ABSTRACT
A Computational Fluid Dynamics (CFD) study of heat enhancement in helically grooved tubes was carried out by using a 3-dimensional simulation with the STARCCM+ simulation package software. The k-ε model selected for turbulent flow simulation and the governing equations were solved by using the finite volume method. Geometric models of the current study include 3 rectangular grooved tubes with different groove width (w) and depth (e) which varies from 0.2 mm to 0.6 mm for the same tube length of 2.0m and diameter of 7.1 mm. The simulations were performed in the Reynolds number (Re) range of 4000-10000 with a uniform wall heat flux of 3150 w/m² applied as a boundary condition on the surface of each tube. The purpose of this research is to investigate the effect of different groove dimensions on the thermal performance and pressure drop of water inside the grooved tubes and clarify the structural nature of the flow in regards to flow swirl and turbulent kinetic energy distributions. It was found that the highest performance belongs to the groove with these dimensions (w=0.2 mm and e=0.2 mm) which was considered for further study. Then, for these same groove dimensions four pitch size to tube diameter (p/D) ratios ranging from 1 to 18 were simulated for the same 2.0 m length tube. The results for Nusselt number (Nu) and friction factor (f) showed that by increasing the (p/D) ratio both the Nu numbers and the friction factors (f) values decrease. With a smaller pitch length (p) the turbulence intensity generated by the internal groove was also found to increase. The physical behavior of the turbulent flow and heat transfer characteristics were observed by contour plots which showed an increasing swirl flow and turbulent kinetic energy as p/D decreases. With an increase of the Nu number for smaller p/D ratio, a penalty of a higher pressure drop was obtained. The results were validated with a previous experimental work and the average error between the experimental and CFD Nu numbers and f were 13% and 8% respectively. A higher level of turbulent kinetic energy is observed near the grooves, as compared to the smooth areas of the pipe surface away from the grooves, which are expected to lead to higher levels of heat transfer. The effect of pitch length (p) on the flow pattern were plotted by streamlines along the tubes, by decreasing the pitch size (p/D ratio) an increase in the swirl is noticed as evidenced by the plots of the path lines. Finally, empirical correlations for Nusselt number and friction factor were provided as a function of p/D and Re number. This study indicates that the incorporation of the internal groove, of particular dimensions, can lead to an improvement of performance in heat exchanger devices. A limited variation of the groove dimensions was conducted and it was found that the values of Nu and f do not improve with an increase of (w) nor with that of (e) from 0.2-0.6 mm.

INTRODUCTION
Enhancement techniques are generally used in heat exchanger devices to increase the thermal performance in different engineering applications such as HVAC systems (heating, ventilating and air conditioning), chemical and power plants, refrigeration systems, petroleum plants, etc [1-4]. Increasing the heat transfer by modifying the surface and geometry design of the heat exchangers or adding inserts to promoting the turbulent flow called passive techniques which do not need external power. Internally grooved tubes categorized as passive techniques and widely used in commercial applications [5]. Most of the heat exchangers have the potential to be considered for heat enhancement, however, each potential application should be tested to see if their thermal performance is practical [6]. Aroonrat et al. [7] studied the turbulent flow in stainless steel pipes with an internal groove. They experimentally investigated the effect of different helical pitch sizes on the Nu number and f and their data showed a significant increase in heat transfer in grooved tube compare to the smooth tube. Zheng et al [8] numerically studied novel grooved design in circular tubes. They evaluate the effect of inclination angle on thermal performance of discrete grooves. Maximum thermal
performance belongs to the angle of 30 in tubes with lower Re number [8]. Bilen et al. [9] experimentally investigated the effect of groove geometry on heat transfer properties of turbulent flow in transversely placed grooved tubes with different shapes (circular, trapezoidal, and rectangular). The highest value for heat-transfer enhancement was reported up to 63% for tubes having circular grooves, compared to smooth tubes. A correlation equation was then developed experimentally for the Nu number and friction factor for each tube. Thermal performance ($\eta$) for all grooved pipes in that study was in the range of 1.13-1.28. In another study by Celik et al the performance of refrigerant (R-600a) in a regular tube is experimentally compared with a grooved tube in refrigerating systems. In general, heat transfer performance coefficient and efficiency of the refrigeration system were higher for the grooved tube compared to the regular tube [3]. In 2013, Rahman, Zhen, and Kadir [13] carried out a computational fluid dynamics (CFD) analysis of copper tube with multi-start inner grooves with a refrigerant (R22) as an internal flow. The results indicated the enhancement was higher for grooved tube compare with smooth tube and also the outcomes were verified by experimental data [15]. In 2015, Liu, Xie and Simon [14] studied the heat transfer performance of turbulent flow (Re: 10,000 - 25,000) in a square duct with cylindrical grooves. Their objective was to find a desirable design for a better enhancement rate and minimum pressure drop penalties. The purpose of the current research was to show the thermal performance of the grooves on the turbulent flow by changing the groove dimensions ($e$ and $w$) and $p/D$ ratio. Also, providing the empirical equation for heat transfer characteristics; Nu number and $f$.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$C_p$</td>
<td>Specific heat at constant pressure (J/kg K)</td>
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<tr>
<td>$D$</td>
<td>Tube inner diameter (mm)</td>
</tr>
<tr>
<td>$L$</td>
<td>Length of the test tube (mm)</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure (Pa)</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature (°C)</td>
</tr>
<tr>
<td>$U$</td>
<td>Flow velocity at the inlet of test tube (m/s)</td>
</tr>
<tr>
<td>$e$</td>
<td>Groove depth (mm)</td>
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<tr>
<td>$f$</td>
<td>Friction factor</td>
</tr>
<tr>
<td>$h$</td>
<td>Heat transfer coefficient (W/m² K)</td>
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<tr>
<td>$k$</td>
<td>Thermal conductivity (W/m K)</td>
</tr>
<tr>
<td>$p$</td>
<td>Grooved pitch (mm)</td>
</tr>
<tr>
<td>$q''$</td>
<td>Heat flux (W/m²)</td>
</tr>
<tr>
<td>$w$</td>
<td>Grooved width (mm)</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Dynamic viscosity (kg/m s)</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density (kg/m³)</td>
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</table>

**CFD SIMULATION**

The effect of groove dimensions on thermal performance of three different rectangular shaped grooves was studied. The variation of groove’s width was $w=0.2, 0.4, 0.6$ mm and depth $e=0.2, 0.4, 0.6$ mm with the same tube length of 2 m, tube diameter of 7.1 mm and pitch to diameter ratio $p/D$ ratio of 1 were investigated by CFD simulation with Re varies from 4000-10000. Then the results compared to each other. Next, the geometrical models of four internal helical rectangular grooved tube with the best groove dimension in regards to thermal performance are simulated to show the effect of $p/D$ ratio on heat enhancement in internally rectangular grooved tubes. The $p/D$ ratio varies from 1 to 18 in the test section with the length of 2 (m) in the turbulent range of Re number 4000-10000. Schematic diagrams of grooved tubes are provided in Fig.1 and 2. A CFD package STARCCM+ software was used for simulation to solve the mean time-averaged Navier-Stokes equation for turbulent flow [10]. Before running the production simulation cases, the best mesh was chosen by a grid independence study [12]. A polyhedral mesh was used in the core region of the tube flow and a prism layer mesh in the adjacent wall region were created. The $y+$ wall treatment function was chosen to deal with near wall layer to increase the accuracy of the solution and also to reduce the computational time by using a lower density mesh. The water flow was considered to be steady, incompressible and a realizable $k$-$e$ model was chosen to solve the flow equations.

**DATA REDUCTION**

From the work in [12] the Nu number and friction factor were calculated for different $p/D$ ratio for the different Reynolds number as follows:

$$ Nu = \frac{h_{ave}D}{k} $$

$$ f = \frac{\Delta P}{\frac{1}{2} \rho U^2} $$

The average heat transfer, $h_{ave}$, is calculated as:

$$ h_{ave} = \frac{q''}{(T_{ave,w}-T_{ave,b})} $$

$T_{ave, w}$ is the average temperature on the surface of the tube and $T_{ave,b}$ is a mean bulk temperature which is computed by the software and $q''$ is constant heat flux of 3150 (w/m²) which is applied on the tube surface. The contour plots of turbulent kinetic energy (TKE) and flow swirl effect were obtained to investigate the effect of pitch size and groove dimensions on heat transfer and flow characteristics of water.
RESULTS AND DISCUSSION

Validation of Grooved Tubes Simulation

Some of the results of the current study were validated with K.Aroonat et al. [7] experimental work and replotted in a new way to compare with a previous study of the authors [12]. The uncertainties of experimental Nu numbers and f were reported ±18% and ± 1% consecutively in the experimental study by Aroonat et al [7]. In the current work, it was found that the average error between the experimental and CFD Nusselt numbers and friction factors for three rectangular grooved tubes with different pitch size are 13% and 8% consecutively, as shown in Fig.3 and 4, which are deemed to be quantitatively close.

Effect of Groove Width (w) and Depth (e) on Nu and Friction Factor f

Figure. 5 shows a comparison of the Nu number and f results for tubes models with same length and pitch size but different groove width and depth consecutively. It shows that the Nu number seems to be slightly reduced (about 10% reduction) when (w) changes between 0.2-0.6 mm; however, there seems to be very little change in the f values as shown in Fig.6 with the same variation in (w). The decrease in the Nu number could be attributed to changes in the flow inside that partial cavity which might increase the recirculating flow in it. Figures 7&8 show the results of varying the depth of the groove from 0.2-0.6 mm for various Re numbers. Figure. 7 shows a more dramatic decrease of the Nu number as (e) increases and could be partially explained by the fact of an increase in the recirculating zone inside the cavity reducing; hence the boundary layer disturbance in this area. On the other hand, Figure.8 shows very little change in the value of f due to changes in (e) as was seen also in Fig. 6. Therefore, for all other remaining runs, it was decided to use the groove whose dimensions are e=0.2mm and w=0.2 mm as it showed from Fig. 5 and 7 to be the one that has the highest values of Nu for each of the Re number runs. Also, it was shown in these figures (i.e. 5&7) that for all groove dimensions chosen the Nu number increases with the Re number as would be expected.
similar to the case of a smooth pipe. Also, the friction factor plots (i.e. figures 6&8) show practically no change in \( f \) by increasing the width and depth in the grooved tube. The trends in these results are also in agreement with Moody diagram which showed that for similar pipe roughness factors in the pipe the frictional factors will not change much.

**Effect of \( p/D \) Ratio on Heat Transfer and Friction Factors**

The effect on the Nu number and friction factor of four different \( (p/D) \) ratio for the optimum groove dimensions in this study with \((e=0.2\text{mm} \text{ and } w=0.2\text{mm width}) \) size and length of 2.0m were studied and shown in Figure 8 and 9. It is seen that by increasing the \( p/D \) ratio the Nu number will decrease indicating less of the turbulent mixing and flow swirl (as seen later in the paper). Figure 9 shows the value of Nu numbers increases for all cases as the Re numbers increase similar to the results of turbulent flow in a smooth pipe. The friction factor plots of figure 10 depicts a higher pressure drop as the value of \( p/D \) ratio decreases and a decrease of \( f \) as the Re number increases similar to what has been found for smooth pipes in turbulent flow.

![Figure 5](image5.png)  
**Figure 5. Nu NUMBER AS A FUNCTION OF Re NUMBER FOR DIFFERENT GROOVE WIDTHS (\( p/D \) ratio=1)**

![Figure 6](image6.png)  
**Figure 6. FRICTION FACTOR AS A FUNCTION OF Re NUMBER FOR DIFFERENT GROOVE WIDTHS (\( p/D \) ratio=1)**

![Figure 7](image7.png)  
**Figure 7. Nu NUMBER AS A FUNCTION OF Re NUMBER FOR DIFFERENT GROOVE DEPTHS (\( p/D \) ratio=1)**

![Figure 8](image8.png)  
**Figure 8. FRICTION FACTOR AS A FUNCTION OF Re NUMBER FOR DIFFERENT GROOVE DEPTHS (\( p/D \) ratio=1)**

![Figure 9](image9.png)  
**Figure 9. Nu NUMBER AS A FUNCTION OF p/D RATIO FOR Re RANGING FROM 4000-10000**
Streamline and Flow Swirl

To visualize the effect of pitch size on the flow pattern, streamlines were plotted for smooth tube and grooved tubes for a Re number of 10000 with different p/D ratio. Streamlines were plotted along the entire length of the heat exchanger. Figures 11 to 14 depict the generation of a swirl in the flow inside the tubes due to these grooves. The starting point of this plot is at the inlet of the tube and locations are chosen randomly on the cross-sectional area at the inlet. Streamlines observed near the central axis of the tube show smoother (less wavy) paths because they were not as affected by the grooves. Since the Re number is the same in all cases, by decreasing the p/D ratio an increase in the swirl in path lines is noticed and also the velocity magnitude is increased somewhat indicating that there is an increase in the transverse component of the flow, while the average axial velocity should be the same for these cases since the Re number is the same. In cases with a larger pitch size such as tubes with p/D of 7 and 18, the streamlines were almost parallel such as in a smooth tube and swirl in the flow was not as noticeable. In addition, the vector velocity plots normal to the pipe’s axis are shown in figures 16 &17 to better demonstrate the effect of the swirl motion for tubes with p/D ratio of 1 and 2, which is shown for that plane at 1.0 (m) distance from the inlet. The vectors show the rate of change in the position of each volume of cells. Also, the color bar illustrates the magnitude of the speed at each position and the arrow to its direction in this transverse plane. For tubes with higher pitch size, the swirl effect is shown to be insignificant.

Figure 11. STREAMLINES (Velocity Magnitude) WITHIN THE FLUID REGION IN TUBE WITHOUT GROOVE, (Re=10000).

Figure 12. STREAMLINES WITHIN THE FLUID REGION IN TUBE WITH p/D RATIO OF 1, (Re=10000)

Figure 13. STREAMLINES WITHIN THE FLUID REGION IN TUBE WITH p/D RATIO OF 2, (Re=10000)

Figure 14. STREAMLINES WITHIN THE FLUID REGION IN TUBE WITH p/D RATIO OF 7, (Re=10000)

Figure 15. STREAMLINES WITHIN THE FLUID REGION IN TUBE WITH p/D RATIO OF 18, (Re=10000)
Turbulent Kinetic Energy

Turbulent Kinetic Energy (TKE) values in the flow field can be considered an indicator of increased turbulence and hence increased heat transfer. An increasingly monotonic value of the average TKE value is observed as a function of increasing Re number. This can be discerned from the maximum TKE values (i.e. red color values) presented in the legend of colors underneath each plot. It has already been revealed that an increased Re is shown to increase the Nu and hence the heat transfer between the tube surface and the bulk of the fluid. In Figures 18 through 21, the TKE pattern of flow in a tube with $p/D$ ratio of (2) has been shown for various Re numbers. The turbulence intensity is predicted by the $k-\varepsilon$ model and in subsequent figures indicates that the peak values are in the downstream region of the grooves and the minimum turbulence intensity is observed just upstream of them. The lowest TKE zone is found to be in the core region of the tube flow as would be expected as it is furthest from the wall region. The basic structure of the contours of turbulent kinetic energy remains the same but the scale changes with the change in Reynolds number. It can be seen that the maximum turbulent kinetic energy increases from 0.0058 to 0.20 (J/kg) with an increase in Re number from 4000 to 100000.
SUGGESTED EMPIRICAL CORRELATIONS

In conclusion, a couple of empirical correlations for Nu number and friction factor $f$ as a function of $p/D$ ratio and Re number range studied are provided here to summarize the general variation trends.

$$\text{Nu} = 0.0009(\text{Re}) + 23.4 \frac{p}{D}^{-0.04} \quad (R^2=0.97) \quad (4)$$

$$f = 1.62 \text{Re}^{-0.44} \frac{p}{D}^{-0.05} \quad (R^2=0.99) \quad (5)$$

CONCLUSION

The result of this study showed that the thermal performance of groove with ($e=0.2$ mm and $w=0.2$mm) dimensions seems to deliver the highest thermal performance than the other groove dimensions considered in the current simulation. Thermal performance did not change by altering the groove width at same groove depth but is decreased drastically due to change of groove depth at a constant width. The frictional factor $f$ was not affected significantly by those changes. The flow pattern indicates significant swirl effect and flows mixing inside the grooved tube with ($w=0.2$ mm, $e=0.2$mm) dimensions by decreasing the $p/D$ ratio at a penalty of higher pressure drop. In addition, the turbulent kinetic energy is increased with increasing the Re number from 4000 to 10000 and the highest value for this energy is observed in the area adjacent to the front of the groove as demonstrated in TKE plots. In conclusion, the empirical correlations provided in this study seem to provide a reasonable fit for the variations of Nu and $f$ obtained numerically from this study.

REFERENCES

