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Thermal Performance Investigation of Grooved Tube with Twisted Tape Insert and Al_2O_3 Nanofluid

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Abstract - *In this paper, experimentally investigated for heat transfer and friction characteristics of dual-pipe heat exchanger for single-phase forced convective flow of Al_2O_3 nanofluid with typical twisted tape (TT tape) inserted in the inner tube having grooves on both side. And effects of twisted tapes on heat transfer enhancement were studied through a circular grooved tube using Al_2O_3 nanofluid as testing fluid in turbulent conditions with a range of Reynolds number between 3000 and 20000. In the dual-pipe heat exchanger, hot water was cooled in the inner tube and cold water was as cooling fluid between the inner tube and the outer tube. The results showed that the combined use of twisted tape insert and Al_2O_3 nanofluid in grooved tube is performed better on heat transfer enhancement than the single enhancement techniques. The maximum values of TPF (Thermal Performance Factor) with the 0.15% Al_2O_3 nanofluid and twisted tape in grooved tube reached 1.3823 in experimental flowing conditions.*

***Key Words*: Twisted Tape, Nanofluid, Grooved Tube, Tabular Heat Exchanger, Thermal Performance Factor**

1. INTRODUCTION

A heat exchanger is a device that provides the transfer of thermal energy between two or more fluids, which are at different temperatures and are in thermal contact with each other. The word "Exchanger" really applied to all types of equipment in which heat is exchanged but it is often used specially to the equipment in which heat is exchanged between two process streams that are at different temperature and are separated by a solid wall. Heat exchangers are widely used as the essential units in heat extraction and recovery systems in industries. Heat exchangers play an important role in various fields such as chemical engineering, metallurgy, electric power generation, refrigeration and air-conditioning, aerospace industries, oil and petrochemical industries, sugar industries, pharmaceutical industries, chemical reactors etc. The effectiveness of heat exchanger is low i.e. actual heat transfer is low as compared to maximum heat transfer. Energy recovery is of prime importance to optimize the energy consumption in industry. To achieve maximum utilization of thermal energy, several heat transfer enhancement techniques have been used in many thermal engineering applications such as nuclear reactor, chemical reactor, chemical process, automotive cooling, refrigeration, and heat exchanger, etc. Heat transfer enhancement techniques are powerful tools to increase heat transfer rate and thermal performance of heat exchangers. The purpose behind the augmentation is to reduce the size of the heat exchanger required for specified heat duty, to upgrade the capacity of an existing heat exchanger, or to reduce the pumping power. Another advantage is the reduction of temperature driving force, which reduces the entropy generation and increases the second law efficiency. In addition, the heat transfer enhancement enables heat exchangers to operate at smaller velocity, but still achieve the same or even higher heat transfer coefficient. This means that a reduction of pressure drop, corresponding to less operating cost, may be achieved. All these advantages have made heat transfer enhancement technology attractive in heat exchanger applications. The design procedure of heat exchangers is complex because it needs the analysis of heat transfer rate, pressure drop and efficiency plus issues like long term endurance and easy maintenance. One of the main categories of increasing heat transfer methods is called as passive technique. It means that there is no need for any kind of extra power source and the heat transfer can increase just using modified surfaces or modified geometries. This method includes the techniques such as treated surface , extended surfaces, rough surfaces, coiled tubes , vortex generator devices, displaced enhancement devices, and additives to the fluids . Also twisted tape (TT), grooved tube and combined of this is one of which can improve heat transfer rate and it is studied completely through the current paper. At any time that twisted tape (TT) inserts are used, in conjunction with the enhancement in the heat transfer rate, the pressure drop increases.

1.1 Properties of Al_2O_3 Nanofluid

A. Density

The values from the regression equations developed by Pak and Cho. Hence for finding the density value of all the concentrations of Al_2O_3 nanofluid, the following correlation is used[15].

$$\rho_{nf} = \phi \, \rho_p + (1-\phi) \, \rho_w$$

B. Specific Heat

The specific heat is one of the important properties and plays an important role in influencing heat transfer rate of nanofluids. Xuan and Roetzel gives correlation for finding specific heat. Hence for finding the Specific heat of all the concentrations of Al_2O_3 nanofluid, following correlation is used[16].

$$C_{nf} = \frac{\varphi(\rho\, C)_p + (1-\varphi)(\rho\, C)_w}{\varphi\rho_p + (1-\varphi)\rho_w}$$

C. Thermal Conductivity

The experimental studies on nanofluids containing nanoparticles are expected to give more thermal conductivity and lower specific heats over conventional fluids. Thermal conductivity of nanofluid is obtained by F.J. Wasp [17].

$$\frac{k_{nf}}{k_w} = \frac{k_p + 2k_w - 2\varphi(k_w - k_p)}{k_p + 2k_w + \varphi(k_w - k_p)}$$

D. Viscosity

From the literature review it is found that the viscosity value obtained by the correlation is very nearer to the experimentally determined value. Hence for finding viscosity value of all the concentration of nanofluid the following correlation is used. It is given by Einstein [18].

$$\frac{\mu_{nf}}{\mu_w} = 1 + 2.5\varphi$$

Table 1. Shows the thermo physical properties of $A1_2O_3$ nanofluid for each concentration of Al_2O_3 nanofluid obtained by using above correlation at 300K.

Table-1: Thermophysical properties of Al_2O_3 nanofluid

Sr. No.	% Volume Concentration	ρ_{nf} (kg/m³)	C_{pnf} (J/kg k)	μ_{nf} (Ns/m²)	k_{nf} (W/mK)
1.	0.00	997.00	4179.00	0.855	0.613
2.	0.05	998.45	4172.39	0.856	0.614
3.	0.10	999.89	4165.79	0.858	0.615
4.	0.15	1001.34	4159.22	0.859	0.616

2. EXPERIMENTAL SET-UP

The experimental set-up consists of tube-in-tube heat exchangers of four types as a test section. 1. Both tube plain tube, 2. Inner tube grooves on inner side, 3. Inner tube grooves on both side, 4. Inner tube grooves on both side and outer tube grooves on inner side. The cold fluid enters the heat exchanger through the outer tube and hot fluid flowing through the inner tube. Cold and Hot fluid flowing opposite to each other i.e. Counter flow heat exchanger. A schematic diagram of the experimental set-up is shown in fig. 1.

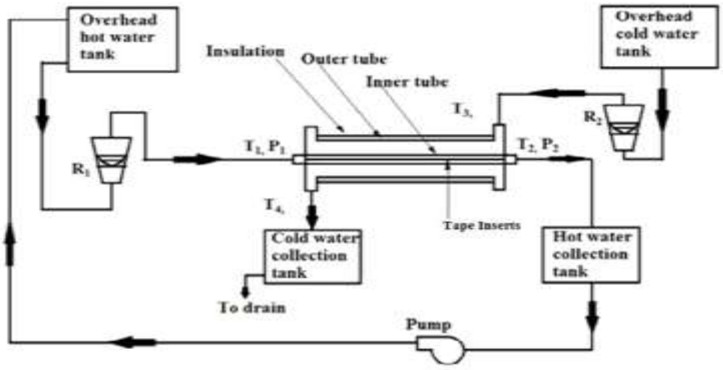

Fig-1: Block Diagram of Experimental Setup

The outer tube is having cylindrical cross section with 18mm inside diameter, 22.5mm outside diameter and length is 750mm. As the cold fluid passes through the tubular heat exchanger, its temperature increases then the heated cold fluid is release in basin where it rejects heat. The hot water flowing into the inner tube is having cylindrical cross section with 9mm inside diameter, 13mm outside diameter and length is 850mm supplied with the help of pump from the hot fluid tank. The hot fluid tank consists of one immersion type heaters having capacity 2000W which is connected to the on/off controller. Two rotameters (R_1 and R_2) are used in the experimental set-up for measuring the volume flow rate of hot and cold fluid. These two rotameters are connected in series with the direction of flowing fluid to measure the volume flow rate of both the fluids. Both the rotameters are calibrated and having the range of 0 to 5 lpm. The connections of fluid flowing through the inner and outer tube are made by using T joints and pipes which connects the rotameters at the heat exchanger inlet sections. Four K-type thermocouples (T_1, T_2, T_3 and T_4) are used for measurement of hot and cold fluid temperatures with an accuracy of 0.1ºC. All the thermocouples are calibrated and connected to the digital temperature indicator. Pressure drop at inlet and outlet (P_1 and P_2) of hot fluid is measured by U-tube differential manometer. For the experimentation purpose at a time, only one tube-in-tube heat exchanger is in working. Bypass line is placed for both hot fluids so that only desired quantity of fluid flow should take place and excess quantity of fluid is return to the respective tanks. The physical properties of fluids flowing through the tubes of test section are assumed to be constant along the tube length and evaluated at the average bulk temperature for each run.

Table-2: Specifications of Instrument Used

Sr. No.	Equipment/ Instrument Name	Dimension/ Range/ Capacity	Least Count
1	Hot Fluid Tank	10 Litres	-
2	Pump	0.5 hp	-
3	Heater	2000 W	-
4	Rotameter	0-5 lpm	0.5 lpm
5	U-tube differential manometer	0-300 mm	1mm
6	Thermocouple (Cr-Al) K type	0 to 900 ºC, 5 Nos.	±0.1 ºC
7	Digital Temperature Indicator	0-200 ºC, 6 channels	±0.1 ºC
8	Temperature Controller	ON/OFF type, k-type thermocouple	±1 ºC

3. DATA REDUCTION

Several formula used in this paper are shown as follows. The energy balance between the heat supplied by hot fluid and energy absorbed by the flowing liquid is established using Eqns. (1) and (2) for every set of data and the experimental heat transfer coefficient is estimated with Eqn. (3), all properties is calculated at bulk mean temp.

Bulk mean temp

$$T_{mean} = \frac{T_{in} + T_{out}}{2}$$

Heat Transfer rates of two fluid stream are calculated as

$$Q_h = m_h \times C_p \times (T_{in} - T_{out})$$

$$Q_c = m_c \times C_p \times (T_{out} - T_{in})$$

$$Q = h \times A_s \times (T_{wi} - T_{mean})$$

The average Heat transfer coefficient and Nusslet number can be calculated as follows

$$h = \frac{Q_h + Q_c}{2 \times A_s \times (T_b - T_w)} \qquad Nu = \frac{h\, d_i}{k_f}$$

Where, $A_s = \pi \times di \times L$

$$T_{wi} = T_{wo} - \frac{Q.\,\mathrm{In}\left(\frac{d_o}{d_i}\right)}{2\pi kL}$$

$$T_{wo} = \sum_{i=n}^{n} \left(\frac{T_i}{n}\right)$$

With the measured parameter of pressure drop, the Friction factor can be calculated using formula:

$$f = \frac{\Delta p}{\left(\frac{1}{2}\rho v^2\right)\left(\frac{L}{d_i}\right)}$$

Velocity of flow can be calculated as,

$$V = \frac{\dot{m}}{\rho\, A_c}$$

Where, $A_c = \frac{\pi \times {d_i}^2}{4}$

The thermal performance factor (η):

The thermal performance factor is defined as the ratio of the Nusselt number ratio to the friction factor ratio at the same pumping power. It is given by expression:

$$\eta = \frac{Nu/Nup}{\left(\frac{f}{fp}\right)^{1/3}}$$

4. RESULT AND DISCUSSION

4.1 Verification of Experimental Results

A. Plain Tube without Twisted Tape

To gain confidence on experimental data throughout the research, the experimental data of the plain tube with water as the working fluid were firstly compared with those from the open literatures which are Dittus–Boelter correlations for Nusselt number and Blasius correlations for the friction factor as under.

Nusselt number correlations for the plain tube:

Correlation of Dittus-Boelter:

$$Nu = 0.023Re^{0.8}Pr^{0.4}$$

Correlation of Gnielinski:

$$Nu = \frac{\left(f/_8\right)(Re - 1000)Pr}{1 + 12.7\left(f/_8\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)}$$

Friction factor correlation for the plain tube:

Correlation of Blasius:

$$f = 0.318\, Re^{-0.25}$$

Verification of the Nusselt Number and friction factor in the plain tube is shown in chart-1

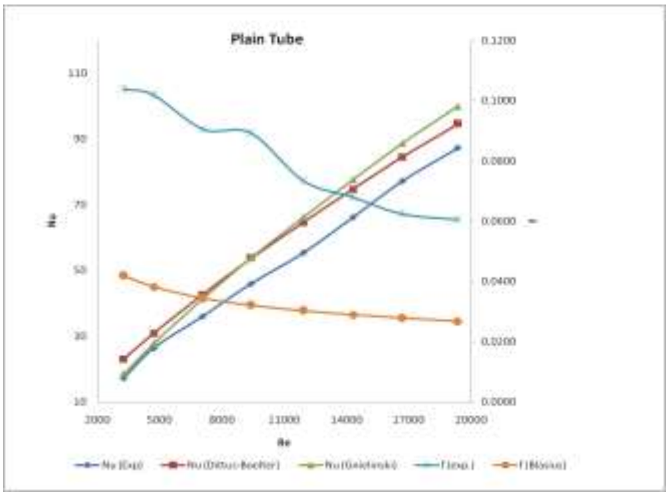

Chart-1: Reynolds Number variation with Nusselt Number

The experimental Nusselt number was in satisfactory agreement, the mean experimental Nusselt number of plain tube were 7.76-24.95% and 6.54-16.34% lower than that of the Dittus-Boelter correlations and Gnielinski Correlations respectively. Experimental mean friction factor of plain tube were 2.24-2.77 and 1.22-1.83 times higher than that of the Blasius correlations and numerical data. According to the comparative results mentioned above, it can be concluded that the present facility was reliable and experimental data was accurate enough. These provide a strong confidence in the present investigation of the heat transfer and flow friction in the tube.

B. Effect of Twisted tape insert and nanofluid in grooved tube.

i. Heat transfer

The Al_2O_3 nanoparticles dispersed in water with volume concentrations of 0.05%, 0.10% and 0.15% and used as working fluid. Chart-2 shows heat transfer enhancement effect due to both, twisted tape and nanofluid in grooved tube. It was found that heat transfer (Nusselt number) increased when increasing nanofluid concentration. For the tubes with twisted tapes, the average Nusselt number of nanofluid with Al_2O_3 concentrations of 0.05%, 0.10% and 0.15% by volume, was around 30.37-72.80%, 38.86-79.86% and 44.42-84.24% higher than the base fluid (water) in plain tube.

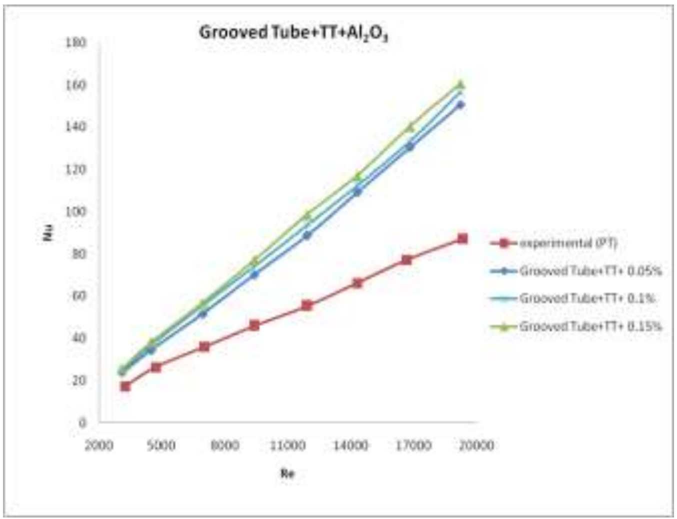

Chart-2: Reynolds Number variation with Nusselt number for Nanofluid

In other words, nanofluid with Al_2O_3 concentration of 0.15% by volume gave 6.62-11.74% and 2.07-5.35% higher Nusselt number than the ones with Al_2O_3 concentrations of 0.05% and 0.10% by volume, respectively. Moreover, the increase of nanoparticle loading also offers higher contact area between nanoparticles and the base fluid as well as twisted tapes. The enhanced heat transfer is due to thermal conductivity and collision of nanoparticles. However, loading too much nanoparticles into nanofluid beyond the optimum concentration may diminish the fluid movement and heat transfer rate due to increased fluid viscosity. In the present work, the decrease of Nusselt number was not found when increasing in concentration. This implies the present nanofluid concentration range did not exceed the optimum level. Therefore, the effect of Al_2O_3 nanoparticles was more pronounced for thermal conductivity and the collision than viscosity.

ii. Friction Factor

Chart-3 shows the effect on friction factor by addition Al_2O_3 nanoparticles in water at different concentration (volume fraction 0.05%, 0.10%, 0.15%). It was found that friction factor increases by increasing nanoparticle concentration. For the present range, nanofluid with Al_2O_3 concentrations of 0.05%, 0.10% and 0.15% by volume respectively caused 1.9119-5.9787, 1.9151-6.9751 and 2.1613-7.9716 times higher friction factor compared to those of the base fluid in plain tube. In other words, nanofluid with Al_2O_3 concentration of 0.15% by volume caused 7.14-33.33% and 5.56-14.29% higher friction factors than the ones with Al_2O_3 concentrations of 0.05% and 0.10% by volume, respectively.

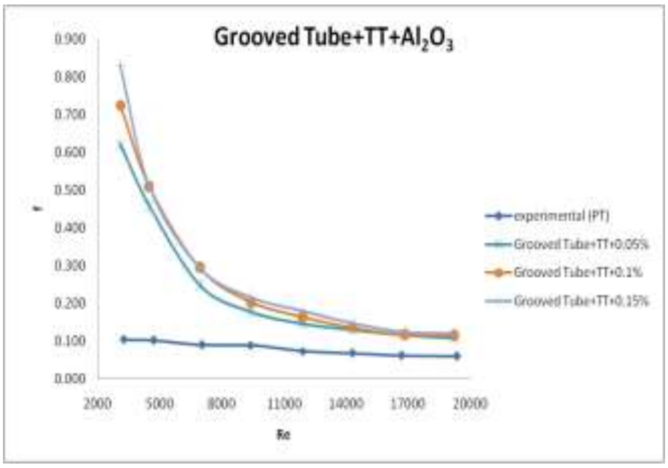

Chart-3: Reynolds Number variation with Friction Factor for Nanofluid

The increase of friction loss is directly caused by the increases of fluid viscosity and shear force on tube wall acted by nanoparticles. In particular, at low Reynolds number all nanofluid yielded higher pressure loss than the base fluid. However, the results indicate that utilizing twisted tape and nanofluid in the present concentration range is an insignificant friction loss penalty.

iii. Thermal performance factor

Chart-4 shows the effect of nanofluid concentration on thermal performance factor. Evidently, nanofluid with higher Al2O3 concentrations yielded higher thermal performance factors. Depending on Reynolds number, thermal performance factors given by Al_2O_3 nanofluid with concentrations of 0.05%, 0.10% and 0.15% by volume were 0.7626-1.3454, 0.7592-1.3740 and 0.7442-1.3823 respectively. Comparatively, nanofluid with Al2O3 concentration of 0.15% by volume offered 2.74-6.95% and 1.39-4.00% higher thermal performance factors than the ones with Al_2O_3 concentrations of 0.05% and 0.10% by volume, respectively.

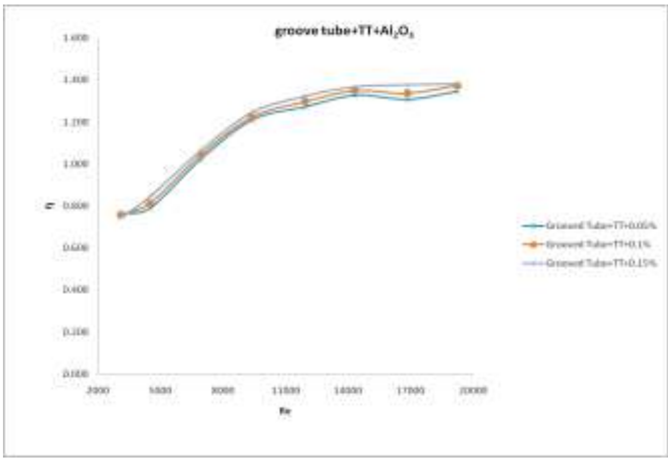

Chart-4: Reynolds Number variation with Thermal Performance Factor for Nanofluid

5. CONCLUSIONS

Experimental investigations of heat transfer, friction factor and thermal performance factor characteristics of tube in tube heat exchanger fitted with plain twisted tape and grooved tube with different configurations were tested. In present work, used water and Al_2O_3 nanofluid as working fluid at different mass flow rates and varied Reynolds number in between 3,000-20,000. From experimentation the following conclusions can be drawn:

1. The Nusselt number for twisted tape and nanofluid with Al_2O_3 concentration of 0.05%, 0.10% and 0.15% by volume in grooved tube (test section 3) were respectively 30.37-72.80%, 38.86-79.86% and 44.42-84.24% higher than that of the base fluid (Water) in plain tube.
2. The friction factor for twisted tape and nanofluid with Al_2O_3 concentration of 0.05%, 0.10% and 0.15% by volume in grooved tube (test section 3) were respectively 1.9119-5.9787, 1.9951-6.9751 and 2.1613-7.9716 times higher than that of the base fluid (Water) in plain tube.
3. The Thermal performance factor for twisted tape and nanofluid with Al_2O_3 concentration of 0.05%, 0.10% and 0.15% by volume in grooved tube (test section 3) were respectively 4.42-34.54%, 4.42-37.40% and 6.58-38.23% higher than that of the base fluid (Water) in plain tube.

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