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Investigation of Heat Transfer and Friction Factor Characteristics of Two Phase Nano Fluids by Inserting Twisted Tape and Helical Inserts in the Tube

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Abstract: *The heat transfer is an important aspect to be studied especially in power plants, chemical industries, process plants, air conditioning and automotive industries. The conventional liquids such as distilled water, glycols and mineral oils are very often used as heat transfer fluids. It is proposed to compare the experimental heat transfer and friction factor of nanofluids of three different concentrations with the correlations available, the heat transfer and friction factor characteristics of the same CuO nanofluids will be investigated by inserting twisted tape and helical inserts in the tube. The influence of nanoparticle concentration, mass flow rate and the twist ratio of the tape on Nusselt number will be studied and discussed. Finally, generalized regression equations to predict Nusselt number and friction factor will be developed for the data that will be obtained in the experimental investigation. The experiments are proposed to be conducted for laminar and transition flow conditions in the same Reynolds number range for both the cases of plain tube and with inserts. The Properties of nanofluids can be advantageously altered to make them suitable particularly in heat transfer applications. The two phase nanofluids are having their thermal conductivities comparatively higher than those of base fluids. The thermo physical properties are required for the calculation of convective heat transfer coefficient and Nusselt number are thermal conductivity, viscosity, specific heat and density. Properties of single phase fluids are well documented and are available in heat transfer data books. But on the other hand the thermo physical properties of two phase fluids are not much available and needs to be measured by conducting experiments. The experimental procedure for measurement of nanofluid properties vary with temperature and the particle volume concentration in the basefluid. Ethylene glycol and propylene glycols are anti-freeze liquids and normally mixed in water in different proportions. Such blends are used as a heat transfer fluid in cold climatic conditions, to lower the aqueous freezing point of heat transfer fluids in automobile radiators and similar heat exchangers.*
Keywords: *Nanofluid, heat transfer, twisted tapes and helical inserts*

I. Introduction

With the advent of recent advances in the nanotechnology, nano material synthesis, processing and characterization was made possible and production of both metallic and non-metallic particles in nano dimension with controlled structure is realized. In nano materials several parameters like shape, size and rapid interface alter the properties of nano particles which are quite different for the same materials in macroscopic dimension. The base liquids containing nano particles are known as Nano fluids.

The thermo physical properties of two phase fluids are not much available and needs to be measured by conducting experiments. The elaborated experimental procedure for measurement of nanofluid properties reports that nanofluid properties vary with temperature and the particle volume concentration in the base fluid.

1.1 Introduction of Nano Fluids: With the progressive development of thermo science and thermal engineering, many efforts have been devoted to heat transfer enhancement. Among them, application of additives to liquids is often involved. Since the flow media themselves may be the controlling factor of limiting heat transfer performance, solid additives are suspended in the base liquids in order to change transport properties, flow and heat transfer features of the liquids. Numerical and experimental studies have been conducted in order to improve heat transferred by these techniques.

The demand of reduction of the cost and dimensions of heat exchanger has motivated the searchers to investigate different ways of heat transfer enhancement. Passive heat transfer enhancement techniques are mostly preferred due to their simplicity and applicability in many applications. Furthermore, in passive techniques, there is no need of any external power input except to move the fluid. The devices in this category include surface coating, rough surfaces, extended surfaces, turbulent / swirl flow devices, convoluted (twisted) tube, and tube inserts. Various kinds of inserts have been employed in the heat exchangers such as helical/twisted tapes, coiled wires, ribs / fins / baffles and winglets. Enhanced tubes with different inserts are used extensively in the refrigeration,air-conditioning and commercial heat pump industries as well as in the chemical, petroleum, and numerous other industries. Using inserts in tubular heat exchangers not only reduced the heat exchanger size but also provided thermal, mechanical, and economic advantages in heat exchangers. The quantities of the two fluids resident in heat exchangers as an important safety consideration, have been greatly decreased by compact enhanced designs. It is observed from the literature review that thermo physical properties of nanofluids are evaluated by different research groups. Nanofluids with different metals and metal oxide nanoparticles were prepared using water as the base fluid in majority of earlier experimental studies. Heat transfer and friction factor studies on different nanofluids were carried out by researchers at and above room temperature by varying nanofluid concentration. The variation of heat transfer coefficient with nanoparticle concentration and the Reynolds number for nanofluids flowing in a circular tube, in laminar and turbulent flow regimes were investigated as per the literature survey.

II. Preparation of Nanofluids

It is proposed to investigate the heat transfer and friction factor characteristics of propylene glycol based CuO nanofluids in different flow regimes by preparing CuO nanofluids and carryout experimental work and to measure temperature dependent thermo physical properties of CuO Nanofluids viz., thermal conductivity, density, absolute viscosity and specific heat. It is also proposed to study the effects of nanoparticle volume concentration and temperature on properties of CuO nanofluids.

In the present investigation it is also intended to study the heat transfer and friction factor of CuO nanofluids flowing in a plain tube under constant heat flux as the boundary condition. For this it is proposed to prepare CuO nanofluids of three different volume concentrations using water-propylene glycol (70:30 by volume percentage) blend as the host fluid and to conduct experiments in laminar to transition conditions in the Reynolds number ranging from $1000<Re<10000$ under constant heat flux as boundary conditions. It is proposed to compare the experimental heat transfer and friction factor of nanofluids of three different concentrations with the correlations available in the literature. Heat transfer and friction factor characteristics of the same CuO nanofluids will be investigated by inserting twisted tape and helical inserts in the tube. The influence of nanoparticle concentration, mass flor rate and tape twist ratio on Nusselt number will be studied and discussed. Finally, generalized regression equations to predict Nusselt number and friction factor will be developed for the data that will be obtained in the experimental investigation. The experiments are proposed to be conducted for laminar and transition flow conditions in the same Reynolds number range for both the cases of plain tube and with inserts.

2.2 Nanofluids Properties

Before the study of the convective heat transfer performance of the nanofluids the properties of nanofluid must be known accurately.By assuming that the nanoparticles are well dispersed in the fluid, the concentration of nanoparticles may be considered uniform throughout the tube. Although this assumption may be not true in real systems because of some physical phenomena such as particle migration, it can be a useful tool to evaluate the physical properties of a nanofluid. The density of nanofluid is calculated by the mixing theory as:

$$\rho = \emptyset \rho_P + (1 - \emptyset)\rho_{bf}$$

The specific heat capacity of nanofluid can be calculated based on the thermal equilibrium model as follows

III. Methodology

The two phase approach can be determined by mixture model, Eulerian-Eulerian, Eulerian-Lagrangian models. The experimental work is also carried out to study on passive heat transfer enhancement in pure fluids using inserts. Ethylene glycol and propylene glycols are anti-freeze liquids and normally mixed in water in different proportions. Such blends are used as a heat transfer fluid in cold climatic conditions, to lower the aqueous freezing point of heat transfer fluids in automobile radiators and similar heat exchangers. Limited data is available on thermo physical properties and heat transfer characteristics of ethylene glycol based nanofluids at present. Heat transfer and friction factor studies on propylene glycol based nanofluids are not explored. Even presence of 30 % propylene in water freezes only at -13 ˚ C. Experimental work on passive heat transfer enhancement using inserts in propylene glycol based nanofluids for further augmentation of heat transfer is a grey area to be explored.

Propylene glycol is an anti-freezing liquid; it is chemically more stable, non-toxic and gives a stabile suspension when compared to ethylene glycol. Propylene glycol and water mixture can be used as base or host fluid. No work is reported so far on the properties and heat transfer characteristics of Propylene glycol based CuO nanofluids.

3.1 Convective Heat Transfer Experiment and Correlation

The experimental heat transfer data for pure water with twisted tape and longitudinal strip inserts in laminar and turbulent flow conditions are also available in the literature. But, no experimental work is reported so far on heat transfer using propylene glycol as base liquid with inserts.

The convective heat transfer feature and flow performance of two phase nanofluids in a tube have experimentally been investigated. The suspended nanoparticles remarkably enhance heat transfer process and the nanofluid has larger heat transfer coefficient than that of the original base liquid under the same Reynolds number. The heat transfer feature of a nanofluid increases with the volume fraction of nanoparticles.

By considering the microconvection and microdiffusion effects of the suspended nanoparticles, a new type of the convective heat transfer correlation for nanofluids in a tube has been proposed as

$$Nu_{nf} = c1(1.0 + c_2 \phi^{m1} Pe_d^{\ m2}) Re_{nf}^{m3} Pr_{nf}^{0.4} .$$

This correlation correctly takes the main factors of affecting heat transfer of the nanofluid into account.

3.2 Experimental Apparatus

Figure 1 illustrates the experimental set-up used in the present study. The test section is a tube in tube heat exchanger. An experimental apparatus is conducted to study the heat transfer performance and friction factor in a tube with twisted tapes and helical inserts. It is composed of cold water tank, hot water tank, temperature sensors, flow control valve, temperature display, mono block pump, thermostatic water heater, annulus, different geometries of inserts and involving.

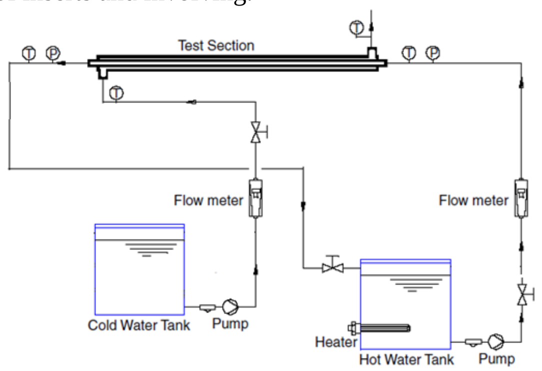

Fig.3.1 experimental setup of double pipe heat exchanger

A double pipe heat exchanger is utilized as the main heat transfer test section which is insulated using asbestos to minimize heat loss to the surrounding. It consists of two concentric tubes in which hot water flows through the inner tube and cold water flows in counter flow through annulus. The outer tube made of a stainless steel having inside and outside diameters of 30 mm and 32mm respectively. The inner tube made of a copper having inside and outside diameters of 18.5 mm and 20 mm respectively. It has a heat transfer section of a length of 1.5m.

The two flow meters are used to maintain shell side and tube side mass flow rates of water. The working range of flow meter is from 0.0048 kg/sec to 0.032kg/sec. The two FCV is used to controlled tube side and annulus side mass flow rate. One flow meter used to measure hot water mass flow rate and another flow meter is to measure annulus side cold-water mass flow rates. A PT100 type temperature sensor is directly inserted into inner and outer tube to measure inlet and outlet temperatures of both the fluids. Temperature data was recorded using data acquisition/switch unit.

In the experimentation, the various specifications of twisted tape and helical inserts having 8.5 mm width, 8mm rod, twist ratios of 2.93, 3.91 and 4.89 are used and helical inserts with the central axis of the inner tube are inserted inside the inner tube.

These different inserts create turbulence inside the inner pipe which enhances the heat transfer rate.

Investigating the effect of twist ratio (H/D) of twisted tapes on heat transfer enhancement showed that, for all Reynolds numbers and all kinds of working fluids, the heat transfer enhancement was increased as the twist ratio increased. The twist ratio also can affect the friction factor, Nusselt number, pressure drop, and velocity fields.

Table 3.1 Dimensions of Double heat exchanger

Dimensions	Outer Tube	Inner Tube
Outer Tube(m)	0.032	0.030
Wall thickness(m)	0.0015	0.0015
Length of tube(m)	1.5	1.5
Reynolds No	4100-9000	4100-11000
Prandtyl Number	3-5	3-5
Twisted tape with helical inserts	8.5 mm width, 8mm rod	8.5 mm width, 8mm rod
Twist ratio (H/D)	2.93, 3.91 and 4.89	2.93, 3.91 and 4.89

3.3 Calculation of tube side heat transfer coefficients: In present investigation, the heat transfer coefficient was determined based on the measured temperature data.

Tube Side Heat transfer

$Q = m_h.\ C_{ph}\ (T_{hi}-T_{ho})$

Shell Side Heat Transfer

$Q = m_c.\ C_{pc}\ (T_{ci}-T_{co})$

The physical properties of taken on average temperature

$T_m=(\ T_{in} + T_{out}\ )/2$

The heat transfer coefficient was calculated with

$$U_0 = \frac{Q}{A_0.\ \Delta T_{\ LMTD}}$$

The overall heat transfer surface area was determined based on the tube diameter and developed area of tube diameter,

$A_{Total}=\prod L\ D_0$ is the total convective area of the tube ($\prod$LD) constant for various geometries of inserts of heat exchanger.

$A_{Total\ Convective\ Area} = \prod L\ D_0 = 0.01\ mm^2$

LMTD is the log mean temperature difference, based on the inlet temperature difference ΔT1, and outlet temperature difference ΔT2,

$$LMTD = \frac{(\Delta T2 - \Delta T1)}{\ln(\Delta T2 / \Delta T1)}$$

The flow rate in shell side was varying with combination to tube side flow rate. The overall heat transfer coefficient can be related to the inner and outer heat transfer coefficient from following equation

$$\frac{1}{U_0} = \frac{Ao}{A_i\ h_i} + \frac{Ao\ \ln(d0 / di)}{2\prod kL} + \frac{1}{h_o}$$

Where, di and do are inner and outer diameters of the tube respectively. k is thermal conductivity of wall material and L, length of tube (stretch length) of heat exchanger. After calculating overall heat transfer coefficient, only unknown variables are hi and ho convective heat transfer coefficient inner and outer side respectively, by keeping mass flow rate in annulus side is constant and tube side mass flow rate varying.

$\mathbf{h_i = C\ V_i^{\ n}}$ Where Vi are the tube side fluid velocity m/sec. Substituting Equation 4.7 into Equation 4.6, the values for the constant, C, and the exponent, n,were determined through curve fitting. The inner heat transfer could be calculated for both circular and square coil by using Wilson plot method. This procedure is repeated for tube side and annulus side for each mass flow rate on both helical coils

3.4 Hydrodynamic Performance

The tube side friction factor can be calculated by using the formula,

$$f = \frac{\Delta p\ dh}{2\rho\ V^2\ L}$$

Where, dh is the hydraulic diameter in mm, L, length of heat exchanger in mm, v, velocity of the flowing fluid. As changes the geometry of the strip it directly affect on the hydraulic diameter because the volume of material required is more for the 10mm pitch strip and volume of material required is less for the 25 mm pitch tape inserts.

IV. Results and Discussions

For measurement of the pressure drop U tube manometer is used having the range 150-0-150 mm. The mercury is an media used in the U tube manometer to measure the height of mercury across the heat exchanger. Pressure drop and friction factor is calculated for the various geometries of twisted tapes and for various mass flow rates. Pressure drop is increases with increase in the velocity of the flow. The pressure drop reduces with increasing strip pitch and very less pressure drop found in the plain tube. The height and length of the inserted strip is same. The effect of bended strip on the pressure drop and friction factor is given in the figure 3.2 and 3.3

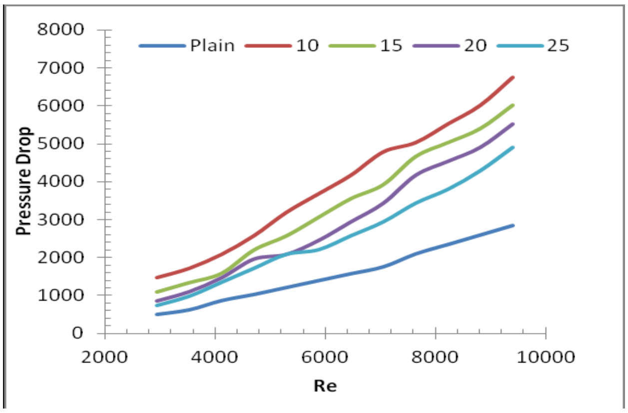

Fig. 3.2 Pressure drop for the various geometries of inserts

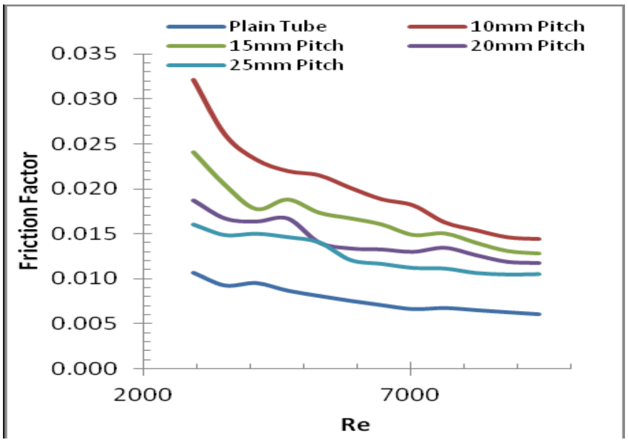

Fig. 3.3 Friction factor variation for various geometries of inserts

Pressure drop of the plain tube is less as compared to the other inserted geometries. The plain tube has less obstruction to flow pattern and less turbulence are occurred hence this effect on the heat transfer performance. In tube-in tube heat exchanger various patterns and geometries are inserted for the enhancing heat transfer rate but same time its affect on the pressure drop. Figure 3.2 and 3.3 presented the variation of the pressure drop and friction factor with change in inner tube side mass flow rate.

V. Conclusion

Experimental study of tube in tube pipe heat exchanger was performed using various twisted ratio inserts. The mass flow rates in inner tube and in the annulus were both varied. The two-phase propylene glycol based CuO nano fluids heat transfer study performed and tested for counter flow configuration. The heat transfer rates in 10 mm pitch were higher than other twisted tape. As increases the strip pitch, reduces the heat transfer rate. In present study found that, very low overall heat transfer in plain tube, tube side heat transfer rate progressively increases with decreasing inserted strip pitch. In hydrodynamic analysis large pressure drop for the minimum pitch i.e. 10mm and very less pressure drop found in the plain tube. In results, various twisted ratio inserts are compared with each other for various operating parameters like heat transfer rate, overall heat transfer coefficients, LMTD and Nusselt number with Varying tube side hot water flow rates. The friction factor is on higher side for the minimum pitch of twisted tape.

The effects of parameters such as modified wave-width, Reynolds number on the heat transfer and overall enhancement ratio are studied.The following conclusions can be drawn:

A. The enhancement of heat transfer with wavy twisted tape inserts as compared to plain tube varied from 12 to 43% for wave width of 13 mm and 9 to 38% for wave-width 24 mm. This enhancement is mainly due to the centrifugal forces resulting from the spiral motion of the fluid.
B. Reduction in tape width causes rise in Nusselt numbers as well as friction factors. The maximum friction factor rise was about 175% for 13mm and 150% for 24mm wave width
twisted tape inserts compared to plain tube.
C. The overall enhancement for the tubes with wavy twisted tape inserts is 1.43 for wave-width 13 mm and 1.38 for wave-width-24mm wavy twisted tape insert. Thus the enhanced performance can be achieved using wavy twisted tapes as compared to plane twisted tape.
Thus, from the considerations of enhanced heat transfer and savings in pumping power wavy-width tape inserts are seen to be attractive for enhancing turbulent flow heat transfer in a horizontal circular tube.

5.1 Nomenclature
A0 - area of orifice, (m2)
A - test section inner tube area, (*p*/4 D2) (m2)
Cp **-** specific heat of air, (J/kg K)
Qa **-** air discharge through test section (m3/sec)
D - Inner diameter of test section, (m)
H - pitch, (mm)
w - width of wavy tape insert,(mm)
H/D - twist ratio
fth **-** friction factor(theoretical) for plain tube
f - friction factor(experimental) for plain tube
fi **-** friction factor obtained using tape inserts
h - experimental convective heat transfer coefficient, (W/m2K)
hw **-** manometer level difference,(m)
hair - equivalent height of air column, (m)
k - thermal conductivity, (W/mK)
L - length of test section, (m)
m - mass flow rate of air, (Kg/sec)
Nui **-** Nusselt number (experimental) with tape inserts, (hD/k)
Nu **-** Nusselt number (experimental) for plain tube
Nuth **-** Nusselt number for plain tube (theoretical)
Pr **-** Prandtl number
p - pitch, (m)
D**P -** pressure drop across the test section, (Pa)
Q - total heat transferred to air (W)
Re **-** Reynolds number, (*r* V D/*m*)
T1, T8 - air temperature at inlet and outlet, (_k)
T2, T3, T4, T5 - tube wall temperatures, (_K)
Ts **-** average Surface temperature of the working fluid, (_K)
Tb **-** bulk temperature, (_K)
V - air velocity through test section, (m/sec)

5.2 Greek Symbols
v **-** Kinematic viscosity of air, (m2/sec)
m **-** dynamic viscosity, (kg/m s)
h **-** Over all enhancement
 w - density of water, (Kg/m3)
 a - density of air (Kg/m3)

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