EXPERIMENTAL ANALYSIS OF SHELL SIDE HEAT TRANSFER COEFFICIENT AND PRESSURE DROP FOR AN SHELL-AND-TUBE HEAT EXCHANGER USING SIC NANOFLUID

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Abstract

Recent studies on heat transfer in shell and tube heat exchangers have pointed out the need for numerical expressions for pressure drop. Overall heat transfer, coefficient can be increased by maximizing shell side and tube side flow velocities, which, in turn, is directed by the allowable pressure drop as higher velocity means higher pressure drop. This paper presents the results of experimental studies to obtain pressure drop relations in designing shell and tube heat exchanger. The effect of the parameters such as mean temperature, effectiveness and overall heat transfer variations of Silicon Carbide (Sic) Nano fluid of (0.1 & 0.2 wt% concentrations) is also compared with water to investigate the heat transfer characteristics.

Keywords: *Shell and tube heat exchanger, Pressure drop, Sic Nano fluid, Heat transfer coefficient*

1. Introduction

A heat exchanger is known to be equipment which transfers the energy from a hot fluid to a cold fluid, with maximum rate and minimum investment and running costs. It is a device used for affecting the process of heat exchanger between two fluids that are at different temperatures. Heat exchangers are useful in many engineering process like those in refrigerating and air conditioning systems, power systems, food processing systems, chemical reactors and space or aeronautical applications.

In heat exchangers the temperature of each fluid changes as it passes through the exchangers, and hence the temperature of the dividing wall between the fluids also changes along the length of the heat exchanger. Heat exchangers are designed to deliver certain heat transfer rate for a certain specified conditions of flow rates and temperature. To get substantial heat transfer area from a double pipe exchanger, it must be long. The result is a high pressure drop, increased pumping costs, and large amounts of metal. This means we need a more compact arrangement that still simulates counter current flow the Shell and Tube exchanger. Shell tube exchangers are used when a process requires large amounts of fluid to be heated or cooled is suited for higher-pressure applications.

There are many different types or designs of shell and tube heat exchanger to meet various process requirements. U-tube type heat exchanger has only one tube sheet and as each tube is free to move with respect to shell, the problem of the differential movement is eliminated.

Tubes can be cleaned mechanically, in applications where the tube side fluid is virtually non-fouling fluid. The advantage of U-tube heat exchanger is, as one end is free the bundle can expand or contract in response to the stress differentials. The disadvantage of U-tube construction is that the inside of the tube cannot be cleaned effectively since the U bends require flexible end drill shafts for cleaning. The applications of single-phase shell-and-tube heat exchangers are quite large because these are widely used in chemical, petroleum, power generation and process industries. In these heat exchangers, one fluid flows through tubes while the other fluid flows in the shell across the tube bundle. The design of a heat exchanger requires a balanced approach between the thermal design and pres-sure drop. The pressure drop results in the increase of the operating cost of fluid moving devices such as pumps and fans. This shows that along with the design for the capacity for heat transfer, the pressure drop determinations across the heat exchanger are equally important. The estimations for pressure loss for the fluids flowing inside the tubes are relatively simple, but complex in the shell-side flow.

To evaluate the pressure drop in the shell, there is a need to know the various internal flow paths and their individual effects.

The flow direction of the main stream relative to the tubes is different in the window section created by the baffle cut from that in interior cross-flow section existing between the segmental baffles. This requires the use of different approaches to compute the pressure drop in window section than the flow across the tube bundle (cross-flow section). Similarly in the end crosses, the flow across the tube bundle is different than interior cross-flow which also has to be taken into account.

In the present work an effort has been made to develop the simple pres-sure loss model. The results of this are compared with the results available in the literature. The present model results are quite good and designers can use these confidently. The work is mainly concerned with liquid flows in the shell.

2. Pressure Drop Model Development

The present work aims to determine the overall pressure loss in the shell from the point of entry of the fluid to the outlet point of fluid. The total pressure drop has been divided into the various components listed below:

- Pressure drop in the inlet and outlet nozzles at the end cross-sections of heat exchanger.
- The pressure drop in the interior cross sections. In these sections, the pressure loss is determined for the flow across the tube bundle and for the flow from interior section through window section to the next consecutive interior section.
- The pressure drop due to flow pattern in the inlet end cross-section across the tube bundle up to the level of base height and thereafter for the flow through window section. Similarly for the outlet end cross-section the pressure loss is first computed for the flow coming from

the previous interior cross-section through window section and thereafter for the flow across the tube bundle.

The main contribution of the present work is concerned with developing the model to compute the pressure drop in the interior section and window section. For rest of the pressure drop components the expressions already available in the literature have been used.

The computation of the pressure drops across the tube bundle and through window section, efforts have been made to consider the actual flow pattern. The pressure losses for each of the above components have been presented one by one in the following text of the paper.

3. Objective of Work

The present work is done by using 1-shell and 2-tube pass heat exchanger having counter flow. The tube side the cold fluid is allowed and the hot fluid on shell side. The various cold fluids which are water based silicon carbide 1% and 2% volume fractions are used. The outlet temperatures of hot and cold fluids are recorded for all the working fluids and the LMTD, effectiveness and the overall heat transfer coefficients are calculated.

The thermal performance analysis is done using solid works, after modeling the one shell two tube pass heat exchanger in solid works. Various thermal parameters like temperature rise, LMTD, effectiveness, over all heat transfer and pressure drop along the length of tube and shell are observed and compared with the experimental results. The work is further extended with insertion of rectangular cross-sectional fins in the tubes to enhance the heat transfer. The thermal analysis results with fins are compared to the results without fins and the performance relations are plotted on graphs.

3.1. Pressure Drop

In any heat exchanger, not only a higher overall heat transfer coefficient is desirable, but a small pressure drop is expected in order to reduce pumping power. In this experiment, pressure and pressure drop were both measured with installed pressure gauges both at the inlet and the outlet of the shell sides and tubes. At the shell side where oil flows, due to the baffle plates the flow was regarded as a cross flow over a tube bundle. Cross flow in the shell side is quite complicated and the flow on the first tubes is quite different with those inside the bundle. Turbulence in the main stream will be enhanced by the first row of the tubes and is stabilized almost at the fourth row. In case of micro-fin tubes the conditions are more complicated than smooth and corrugated ones, since the temperature and pressure fields are dependent on several geometrical hydrodynamics and thermal parameters. Since the aim is to compare the pressure drops across different external surfaces similar arrangements, the cross flow for the oil on the shell side seems to be a reasonable assumption. The shell-side pressure drop can be estimated with Bell's method, which attempts to provide a more reasonable representation of the pressure drop by summing the calculated pressure drops for individual parts of the exchanger as given in below eqn.

$$\partial P_t = (N_b - 1)\, \partial P_c + N_b\, \partial P_w + \partial P_{ei} + \partial P_{eo} + \partial P_{ni} + \partial P_{no}$$

4. Experimental Setup

The experiment set up consists of one shell that is fabricated from steel pipe with nominal IPS diameter up to 12 in. shells are fabricated by rolling steel plate. It is apparent that higher heat transfer coefficient results when a liquid is maintained at a state of turbulence. To induce turbulence outside the tubes it is customary to employ a baffle which causes the liquid flow through the shell at right angles to the axis of the tubes. Two baffle plates are inculcated to the tube bundle this caused considerable turbulence even when a small quantity of liquid flows through the shell.

Heat exchanger tubes are also referred as condenser tubes and are made up of brass. The brass having a high thermal conductivity has the capacity to transfer more heat from the hot fluid in shell to the cold fluid in tubes. There are two u-tubes that are kept with square pitch. The tube sheets inserted at ends of the tubes in order to avoid the instability of tubes in the shell and to provide support to withstand pressures. A separate partition has been created with a non - conducting material in between the cold fluid inlet and cold fluid outlet sections in order to avoid the mixing of fluids.

Table 4.1: List of Components and Components

Components	Material	Specifications
Shell	Mild steel	Diameter = 250 mm
		Thickness = 4mm
		length = 3.7 feet
Tube bundle	Brass	length = 3.5 feet
		inner diameter =10 mm
		outer diameter = 12mm
Tube sheet	Mild steel	thickness= 2mm

The various working fluids used in this experiment are water, 1% volume fraction of Sic and 2% volume fraction of Sic with water as base fluid. The working fluid silicon carbide Nano fluid is used with the advantages of having low density with high thermal conductivity among the ceramic compounds. It is comparatively cheap and commercially available. The colour of nanoparticles is grey and black. With the cheap potential availability the black coloured 100nm sized have been used for the preparation of Nano fluid.

The thermal conductivity of the Nano fluid increases when compared with the base fluid. The Nano fluids are prepared by two-step method by the ultrasonicator and the stirrer break down the aggregates of the nanoparticles. The suspension and addition of surfactants help in adjusting the PH value of Nano fluids. The adjustment of Nano fluids prevented the re-aggregation of nanoparticles and improved the stability of Nano fluids.

The Nano fluid is prepared by stirring for 8 hours and the settlement of the particles is avoid for five hours. Always the hot fluid water is kept used in the shell with a temperature of 331K. On the other side the tube side various working fluids are varied and those are water, 1%,2% volume concentrations of Sic with water as base fluid are allowed at a temperature of 300K.

a. Properties of Working Fluid

The hot fluid considered is water on the shell side and the various cold fluids used is water, ethylene glycol (pure, 75%, 25%) and Nano fluid Sic on the tube side.

Required properties are shown in the table.

The thermo physical properties of the fluids are taken at different concentrations are calculated using the following equations.

Density: ρnf = (1-Ø) ρcf + Ø ρp

Heat capacity: (ρc) nf = (1-Ø) (ρcp) + Ø

(ρcp)p *Thermal conductivity :* knf =

[kp+2kbf+2(kp-kbf) Ø]/ [kp+2kbf-(kp-kbf) Ø]

Where Ø is particle volume fraction, the subscript “nf” refers to Nano fluid, “bf” refers to base fluid, and “p” refers to particle.

Table 4.2: Thermal Properties of Working Fluids

Fluids	Density (kg/m^3)	Heat capacity (J/Kg K)	Viscosity (Ns/m^3)	Thermal conductivity (W/m K)
water	998	4182	0.001003	0.600
Sic-water (1%)	1015	4812	0.001	0.661
Sic-water (2%)	1037	4812	0.001	0.680

3D Modeling of 1-Shell 2-Pass Heat Exchanger:

The modeling of 1-shell and 2-tube pass heat exchanger with u-bend counter flow has been modeled in solid works.

Boundary Conditions

The desired mass flow rate and temperature values are assigned to the inlet nozzle of the heat exchanger. The shell inlet temperature is set to 331K. Zero gauge pressure is assigned to the outlet nozzle, in order to obtain the relative pressure drop between inlet and outlet. The velocity of hot fluid is considered as 0.19kg/sec flowing into the shell. The inlet velocity profile is assumed to be uniform. No slip condition is assigned to all surfaces. The zero heat flux boundary condition is assigned to the shell outer wall, assuming the shell is perfectly insulated outside. On the other side in the tubes the cold fluid inlet temperature is set to 300K with a mass flow rate of 0.071428kg/sec.

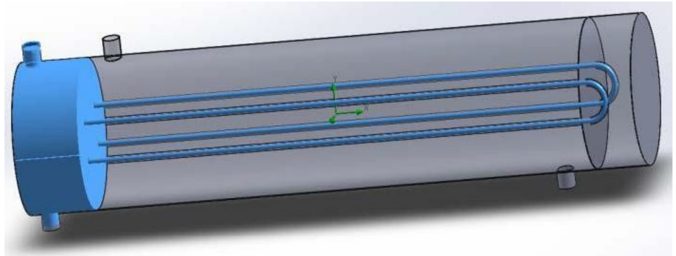


Figure 4.1.2: Cold Working Fluids in the Tubes

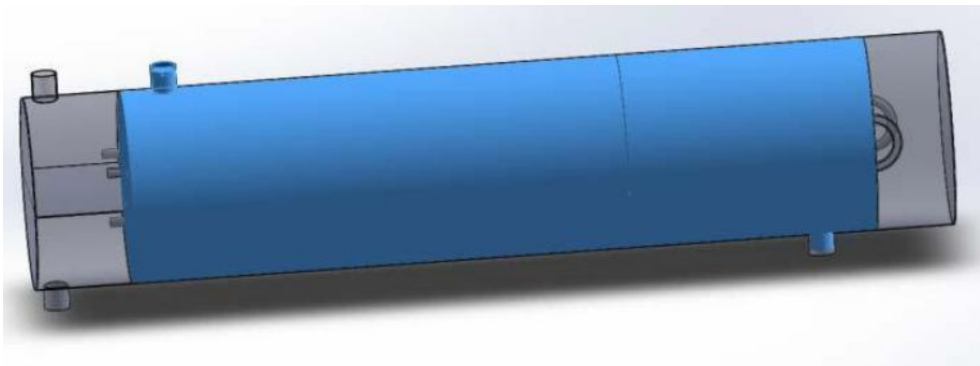


Figure 4.1.3: Hot Working Fluid in the Shell

Calculations

Heat transfer general equation is as follows:

$Q = m\ cp\ (T1-T2)$

Heat removed on tube side : $Qc = mc\ cpc\ (Tc2-Tc1)$ Heat removed on shell side: $Qh = mhcph\ (Th1- Th2)$ Logarithmic Mean Temperature Difference

(LMTD):$LMTD = [(T1- T2)/Ln (T1/ T2)]$

$T1 = Th1-Tc2$

$T2 = Th2-Tc1$

Effectiveness ( ):

1 = [Th1-Th2]/[Th1-

Tc1] 2 =[Tc2-Tc1]/

[Th1-Tc1]

Overall heat transfer coefficient: $Uo = Q/ [A \times LMTD]$

5. Results and Discussions

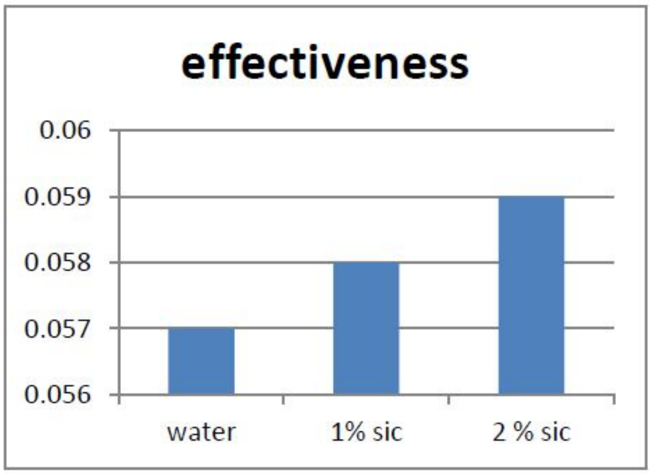
Experimental. Results

The experimental results are recorded by using the thermocouples inserted at the outside fluid flow of both cold side and hot side. The hot side, that is at the shell side inlet temperature is considered as 331K and while on the other hand cold side tubes inlet temperature considered to be 300K.The outside temperatures of both cold fluid and the hot fluid by imposing different working fluids are recorded as follows:

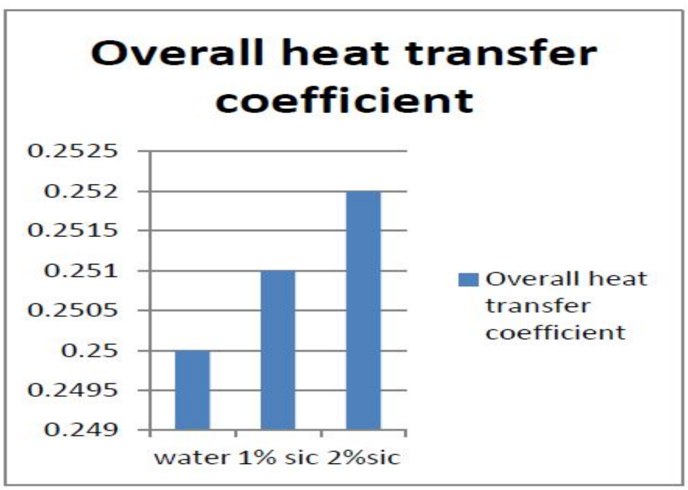
Table 5.1.1: Temperatures of Fluids

Working Fluid (tube side)	Cold fluid inlet (^{0}C)	Cold fluid Outlet (^{0}C)	Working Fluid (shell side)	Hot fluid Inlet (^{0}C)	Hot fluid Outlet (^{0}C)
Water	26.7	28.4	water	58	54.8
Sic 1%	26.7	29.6	Water	58	54.7
Sic 2%	26.7	29.7	water	58	54.8

Experimental effectiveness of Heat exchanger shows in bar chart bellow,

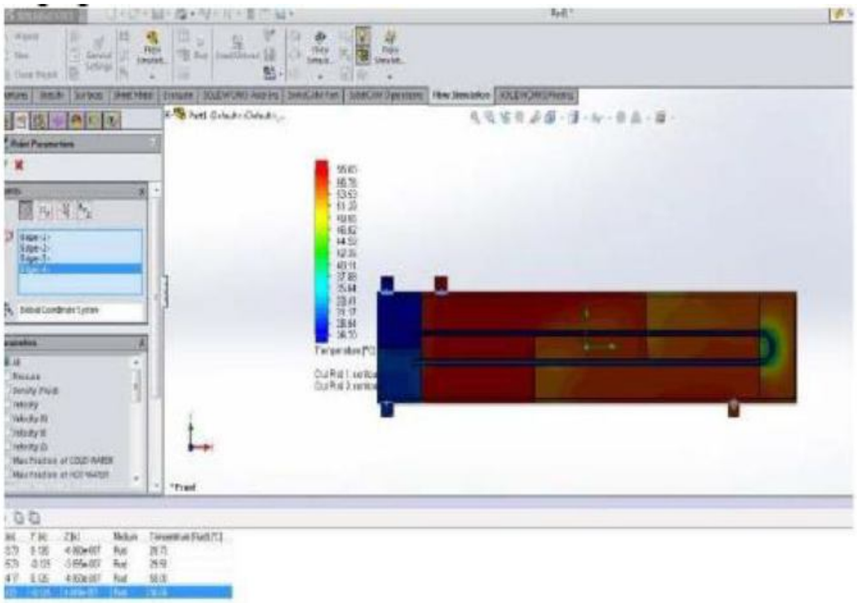


And overall heat transfer coefficients of heat exchangers shows in bar chart below.



ThermalField Analysis of Shell and Tube Heat Exchanger

The analysis has been carried out in solid works by importing the model that is been created and assembled in the solid works. In the experimental and analysis, the temperatures of both the cold fluid and hot fluid temperatures of inlet and outlet are compared as result showed a much closer compatibility. The results obtained were similar to that of analysis with a minute negligible difference.



Temperature counters of water in solid works

Conclusion

The experiment analysis and simulation are carried on one shell two tube pass heat exchanger. With an aim to enhance the overall heat transfer coefficient by using various fluids with high thermal conductivity have yielded a potential increase in overall performance of heat exchanger with the use of Nano fluid SiC. The results were analyzed and the conclusions drawn are as follows:

- The overall heat transfer coefficient of water when compared with respect to Sic is more while the LMTD of Sic compositions are higher than water.
- The heat transfer rate has been increased from 4-6% by using the SiC Nano fluid with volume fractions of 1% and 2% when compared with base fluid that is water. Similarly the overall thermal performance of the heat exchanger has enhanced with the use of Nano fluids.
- The Nano fluids thermal conductivity and the characteristics thermal performance has been increased with the increase in concentration of volume fractions. The SiC Nano fluid with 2% volume fraction has a better performance to SiC 1% volume fraction.
- IV. With increasing the surface area of contact of fluid flow by imposing rectangular stripped fins inside the tubes. There has been an extreme increase of 60% to 70% in the overall heat transfer coefficient of working fluids when compared with flow field analysis without using fins.
- Compared with Reynolds number, it increases means pressure drop will increase. In order to reduce the pressure drop means we reduce the Reynolds number.

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