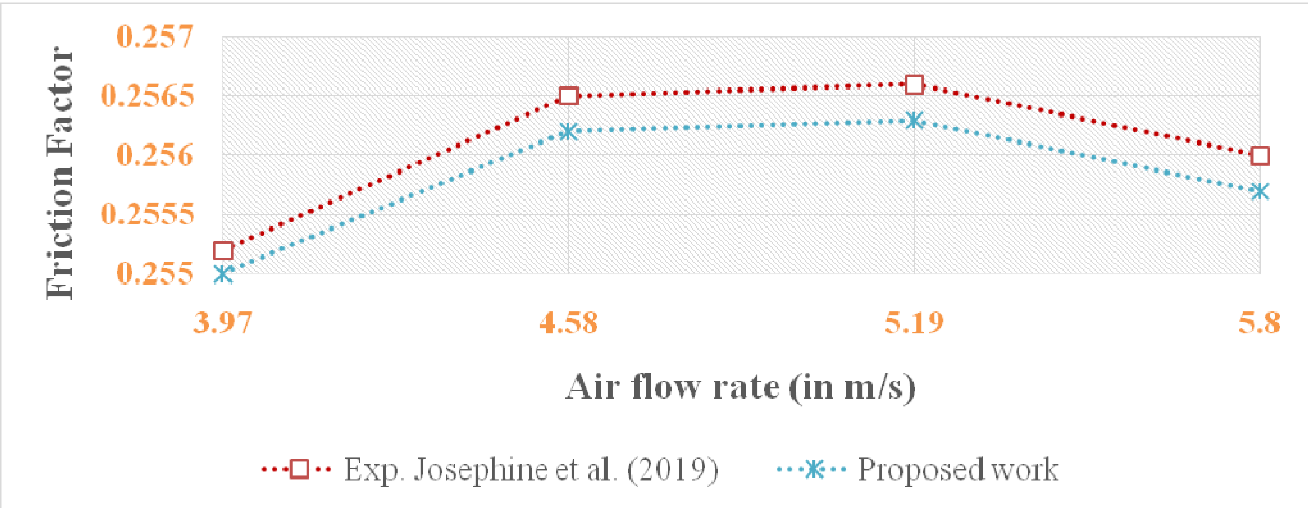

Figure 11. Comparison of friction factor for previous work and proposed work.

V. CONCLUSIONS

The present study explores the ability to predict flow and heat transfer characteristics of a commercial CFD code on a Plate Heat Exchanger with spherical dimples created on the upper and having rib on the lower side of the plate, air is used for the functioning fluid, and the flow through the channel was varied between 3.97 and 5.80 m/s. A standardized heat stream of 400 W was added.

The following are the study's findings:

- The numerical results showed that the combination of dimple ribs enhanced heat transfer significantly more than the dimple.
- In comparison, the average Nusselt number of a combination of dimple ribs is 4.28 percent higher than that of a channel with dimpled Plate.
- In comparison, the friction factor of a combination of dimple ribs is 8.12 percent lower than that of a channel with dimpled Plate

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