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Thermal Performance Enhancement of Double Pipe Heat Exchanger with Wavy Tape

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***Abstract* - CFD simulation will be carried out to investigate the effects of different tape angles on the thermal-hydraulic performance of pipe. The effect of wavy strip tarbulators with different angles on the NU, friction factor and thermal performance enhancement factor of double tube heat exchanger was studied. The configuration, gain in heat transfer augmentation, cost in pressure loss, generated swirl flow character, heat transfer enhancement mechanism, optimal geometric parameters and overall thermal hydraulic performance of wavy-tape are successively expounded in this work. It was seen that the results obtained from the simulation for the plain tube are in a good agreement with those predicted by the standard correlations. It is noted that, in this case the discrepancy between the results is less than 9% for Nusse. lt number and 11% for the friction factor. The wavy-tape induces significant swirl flow inside the pipe. Particularly, pairs of tangential vortices are generated on the flanks of the tape. These vortices break the thermal boundary layer and intensify the fluid advection, which consequently lead to highly localized heat transfer enhancement on the pipe surface.**

***Keywords*- Taguchi method, DPHE, Computational Fluid, Dynamics, Reynolds Averaged Navier-Stokes.**

I. INTRODUCTION

Various techniques have been tested on heat transfer enhancement to upgrade the involving equipment, mainly in thermal transport devices. These techniques unveiled significant effects when utilized in heat exchangers. One of the most essential techniques used is the passive heat transfer technique. Corrugations represent a passive technique. In addition, it provides effective heat transfer enhancement because it combined the features of extended surfaces, turbulators and artificial roughness. Basically, three approaches are available yet to enhance the rate of heat transfer, active method, passive method and the compound method [1].

A power source is essential for the active, certain surface modifications or extension, and inserts or fluid additives are used in the passive method, while the compound method is a combination of the above two methods such as surface modification with fluid vibration [2].The motivation behind this activity is the desire to obtain more effective heat exchangers and other industrial applications [3], with the major objectives being to provide energy, material, and economic savings for the users of heat transfer enhancement technology.

In heat exchangers, corrugation and other surface modifications are commonly used because they are very effective in the heat transfer enhancement; also it is appearing very interesting for practical applications because it is a technique that promotes secondary recirculation flow, by inducing non-axial velocity components [4]. Recently, a swirl or helical flow pattern produced by employing surface modifications or any other passive technique for heat transfer enhancement is very interesting [5]. Also, Spiral corrugation increases heat transfer enhancement due to secondary flow swirls and surface curvatures pass by fluid layers, which also causes pressure losses [6].

The main reason for employing heat transfer enhanced techniques is for cutting costs as well as for practical purposes. The major roles of corrugations are for enhancing the secondary re-circulation flows, via induction of the component the radial velocities as well as the mixing of the flow layer. These techniques have been widely utilized in recent heat exchangers [7]. The outcome generated from the surface area modifications or the manipulations of heat transfers, which has been demonstrated to induce swirls or spirally flowing patterns has attracted increasing interests [8]. Additionally, corrugation enhances heat transfer owing to the existence of mixing fluids generated through separations and re-attachments [9].

II. METHODOLOGY

1.Specifications of Double Pipe Heat Exchanger used

The study is done in a double pipe heat exchanger having the specifications as shown in table below:-

In present study, wavy-tape inserts are introduced to help establish swirl flow inside a straight pipe. Threedimensional models will be created to simulate the flow and heat transfer characteristics inside the pipe with wavy-tape inserts. CFD simulation will be carried out to investigate the effects of different tape angles on the thermal-hydraulic performance of pipe.

Table 1: Structural parameters (Zhu et al. 2016)

Diameter of pipe	30 mm
Pipe length, L	500 mm
Thickness of tape, t	1 mm

III. DATA PROCESSING

Heat transferred from hot water, and cold water, can be calculated by following equations, respectively:

$$q_h = \dot{m}_h c_{p,w}(T_{h,i} - T_{h,o})$$

$$q_c = \dot{m}_c c_{p,w}(T_{c,o} - T_{c,i})$$

All the calculations are based on average heat transfer rate that can be written as follow

$$q_{ave} = \frac{q_c + q_h}{2} = UA_i \Delta T_{LMTD}$$

Then experimental overall heat transfer coefficient (U) in a double pipe heat exchanger can be calculated with:

$$U = q_{ave} / A_i \Delta T_{LMTD}$$

Maximum possible heat transfer rate is expressed as:

$$q_{max} = C_{min}(T_{h,i} - T_{c,i})$$

Where, Cmin is the minimum thermal capacity and it is defined as below:

$$C_h = \dot{m}_h c_{p,w}$$

$$C_c = \dot{m}_c c_{p,w}$$

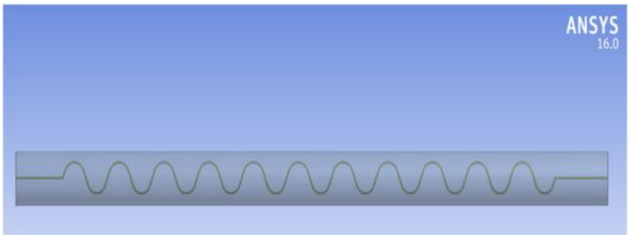
$$C_{min} = \min[C_h, C_c]$$

Friction factor can be calculated by

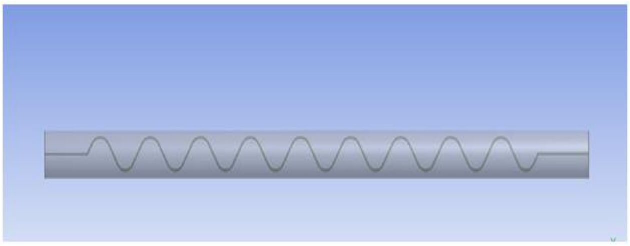
$$f = 2D_h \Delta P / (L\rho \overline{U}^2)$$

IV. COMPUTATIONAL DOMAIN

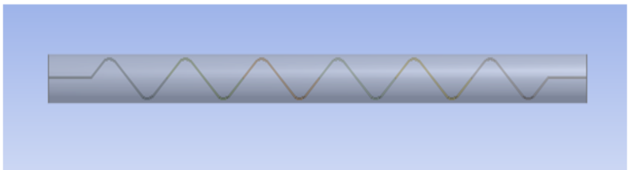
The aim of this research to numerically study and compare different configurations of wavy tape in a double pipe heat exchanger. The working fluid is water. Five double pipe heat exchangers are considered, different configurations of wavy tape.



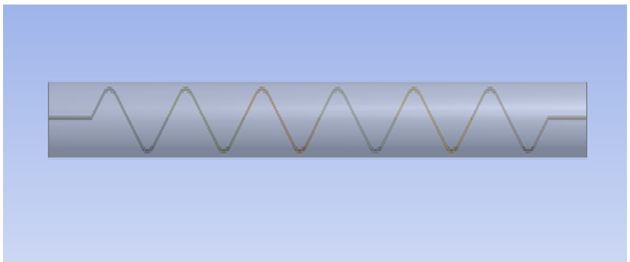
Wave angle = 45°
(a)



Wave angle = 60°
(b)



Wave angle = 90°
(c)



Wave angle = 150°
(d)

Fig.1 CFD domain.

V. BOUNDARY CONDITIONS

In order to evaluate the heat and momentum transfer of DPHEs, some preliminary conditions of the physical model have to be defined appropriately. For the numerical approach to the problem, the boundary conditions are required to set for all boundaries of the computational domain. At the upstream boundary conditions, the water

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entering the computational domain is assumed to have uniform velocity Uin, temperature Tin (293.15 K) and turbulent intensity I (1 %). The velocity components in the y and z directions are considered to be zero. The fluid region consists of the entrance, outlet, and bundle zone. The solid region includes the fin. At the solid surfaces, no-slip conditions for the velocity are specified. Heat convection to the fin and heat conduction in the fin is considered.

At the symmetry planes assume a zero heat flux. The normal velocity component at the symmetry plane is zero, i.e. no convective flux across that symmetry plane. Thus, the temperature gradients and tangential components of the velocity gradients in normal direction are set to be zero.

- Inlet:

$$U = U_{in} = \text{constant}$$

$$T = T_{in} = \text{constant}$$

$$I = 1\%$$

- Outlet:

Static pressure

- Tube

no-slip condition

$$T = T_w = \text{constant}$$

- Fin:

no-slip condition

Coupling of conduction and convecti

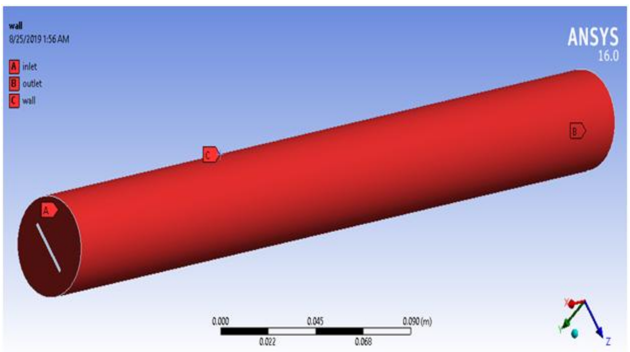

Fig.1. Boundary condition in CFD.

VII. MESHING OF DOMAIN

In this study, a general curve linear coordinate grid generation system based on body–fitted coordinates was used to discrete the computational domain into a finite number of control volumes. The geometries of the problems are carefully constructed. All cases were modeled and meshed with the GAMBIT. FLUENT also comes with the CFD program that allows the user to exercise the complete flexibility to accommodate the compatible complex geometries.

The refinement and generation of the grid system is important to predict the heat transfer in complex geometries. In other words, density and distribution of the grid lines play a pivotal role to generate accuracy. Due to the strong interaction of mean flow and turbulence, the numerical results for turbulent flows tend to be more dependent on grid optimization than those for laminar flows.

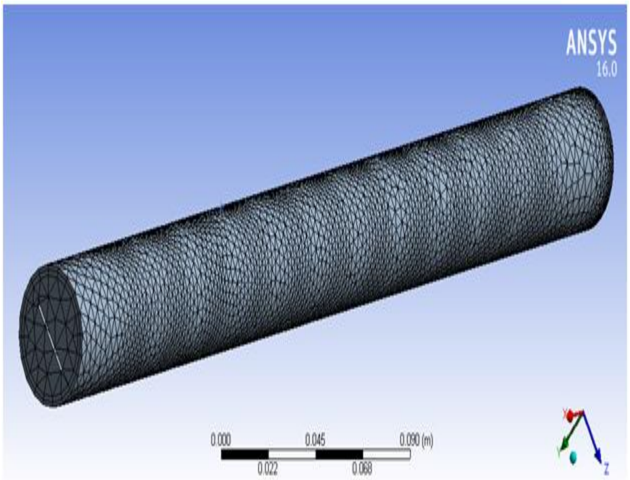

Fig.2. Mesh model.

VIII. RESULT

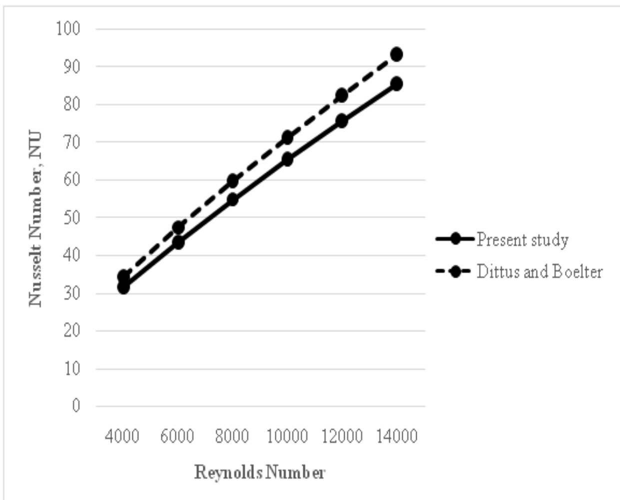

Fig.3. Validation of Nu number for the plain tube.

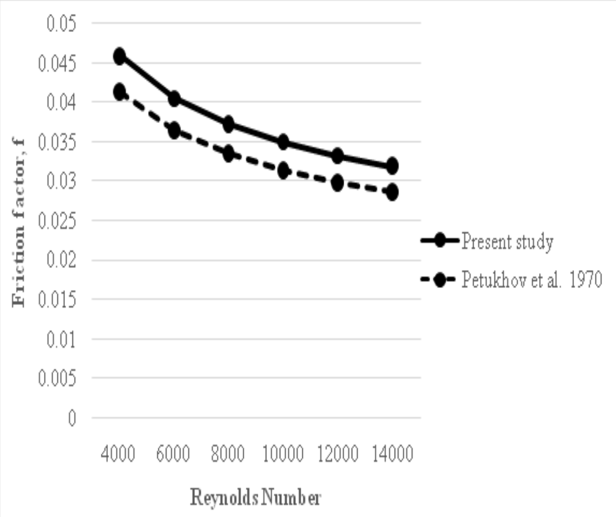

Fig.4. Validation of friction factor for the plain tube.

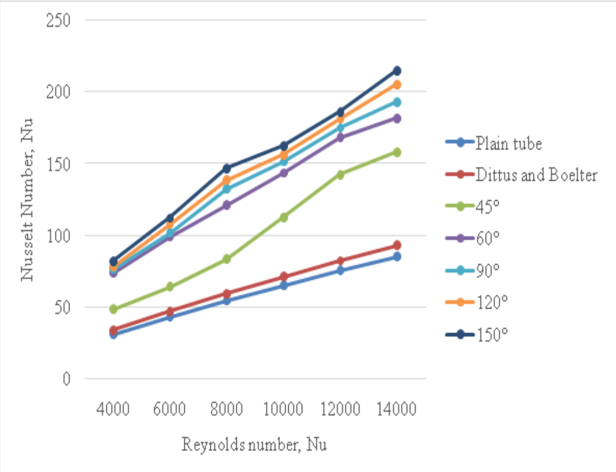

Fig.5. Variation of Nusselt number with Reynolds number at different angles.

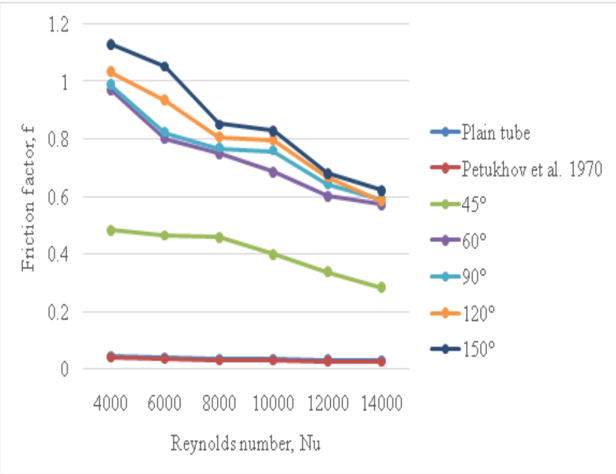

Fig.6. Variation of friction factor at different angles.

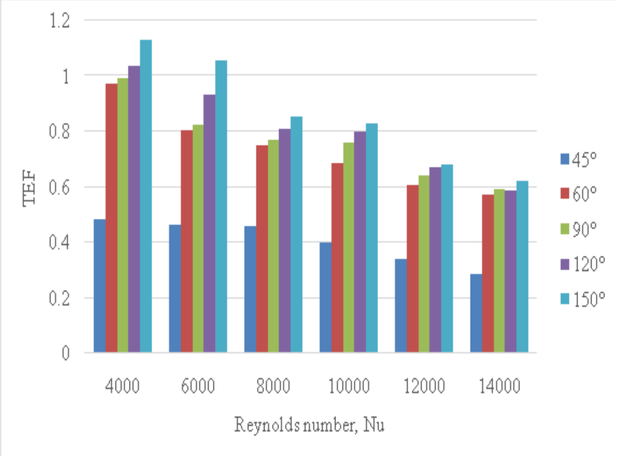

Fig.7 Thermal performance enhancement factor for DPHE with different wavy angle versus Re.

VIII. CONCLUSION

Following are the main findings of the present study.
It was seen that the results obtained from the simulation for the plain tube are in a good agreement with those predicted by the standard correlations. It is noted that, in this case the discrepancy between the results is less than 9% for Nusse. It number and 11% for the friction factor. The wavy-tape induces significant swirl flow inside the pipe. Particularly, pairs of tangential vortices are generated on the flanks of the tape. These vortices break the thermal boundary layer and intensify the fluid advection, which consequently lead to highly localized heat transfer enhancement on the pipe surface.

The Nusse. It number increases with the increase of wavy-tape angle. So the optimal design parameters for the wavy-tape should be determined by taking the heat transfer enhancement and thermo-hydraulic performance into account which was found to be 150°.

The improvement of heat transfer with increasing Reynolds number is responsible by a decrease of thermal boundary layer thickness due to the promoted turbulent intensity. The turbulence augmentation has a great effect on pressure drop due to its action on wall shear stress. Therefore, the friction factors considerably decrease with increasing Reynolds. Overall, it can be said that a double-pipe heat exchanger with a wavy angle of 150° shows better thermo-hydraulic performance than that with longitudinal fins.

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