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CFD Investigation of Heat Transfer and Flow Patterns in Tube Side Laminar Flow and the Potential for Enhancement

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Heating and cooling of process streams is a standard operation in many industries. This operation is often performed in heat exchangers where the heated or cooled fluid flows under laminar conditions inside the tubes. The mechanisms of heat transfer under those flow conditions are complex and poorly understood, since they can involve both forced and natural convection making accurate prediction for heat exchanger design a challenge. In this paper Computational Fluid Dynamics (CFD) techniques in conjunction with heat transfer measurements are employed in order to investigate laminar flow behaviour in heating and cooling cases.
Heat transfer in the laminar flow regime is low by default but can be greatly increased by the use of passive heat transfer enhancement techniques such as tube inserts. Tube side enhancement in laminar flow is commonly used in heat exchanger design and leads to much smaller more efficient heat exchangers.
CFD will also be used to investigate the heat transfer mechanism found in enhanced tubes. Devices investigated are wire matrix turbulators (hiTRAN), twisted tapes and coils. The results from the CFD simulations are compared with experimentally measured data.
The CFD simulation results show good agreement with the experimental data. The Nusselt number was found to have increased by several times over the empty tube when using enhancement devices, with different improvement levels depending on the device used. The perceived mechanism for this increase was the greater movement of fluid evident from the CFD simulations.

1. Introduction

The mechanisms for laminar heat transfer in horizontal tubes are complex as they can be forced, natural and mixed convection. The dominant mechanism depends on the conditions and physical properties of the fluid being heated or cooled. This is different to the conditions in turbulent flow, where the heat transfer mechanism is dominated by forced convection (Holman, 1992).
Natural convection is where density changes in the fluid caused by the heating or cooling process causes motion within the fluid. This is due to buoyancy forces; in cases where the fluid is heated it becomes less dense near the tube wall so it rises to the top of the tube which pushes the fluid that was there down through the centre thus creating a recirculation inside the tube. Forced convection is where an external force is applied to the fluid to increase convection such as increasing the velocity of the fluid through the tube. Mixed convection describes a situation where the convection is a combination of the forced and natural convection mechanism. Therefore the fluid is forced through the tube at low enough velocities that the natural convection buoyancy forces still have an effect on the flow patterns inside the tube. Metais and Eckert (1964) have proposed the forced, mixed and free convection regimes in horizontal tubes; this will be looked at in detail in section 5.3. Due to the complexity of the heat transfer mechanism a variety of correlations to calculate the tube side heat transfer can be found in literature. The results of those differ considerably. In this paper the Nusselt number correlations by Sieder and Tate (1936) and Oliver (1962) for the laminar forced and mixed convection will be used to compare the results from the CFD and experimental empty tube results.

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Heat Transfer in the laminar flow regime can benefit from heat transfer enhancement devices, this paper will focus on three types of tube inserts, wire matrix (hiTRAN), twisted tape and coils, generally known as passive heat transfer enhancement devices. (Webb and Kim (2005))

Twisted tapes have been investigated in laminar flow by DuPlessis and Kröger (1987). The flow patterns have been numerical investigated in the turbulent region using CFD by Eiamsa-ard et al (2012) and experimentally by Thianpong et al (2012). They found that twisted tape inserts enhance the heat transfer by increasing the swirl flow inside the tube.

Comprehensive research of various Coil geometries in tube side flow has been carried out by García, et al (2007) and Solano et al (2012). They describe flow patterns that coil inserts create and investigated experimentally and numerically in the laminar and transitional regime the heat transfer and pressure drop characteristics. Coils enhance heat transfer by disrupting the boundary flow and by increasing swirl flow inside the tube.

hiTRAN is a wire matrix type insert manufactured by CALGAVIN who have experimentally investigated its heat transfer enhancement capabilities.

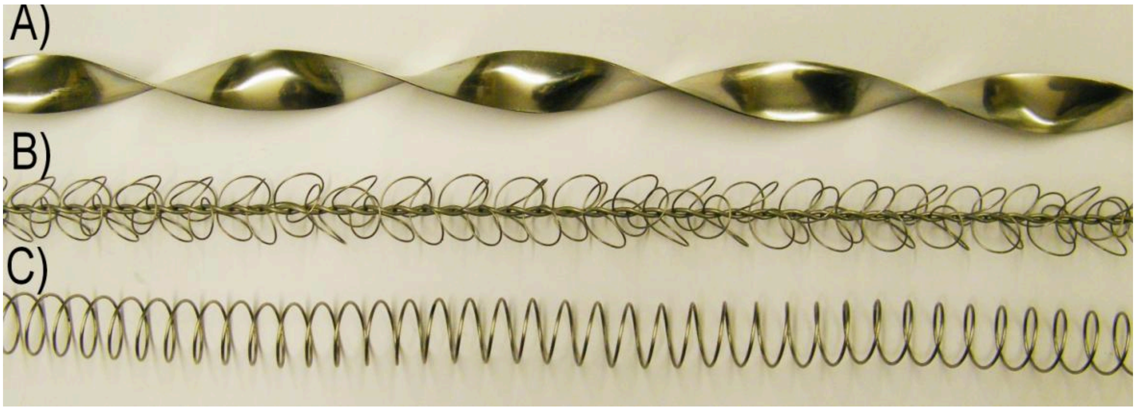

Figure 1: A) Twisted Tape, B) hiTRAN and C) Coil

CFD works by splitting a fluid domain (in this case a tube), into small cells creating a mesh. The computer program then solves the heat transfer and transport equations for each of the cell until it converges to a stable answerer. The advantage of using CFD is that the flow patterns inside the tube can be observed without having an effect on the result. (Versteeg and Malalasekera (2007))

2. CFD Models

The CFD package ANSYS CFX was used to carry out the simulations presented in this paper. The Inserts used are shown in figure 1. All inserts were manufactured to fit into a 22.1mm ID tube. The twisted tape insert was manufactured with a tape thickness of 0.8mm and a ratio L/D of 8.1. The low density hiTRAN insert had a voidage of about 97.6 %. In other words 2.4 % is taken up by wire material. The coil can be characterised by a wire thickness of 1.22mm and a pitch between adjacent loops of 10mm

The insert geometries were drawn using ANSYS DesignModeler. Some simplifications were made to the insert geometries to make them simpler to mesh. For the real geometries where a wire touches the tube wall a very acute angle would be formed which leads to poor quality mesh. Therefore these were drawn so that the wire touches the wall in a 90 degree angle. The geometries were then meshed using the ANSYS meshing software.

A mesh independence study was carried out on each of the geometries to ensure that the mesh was of high enough quality to accurate model each of the inserts. But at the same time does not require an excessive amount of computational time.

The simulations were carried out in the laminar flow regime from Reynolds number (Re) 50 to 2100. The Reynolds number is given Eq (1), where ρ is density (kg/m^3), D is tube diameter (m), v is velocity (m/s) and μ is dynamic viscosity (Pa s).

$$Re = \frac{\rho D v}{\mu} \tag{1}$$

Therefore the laminar flow model was used for the majority of models, however there were some CFD results especially towards the transitional flow region, that were closer to the experimental data by using the k-ε turbulence model. The fluid used for the experiments and CFD model was the heat transfer oil Transcal N. Expressions for density, viscosity, thermal conductivity and specific heat capacity change due to temperature were written into the CFD model. Of special interest is the temperature dependency of density, since this is the driving force for natural convection

This effect was included in the CFD models. The simulations where then set up using the experimental inlet and wall temperature and mass flow as the boundary conditions. An isothermal inlet section for each of the geometries was also made to develop the velocity profile with in the fluid before the heat transfer section. This was done as the experimental test rig also has an isothermal section before the heat transfer section.

3. Heat Transfer Calculations

The Nusselt number (Nu) was calculated as follows: with the Bulk fluid properties are taken as an average between the inlet and outlet of the tube. First Eq(2) is used to Calculate Q, the duty of the tube (W), $\dot{m}$ is mass flow (kg/s), A is tube surface area (m^2), C_p is specific heat capacity (J/kg °C) and ΔT (°C) is the temperature difference between the inlet and outlet of the tube.

$$Q = \dot{m} * C_p * \Delta T \tag{2}$$

Next U, the overall heat transfer coefficient (W/m^2 K) of the tube is calculated using Eq(3). With ΔT_{LMTD}, the log mean temperature difference given by Eq(4). Since in the experiments the annulus flow rate was very high the resulting wall temperature was almost constant over the whole tube length. An arithmetic average was used for the wall boundary condition in CFD.

$$U = Q \big/ (A * \Delta T_{LMTD}) \tag{3}$$

$$\Delta T_{LMTD} = \left(T_{WALL} - T_{OUT}\right) - \left(T_{WALL} - T_{IN}\right) \Big/ \ln\left(\frac{\left(T_{WALL} - T_{OUT}\right)}{\left(T_{WALL} - T_{IN}\right)}\right) \tag{4}$$

For the CFD calculation with constant Wall temperature the overall coefficient U equals the tube side coefficient hi. Finally Nu is calculated using Eq(5), D is tube diameter (m) and k is thermal conductivity (W/m K).

$$Nu = \left(hi * D\right) \big/ k \tag{5}$$

f the friction factor is calculated by Eq(6), where ΔP is pressure drop (Pa), ρ is density (kg/m^3) and L is length of tube (m).

$$f = \left(\Delta P * D^5 * \pi^2 * \rho\right) \Big/ \left(32 * L * \dot{m}^2\right) \tag{6}$$

4. Empirical Correlations

A number of different correlations for forced and mixed convection in laminar flow can be found in Literature. The most common one for forced convection laminar flow is by Sieder and Tate (1936) and for mixed convection laminar flow is by Oliver (1962).

The experimental and CFD Nu results were compared with the two correlations, in general the theory showed good agreement with CFD and experimental results. For low Reynolds numbers where natural convection start to become more dominant, as expected the forced convection only equation underestimates the tube side heat transfer. For experiments conducted towards forced convection at higher Reynolds numbers (Re > 400) the theoretical correlations over predict the coefficient by more than 10% compared to the CFD results and even more when comparing to the experimental results.

5. Results and Discussion

5.1 Heat Transfer

Next CFD was applied in order to compare measured outlet temperatures for enhanced tubes with simulated outlet temperatures. All experiments were conducted with the test fluid cooled inside the tube. Heat transfer experiments were conducted in a double pipe heat exchanger operating with heat transfer oil Transcal N on the tube side and cooling water flow on the shell side. All CFD simulated outlet temperatures are within ±2 % of the experimental data. This shows that CFD calculations can accurately

simulate the conditions in enhanced non adiabatic tube flows. This also means that the flow pattern seen in the CFD results should mirror what is happening in the experiments.
The results show that the heat transfer has been increased by the use of tube inserts. The greatest increase in Nu is achieved by the hiTRAN insert, giving a 420 % increase over the empty tube. Twisted tape gives a 257 % increase over the empty tube. Both these types of insert show a gradual change in Nu as the Re increase. Whereas the wire coil shows a large jump up in Nu above a Re of 1,600, an explanation for this can be found when the flow patterns are investigated.

5.2 Friction Factor

The friction factor results for the CFD simulations were compared to the theoretical plain empty tube friction factor for adiabatic flow. The plain empty tube simulation indicates an increase of friction factor compared to the adiabatic case due to the fact that near to the wall a more viscous fluid layer can be found. The overall results show that all the types of tube inserts increase the friction factor above that of an empty tube. The greatest increase is seen with the hiTRAN wire matrix inserts and twisted tape inserts with a 405 % and 223 % increase. However the increase in friction factor is balanced by having an increase in tube side heat transfer.

5.3 Flow Patterns

The flow patterns found in each of the inserts and empty tube flow were found by using the streamline tool in CFX. This tool traces were a “massless” particle would flow if released from the specified point inside the tube.

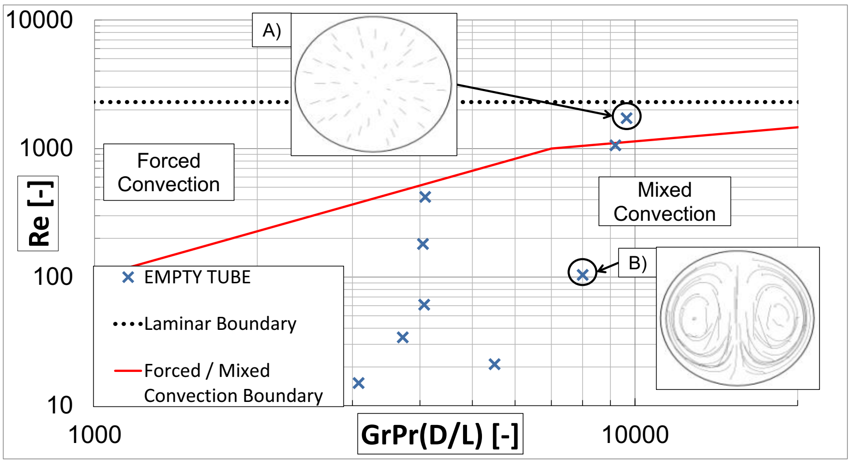

Figure 2: Forced and mixed convection regions, Metais and Eckert (1964). A) CFD forced convection flow patterns and B) CFD mixed convection flow patterns

Figure 2 shows the regions of Forced and mixed convection proposed by Metais and Eckert (1964). Where Grashof Number (Gr) is Eq(9) and Prandtl number (Pr) is Eq(10), where g is gravity (m/s^2), β is thermal expansion coefficient (C^{-1}) and ν is Kinematic viscosity (m^2/s).

$$Gr = \frac{g * \beta * \Delta T * D^3}{\upsilon^2} \tag{9}$$

$$\Pr = \frac{Cp * \mu}{k} \tag{10}$$

The CFD results for the empty tube have been plotted on to the figure. Point A is highlighted to show the flow patterns in the forced convection region and Point B is highlighted to show the mixed convection flow patterns. At point A there is very little fluid movement due to natural convection, however at Point B there is increased axial movement in the flow due to natural convection in the fluid. This shows there has been a change from Forced to mixed convection as Re decrease.
The flow patterns for the Coil are shown in Figure 3A. It is interesting to see that at low Re it has a similar pattern to an empty tube with natural convection dominating this explains the relative poor performance of this enhancement device at low flow rates. However as the Re increases the coil inserts has more of an effect on the flow pattern, with a swirl flow pattern taking over from the natural convection pattern. Also as seen in the heat transfer results there is a step up in Nu for the highest two Re it can be seen in figure 3A that the swirl flow produced by the inserts have greatly increased for those cases.

The twisted tape flow patterns shown in figure 3B shows that the swirl flow pattern is found over the whole Reynolds range investigated. This ties in with the heat transfer results as they show a gradual increase and no step change as found with the coil insert. Also from Figure 3B it can be seen as the Reynolds number increases the streamlines are closer to the tube wall.

hiTRAN flow pattern are shown in Figure 3C, as with twisted tape the flow patterns remains constant over the whole range of Reynolds, giving a gradual increase of heat transfer. However there is a greater amount of fluid movement seen in the hiTRAN flow patterns that any of the others. The greater fluid movement generated by the hiTRAN insert when compared with twisted tape and coil can explain the higher increase in heat transfer compared to the other devices.

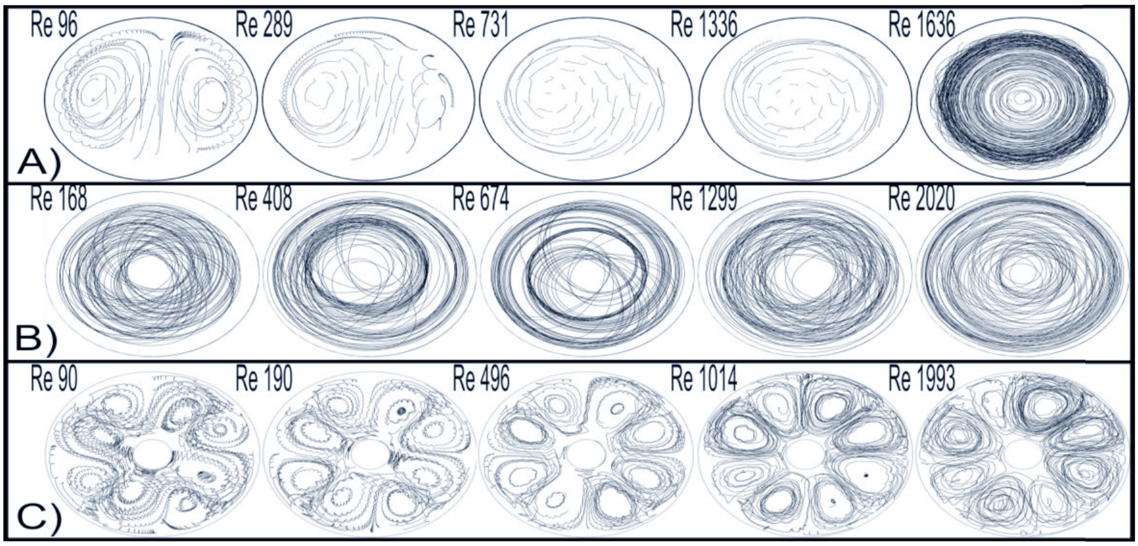

Figure 3: A) Coil insert flow patterns, B) twisted tape flow patterns and C) hiTRAN flow patterns

5.4 Stratification

Chapter 2 From the CFD images in Figure 4A and 4B is can be seen for empty tubes that an increased amount of temperature and velocity stratification can be found when the fluid operates in the mixed convention region compared to the forced convection region. Because of the natural convection movements the cold dense liquid moves towards the bottom of the tube and the hot less dense warm fluid rises to the top. Since the warm liquid has a lower viscosity, the velocity in the top region is also higher as can be seen in figure 5. Stratification can also lead to areas with greatly reduced heat transfer. When the liquid is cooled stratified flow at the bottom of the tube will show nearly the same fluid temperature as the cooling fluid, for that reason there is no driving temperature difference between those two fluids. As a consequence this area is lost for effective heat transfer. The same is valid when the flow is stratified due to heating. This is important as it reduces the effective length of the tube. Stratification can be greatly reduced by the use of hiTRAN this is evident by figure 4C, leading to much smaller more efficient heat exchangers.

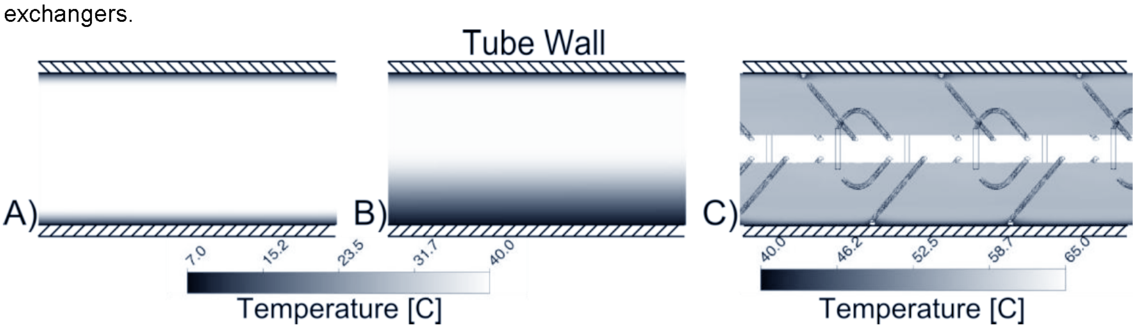

Figure 4: Temperature profile after 2500 mm a) empty tube in forced convection region, b) empty tube in mixed convection region and C) hiTRAN insert in mixed convection region

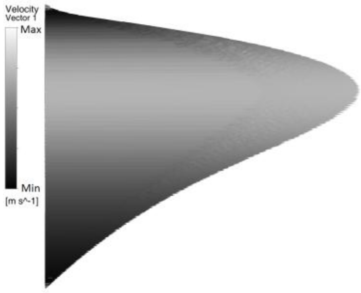

Figure 5: Mixed convection velocity profile, low velocity at bottom of tube and high velocity at top

6. Conclusions

This paper has demonstrated that CFD is a useful tool to analysis fluid flows through complex geometries. It can produce informative flow patterns of the internal flow that would not be easily obtained with traditional experimental techniques. The quantitative results with respect to tube side outlet temperatures and heat transfer performance are verified reliable.Of the inserts investigated in this paper it was found the hiTRAN wire matrix insert show the greatest enhancement in the laminar flow regime studied, with each of the other enhancement devices also showing some benefit over an empty tube. Changes in simulated and measured heat transfer results can be explained with changes in flow patterns

It is also shown that in plain empty tubes stratification, can be expected when operating in a laminar mixed convection regime. This investigation demonstrates that passive enhancement techniques are a suitable tool in order to avoid this flow pattern, with all its negative implications.

This demonstrates that CFD is a powerful reliable tool at predicting heat transfer in conjunction with flow patterns for fluid flow through tubes with complex internals.

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