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Heat Transfer Augmentation Techniques for Double Pipe Heat Exchanger

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***Abstract*- Nowadays for enhancing the convective heat transfer in various industries such as thermal power plants, chemical processing plants, air conditioning equipment, refrigerators, petrochemical, biomedical and food processing plants heat exchangers are employed with twisted-tape inserts have widely been applied. By employing, twisted tape insert bring together swirl into the fluid flow which subsequently interrupts a thermal boundary layer on the tube surface. The thermal performance of heat exchangers can be improved by heat transfer augmentation techniques. Tape insert is one of the passive heat transfer augmentation technique and used in utmost heat transfer application, for example, air conditioning and refrigeration systems food processes. Therefore, various enhancement techniques have been reviewed.**

***Keywords*-Heat Transfer, Augmentation Techniques, DPHE, Twisted Tape**

I.INTRODUCTION

Heat exchangers were used in a wide-ranging of applications including power generation plants, nuclear reactors for generation of electricity, Refrigeration & Air Conditioning (RAC) systems, self-propelled industries, food industries, heat retrieval systems, and chemical handling. The upgrading methods can be distributed into two groups: active and passive methods. The active method requires peripheral forces. The passive methods need discrete surface geometries. Both methods have been commonly used to improve performance of heat exchangers. Due to their compact structure and high heat transfer coefficient helical tubes have been declared as one of the passive heat transfer improvement method and they are broadly used in many industrial applications [1, 4, 16].

The development of high performance thermal systems has stimulated interest in methods to improve heat transfer. In heat exchangers, enhancement of heat transfer is achieved by increasing the convection heat transfer coefficient or by increasing the convection surface area. One of the method to increase the convection coefficient within a heat exchanger is by introduces inserts within the pipes/tubes.

Heat Exchanger is a device in which the exchange of energy takes place between two fluids at different temperature. A heat exchanger utilizes the fact that, where ever there is a temperature difference, flow of energy occurs. So, that Heat will Flow from higher Temperature heat reservoir to the Lower Temperature heat Reservoir. The flowing fluids provide the necessary temperature difference and thus force the energy to flow between them. The energy flowing in a heat exchanger may be either sensible energy or latent heat of flowing fluids. The fluid which gives its energy is known as hot fluid. The fluid which receives energy is known as cold fluid. It is but obvious that, Temperature of hot fluid will decrease while the temperature of cold fluid will increase in heat exchanger. The purpose of heat exchanger is either to heat or cool the desired fluid.

In a special case, when one of fluid undergoes change in its phase, its temperature remains unchanged. These types of heat exchanger are known as condensers or evaporators. Heat exchangers with the convective heat transfer of fluid inside the tubes are frequently used in many engineering application. The techniques of heat transfer enhancement to accommodate high heat flux i.e., to reduce size and cost of heat exchangers have received serious attention passed years. Enhancement of heat transfer Rate in all types of thermos-technical apparatus is of great significance for industry. Beside the savings of primary energy, it also leads to a reduction in size and weight. Up to the present, several heat transfer enhancement techniques have been developed. Twisted-tape is one of the most important members of enhancement techniques, which employed extensively in heat exchangers.

II. LITERATURE REVIEW

Shailesh Dewangan (2018) made helical ribs on the tube surface by machining the surface on the lathe so that artificial roughness can be created. The artificial roughness that results in an undesirable increase in the pressure drop due to the increased friction; thus the design of the tubes surface of heat exchanger should be executed with the objectives of high heat transfer rates. N Sreenivasalu Reddy (2017) inspected the heat transfer analysis in the horizontal double pipes with helical fins in the annulus side. The material is copper with inner tube internal diameter 10 mm, inner tube thickness 1 mm, outer tube external diameter 40 mm, outer tube thickness 1.5 mm, helical pitch of 50mm, 75mm and 100mm, heat exchanger length 1100 mm. The experimental results of plain tube are validated with numerical results. The results obtained for helical fins in the annulus side provide enhanced heat transfer performance compared to the simple double-pipe exchangers. Riddheshwar R. Bilawane (2017) presented a review of one of the passive augmentation techniques used in a concentric tube heat exchanger using inner wavy tube. The performance of counter flow heat exchanger will be studied with inner plain tube and inner wavy tube. Then this enhanced performance due to inner wavy tube will be compared with performance of heat exchanger with inner plain tube and percentage of enhancement will be calculated in different hot fluid temperature input and different mass flow rates of hot as well as cold water. Experimentally, Overall heat transfer enhancement will be studied and also, the experimental results will be validated with CFD simulation.

Patel Yogeshwari (2017) discussed analytical solution of the compartment based double pipe heat exchanger model obtained using Differential Transform Method for parallel flow with theoretical varying initial and boundary condition. The working fluid is transformer oil i.e. hot fluid and water act as coolant. Convergence analysis of solution is also discussed.

Pourahmad and Pesteei (2016) experimentally investigated on double pipe heat exchanger by inserting wavy strip turbulators in the inner pipe, their findings are on considerable improvements in enhancement of heat transfer characteristics.

Dhanraj S. Pimple (2016) investigated the heat transfer and friction factor data for single -phase flow in a shell and tube heat exchanger fitted with a helical tape insert. In the double concentric tube heat exchanger, hot air was passed through the inner tube while the cold water was flowed through the annulus. The influences of the helical insert on heat transfer rate and friction factor were studied for counter flow, and Nusselt numbers and friction factor obtained were compared with previous data for axial flows in the plain tube. The flow considered is in a low Reynolds number range between 2300 and 8800. A maximum percentage gain of 165% in heat transfer rate is obtained for using the helical insert in comparison with the plain tube.

K.A. Goudiya (2016) presented the literature survey of enhancement techniques in heat transfer using inserts. Ayush Kumar (2015) discussed with different configurations. Here CDD (convergent divergent spring tabulators) CDDSTs were placed in the inner tube of double pipe heat exchanger and effect on heat enhancement and friction factor was experimentally investigated. CDDSTs at various pitches i.e (p=0, p=15, p=16) were used for the different ranges of Reynolds number. For cold water its ranges between 9000 to 17000 and for hot water 18000 to 24000. Finally results from CDDSTs were compared with plane tube and results showed that Nusselt number increased while friction factor decreased with increased in Reynolds number. Friction factor was increased by 287% while Nusselt no. increased by 28%. However thermal performance factor was maximum for CDDSTs (p = 15) with value 0.319.

Abhishek Tripathi (2015) presented a review on different arrangement of finned tube bundles placed on inline arrangement and staggered arrangement in cross flow. A large number of experimental and numerical works had been performed for enhancement of air-side heat transfer. A brief discussion is done on the effect of local heat transfer behaviour of circular finned tube and analysis of geometric and flow parameters included in this paper. Different parameters like fin height, fin spacing, fin thickness, tube diameter, tube spacing, effects of row and arrangement of tube bundles affect directly on the performance of solid circular finned tube. All these parameters are briefly discussed in this paper. Discussions on some important points which affect the performance of tube bundles (i.e. inline and staggered arrangement) from various authors and their problem and related issues are presented in this paper. The flow profiles and the related heat transfer characteristics in the complex geometries are still needed to be verified.

PatnalaSankara Rao (2014) studied the performance of (i) bare tube-in-tube heat exchanger, (ii) tube in tube with twisted tape insert and (iii) helical insert at annulus and twisted tape insert inside the inner tube of the heat exchanger. Numerical results have been compared with the available analytical solution. It has been observed that there is a good agreement between these two results: within ±19.78 percentage error limit for Nusselt number measurement and ±25 percentage error for friction factor.

Antony luki. A (2013) investigated augmented surface has been achieved with dimples strategically located in a pattern along the tube of a concentric tube heat exchanger with the increased area on the tube side. Augmented surfaces to increasing the heat transfer coefficient with a consequent increase in the friction factor. In this analysis to modify the inner tube of double pipe heat exchanger using dimpled tube. In this design the inner tubes consider as the hot flue gas and outer tube is nano fluid. Here In this study the properties of nano fluid from the alumina as the nano fluid with ethyl glycol as the base fluid. a. From this design calculation the heat transfer coefficient is increased compared to plain concentric tube heat exchanger. Similarly the effectiveness is 8% increase compared to plain concentric tube heat exchanger. The theoretical results show that the using dimpled tube in concentric tube heat exchanger gives better performance.

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The modeling and analysis is carried out to vary the dimple tube cross sections, ellipsoidal and spherical shapes using CFD. Finally the enhanced dimple tube is compare with the theoretical, analytical and analysis the results.

M. Kannan (2012) made comparison of different types of heat transfer augmentation techniques or methods in heat exchangers by extended surfaces, obstruction devices and swirl flow device. The system has followed different geometric profiles for attainable heat transferred in experimental result and compare with simulation result. The objective of these Experiments is to assist the general heat transfer processes and the methods and devices that can be implemented to enhance more heat transfer rate. The experimental setup and apparatus required to carry out the double pipe heat exchanger experiment.

The apparatus includes tube-within-a-tube heat exchangers with threaded thermometer at each end, measuring flask, a water pump and electric geyser device. Three of the four heat exchangers are modified by one type of the above-mentioned heat transfer enhancement techniques. These methods used to found out the heat loss from the surface and related temperature of fluid motions also used to found the effectiveness, the effectiveness are having to compare the different flow rates for which one is maximum possible heat transfer in double pipe heat exchanger. Annular method is higher rate of heat transfer than other three methods.

III. CFD MODELLING

1. 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 analyse not only for fluid flow behaviour but also the processes of heat and mass transfer.

2. 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:

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

In conservative form, the partial differential equations for the RNG k−ε model are

Turbulent kinetic equation:

Turbulent kinetic dissipation equation

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

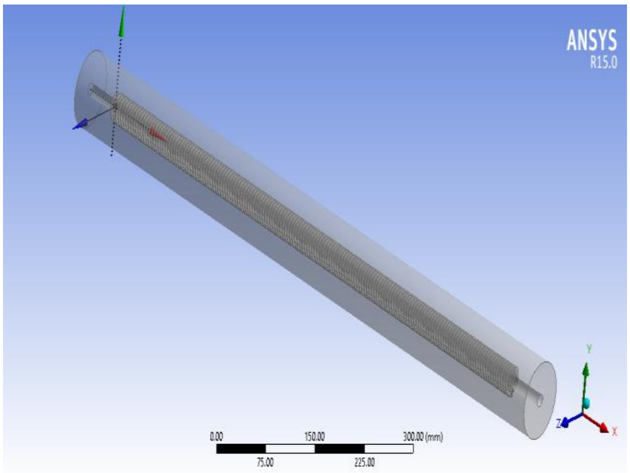
The above Reynolds Averaged Navier-Stokes (RANS) turbulence models offer the most economic approach for computing complex turbulent industrial flows.

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:

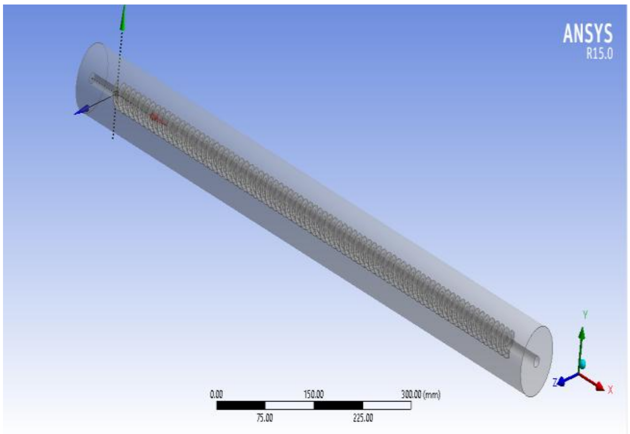
- Large Eddy Simulation (LES) where the large energy containing eddies are simulated directly while the small eddies are accounted for by averaging. The separation of large and small eddies requires following,
- Reynolds averaging (RANS) where all eddies are accounted for by Reynolds stresses obtained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

3. Computational Domain

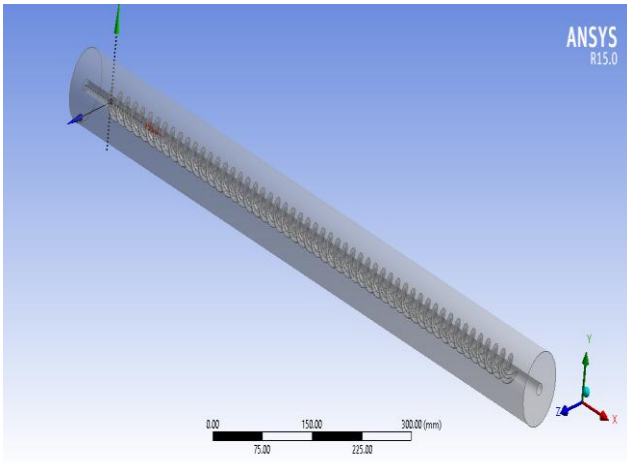
For hydrothermal performance of a double pipe heat exchanger using double helical tape insert, we have select a base paper M.R. Salem et al (2018) in this paper they examines the hydrothermal performance of horizontal Double Pipe Heat Exchangers (DPHEs) with and without continuous Helical Tape Insert (HTI) conducted on the outer surface of the internal pipe. Ten DPHEs of counter-flow configurations were constructed; nine of them were with HTI fabricated with different ratios of HTI height to the clearance between the two pipes (δ), and different ratios of HTI pitch to HTI diameter. The experiments were performed with pure water in both sides with 2050 ≤ Rean ≤ 15925, and Ret ≈ 26700.



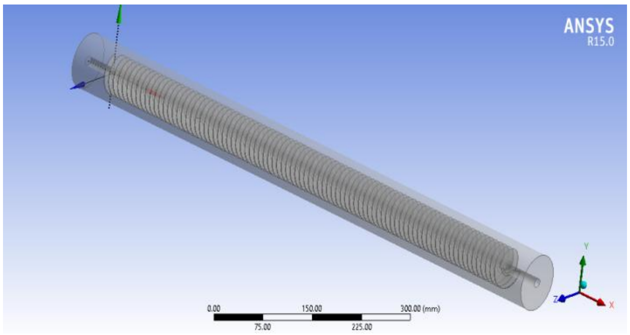
HFT no. 2, Nt=80, p=10



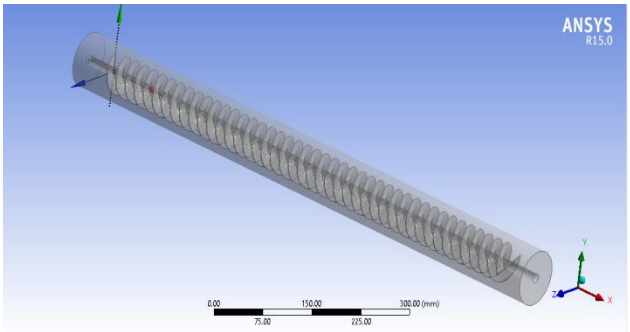
HFT no. 3, Nt=40, p=20



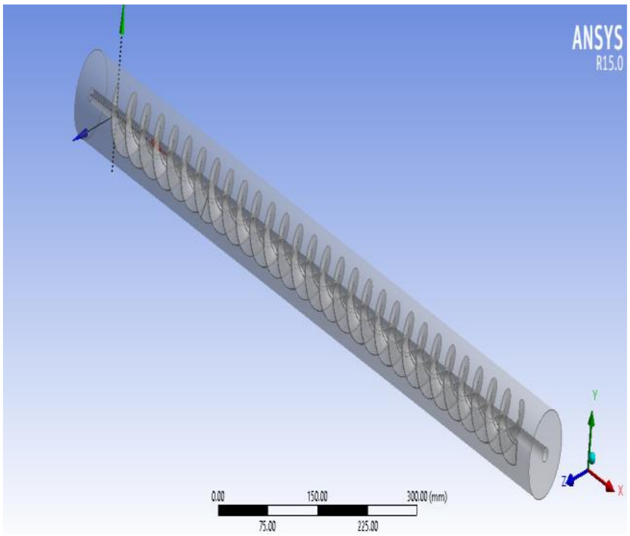
HFT no. 4, Nt=26.7, p=30



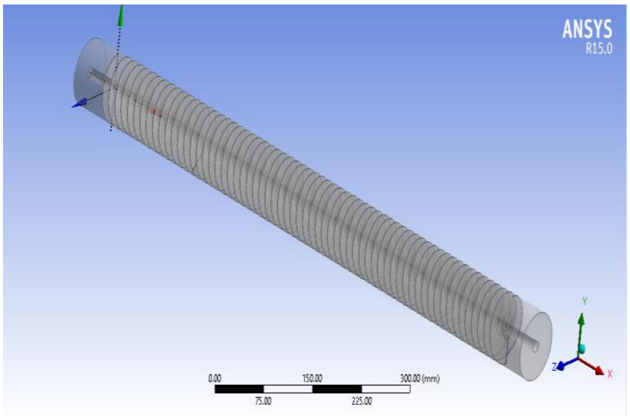
e) HFT no. 5, Nt=43.7, p=18.3



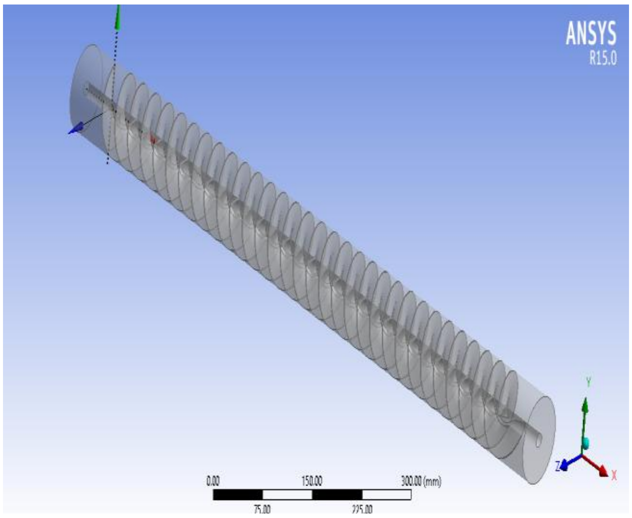
HFT no. 6, Nt=21.8, p=36.7



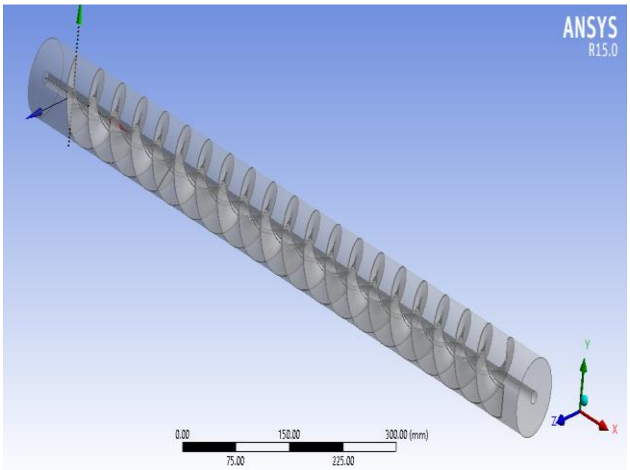
HFT no. 7, Nt=14.5, p=55



HFT no. 8, Nt=31.5, p=25.4



HFT no. 9, Nt=15.7, p=50.8



HFT no. 10, Nt=10.5, p=76.2

Fig. 1 Computational domain.

A sketch of the test section is shown in Fig. 3. Ten DPHEs of counter-flow configurations were constructed; nine of them were with HTI in the annulus side, while the first is without any inserts. The HTIs were constructed with different height and pitch ratios, divided into three groups to investigate the effect their geometry (δ, λ). The characteristic dimensions of the different configurations of the DPHEs and their HTIs are revealed in Table 1.

Table 1 Characteristic dimensions of the used DPHEs and their HTIs

HFT no.	d_{TPI} (mm)	H_{TPI} (mm)	c (mm)	$d_{an,\ h}$ (mm)	p (mm)	N_T	λ	δ
1	...	...	...	63.7	...	00.0	...	...
2	30	8.8	23.1	44.9	10	80.0	0.333	0.275
3				52.7	20	40.0	0.667	
4				55.9	30	26.7	1.000	
5	55	21.3	10.6	33.8	18.3	43.7	0.333	0.667
6				44.2	36.7	21.8	0.667	
7				49.2	55.0	14.5	1.000	
8	76.2	31.9	00.0	28.2	25.4	31.5	0.333	1.000
9				39.2	50.8	15.7	0.667	
10				44.9	76.2	10.5	1.000	

IV. 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 Uin, temperature Tin (15ºC, 20ºC and 25ºC) 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. Constant temperature Tw (50ºC) is assigned at the tube surface and all velocity components are considered to be zero. 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

Table 2 Range of both fluids operating conditions.

Parameters/operating conditions	Range or Value
Annulus-side	
Water flow rate, l/min	6.01–18.26 ($2050 \leq Re_{an} \leq 15925$)
Inlet temperature, °C	15, 20, 25 ($5.8 \leq Pr_{an} \leq 8$)
Internal pipe-side	
Water flow rate, l/min	8.07 ($Re_t \approx 26700$)
Inlet temperature, °C	50 ($Pr_t \approx 3.71$)

V. 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 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 optimisation than those for laminar flows [11].

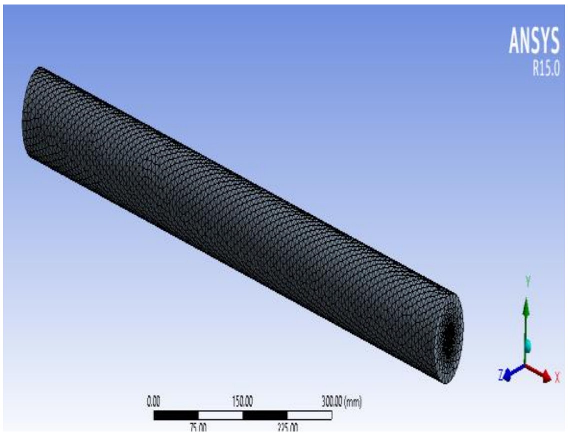


Fig. 2 Mesh model.

VI. DATA PROCESSING

FLUENT evaluates the Nusselt number as follows
The outlet air temperature Tout was calculated as a mass average temperature at the outlet position of the calculation domain.

$$T_{out} = \frac{\int T\rho\vec{u}.d\vec{A}}{\int \rho\vec{u}.d\vec{A}} = \frac{\sum_{i=1}^{n} T_i \rho_i \vec{u}_i .\vec{A}_i}{\sum_{i=1}^{n} \rho_i \vec{u}_i \vec{A}_i}$$

The dimensionless number for air-side heat transfer in the finned-tube bank was defined and calculated depending on the Reynolds number and geometric parameters. For many cases the Nusselt number and Stanton number are used to express the heat transfer coefficient and the characteristic length is not the same.
In here the Nu number was used as

$$Nu = \frac{hd}{k_a}$$

The static pressure at the inlet and outlet of the computational domain were evaluated as

$$p_{in,out} = \frac{\int p d\vec{A}}{\int d\vec{A}} = \frac{\sum_{i=1}^{n} p_i \vec{A}_i}{\sum_{i=1}^{n} \vec{A}_i}$$

VII. RESULTS AND DISCUSSION

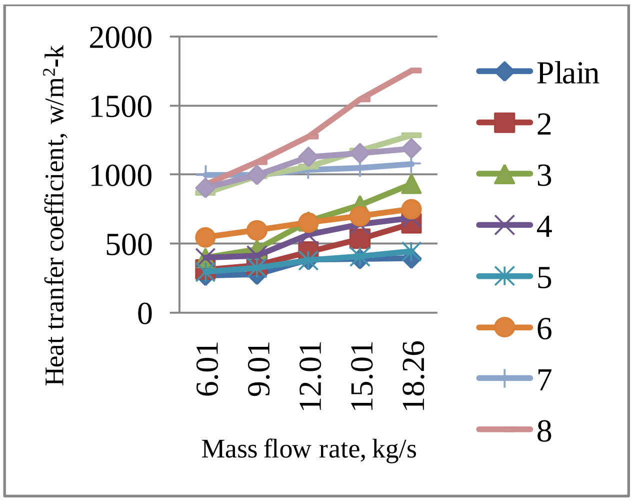


Fig. 3 Variation of heat transfer coefficient for DPHE at water inlet temperature 15°C.

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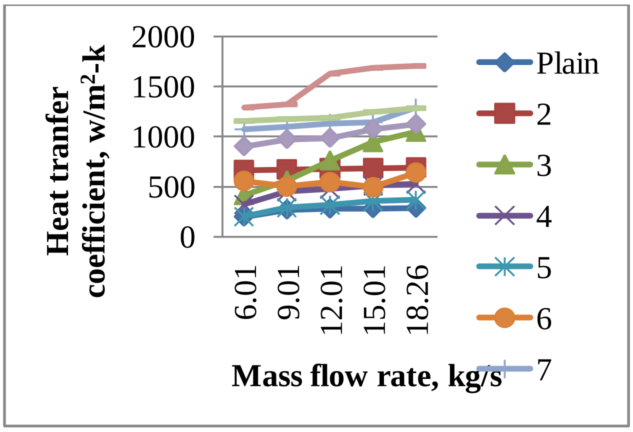

Fig. 4 Variation of heat transfer coefficient for DPHE at water inlet temperature 20ºC.

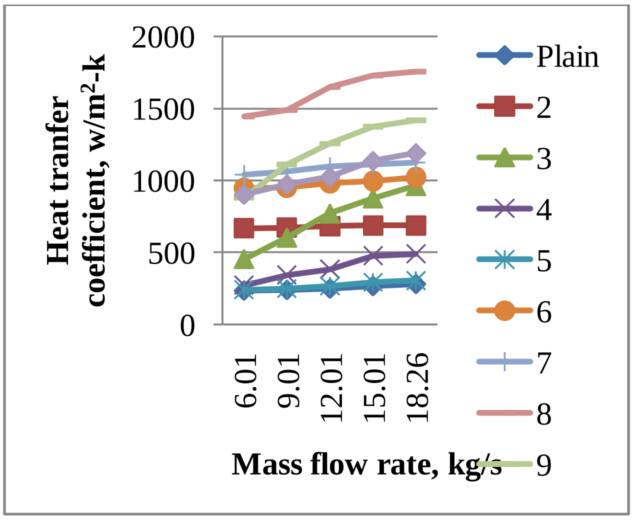

Fig. 5 Variation of heat transfer coefficient for DPHE at water inlet temperature 25ºC.

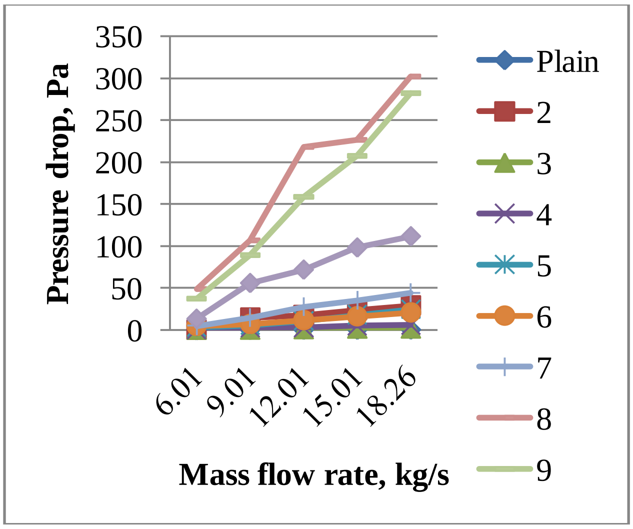

Fig. 6 Variation of pressure drop for DPHE at water inlet temperature 15ºC.

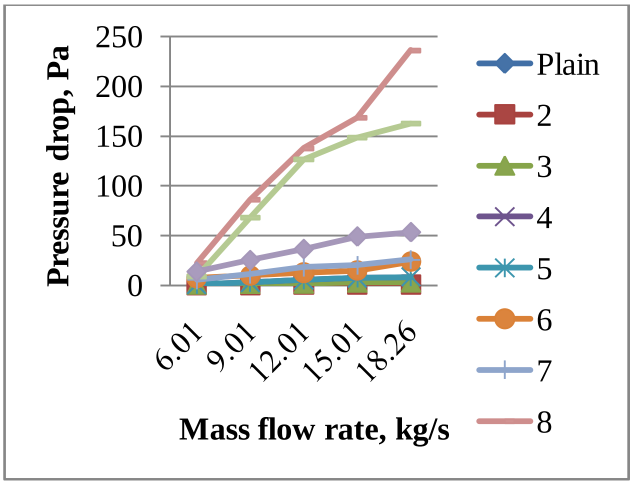

Fig. 7 Variation of pressure drop for DPHE at water inlet temperature 20ºC.

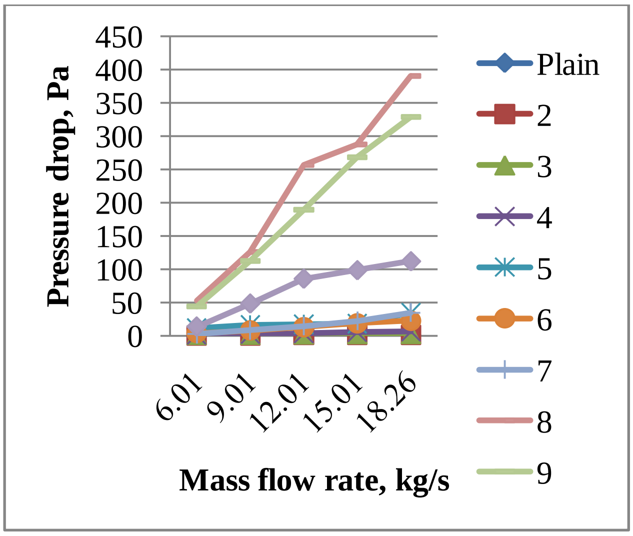

Fig. 8 Variation of pressure drop for DPHE at water inlet temperature 25ºC.

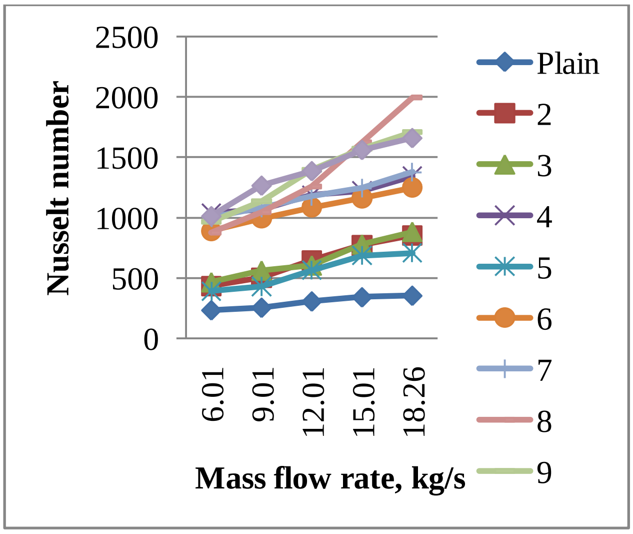

Fig. 9 Variation of Nusselt number for DPHE at water inlet temperature 15ºC.

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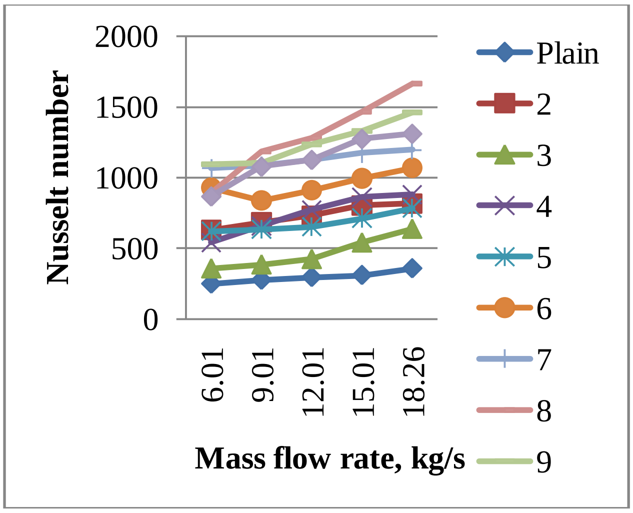

Fig. 10 Variation of Nusselt number for DPHE at water inlet temperature 20ºC.

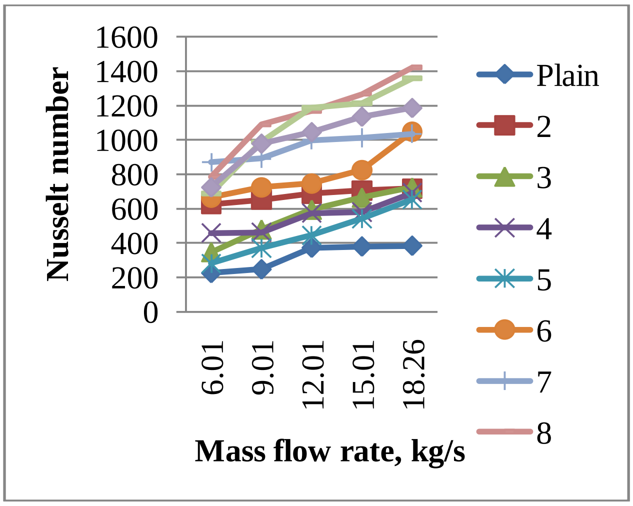

Fig. 11 Variation of Nusselt number for DPHE at water inlet temperature 25ºC.

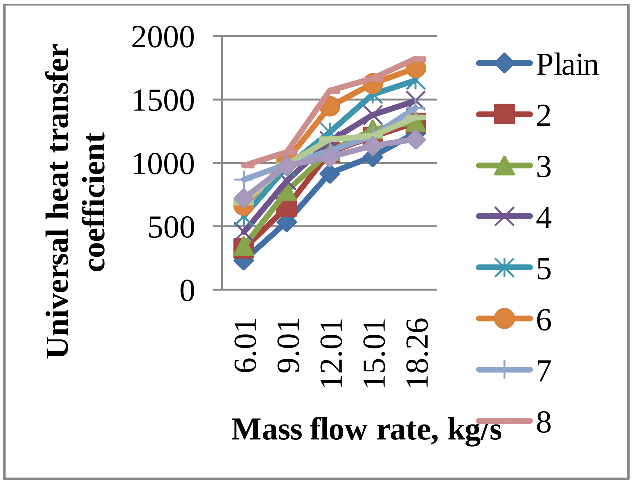

Fig. 12 Variation of universal heat transfer coefficient for DPHE at water inlet temperature 15ºC.

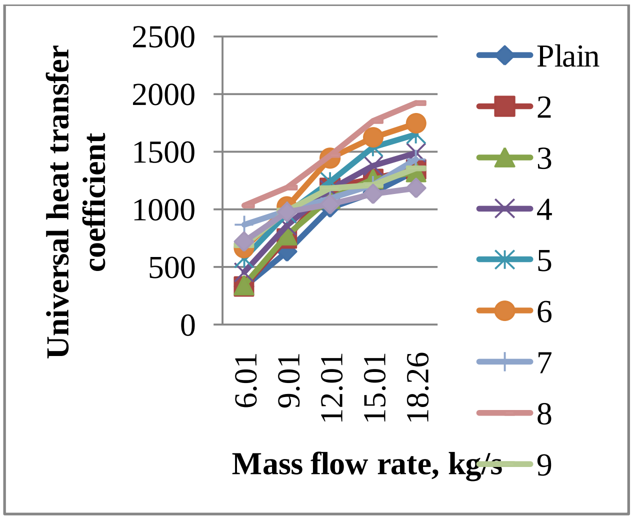

Fig. 13 Variation of universal heat transfer coefficient for DPHE at water inlet temperature 20ºC.

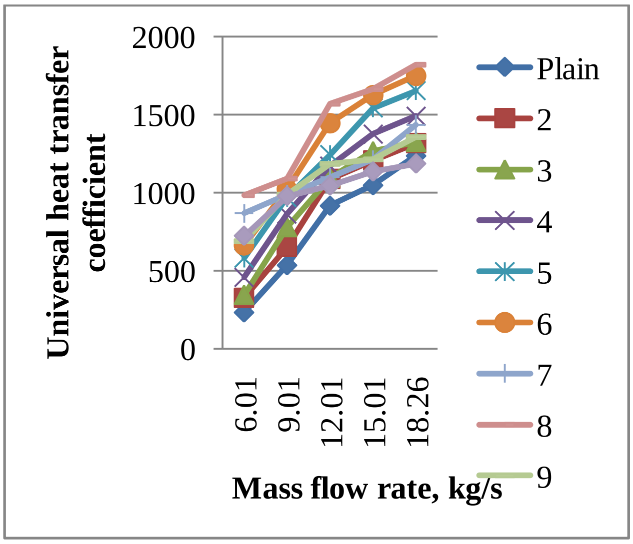

Fig. 14 Variation of universal heat transfer coefficient for DPHE at water inlet temperature 25ºC.

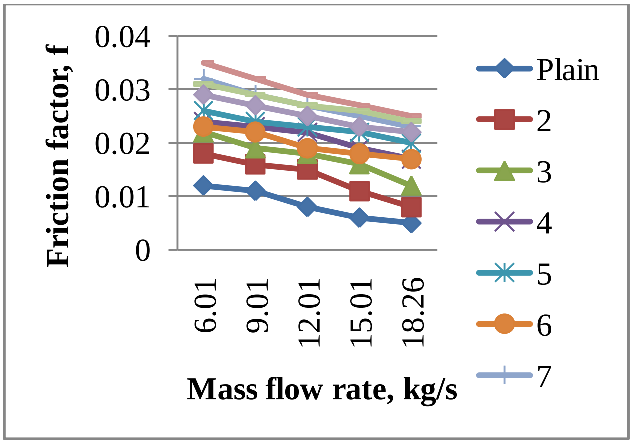

Fig. 15 Variation of friction factor for DPHE at water inlet temperature 15ºC.

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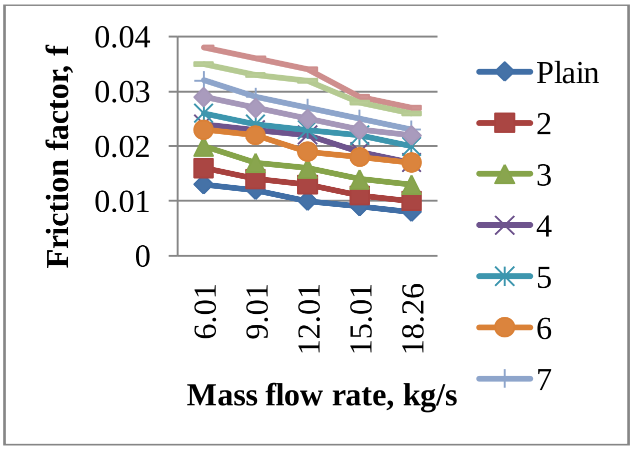

Fig. 16 Variation of friction factor for DPHE at water inlet temperature 20ºC.

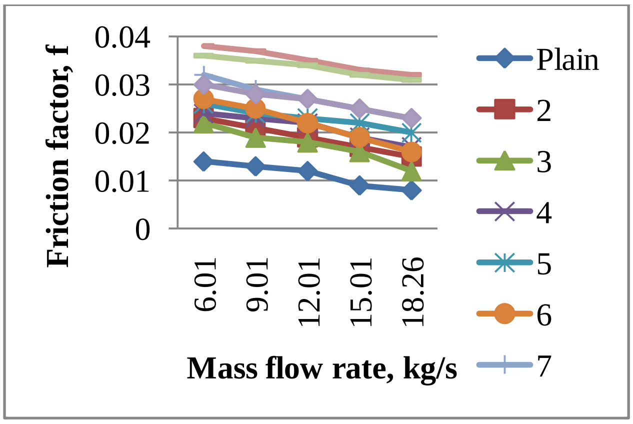

Fig. 17 Variation of friction factor for DPHE at water inlet temperature 25ºC.

It is revealed that just the presence of HTI in the heat exchanger decreases the fan as shown in fig.. These performance parameters significantly decrease with increasing mass flow rate. The experimental results due to increasing the mass flow rate from 4 lpm to 10 lpm at average annulus-side inlet temperatures and average annulus Reynolds numbers. This increase may be due to increasing the heat transfer area in addition to decreasing the annulus-side hydraulic diameter compared with the plain case, which leads to a dramatic increase in the throttling for the annulus flow. Additionally, increasing the helical path and consequently growths the swirl and secondary flows as a result of increasing the centrifugal force. These increase the turbulence level around the outer surface of the internal tube and break the water boundary layer.

In the smooth pipe friction factors are produced by skin-friction. The friction factors are reduced by increasing Reynolds number. It is seen that the value of friction factor decreases with increase in Reynold number. This may be due to the fact that as the Reynold number increases, the thickness of boundary layer decreases therefore, friction factor decreases with increase in Reynold number. It is revealed that just the presence of HTI in the heat exchanger increases the Uan. These performance parameters significantly increase with increasing mass flow rate. The experimental results due to increasing the mass flow rate from 4 lpm to 10 lpm at average annulus-side inlet temperatures and average annulus Reynolds numbers. This increase may be due to increasing the heat transfer area in addition to decreasing the annulus-side hydraulic diameter compared with the plain case, which leads to a dramatic increase in the throttling for the annulus flow. Additionally, increasing the helical path and consequently growths the swirl and secondary flows as a result of increasing the centrifugal force. These increase the turbulence level around the outer surface of the internal tube and break the water boundary layer. Consequently, a significant enhancement of heat transfer in addition to a significant increase in the pressure drop is resulted.

VIII. CONCLUSION

The current study was performed to examine experimentally the hydrothermal performance of horizontal DPHEs with/without a continuous HTI conducted on the outer surface of the inner pipe. The HTI geometrical parameters and operating conditions of the annulus-side were the main parameters throughout this investigation. Ten DPHEs of counter-flow configurations were constructed with/without different HTI height and pitch ratios, and tested at different water flow rates and inlet temperatures in the annulus-side. In the experimental runs, the investigated operating parameters were 6.01 lpm ≤ man ≤ 18.26 lpm. According to the obtained results, the following conclusions can be expressed:

Installing a continuous HTI around the outer surface of the inner pipe of DPHEs significantly increases the heat transfer rate in addition to the pressure drop in the annulus-side when compared with that in the plain annulus heat exchangers.

- The annulus average Nusselt number and friction factor increase with increasing mass flow rate.
- It is obvious that the HTPI for tube with helical tape is more than tube without helical tape and it is unity for all ranges of mass flow rates.
- The friction factors are reduced by increasing Reynolds number. It is seen that the value of friction factor decreases with increase in Reynold number. This may be due to the fact that as the Reynold number increases, the thickness of boundary layer decreases therefore, friction factor decreases with increase in Reynold number.
- It is revealed that just the presence of HTI in the heat exchanger increases the Nuan, han and Uan. These performance parameters significantly increase with increasing mass flow rate. This increase may be due to increasing the heat transfer area in addition to decreasing the annulus-side hydraulic diameter compared with the

plain case, which leads to a dramatic increase in the throttling for the annulus flow.

- Additionally, increasing the helical path and consequently growths the swirl and secondary flows as a result of increasing the centrifugal force. These increase the turbulence level around the outer surface of the internal tube and break the water boundary layer. Consequently, a significant enhancement of heat transfer in addition to a significant increase in the pressure drop is resulted.

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