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Simulation of laminar flow heat transfer in oil coolers

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ABSTRACT

A CFD analysis was done to numerically study the heat transfer characteristics of an oil cooler under laminar flow conditions with varying Reynolds number from 250 to 2400 with ISOVG46 Turbinol on tube side and water on shell side for different flow rates. Simulated results of Nusselt number and friction factor are in good agreement with the available experimental results and with the Sieder and Tate equation for plain tube. Results show that the strip inserts led to a higher heat transfer rates over the plain tube with increase in Nusselt number, friction factor and over all enhancement ratios.

Introduction

eat transfer processes of viscous fluids usually take place in laminar or transitional regimes, where transfer rates are particularly low. Heat exchangers that work under these flow conditions are required to undergo enhancement techniques. Among the different techniques, which are effective to improve the thermo hydraulic behavior in the tube-side in single-phase laminar flow, the insert devices are important. The main advantage of inserts with respect to other enhancement techniques is that they allow an easy installation in an existing smooth-tube heat exchanger. A comprehensive experimental study was carried out on six wire coils inserted in a smooth tube, covering the laminar, transition and turbulent regimes by Garcia et al [1]. Experimental investigations were carried out by Promvonge [2] to investigate the air flow friction and heat transfer characteristics in a round tube fitted with coiled square wire turbulators for the turbulent regime. Smith Eiamsa-ard and Pongjet Promvonge [3] experimentally investigated heat transfer and flow friction characteristics of insertion of a helical screw tape with or without core rod in a concentric double tube heat exchanger. Wen et al. [4] carried out experimental study on the heat transfer enhancement and pressure drop in small tubes. Sreenivasulu and Prasad [5] presented a computational study on convective heat transfer in an annulus with its inner cylinder wrapped by a helical wire. Pavel et al [6] experimentally investigated the effect of metallic porous inserts in a pipe subjected to constant and uniform heat flux at a Reynolds number range of 1000-4500. Mehmet Sozen and T M Kuzay [7] numerically studied the enhanced heat transfer in round tubes filled with rolled copper mesh at Reynolds number range of 5000-19,000 with water as the energy transport fluid. Heat transfer, friction factor and enhancement efficiency characteristics of a circular tube fitted with conical ring turbulators and a twisted-tape swirl generator were investigated experimentally by Promvonge and Eiamsa-ard [8]. The effect of two tube inserts, wire coil and wire mesh on the heat transfer enhancement, pressure drop and mineral salts fouling mitigation in tube of a heat exchanger were investigated experimentally by H. Pahlavanzadeh et al [9] with water as working fluid. Heat transfer and friction characteristics were investigated experimentally by using louvered strip inserts inserted in the inner tube of a concentric tube heat exchanger by Smith et al. [10]. The flow rate of the tube was in the Reynolds number range of 6000 to 42000.

In the above literature review, most studies were focused on the turbulent flow heat transfer enhancement with either air or water as working fluids while numerical simulations are scarce. Therefore, the present study focuses on laminar flow heat transfer enhancement using louvered strip inserts with ISOVG46 turbinol (with high prandtl number of 220 at 700 C) as the working medium by numerical modeling using computational fluid dynamics.

Details of the experimental setup

The experimental oil cooler test rig consists of oil circulation system, water circulation system, shell and tube heat exchanger, cooling tower and instrumentation system to measure oil and water flow rates, pressures and temperatures. Oil circulation system consists of an oil tank, strainer, canned type centrifugal oil pump, flow regulation valves and heating system. Heating system consists of an oil tank with four coiled ceramic insulated electric heaters as shown in Figure 1. Similarly water circulation system consists of water sump, cooling tower, pump and flow regulation valves. Water pump with a maximum capacity of $10m^3$/ hr was used for water pumping through the

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shell of the heat exchanger. The warm water from the outlet of the heat exchanger was cooled in an induced draught cooling tower and sent back to sump. Shell and tube heat exchanger consists of four tubes enclosed inside the shell. Hot oil is made to flow inside the tubes with water flowing in counter flow direction inside the shell. The outer surface of the shell is wrapped with insulation to minimize heat losses to surroundings. Schematic diagram of oil cooler test rig is shown in figure 1. Temperature Measurement was done by Resistance Temperature Detectors. RTDs were used for the measurement of inlet and outlet temperatures of water and oil. Along the length of tubes, exclusively made Copper-Constantan thermocouples (T-type) were used for the measurement of tube wall temperatures and bulk oil temperature inside the tank. Pressure transmitter was used to measure the pressure at a single point. Differential pressure transmitter was used to measure the pressure drop of oil across the tubes. In addition, two pressure transmitters were utilized for measurement of inlet pressures of oil and water. An oil flow rotameter and orifice plate was used to measure the oil flow rate. A water flow rotameter was used for measuring water flow rate.

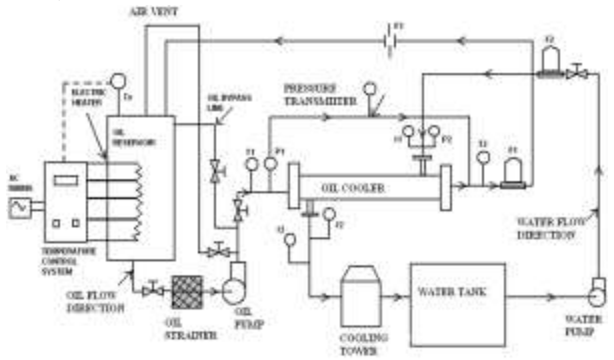

Fig 1: Schematic diagram of oil cooler test rig

Procedure

Oil is heated by switching on electric heaters. During heating of oil, the bye-pass valve is fully opened and the other valves are fully closed so that the oil temperature in the tank rises in short time. When the tank temperature reached the desired temperature of 70^0C, the inlet valve to the heat exchanger was gradually opened till the required flow rate was reached in the oil flow rotameter. Water pump was put on and the water flow rate was adjusted slowly to a value at which the water temperature rise is around 2 to 3^0C. Once the steady state was reached, data recording was initiated. This data consisted of wall temperatures, oil inlet and outlet temperatures, water inlet and outlet temperatures and pressure measurements. Experiments were conducted (plain tube experiment) without inserts for different oil flow rates. All the experiments were carried out for laminar flow conditions with Reynolds number varying from 250 to 2450.

Heat loads were evaluated by

$Q_h = m_h \, cp_h \, (Th_{in} - Th_{out})$ (1)

$Q_c = m_c \, cp_c \, (Tc_{out} - Tc_{in})$ (2)

$V = m_h/(\rho_h \; A_c)$ (3)

$Re = \rho \; V \, D_i / \mu$ (4)

Nusselt numbers calculated from the experimental data for plain tube (tube side) were compared with the correlation recommended by Sieder and Tate

$Nu_0 = 1.86(RePr(D_i/L))^{1/3}(\mu \;/\mu_{\,w})^{0.14}$ (5)

Properties of oil were considered at the local mean bulk temperature.

Equation (5) gives theoretical Nusselt number.

$f_{the} = 64/Re$ (6)

$f_0 = \Delta P/ \; ((L/D_i) \; (\rho \; V^2/2))$ (7)

Numerical experiment

The geometry used for numerical modeling is same as the experimental set up. The heat exchanger geometry is created and meshed using the software package ICEMCFD and analyzed using ANSYS CFX 11.0 for oil to water heat transfer through the metal wall. Tube wall thickness is considered while modeling the heat exchanger. Heat exchange from hot fluid inside the tubes to cold fluid in shell was modeled with convective heat transfer in the tube, conduction through the tube wall and convective heat transfer to the shell fluid. Structured grids were used to mesh the tube fluid volume and tube solid volume. For momentum equation, the inner and outer wall of the tubes was treated as no-slip ones. The inner and outer walls of shell were considered as no-slip with adiabatic condition. Hot fluid inlet is taken with mass flow inlet boundary condition. Hot fluid outlet was specified as atmospheric pressure. Cold fluid enters the heat exchanger in the opposite direction (counter flow) with mass flow inlet and pressure outlet condition. It has been found that the heat transfer coefficients predicted by the Sieder and Tate equation for plain tube are comparable with those calculated by the computer code CFX, with a maximum deviation of 9%.

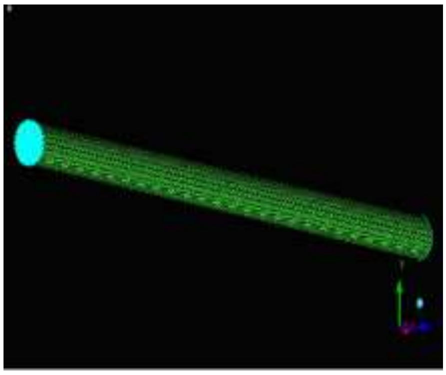

Fig 2: Meshed model of oil cooler

The meshed model of oil cooler (shell and tube heat exchanger) is shown in figure 2. Structured grids were used to mesh the tube fluid volume and tube solid volume.

CFD analysis was carried out varying the inlet flow rate of oil. Before carrying out the actual analysis, a grid independency of the solution was established. The optimum mesh was chosen for further analysis and has 73600 nodes and 129523 elements for plain tube.

The sensitivity of the results on the value of the imposed boundary condition, constant wall heat flux was studied. On changing the wall heat flux from 2000 to 6000 W/m^2, changes in Nusselt number was marginal. The change is attributable to differences in the fluid temperature distribution along the tubes.

CFD Modelling

The mass flow rate of the cold water was kept constant as 0.3 kg /sec. Mass flow rate of hot fluid varied from 0.16 to 1.56kg/sec, which is same as the one used in the experiment.

A convergence criterion of $1.0e^{-05}$ was used for continuity, momentum and energy equations. Inlet temperature of oil is kept constant at 347.5 K. Streamlines plotted with velocity as variable are shown in figure 3.

Velocity of oil at inlet varied from 0.12m/sec to 1.21m/sec with respect to variation in mass flow rates of oil for plain tube experiment.

The results of the analysis of the CFD simulation are used to estimate overall heat transfer coefficient. It is found that the deviation between CFD and experimental results is within 5%.

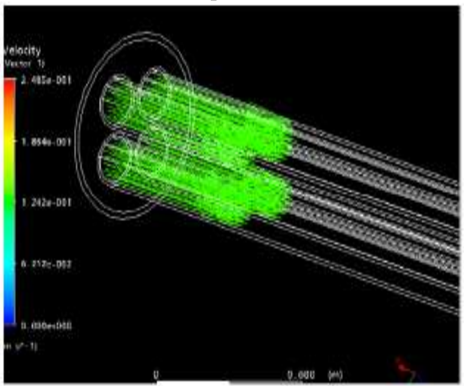

Fig 3: Plain tube velocity streamlines at inlet

Nusselt number obtained by CFD simulation is in close agreement with that of experimental and theoretical (Sieder and Tate equation) Nusselt number as shown in figure 4. Figure 5 shows the validation of friction factor obtained from simulation. The maximum deviation in simulated Nusselt number and friction factor was observed to be $\pm$4.9% and $\pm$1.25% respectively compared to the theoretical values.

Enhancement of heat transfer on oil side can be obtained by using tube inserts. The inserts used in the present study are louvered strip inserts with elliptical and diamond shaped leaves.

The strips are mounted on a core rod as shown in figure 6. These strip inserts are inserted in the four tubes of heat exchanger.

The louvered strip insert material considered for CFD analysis is stainless steel with an inclined angle of 30^0 with axis of tube. Distance between two consecutive leaves (pitch) is taken as 90mm.

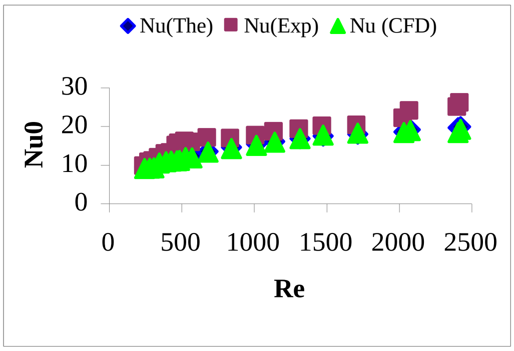

Fig 4: Verification of Nusselt number for plain tube

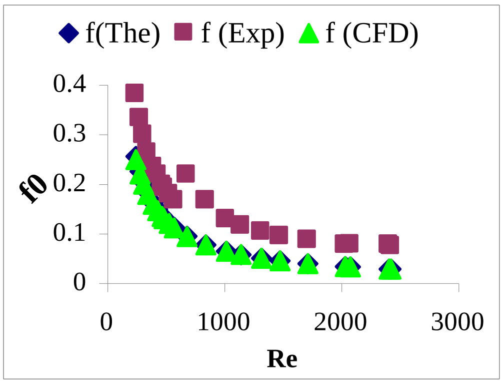

Fig 5: Verification of friction factor for plain tube

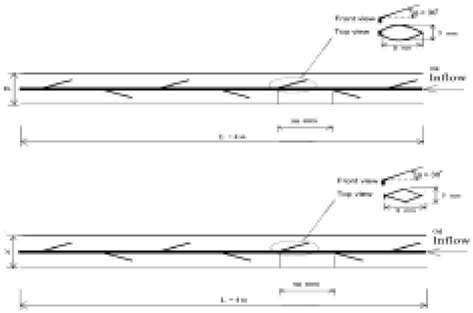

Fig 6: Louvered strip inserts with elliptical/ diamond shaped leaves

Figure 6 shows louvered strip insert with elliptical/diamond shaped leaves. The optimum mesh for oil cooler with louvered strip inserts with elliptical shaped leaves has 769939 nodes and 1433366 elements and with diamond shaped leaves has 764109 nodes and 1410090.

Each of the run took about 2hrs with Intel Quad core processor, 2.83GHz with 6 GB RAM. Meshed models of strip inserts are shown in figures 7 and 8. Outlet temperature contours of louvered strip inserts with elliptical shaped leaves are shown in figure 9. Due to the presence of strip inserts, tube hydraulic diameter decreases. As the mass flow rate of oil is kept constant for analysis with/without strip inserts, velocity of oil increases inside the tubes due to the presence of tube inserts. This promotes oil side turbulence leading to enhancement of heat transfer.

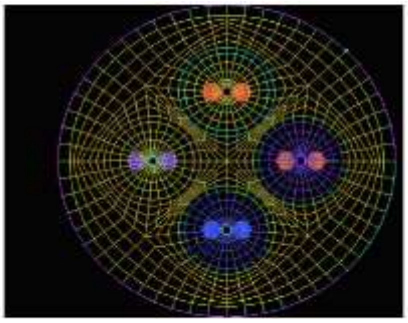

Fig 7: Cross-sectional view of louvered strip inserts with elliptical shaped leaves-meshed model

Inlet temperature of oil is kept constant at 347.5K. By comparing the oil outlet temperatures from figures 9 and 10, they are almost same for both strip inserts with elliptical and diamond shaped leaves.

But pressure drop contours (figures 11 and 12) showed that pressure drop is less for louvered strips with diamond shaped leaves. Maximum pressure drop is found at lowest mass flow rate of oil. Maximum pressure drop is observed for strip with elliptical shaped leaves (558 Pa) compared to strip with diamond shaped leaves (547 Pa) as shown in figures 11 and 12 respectively.

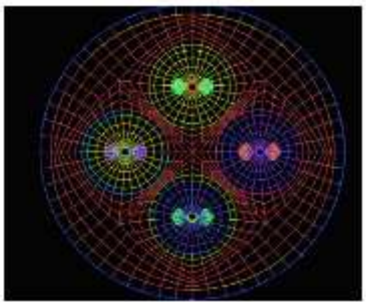

Fig 8: Cross-sectional view of louvered strip inserts with diamond shaped leaves-meshed model

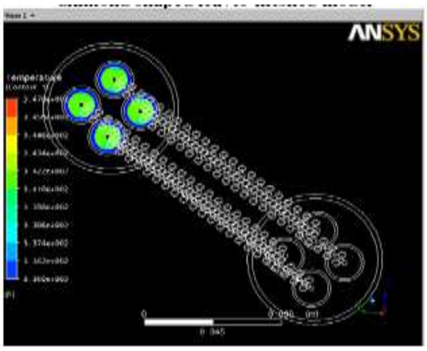

Fig 9: Outlet temperature contours of Louvered strip inserts with elliptical shaped leaves

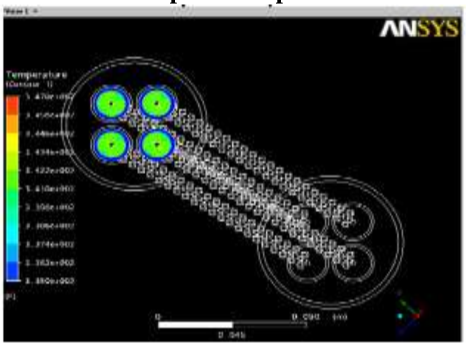

Fig 10: Outlet temperature contours of Louvered strip inserts with diamond shaped leaves

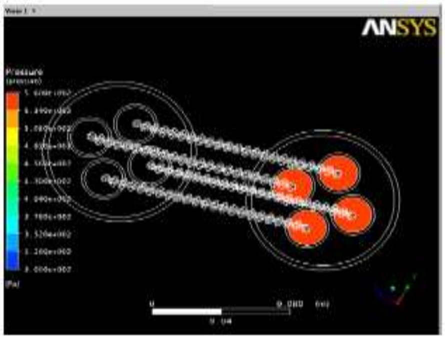

Fig 11: Contours of pressure drop for Louvered strip inserts with elliptical shaped leaves

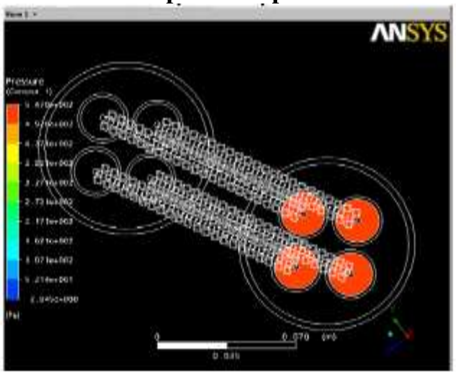

Fig 12: Contours of pressure drop for Louvered strip inserts with diamond shaped leaves

Variation of Nusselt number ratio with the Reynolds number is shown in figure 13. It indicates the ratio of Nusselt number with insert (for ellipse and diamond louvered strip inserts) in comparison to that of plain tube. It is seen that the Nusselt number ratio value is high at lower Reynolds numbers and decreased with increase of Reynolds numbers. The Nusselt number ratios of all cases are higher than unity. This indicates an advantageous gain of using tube inserts over plain tube. The maximum increase in Nusselt number for strips with elliptical and diamond shaped leaves are 431% and 430% respectively compared to plain tube.

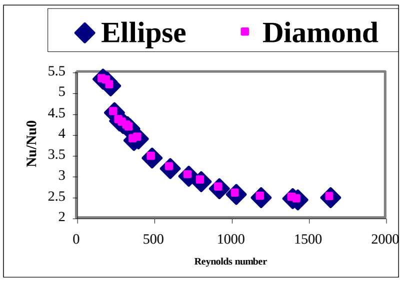

Fig 13: Variation of Nusselt number ratio with Reynolds number

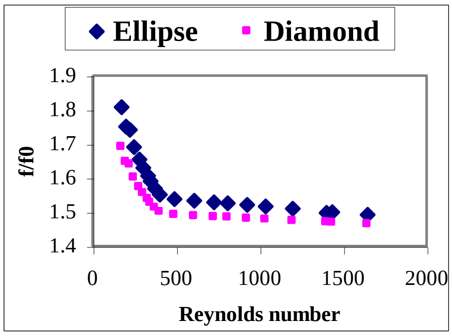

Fig 14: Variation of friction factor ratio with Reynolds number

Figure 14 shows the variation of friction factor ratio with Reynolds number. It indicates the ratio of friction factors with inserts (ellipse and diamond shaped louvered strip inserts) in comparison to that of plain tube. The friction factor ratio is observed to decrease with increase of Reynolds numbers. The friction factor ratio for the case of ellipse insert is observed to be higher than diamond insert. The maximum increase in friction factor for strips with elliptical and diamond shaped leaves are 49.22% and 46.38% respectively compared to plain tube. Overall enhancement ratio is used to evaluate the quality of the enhancement technique used. It is given by

$$\eta = (Nu/Nu_0) / (f/f_0)^{1/3} \qquad (8)$$

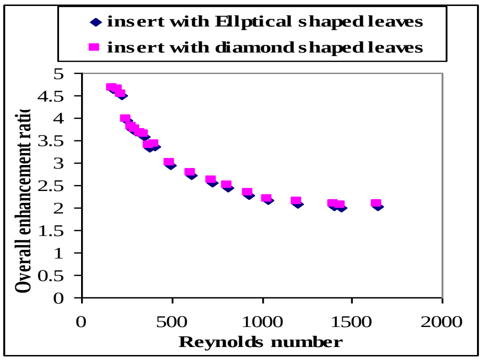

Fig 15: Variation of over all enhancement ratio with Reynolds number

Overall enhancement ratio is found to vary from 2.08 to 4.67 for louvered strip inserts with diamond shaped leaves which is slightly higher than that of louvered strip inserts with elliptical shaped leaves as shown in figure 15. This is due to lesser pressure drop associated with diamond shaped strip insert compared to elliptical shaped strip insert.

Conclusions

Computational Fluid Dynamic simulation of oil cooler using ANSYS CFX 11.0 was carried out with Reynolds number on tube side varying from 250 to 2400 for analyzing the flow behavior and enhancement of heat transfer using louvered strip inserts with elliptical and diamond shaped leaves. The plain tube CFD results like heat transfer coefficient, Nusselt number, friction factor were compared with theoretical and experimental results for validation. A good agreement was found between theoretical correlations and CFD simulated results.

- The louvered strip inserts with elliptical and diamond shaped leaves were found to provide a considerable enhancement in heat transfer with respect to plain tube.
- The maximum percentage increase in Nusselt number of louvered strip with elliptical shaped leaves with respect to plain tube is 431.37% with friction factor increase of 49.22% and the overall enhancement ratio is found to vary from 2.02 to 4.65.
- The maximum %increase in Nusselt number of louvered strip with diamond shaped leaves with respect to plain tube is 430.73% with friction factor increase of 46.38% and the overall enhancement ratio is found to vary from 2.08 to 4.67.
- As the overall enhancement ratio of louvered strip with diamond shaped leaves is marginally greater than that of louvered strip with elliptical shaped leaves.

Nomenclature

A_c	Cross sectional area for tube side fluid, m^2
C_{ph}	Specific heat at constant pressure (oil)
C_{pc}	Specific heat at constant pressure (water)
D_i	Inside diameter of tube, m
D_0	Outside diameter of tube, m
f_{the}	Friction factor(theoritical),tube side without insert
f	Friction factor, tube side with insert
f_0	Friction factor, tube side without insert
K	Thermal conductivity of fluid , W/mK
L	Length of each tube, m
m_h	mass flow rate of hot fluid(kg/sec)
m_c	mass flow rate of cold fluid(kg/sec)
Nu_0	Nusselt number, tube side with out insert
Nu	Nusselt number, tube side with insert
P	Wetted perimeter, $4*\Pi D_0$, m
ΔP	Pressure drop, m
Pr	Prandtl number
Q_h	Heat duty for hot fluid (Oil), W
Qc	Heat duty for cold fluid (Water), W
Re	Reynolds number
Th_{in}	Hot fluid inlet temperature, ^0C
Th_{out}	Hot fluid outlet temperature, ^0C
Tc_{in}	Cold fluid inlet temperature,^0C
Tc_{out}	Cold fluid outlet temperature,^0C
V	Mean velocity in tube, m/s
ρ	Density, kg/m^3
ρ_h	Density of hot fluid, kg/m^3
μ	Dynamic viscosity of oil at bulk temperature, kg/ms
μ_w	Dynamic viscosity of oil at wall temperature, kg/ms
η	l enhancement ratio

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