Intertwined Helical Tube Insertion Impact on Thermal-Hydrodynamic Characteristics Associated with Internal Flows

[1]Ikpotokin I., [2]Uguru-Okorie D. C., [3]Dare A. A..
[1,2]The Department of Mechanical Engineering, Landmark University, Omu Aran, Kwara State, Nigeria.
[3]The Department of Mechanical Engineering, University of Ibadan, Oyo Sate, Nigeria.

Abstract: Heat transfer improvement has attracted a great deal of interest owing to the concern for energy saving and high thermal system performance requirement. For example, heat exchangers that operate in parallel flow are limited in engineering applications because of their inability to recover much heat. Tube inserts such as wire coil and twisted tapes are used to enhance heat transfer. Unfortunately, the attendant increase in pressure drop associated with wire coil and twisted tube inserts has become an increasing concern in industries. Therefore, an experimental investigation was carried out to determine the effect of intertwined helical tube insert on convective heat transfer coefficient and pressure drop for concurrent and countercurrent flow in tube heat exchanger. Pressure drop associated with the use of the tube insert was 10% and 12% greater than that of using plain tube for concurrent and countercurrent flows respectively. Such low resulting pressure drop could be due to the tube insert pitch ratio and well ordered flow within the tube. Thermal improvement results were 3.2 and 3.8 for concurrent and countercurrent flows respectively. This implies that the use of intertwined helical tube insert justify the additional pumping power required. In addition, much energy was recovered in concurrent flow using helical tube inserts.

Keywords: *Intertwined Helical Tube Insert, Tube Heat Exchanger, Concurrent Flow, Countercurrent Flow*

I. INTRODUCTION

The performance of heat transfer equipment in engineering and chemical process applications is influenced by the amount of heat to be transferred, flow rates and arrangement, the driving potential, and the amount of mechanical energy expended to facilitate heat transfer. This is because effective and efficient heat transfer as well as the associated flow is significant for achieving the thermal energy requirement in numerous processes that occur in engineering equipment, natural environment and living organisms. The significance are usually encountered in many engineering field such as power generation, heating, ventilation, air conditioning and refrigeration, chemical processing, pollution, space vehicles and so on. The phenomena of heat transfer and flow is also critical to the design, performance, selection, operation and maintenance of cost effective heat exchangers or heat transfer surfaces [1].

Most conventional and recent heat exchangers employ tube as the heat transfer surfaces. A typical example is the shell-and-tube exchangers. In such equipment, the process fluid is contained within heat conducting tubes. The other fluid is caused to flow over the tubes in order to remove heat. However, heat transfer is usually degraded by flow instability, localized surface heating, dead spots leading to fouling [2, 3]. In addition, structural malfunctioning such as tube failures are orchestrate by flow instability and accelerated fouling of dead zones [4]. These problems have caused waste of energy, high cost of maintenance, and increasing environmental concern. Therefore, there arose the need for saving energy and improving heat transfer systems effectiveness and efficiency.

The quest to improve thermal and hydrodynamic performance of heat exchangers while minimizing energy degradation as well as saving energy and materials had stimulated interest in the use of passive techniques heat promoters in recent years [5]. Before the introduction of heat promoters, flow arrangements (i.e. concurrent and countercurrent) were impressively employed to promote heat transfer. In concurrent flow, the hot fluid cannot fall below the outlet temperature of the cold fluid. Consequently, parallel-flow heat exchanger ability to recover heat is greatly affected [6]. For this reason, a large proportion of heat exchangers employed in industrial applications operate in countercurrent flow. However, the application of tube inserts has promising potential for recovering large amount of heat from concurrent flow as well as countercurrent flow configurations.

Passive method of heat transfer enhancement involves the introduction of metal shaped in a particular form (i.e. tube inserts) into the tube to redirect the flow and reduce the hydraulic diameter. Some examples of well known tube inserts include twisted tapes, wire coils, ribs and dimples. This method is usually preferred in many practical situations compared to active techniques because the tube manufacturing process is simple. In addition, the technique can be employed in an existing heat exchanger, and no external power input is required [7].

In the past decade, studies concerning twisted tapes and wire coils tube insertions have increased significantly owing to their economic and improved heat transfer. A literature survey carried out by Dewan et al [8] shows that twisted tapes were found to perform better in laminar flow regime while wire coils perform excellently well for turbulent flow. It means that for any particular application, efficient design of compact heat exchanger requires suitable tube insertion configuration selected according to the working condition.

By increasing flow intensity through insertion of coil wire or twisted tapes in the tube the heat transfer and flow resistance increase [9]. Higher pressure drop resulting from increase in flow resistance leads to an increase in pump or fan power consumption. This limits the use of tube inserts in industrial applications where the improvement in heat transfer cannot compensate for the additional pumping power cost required [10, 11]. This has been a major disadvantage of tube insertions, and as such an optimized value of pressure drop for optimum heat transfer rate as well as power consumption is needed. Considering the attendant effect of high pressure drop associated with the use of coil wire and twisted tape as well as the flow regime that favours their use, an intertwined helical tube insert was employed in this investigation to determine the thermal-hydrodynamic characteristics performance of the heat exchanger. The interest to use this particular tube insertion in this research was stimulated by the gain in heat transfer and low pressure drop as well as low fouling recorded in shell and tube heat exchangers fitted with helical baffles [12].

The objective of this investigation is thus to evaluate the effect of intertwined helical tube insert on thermo-hydrodynamic behaviour for concurrent and countercurrent flow arrangements.

II. MATERIALS AND METHOD

As shown in figure 1.1, the experimental set up consists of two concentric tubes, base plate, flexible tubing, electronic control valves, thermocouples, personal computer and other accessories. As a major component of the experimental system, the four sets of concentric tubes were arranged in series to reduce the overall length and allow the temperature mid way along both fluid streams to be measured. Each of the inner tubes is a horizontal straight stainless steel tube of 330 mm long, 9.3 mm inner diameter and 9.5 mm outer diameter. The temperatures of both fluid streams were measured using thermocouples. The mass flow rate of hot or cold fluid (water) was measured with flow meters. The signals of thermocouples, pressure sensors and flow meters were recorded by the data acquisition system and processed in the computer.

The passive tube insert employed in this experiment is shown in figure 1.2. The device consists of three binding wire twisted together into a strand of 1.4 mm thickness. The single wire strand was formed into a helical shape with pitch to tube diameter ratio equal to 3.6.

In operation, the hot fluid from the hot water circulator passes through the inner stainless steel tube and cold fluid from the cold water supply passes through the annulus created between each inner metal tube and clear acrylic outer tube. This arrangement minimizes heat loss from the exchanger without the need for additional insulation on the cold water flow line.

The heater was set to 40^{o}C above the ambient temperature of 20^{o}C to ensure that the water temperature in the boiler is at reasonably high temperature. While the flows were set to counter configuration, the control valves for both hot and cold water were set to 100% (maximum) opening to eliminate any trap air bubbles without tube inserts. The system was allowed to stabilize for five minutes before taking readings. The flow rate and temperate were obtained through their respective measuring instrument. A minimum of fifty data points were collected for each experimental runs from which the required heat transfer parameters were calculated. The experiment was repeated four times by reducing the flow rate by 15%.

The experimental procedures were repeated for counter current flow with the introduction of tube insert, as well as for concurrent flow with and without tube insertion.

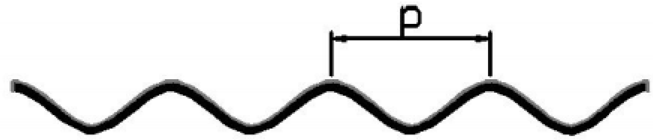

Figure 1.1: Helical tube insertion

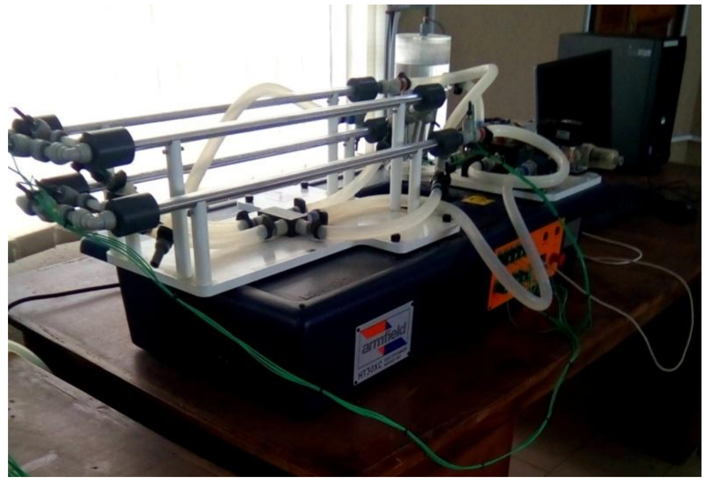

Figure 1.2: Experimental setup

Basic assumptions and calculation equation:

On the basis of the law of conservation of energy,

$$Q_e = Q_a \tag{1}$$

For countercurrent flow

$$Q_e = \dot{m}_h c_h (T_1 - T_5)_h \tag{2}$$

$$Q_a = \dot{m}_c c_c (T_6 - T_{10})_c \tag{3}$$

For concurrent flow

$$Q_e = \dot{m}_h c_h (T_5 - T_1)_h \tag{4}$$

Where Q_e is heat lost by hot water, Q_a is heat absorb by cold water

c_h is specific heat capacity of hot water

c_c is specific heat capacity of cold water

$\dot{m}_h$ is mass flow rate of hot water

$\dot{m}_c$ mass flow rate of cold water

T_1 is hot fluid inlet temperature

T_5 is hot fluid exit temperature

T_6 is cold fluid inlet temperature

T_{10} is cold fluid exit temperature

The logarithm mean temperature difference (LMTD) is determined from

LMTD or $$\Delta T_{lm} = \frac{(T_1 - T_{10}) - (T_5 - T_6)}{\ln((T_1 - T_{10})/(T_5 - T_6))} \tag{5}$$

In this work, equation (5) is the same for both countercurrent and concurrent operation because the temperature measurement points are fixed on the heat exchanger. Convective heat transfer coefficient inside the tube is

$$h = \frac{Q_e}{A_s \Delta T_{lm}} \quad (6)$$

$$Nu = \frac{hD_h}{k} \quad (7)$$

The frictional factor can be calculated from

$$f = \frac{\Delta p}{(\rho u^2/2)/(L/D_h)} \quad (8)$$

According to [13],
Thermal improvement factor is expressed as

$$\eta = \frac{Nu/Nu_o}{(f/f_o)^{1/3}} \quad (9)$$

Where,

h is average heat transfer coefficient
As is heat transfer surface area
D_h is hydraulic diameter
k is fluid thermal conductivity
Nu is Nusselt number associated with the use of tube insert
Nuo is Nusselt number associated without the use of tube insert
f is friction factor obtained using tube insert
f_o is friction factor obtained without the use of tube insert
Δp is pressure drop
L is length of tube
For tube without insertion, $D_h = D$

For tube with insertion,

$$D_h = \frac{LD^2 - Lct^2}{LD - Lct} \quad (10)$$

$$Lc = \frac{\pi DL}{p}$$

$$\tan\alpha = \frac{\pi D}{2p}$$

Reynolds number for flow without insertion

$$Re_o = \frac{4\dot{m}}{\pi D \mu} \quad (11)$$

Reynolds number for flow with insertion

$$Re = \frac{\dot{m} D_h}{A_c \mu} \quad (12)$$

Where,

$$A_c = \frac{\pi D^2}{4} - tD$$

Where
D is tube internal diameter
t is tube thickness
Lc is helical tube insert length
Ac is the flow area
P is helical pitch
α is helical angle

Average values of the inlet and exit temperatures as well as flow rate and pressure drop for the fifty data point was determined. These mean parameters with the fluid properties and tube geometry were used to calculate the performance characteristics of the tube heat exchanger.

III. RESULT AND DISCUSSION

In general, the Reynolds number, pressure drop, convective heat transfer coefficient, rises with increase of valve opening in both counter and concurrent arrangement with and without tube insert and maximum at 100% valve openings. For minimum valve openings, the Reynolds number values were greater than 2300 in this experimental work. This implies that the flow is turbulent. However, the Reynolds number values obtained with the application of tube insertion was about 20% more than using just tube without insertion. This was as a result of the smaller flow cross section or hydraulic diameter, flow disruption and reattachment of boundary layers and the generation helical flows.

Figure 1.3 and 1.4 depict the result of convective heat transfer coefficient for current and concurrent flows respectively. In the case of counter current flow, the average convective heat transfer coefficient realized with tube insert was about 290% higher than using plain tube. For concurrent flow with tube insert, the average convective heat transfer coefficient was about 260% higher than using plain tube. This could be as a result of the enhance turbulent flow contributed by smaller hydraulic diameter, flow disruption and reattachment of boundary layers as well as generation of helical flows.

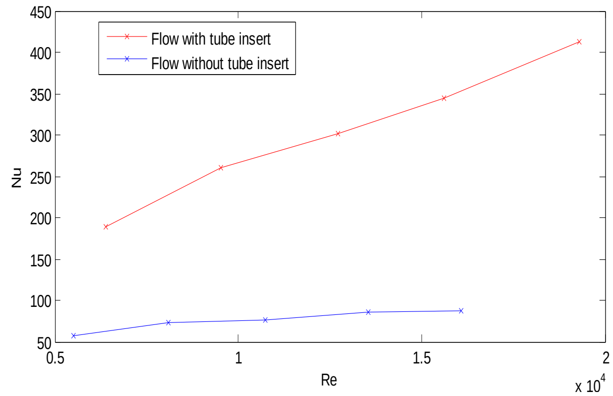

Figure1.3: Heat transfer coefficient in counter flow configuration

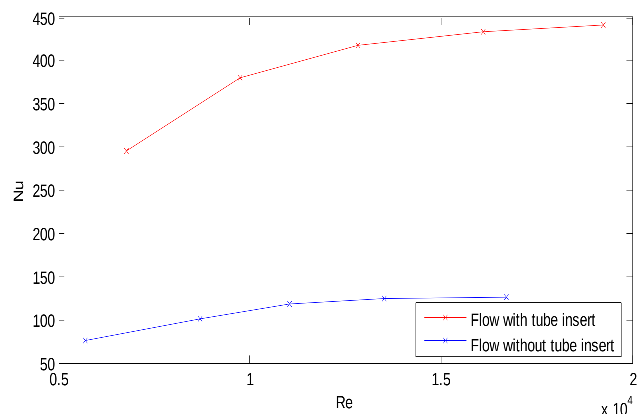

Figure 1.4: Heat transfer coefficient for concurrent in configuration

The friction factor associated with the flow arrangement is shown in figures 1.5 and 1.6 for counter and concurrent flows respectively. The factor decrease with increasing Reynolds number. In counter flow with tube insert, the frictional factor was on the average about 12% greater than that of using plain tube while in the case of concurrent flow with tube insert, the frictional factor was on the average 10% more than what was obtained using plain tube. Such low resulting friction factor could be due to the tube insert pitch ratio and well ordered flow (helical flow) within the tube.

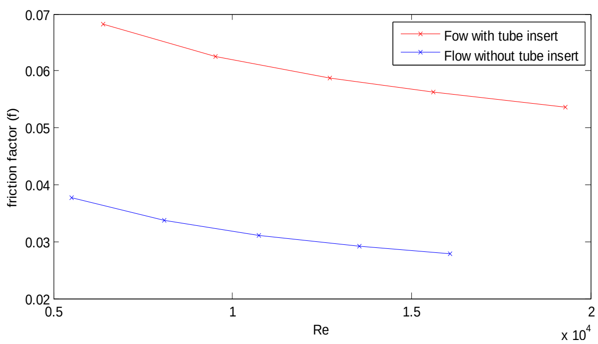

Figure 1.5: Friction coefficient in counter current flow

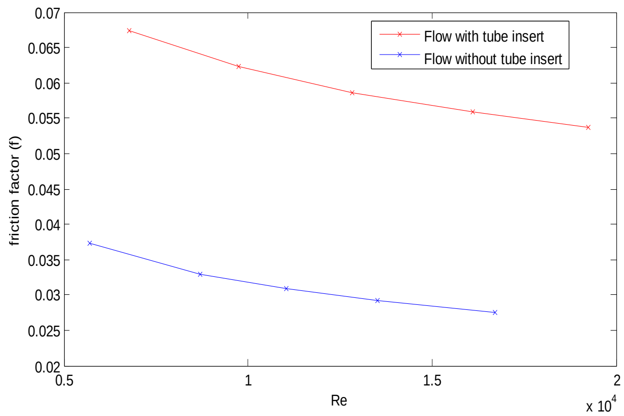

Figure 1.6: Friction factor in concurrent

The thermal improvement factor was determined using equation (9) and the results are shown in figures 1.7 and 1.8 for counter flow and concurrent flow respectively. Thermal enhancement as high as 3.8 was obtained for maximum valve opening in counter flow and 3.2 for concurrent flow showing that the use of helical tube insert is justifiable although additional pumping power required. In addition, much energy can be recover in concurrent flow using helical tube inserts.

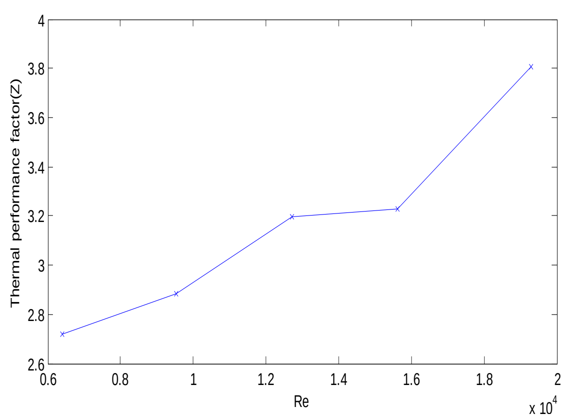

Figure 1.7: Thermal performance factor for counter current flow.

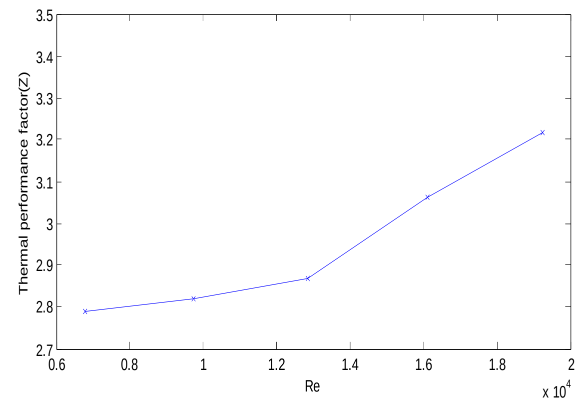

Figure 1.8: Thermal performance factor for concurrent flow.

CONCLUSION

This study has established that the use of intertwined helical tube insert will lead to improved performance in shell and tube heat exchanger due to the high amount of heat recovered and the relatively low pressure drop obtained.

References

[1] Ming Pan, Sara Jamaliniya, Robin Smith, Igor Bulatov, Martin Gough, Tom Higley and Peter Droegemueller, (2013). New insights to implement heat transfer intensification for shell and tube heat exchangers. Energy, Vol. 57, 208-221.

[2] Sirous Zeyninejad Movassag, Farhad Nemati Taher, Kazem Razmi, and Reza Tasouji Azar, (2013). Tube bundle replacement for segmental and helical shell and tube heat exchangers: Performance comparison and fouling investigation on the shell side. Applied Thermal Engineering, Vol. 51, 1162-1169.

[3] Tiejun Zhang, Tao Tong, Je-Young Chang, Yoav Peles, Ravi Prasher, Michael K. Jensen, John T. Wen, and Patrick Phelan, (2009). Ledinegg instability in microchannels. International Journal of Heat and Mass Transfer, Vol. 52, 5661-5674.

[4] J.S. Corte, J.M.A. Rebello, M.C.L. Areiza, S.S.M. Tavares, and M.D. Araujo, (2015). Failure analysis of AISI 321 tubes of heat exchanger. Engineering Failure Analysis, Vol. 56, 170-176.

[5] Usman Salahuddin, Muhammad Bilal, and Haider Ejaz (2015). A review of the advancements made in helical baffles used in shell and tube heat exchangers. International Communications in Heat and Mass Transfer, Vol. 67, 104-108.

[6] Thanhtrung Dang, Jyh-tong Teng, and Jiann-cherng Chu, (2010). Effect of Flow Arrangement on the Heat ransfer Behaviors of a Microchannel Heat Exchanger. Proceedings of the International MultiConference of Engineers and Computer Scientists Vol. 3, Hong Kong.

[7] Mohsen Sheikholeslami, Mofid Gorji-Bandpy, and Davood Domiri Ganji, (2015). Review of heat transfer enhancement methods: Focus on passive methods using swirl flow devices. Renewable and Sustainable Energy Reviews, Vol. 49, 444-469.

[8] A Dewan1, P Mahanta1, K Sumithra Raju1 and P Suresh Kumar, (2004). Review of passive heat transfer augmentation techniques. J. Power and Energy, Vol. 218, 509-527.

[9] Y. You, et al., (2013). Experimental and numerical investigations of shell-side thermohydraulic performances for shell-and-tube heat exchanger with trefoil-hole baffles. Applied Thermal Engineering, Vol. 50, (1) 950–956.

[10] Q. Wang, et al., (2009). Numerical investigation on combined multiple shell-pass shell-and-tube heat exchanger with continuous helical baffles, International Journal of Heat and Mass Transfer, Vol.52, (5-6) 1214–1222.

[11] M. Reppich, J. Kohoutek, (1994). Optimal design of shell-and-tube heat exchangers. Comput. Chemical Engineering, Vol.18, 295–299.

[12] S.B. Genić, et al., (2013). Analysis of fouling factor in district heating heat exchangers with parallel helical tube coils, Int. J. Heat Mass Transf. Vol. 57, (1) 9–15.