Research Results

Hydrothermal Performance Of Dual Pipe Heat Exchanger By Dual Helical Tape Insert

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ABSTRACT

Heat exchangers are being 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 for enhancing the rate of heat transfer. 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.

KEYWORDS

Heat exchanger, Double Helical tape insert, Hydrothermal performance, Passive technique, Heat transfer enhancement.

1.INTRODUCTION

1.1 Introduction

Double pipe heat exchangers have an important role in various engineering processes. A simple double-pipe exchanger consists of two pairs of concentric pipes, the two fluids that are transferring heat flow in the inner and outer pipes, respectively. The fluids usually flow through the exchanger in opposite directions (counter- current flow). Double-pipe exchangers are commonly used in applications involving relatively low flow rates and high temperatures or pressures, for which they are well suited. 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. 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.

1.2 Double Pipe Heat Exchanger A double pipe heat exchanger (also sometimes referred to as a 'pipe-in-pipe' exchanger) is a type of heat exchanger comprising a 'tube in tube' structure. As the name suggests, it consists of two pipes, one within the other. One fluid flows through the inner pipe (analogous to the tube-side in a shell and tube type exchanger) whilst the other flows through the outer pipe, which surrounds the inner pipe (analogous to the shell-side in a shell and tube exchanger). A cross-section of a double pipe exchanger would look something like this:

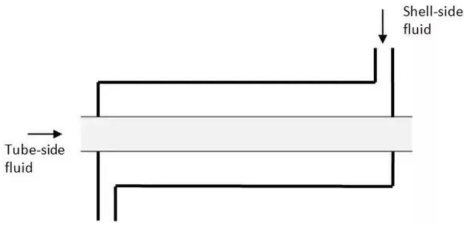

Fig. 1.1: Double pipe heat exchanger

They often have a U-tube structure to accommodate thermal expansion of the tubes without necessitating expansion joints, as illustrated below:

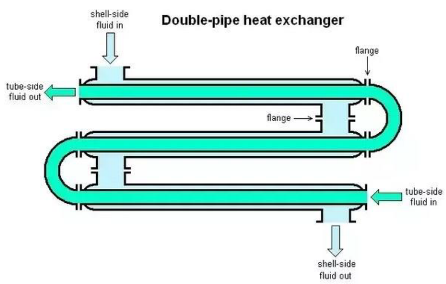

Fig. 1.2: U-type double pipe heat exchanger

1.3 Heat Transfer Augmentation Techniques Heat transfer augmentation techniques are generally classified into three categories namely: Active techniques, Passive techniques and Compound techniques.

1. Active Techniques: Active techniques involve some external power input for enhancement of heat transfer. Example: Mechanical aids, Surface vibrations, Fluid vibrations and Jet impingement.

2. Passive Techniques: Passive techniques do not require any direct input of external power. They generally use geometrical or surface modifications to the flow channel by incorporating inserts or additional devices. Example: Rough surfaces, Extended surfaces, Swirl flow devices and Coiled tubes.

3. Compound Techniques: Combination of active and passive techniques may be employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by any of those techniques separately. This simultaneous utilization is termed compound enhancement.

II. LITERATURE REVIEW

Salem et al (2021) examined 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.

Lachi et al. (2020) studied time constant of a DPHE and a shell and tube heat exchanger. The particular purpose of this investigation was to classify the characteristics of these heat exchangers in a transient condition, especially the time when abrupt changes in inlet velocities are considered. Upon carrying out this study, a model with two parameters of time delay and time constant has been employed. It is also noted that the analytical term was derived by applying energy balance equation. Moreover, it was stated that an experimental method was used to validate the numerical data which the highest observed difference found to be less than ten percent.

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Aicher and Kim (2018) investigated the effect of counter flow in nozzle section of a DPHE which were mounted on the wall of the shell side. It turned out that the counter flow in nozzle section had a significant effect on heat transfer and pressure drop. It was also concluded that the very effect would be more conspicuous, if the heat exchanger were small and also the ratio of free cross section areas were low enough. They also presented experimental correlations to predict heat transfer rate in turbulent flow.

Ma et al. (2018) experimentally investigated the effects of supercritical carbon dioxide (SCO2) in a DPHE in which the effects of pressure, mass flux and buoyancy force of the SCO2-side were broadly studied. On one hand, it was observed that pressure increase of the gas-side conspicuously caused both the overall and the gas-side heat transfer rates to be decreased. On the other hand, it was obvious that the flow rate of the waterside, in comparison with the gas-side, was the key element of the heat transfer rate. Moreover, a mathematical correlation based on Genetic Algorithm was presented for predicting heat transfer rate.

Rennie and Raghavan (2018) investigated a double pipe helical heat exchanger for both parallel and counter flow configurations. The corresponding heat transfer rates of inner tube and the annulus were calculated using Wilson plots. It is well worth noting that the performance evaluation criterion of both configurations was identical, while surely the heat transfer regarding to the counter flow configuration was higher than its counterpart which was due to a higher temperature difference. The above-mentioned performance evaluation criterion (PEC) is the comparison of heat transfer coefficients between the enhanced tube and smooth tube under the same pumping power condition.

Dizaji et al. (2018) did an experimental study of heat transfer and pressure drop of corrugated tubes in a DPHE which turned out to perceive much importance in the field (Fig. 2.1). Both inner and outer tubes were corrugated in concave and convex shapes. Working fluids in the experiments were hot and cold water which flowed in the inner and outer tube of the heat exchanger, respectively. Research findings showed that the highest effectiveness was obtained for a case when the inner tube and the outer tubes had the convex and concave corrugated configurations, respectively.

Bhadouriya et al. (2018) investigated heat transfer and pressure drop of a DPHE both experimentally and numerically in which the major objective was the effect of twist ratio of the inner tube on the flow characteristics (Fig. 2.2). A uniform wall temperature at the inner wall of annulus was a boundary condition for the outer flow. Working fluids in the experiments were water and air which flowed in the inner (square duct) and the annulus of the heat exchanger, respectively. The results showed that this geometry change led to an increase in heat transfer rate and pressure drop in all flow regimes. The results of the present paper will help the engineers design more compact heat exchangers. It was also concluded that, unlike smooth tube, Nusselt number in the laminar flow regime was dependent on the flow characteristics and physical parameters such as Reynolds number and twist ratio.

Tang et al. (2018) investigated the effects of twisted inner tube of a DPHE which was carried out experimentally and numerically. In the experimental process, the inner tube had three different cross section shapes which were circular, oval and tri-lobed (Fig. 2.3); while the outer tube was a simple cylindrical tube. Upon having a higher performance evaluation criterion, an intense concentration was shown to the above-mentioned tri-lobed cross section along with the simple outer tube. Moreover, a broad range of studies were carried out in numerical process of the study, especially in different cross-section shapes.

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.

III. PROBLEM FORMULATION

According to Previous work these are known to be economic heat transfer augmentation tools. The double helical twisted tapes insert is found to be suitable in a laminar flow regime and the latter is suitable for turbulent flow. The thermo hydraulic behaviour of an insert mainly depends on the flow conditions (laminar or turbulent) apart from the insert configurations. Heat transfer augmentation techniques (passive, active or a combination of passive and active methods) are commonly used in areas such as process industries, heating and cooling in evaporators, thermal power plants, air-conditioning equipment, refrigerators, radiators for space vehicles, automobiles, etc. Passive techniques, where inserts are used in the flow passage to augment the heat transfer rate, are advantageous compared with active techniques, because the insert manufacturing process is simple and these techniques can be easily employed in an existing heat exchanger. In design of compact heat exchangers, passive techniques of heat transfer augmentation can play an important role if a proper passive insert configuration can be selected according to the heat exchanger working condition (both flow and heat transfer conditions). In the past decade, several studies on the passive techniques of heat transfer augmentation have been reported. The present work is deal with the passive augmentation techniques in the recent past and will be useful to designers implementing passive augmentation techniques in heat exchange.

IV. PROPOSED METHODOLOGY

4.1 Specifications Of Heat Exchanger Used The experimental study is done in a double pipe heat exchanger having the specifications as listed below:- Specifications of Heat Exchanger: Inner pipe ID = 11 mm Inner pipe OD = 12 mm Outer pipe ID = 76.2 mm Outer pipe OD = 81 mm Material of construction of inner tube= Copper Pipe length= 800 mm Water at room temperature was allowed to flow through the inner pipe while hot water (set point 50°C) flowed through the annulus side in the counter current direction.

Properties	Value Measured
Melting point	1083°C
Density	8.94 X 10^3 kg/m^3 at 20°C
Thermal expansion coefficient	17.7 X 10^{-6} per °K
Thermal conductivity	305 – 355 W/(m.K)
Specific heat capacity	0.385 kJ/(kg.K)
Electrical conductivity (annealed)	75 – 90% IACS
Electrical resistivity (annealed)	0.0192 – 0.0230 microhm at 20°C
Modulus of elasticity	117 Gpa
Modulus of rigidity	44 Gpa

4.2 Experimental Planned

One DPHEs of counter-flow configurations were constructed with DHTI in the annulus side, while the other one is without any inserts. The DHTIs were constructed with different height and pitch ratios. In the paper of salem et al. 2018 the annular pipe of all tested heat exchangers was made up of PVC pipes of 76.2 mm inner diameter and 5 mm wall thickness. Its ends were closed using PVC caps and adhesives to prevent any leakage. In addition, its outer surface was thermally isolated with thick insulation consisting of layers of ceramic fiber, asbestos rope, and glass wool. While the internal pipe of all heat exchangers was a copper tube of 11.1 mm internal diameter and 12.5 mm external diameter with a length of 800 mm for all tested heat exchangers. However in our work we change the PVC pipe to MS pipe for annulus body for checking the performance of parameters.

$$\overline{Nu}_{an} = \frac{\frac{f_{an}}{2}(Re_{an} - 1000)Pr_{an}}{1 + 12.7\sqrt{\frac{f_{an}}{2}}\,(Pr_{an}^{2/3} - 1)}\left[1 + \left(\frac{d_{an,\,h}}{L_{an}}\right)^{2/3}\right]$$

4.3 VALIDATION EXPERIMERNTS The result of the proposed research work will be validated by comparing the experimental results with the results reported in literature Gnielinski and Filonenko. The experiment will be conducted with finned tube with helical tape by varying cold water flow rate and computation of Nusselt number and fanning friction factor with that of plain tube. If the readings are in close agreement it will be concluded that the proposed experimental setup is acceptable.

V. RESULTS AND DISCUSSION

5.1 Validation The validation of the procedures in determining the heat transfer coefficients and friction factors in the annulus side was done by using the above-mentioned analysis methods. The obtained results were compared with the confirmed heat transfer and friction factor correlations.

(Cold) Water flow rate (lpm)	T_{hi} (°C)	T_{ho} (°C)	T_{ci} (°C)	T_{co} (°C)	Nu (Present)	Nu [V. Gnielinski (2016)]
4	50	49.2	29.2	32.4	765.04	778.17
5	50	48.6	29.6	32.5	782.38	798.34
6	50	48.5	30.2	32.6	838.24	852.47
7	50	48.4	30.5	32.5	988.16	1045.04
8	50	48.7	30.6	32.6	1182.32	1234.15
9	50	47.8	30.8	32.8	1224.27	1282.71
10	50	47.5	30.9	32.9	1474.61	1519.62

Table 5.1: Validation of Nusselt number for DPHE without Helical Tape

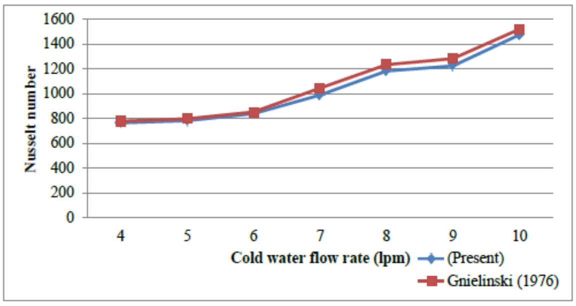

Fig. 5.1: Validation of Nusselt (experimental value) with reference to research work by Genielinski (1976)

In this analysis, experiment is conducted on the internal tube of the DPHEs that are tested at seven different mass flow rates without helical tape for the illustrated operating conditions in table 5.1. Table 5.1 demonstrates a sample of the obtained results for fan for at different flow rates of the annulus-side fluid. Comparisons of the experimental Nuan with those predicted by the proposed correlations are illustrated in Fig. 5.1. From this figure, it is obvious that the proposed correlations are in good agreement with the present experimental data.

(Cold) Water flow rate (lpm)	T_{hi} (°C)	T_{ho} (°C)	T_{ci} (°C)	T_{co} (°C)	f_{an} (Present)	f_{an} (G.K. Filonenko)
4	50	49.2	29.2	32.4	0.065	0.068
5	50	48.6	29.6	32.5	0.048	0.055
6	50	48.5	30.2	32.6	0.035	0.042
7	50	48.4	30.5	32.5	0.022	0.028
8	50	48.7	30.6	32.6	0.018	0.024
9	50	47.8	30.8	32.8	0.017	0.022
10	50	47.5	30.9	32.9	0.013	0.018

Table 5.2: Validation results of fanning friction factor for DPHE without Helical Tape

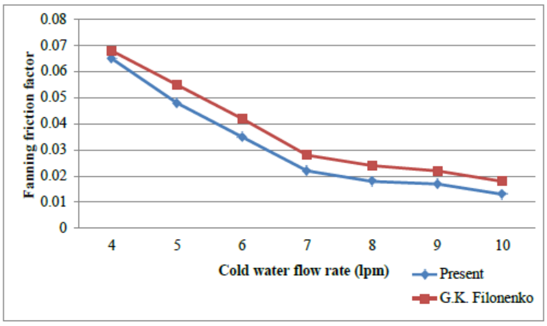

Fig. 5.2: Validation of fanning friction factor (experimental value) with refrence toresearch work by G.K. Filonenko

5.2 Experimental Results Of Dphe Without Helical Tape

5.2.1 Variation of Cold And Hot Outlet Temperature In this work DPHE without helical tape is simulated and results have been presented as shown below. After that comparison will be done between DPHE with or without helical tape.

(Cold) Water flow rate (lpm)	T_{hi} (°C)	T_{ho} (°C)	T_{ci} (°C)	T_{co} (°C)
4	50	49.2	29.2	32.4
5	50	48.6	29.6	32.5
6	50	48.5	30.2	32.6
7	50	48.4	30.5	32.5
8	50	48.7	30.6	32.6
9	50	47.8	30.8	32.8
10	50	47.5	30.9	32.9

Table 5.3: Inlet and exit temperature for hot and cold fluid in case of DPHE without Helical Tape

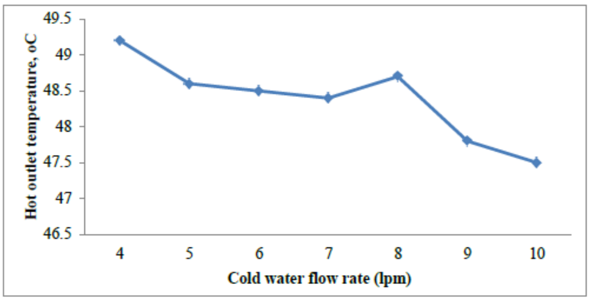

Fig. 5.3: Variation of hot outlet temperature vs cold water flow rate

VI. CONCLUSION AND FUTURE SCOPE

6.1 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. One 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 4 lpm ≤ man ≤ 10 lpm. According to the obtained results, the following conclusions can be expressed:

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

2. The annulus average Nusselt number and friction factor increase with increasing mass flow rate.

3. 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.

4. 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.

6.2 Future Scope

1. Further study can be extended to explain the effects of the HTI pitch and height of helical tape by develop more tubes.

2. Generally, the accuracy of the experimental results hinges on the accuracy of the individual measuring instruments and techniques.

3. The uncertainties in the measured annulus diameters and lengths, in addition to the HTIs pitch, height and diameter were ± 0.5 mm.

4. The uncertainty of the parameters was also calculated based upon the root sum square combination of the effects of each of the individual inputs.

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