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A Novel Wavy-Tape Insert Configuration for Pipe Heat Transfer Augmentation

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***Abstract*-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.The effect of wavy strip turbulators 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.**

***Keywords*- Wavy tape insert, heat transfer, performance, heat exchanger.**

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:

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

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

In present study, wavy-tape inserts are introduced to help establish swirl flow inside a straight pipe. Three-dimensional 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.

2. 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

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

III. NUMERICAL CONSIDERATION

Due to the advances in computational hardware and available numerical methods, CFD is a powerful tool for the prediction of the fluid motion in various situations, thus, enabling a proper design. CFD is a sophisticated way to analyze not only for fluid flow behavior but also the processes of heat and mass transfer.

1. Governing Equation:
The steady-state fluid flow characteristic in the three-dimensional computational domain can be described using the following governing incompressible fluid flow equations.
Continuity equation:

$$\frac{\partial u_i}{\partial x_i} = 0$$

Momentum balance without gravity force:

$$\rho u_i \frac{\partial u_i}{\partial x_i} = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j}$$

Energy equation:

$$\rho c_p \frac{\partial (u_i T)}{\partial x_i} = \frac{\partial}{\partial x}\left(\lambda_{\text{eff}} \frac{\partial T}{\partial x_i}\right)$$

In conservative form, the partial differential equations for the RNG k−ε model areTurbulent kinetic equation:

$$\rho u_j \frac{\partial k}{\partial x_i} = \frac{\partial}{\partial x_j}\left[\mu_{\text{eff}}\, \alpha_k \frac{\partial k}{\partial x_j}\right] + \tau_{ij}\frac{\partial u_i}{\partial x_j} - \rho\varepsilon + G_k + G_b$$

Turbulent kinetic dissipation equation

$$\rho u_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\mu_{\text{eff}}\, \alpha_\varepsilon \frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C^*_{2\varepsilon}\rho\frac{\varepsilon^2}{k}$$

The above Reynolds Averaged Navier-Stokes (RANS) turbulence models offer the most economic approach for computing complex turbulent industrial flows.Generally, the Navier-Stokes equations describe the motion of the turbulent flow. However, it is too costly and time-consuming to solve these equations for complex flow problems [26].

Alternatively, two methods have been suggested in the past:

- Large Eddy Simulation (LES) where the large energy containing eddies are simulated directly while the small eddies are accounted for by averaging. The separation of large and small eddies requires following,
- Reynolds averaging (RANS) where all eddies are accounted for by Reynolds stresses obtained by averaging the Navier-Stokes equations (time averaging for statistically steady flows, ensemble averaging for unsteady flows).

2. 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 fluids is water. Five double pipe heat exchangers are considered, different configurations of wavy tape.

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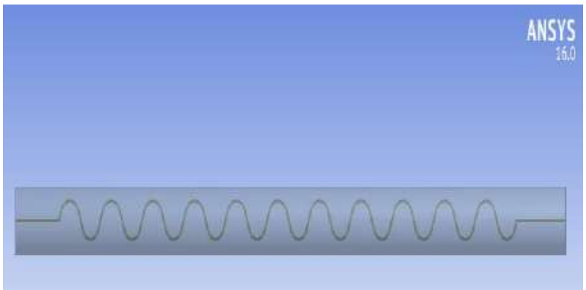
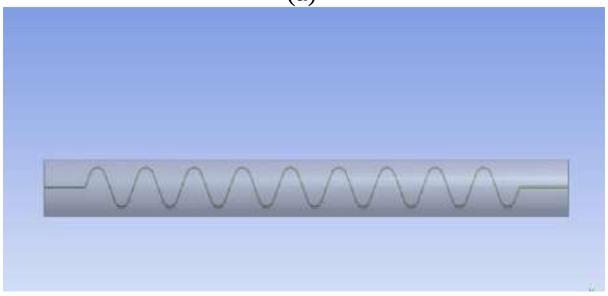
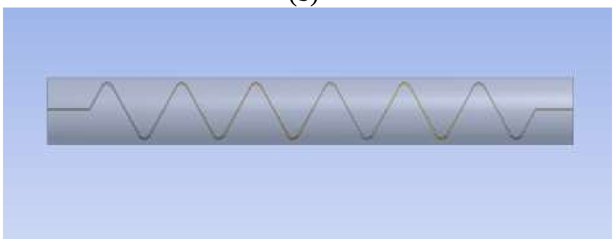


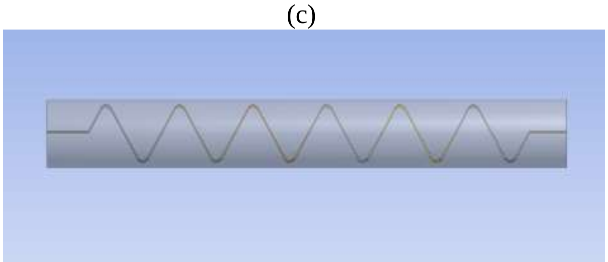
Wave angle = 45°
(a)



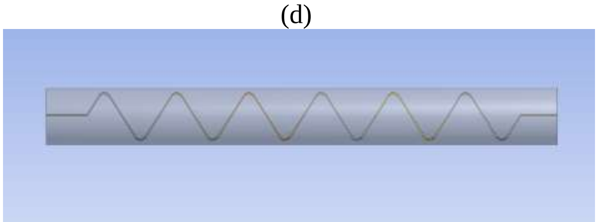
Wave angle = 60°
(b)



Wave angle = 90°
(c)



Wave angle = 120°
(d)



Wave angle = 150°
(d)

Fig 1. CFD domain.

3. 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 entering the computational domain is assumed to have uniform velocity U_{in}, temperature T_{in} (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 convection

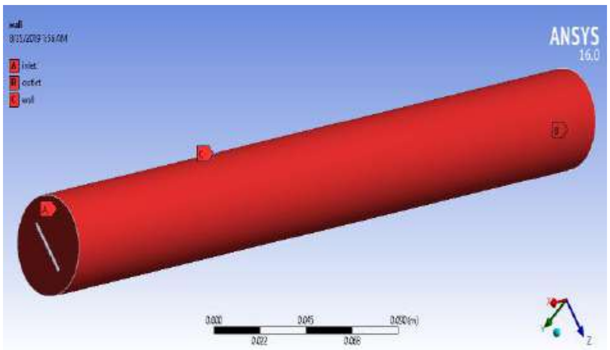


Fig 2. Boundary condition in CFD.

4. 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 modelled and meshed with the GAMBIT [12]. FLUENT

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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 [11].

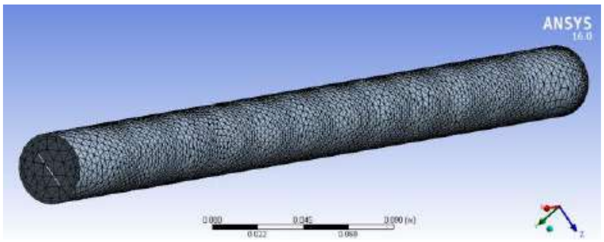

Fig 3. Mesh model.

IV. RESULTS AND DISCUSSION

1. Validation of Plain Tube:
Nusselt numbers are evaluated using traditional ‘‘Wilson plots” as explained in Rose [14] study. Subsequently, to validate the results, a comparison has been done between Nusselt number obtained from the present plain tube and the empirical correlation proposed by Dittus and Boelter [15]. In addition, the friction factor obtained from the plain tube was compared to the correlation presented by Petukhov [16].

$$Nu = 0.023Re^{0.8}Pr^{0.3}$$

$$f = (1.82 \mathrm{Log}Re - 1.64)^{-2}$$

Table 2. Validation table of Nusselt number.

Reynolds number	Nusselt number	
	Present study	Dittus and Boelter
4000	31.39	34.21
6000	43.41	47.32
8000	54.65	59.56
10000	65.33	71.21
12000	75.58	82.39
14000	85.50	93.20

The obtained results were compared with the confirmed heat transfer and friction factor correlations. On the basis of Figs. 3 and 4, 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 Nusselt number and 11% for the friction factor.

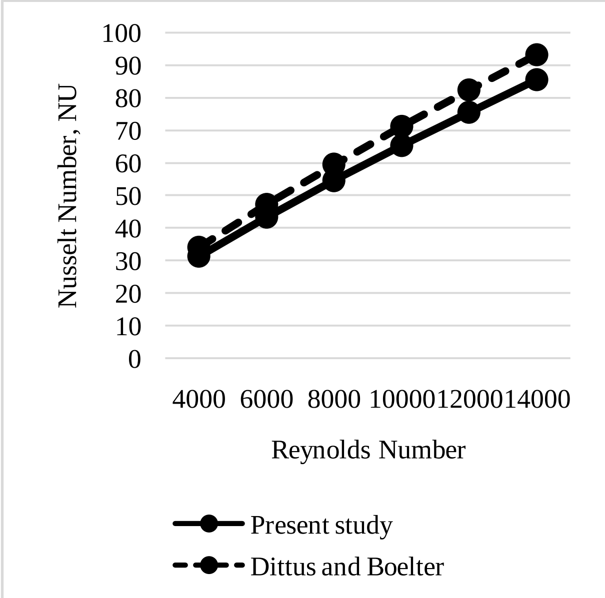

Fig 4. Validation of Nu number for the plain tube.

Table 3: Validation table of friction factor.

Reynolds number	Friction factor, f	
	Present study	Petukhov et al. 1970
4000	0.045935	0.041383
6000	0.040484	0.036472
8000	0.037184	0.0335
10000	0.034895	0.031437
12000	0.033178	0.02989
14000	0.031823	0.028669

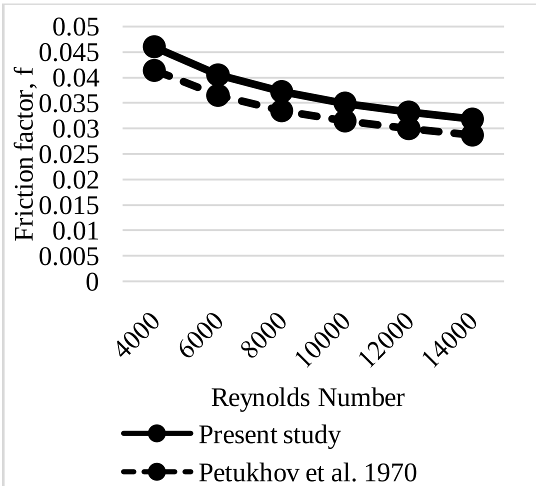

Fig 5. Validation of friction factor for the plain tube.

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.

2. Variation of Nusselt Number:
For heat transfer determinations, the simulation results were validated by comparing the values of Nu_{an} for the water flowing through the pipe for turbulent flow developed by Dittus and Boelter.

Table 4: Nusselt number at different angles.

Reynolds number	Nusselt number						
	Present study	Dittus and Boelter	45°	60°	90°	120°	150°
4000	31.39	34.21	48.36	74.13	76.52	78.41	82.35
6000	43.41	47.32	64.11	99.31	102.15	108.25	112.58
8000	54.65	59.56	83.54	121.42	132.63	138.57	146.67
10000	65.33	71.21	112.49	143.99	151.74	156.42	162.74
12000	75.58	82.39	142.52	168.61	175.39	181.29	186.27
14000	85.50	93.20	158.21	182.14	193.52	205.37	215.07

The simulations were conducted in a double tube heat exchanger fitted with wavy strip turbulators. These turbulators with different angles of 45°, 60°, 90°, 120° and 150° were placed along the tube with the axial direction. Reynold number was varied from 4000 to 14000 to study the effect of wavy strip turbulators considering different angles on the Nusselt number were investigated.

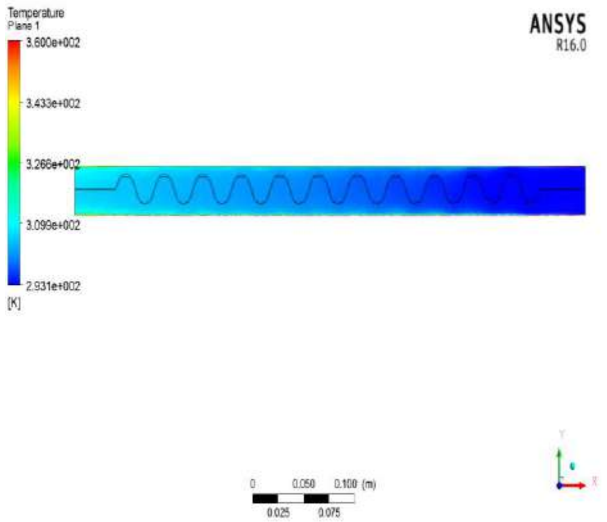

Fig 6. Temperature distribution on a symmetric plane with Re = 4000 for wavy angle 45°.

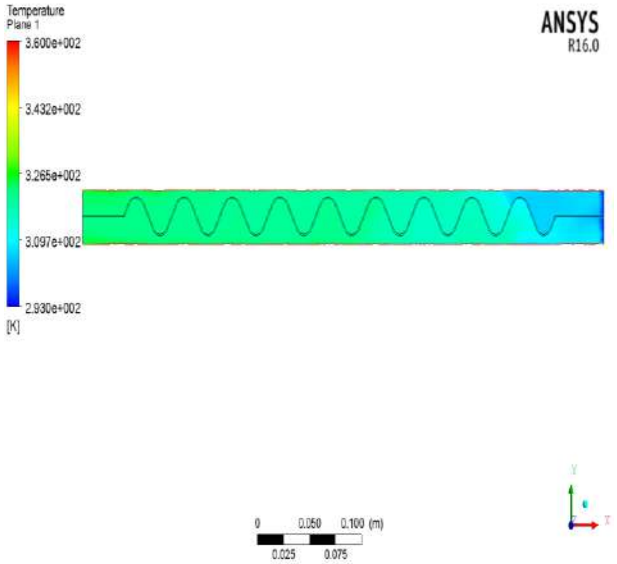

Fig 7. Temperature distribution on a symmetric plane with Re = 4000 for wavy angle 60°.

3. Thermal Performance Enhancement Factor (TEF):
The maximum thermal enhancement factor is obtained with a wavy angle of 150°. Thus, the thermo-hydraulic performance improves with the wavy tape configurations whereas it degrades with the other configuration. Although its η ranks last, the DPHE with helical 12 fin configuration provides the best thermal enhancement, and it should be adopted when the heat transfer capability is considered to be important and the pressure drop is considered to be negligible. If the heat transfer capability can meet the requirement of a particular engineering application, pipe with wavy angle of 150° should be the first choice because of its best TEF value.

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Overall, it can be said that a double-pipe heat exchanger with a wavy angle of 150° shows better thermos-hydraulic performance than that with longitudinal fins.

Using the wavy strip turbulators leads to an enhancement in the heat transfer, however, it increases the pressure drop. Therefore, to evaluate the quality of enhancement concept, thermal performance enhancement factor (TEF) can be calculated as follow:

Table 5. Thermal performance enhancement factor (TEF) at different angles.

Reynolds number	TEF				
	45°	60°	90°	120°	150°
4000	1.532	1.581	1.623	1.714	1.738
6000	1.549	1.623	1.651	1.764	1.842
8000	1.653	1.745	1.783	1.812	1.867
10000	1.780	1.831	1.849	1.860	1.885
12000	1.883	1.926	1.938	1.943	1.962
14000	1.532	1.581	1.623	1.714	1.738

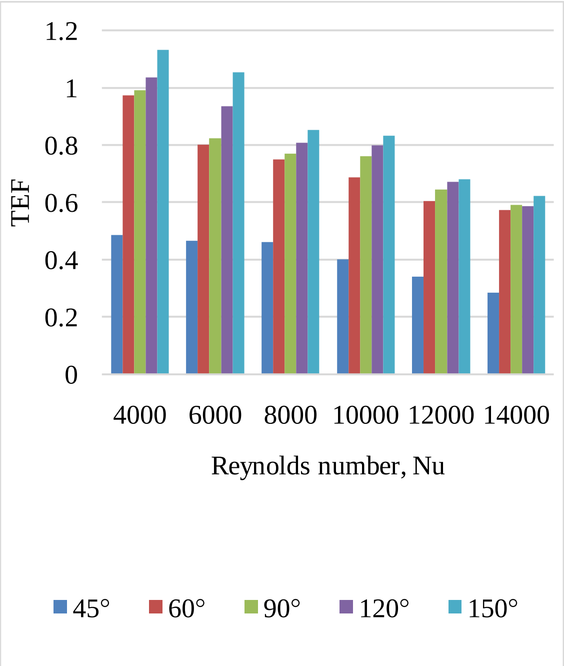

Fig 8. Thermal performance enhancement factor for DPHE with different wavy angle versus Re.

V. CONCLUSION

The effect of wavy strip turbulators 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. 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 Nusselt 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 Nusselt 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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