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Experimental and Numerical Study of Heat Transfer Enhancement in a Shell and Tube Heat Exchanger using Helical Coiled Wire Inserts

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ABSTRACT

An experimental work on heat transfer enhancement in a shell and tube heat exchanger by insertion of helical coiled wires into the tube bundle was conducted. Four different pitch sizes were used (4.8, 6.4, 8.0, 9.6) mm. The working fluid for both shell and tube sides of the exchanger was water. Flow arrangement used was parallel and laminar for both sides with a Reynolds number for the tube inside ranging (100-920). Variables were taken into account, including; convection heat transfer coefficient, number of heat transfer units, thermal effectiveness, and pressure drop. CFD analysis was done in order to validate the experimental results. The study results showed an increase in heat transfer coefficient on the tube side by 54% due to the use of 9.6 mm pitched inserts. Highest thermal effectiveness obtained was in the case of 9.6 mm pitch inserts. Pressure drop rose up to 260% due to inserts.

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دراسة تجريبية وعددية لتحسين انتقال الحرارة في مبادل حراري غلافي انبوبي باستخدام حشوة سلكية حلزونية

الخلاصة

تم اجراء اختبار لتحسين انتقال الحرارة في مبادل غلافي انبوبي بأدخال حشوات سلكية حلزونية الى حزمة الانابيب. استخدم اربع احجام للخطوات بين حلقات الحشوة الحلزونية، (4.8, 6.4, 8.0, 9.6) ملم. المائع المستخدم كان الماء ولكلا جانبي المبادل. ترتيب الجريان كان موازي وصفائحي لكلا الجانبين، ومدى رقم رينولدز لجهة الانابيب يتراوح بين (100-920). تم الاخذ بعين الاعتبار عدة متغيرات وتشمل: معامل انتقال الحرارة بالحمل و-عدد وحدات انتقال الحرارة- الفعالية وانخفاض الضغط. اجري تحليل CFD للمقارنة مع النتائج المختبرية. اظهرت النتائج ان هناك زيادة في معامل انتقال الحرارة لجهة الانابيب وبقيمة 54%، بسبب استخدام الحشوات ذات الخطوة 9.6 ملم. اعلى فعالية تم الحصول عليها كانت في حالة الحشوات ذات الخطوة 9.6 ملم. ان الحشوة زادت من هبوط الضغط بنسبة 260%.

1. INTRODUCTION

The heat exchanger is a device used to transfer thermal energy between two or more fluids with direct or indirect contact between them that is if there is a temperature difference between the fluids. The work principles of a heat exchanger relay on three heat transfer methods, convection, conduction, and radiation. Improving heat exchanger is usually done by modifying the heat transfer surface or the cross sectional area, these modification result in an increase in heat transfer coefficient through the surface or an increase in heat transfer area. There are two main methods for improving heat transfer in a heat exchanger; "passive techniques", which include the use of fins, rough surfaces, helical wire and tape inserts. These techniques do not require the use of external energy during the operational phase of the heat exchanger. On the other hand, techniques that require an external source of energy, as in improving the working the working fluids, using electrical fields, or mechanical mixing, are called "active techniques" [1].

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Nomenclature

A	area, (m^2)
C	heat capacity rate, (W/°C)
C_p	heat capacity at constant pressure
D	diameter, (m)
d	inner diameter, (m)
h	convection heat transfer coefficient, (W/m^2.°C)
k	thermal conductivity, (W/m^2.°C)
L	length (m)
$\dot{m}$	mass flow rate, (kg/s)
ΔP	pressure drop, (Pa)
q	heat transfer rate, (W)
T	temperature, (°C)
ΔT	temperature difference, (°C)
U	overall heat transfer coefficient, (W/m^2.°C)
UA	overall heat transfer coefficient per unit area, (W/°C)
NTU	number of heat transfer units
ε	thermal effectiveness

Subscript

avg	average
b	average bulk
c	cold water
h	hot water
lm	logarithmic
I	heat exchanger fluid entrance
Π	heat exchanger fluid exit
s,c	tube bundle external surface
s,h	tube bundle internal surface
surf	the outside surface of the tube bundle
str	cold water stream near the tube bundle outside surface

Extensive research have been done in the field of improving heat transfer using the passive techniques, these improvements for a tubular heat exchangers mainly include the use of external modifications where the tube or tubes external surface is modified, as done by Abduljalil et al. [2] in the use of external fins for the most inner tube in a triplex tubular heat exchanger with phase change material (PCM), they showed that increasing the number of fins results in a reduction in the time required to melt the (PCM) by 69.5% when compared with a slandered triplex exchanger, also the increase in fin length led to a reduction in melt time for the (PCM) as high as 73.9% in the case of 10mm fins compared to a standard triplex exchanger. Also Kumar et al. [3] used different fin designs (triangular, rectangular, and parabola) that were added on external surface of the inner tube of a double pipe heat exchanger, their results showed that the use of fins increased the effectiveness of the heat exchanger by 21% for the rectangular fins, and pressure drop where the lowest in the case of parabolic fins. Researchers also tried to increase the efficiency of the used passive techniques, such as in the case of Indhe and Bhatkar [4] were they studied the enhancement of the efficiency of longitudinal fins, they in turn concluded that the best height for the fins was 15 mm.

In addition to external improvements, passive techniques also include internal modifications to the flow channels of a heat exchanger to cause disruption in the flow which in turn increases heat transfer. Rozzi et al. [5] done a comparative study between smooth and corrugated tubes in a shell and tube heat exchanger, results showed that in the laminar flow region the improvement effects were negligible, as for turbulent flow an enhancement in overall heat transfer was achieved but at the expense of an increase in pressure drop. Also Zhi-Min et al. [6] numerically showed that the use of twisted tape insert for a tube leads to a steady increase in Nusselt number and absolute vortex flow along the main flow as the twist ration for the tape decreases, moreover increasing the thickness of the tape also led to an increase in Nusselt number and vortex flow but after a certain point the effect became negative. Urkudde and Farkade et al. [7] employed a perforated helical tape insert with different perforation index values, as a result of their study they reached an understanding that an increase in the perforation index leads to a reduction in friction factor as well as a drop in heat transfer values due to lesser eddy and vortex formation, also it was found that heat transfer increases by 4.12 times for a tube fitted with perforated helical insert when compared to a standard smooth tube.

Active techniques also have been studied extensively; these methods include any change or augmentation done to the working fluid. An example of these techniques is the use of nanofluids as in the case of Salman et al. [8] where they studied a laminar flow in a tube fitted with parabolic-cut twisted tape insert, they also used (CuO) and water as a nanofluid with two different concentrations. There numerical studies showed that an increase in the volume fraction of the Nano particles (CuO) causes an increase in Nusselt number, also the results showed that the thermal performance of a tube fitted with parabolic-cut twisted tape is 10% higher than that of a standard twisted tape. Eiamsaard et al. [9] also studied the use of nanofluids in a double pipe heat exchanger equipped with overlapped duel twisted tapes; the nanofluid used was Titanium Oxide (TiO_2).and in three different concentrations. Their results show that the increase in number of Nano particles leads to an increase in Nusselt number and friction factor, also thermal performance was increased by 1.7%-4.5% as compared to distilled water when used as the working fluid, finally increasing the twist ration had a positive effect on thermal performance.

In addition to the use of nanofluids, active techniques can include the use of external power to disrupt the flow inside the heat exchanger channels. Such technique is used by Omkar et al. [10] in the form of rotating inner tube of a double pipe heat exchanger equipped with helical fins on the external surface of the inner tube. Their experimental results revealed that a double pipe heat exchanger with inner rotating finned tube has a Nusselt number 4 times higher than that of a standard double pipe heat exchanger.

The purpose of the study was the demonstration of the ability to increase the rate of heat transfer in a shell and tube heat exchanger by enhancing the inner surface of the tube bundle. This enhancement is achieved by the use of helically twisted wire inserts with four different pitch arrangements. In order to reach to a conclusion regarding the current study, number of variable were accounted for, which include; heat transfer coefficient, number of heat transfer units, effectiveness, and pressure drop.

2. EXPERIMENTAL SETUP AND TEST SECTION

The experimental setup consisted of a shell and tube heat exchanger, electrical water heater, cold water reservoir, pumping and tubing system, and a data logger for experimental data collection, Fig. 1 shows a schematic diagram of the experimental setup. The test section was the shell and tube exchanger, the acrylic shell inside diameter was 150 mm, tube bundle consisted of 18 copper tubes with 320 mm length and an inner and outer diameter 10.5 mm, 12.7 mm respectively. Each inlet and outlet of the exchanger had a K-type thermocouple connected to it in order to measure the temperature during the test period, also 4 more K-type thermocouples were place inside the shell 2 alongside one of the tube bundle and the other 2 in contact with the tube surface, Fig. 2 demonstrates the placement of the thermocouples. Hot water is supplied to the tube side of the exchanger using a centrifugal pump (0.37 KW), and ran in a closed loop, meanwhile the same type pump is used to supply clod water stream to the shell side in open loop style, were cold water enters the shell side and exits to a drain. Volumetric flow rate for both sides was controlled using a needle valve with a range of (0.5-4.0) LPM. Coiled copper inserts were placed inside each tube of the tube bundle. Four pitch lengths of the helical collided wire inserts were used (4.8, 6.4, 8.0, and 9.6) mm, the diameter of the helical inserts was 10.5 mm, Fig. 3.

The test was carried out for each pitch size starting with an empty tube bundle as a reference point, flow rate for the cold (shell) side was fixed while the hot (tube) side flow rate was changed each 15 min by 0.5 LPM intervals for a flow rate range of (0.5-4.0) LPM. Each 15 min temperatures were registered by the thermocouples and sent to the data logger which was in turn connected to a computer for data storage.

3. DATA REDUCTION

The data that have been acquired from the test which include temperatures as well as flow rates for the test section must be reduced to a much comparable form. Assumptions were made for this study:

- Steady state system.
- Environment effects on the heat exchanger are negligible.
- Steady thermal diffusion.
- Tube wall thermal resistance is uniformly distributed

Abiding by the permissioned assumptions the fallowing equations are used to acquire the variables:
First the average heat transfer rate is calculated by [11]:

$$q_h = C_h \times \Delta T_h \tag{1}$$

$$q_c = C_c \times \Delta T_c \tag{2}$$

where;

$$C = \dot{m} \times Cp \tag{3}$$

$$q_{avg} = \frac{q_h + q_c}{2} \tag{4}$$

Secondly the log mean temperature difference is calculated [11],

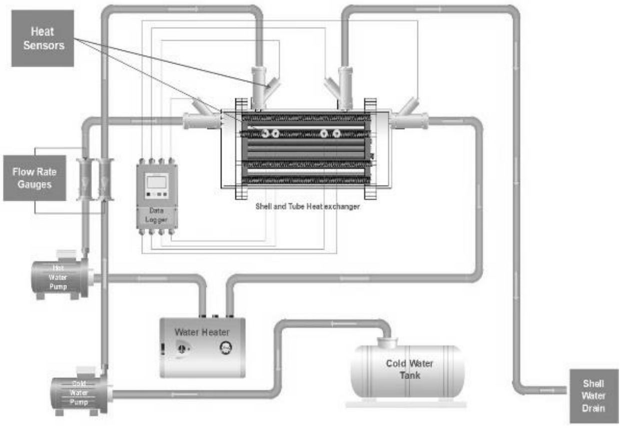


Fig. 1. Schematic diagram of the experimental setup.

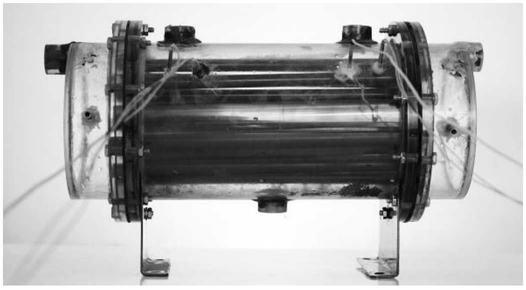


Fig. 2. Placement of the thermocouples along the tube bundle.

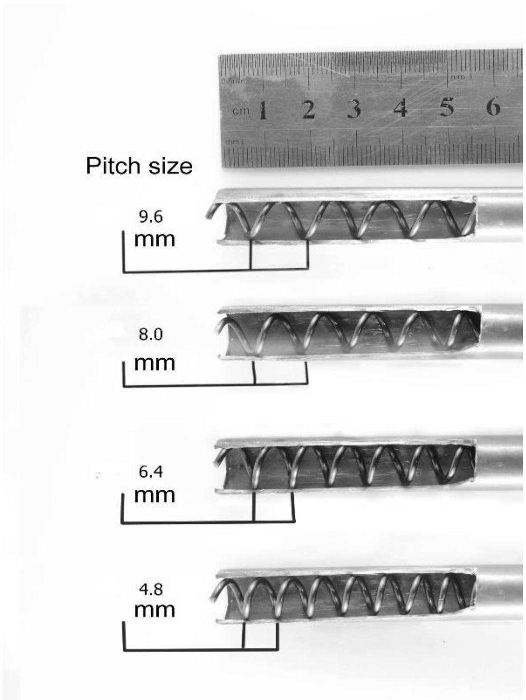


Fig. 3. Different pitch sizes.

$$\mathrm{LMTD} = \Delta T_{lm} = \frac{\Delta T_{I} - \Delta T_{II}}{\ln(\Delta T_{I}/\Delta T_{II})} \tag{5}$$

After that the overall heat transfer coefficient per unit of area can be found as [11]:

$$UA = \frac{q_{avg}}{\Delta T_{lm}} \tag{6}$$

This in turn is used to find the number of transfer units [11]:

$$NTU = \frac{UA}{C_{min=h}} \quad (7)$$

The thermal effectiveness is found by [11]:

$$\varepsilon = \frac{\Delta T_{min=c}}{\Delta T_{max}} \quad (8)$$

Finally, the heat transfer coefficient for both shell and tube sides is obtained by [11]:

$$h_h = \frac{q_{avg}}{A_{s,h}(T_{b,h} - T_w)} \quad (9)$$

$$h_c = \frac{q_{avg}}{A_{s,c}(T_w - T_{b,c})} \quad (10)$$

4. VALIDATION OF EXPERIMENTAL RESULTS

In order to reach an accurate conclusion, part of the experimental results must be compared to the results of a valid source, this is done by a Computational fluid dynamic analysis (CFD), using the commercial program ANSYS FLUENT® 16.0. The design model was one tube from the tube bundle surrounded by a cold water domain, Fig 4. Two cases were studied, the first case was for a smooth tube, and the second was for a tube fitted with helical coiled wire insert with 9.6 mm pitch. A mesh of the design model was generated using ANSYS mesh tool, after experimenting with different mesh sizes, a mesh was reached with an optimum cell count of 367497, average element quality of 0.82, maximum cell skewness of 0.91, and an aspect ratio of 14.98. The ANSYS FLUENT® 16.0 program uses the numerical method to solve the Navier-Stokes equations; the solution of those equations was done on the basis of the fallowing assumptions:

- 3D model.
- Steady laminar flow.

Newtonian and incompressible fluid.

In order to obtain a reasonable solution for the study in point, a boundary conditions must be set for the model which include:

- Hot water inlet at 60°C (333k), steady flow speed.
- Cold water inlet at about 25°C (298).
- Outlet for both hot and cold water where set to atmospheric pressure.

Fig. 5 shows boundary conditions and temperature measuring points along the tube. After the CFD analysis is complete, a comparison was done between the experimental results and the CFD analysis results regarding heat transfer coefficient as shown in Fig 5 and 6.

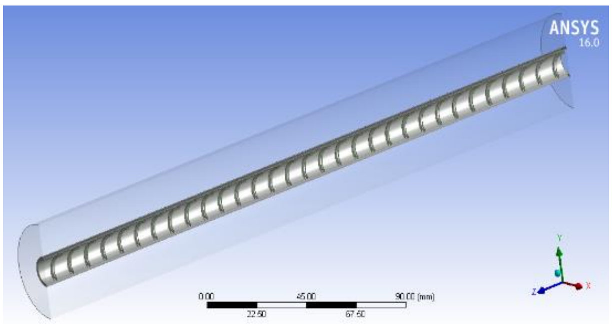


Fig. 4. CFD design model.

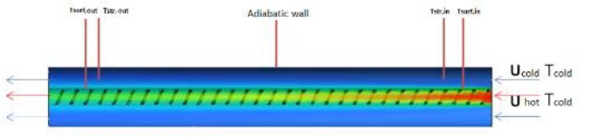


Fig. 5. Boundary conditions and temperature measurement points along the test section.

5. RESULTS AND DISCUSSION

5.1. Heat Transfer Coefficient

The heat transfer coefficient increases with the increase of Reynolds number, also a noticeable jump in heat transfer coefficient is noticed between the case of smooth tubes and the other four cases which represents a tube bundle with inserts. The increase in heat transfer coefficient for tube bundle with inserts over the smooth tube bundle are; 42% for 4.8 mm pitched tubes, 41% for 6.4 mm pitched tubes, 50% for 8.0 mm pitched tubes and 54% for 9.6 mm pitched tubes. That difference is caused by the formation of small turbulence regions along the stream which in turn leads to the renewal of the boundary layer along the tube inside wall. Fig. 7. shows the relation between the heat transfer coefficient and Reynolds number for five different cases, which include a shell and tube heat exchanger fitted with a smooth tube bundle, and the same heat exchanger but it was fitted with a tube bundle that contain four different pitch sizes. The highest average of heat transfer coefficient was for the tube bundle with 9.6 mm pitched helical inserts, and the highest value obtained was 833 W/m^2.°C. As for the smooth tube bundle case, the highest value obtained was 595 W/m^2.°C. The case of 9.6 mm pitch tubes was the highest because it was the longest pitch used in the study, which allows the formation of the most number of turbulence regions along the flow Fig. 8 shows the difference in volume rendering of temperature along a tube of the tube bundle for two cases. As far as goes for the shell side, the highest average of heat transfer the highest average of heat transfer coefficient was in the case of tube bundle fitted with 8.0 mm pitched helical inserts, and the highest value obtained was 1637 W/m^2.°C. The highest heat transfer coefficient value in the case of a smooth tube bundle was 1051 W/m^2.°C. Fig 9 shows the difference in heat transfer coefficient against Reynolds number for all studied cases.

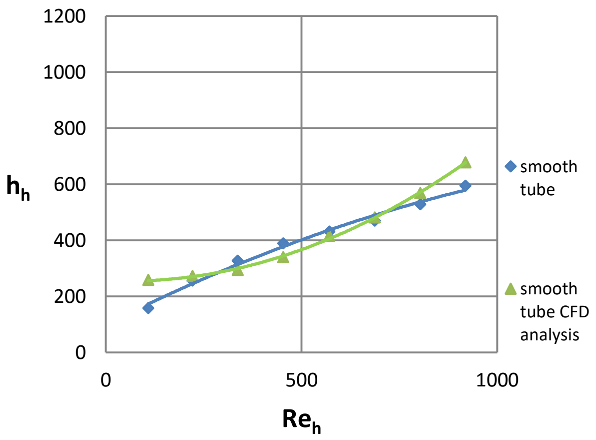


Fig. 6. The relation between Reynolds number and heat transfer coefficient for tube side.

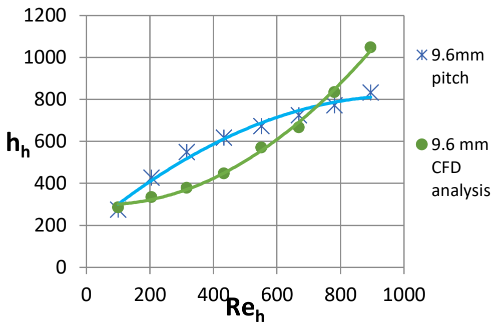


Fig. 7. The relation between Reynolds number and heat transfer coefficient for tube side.

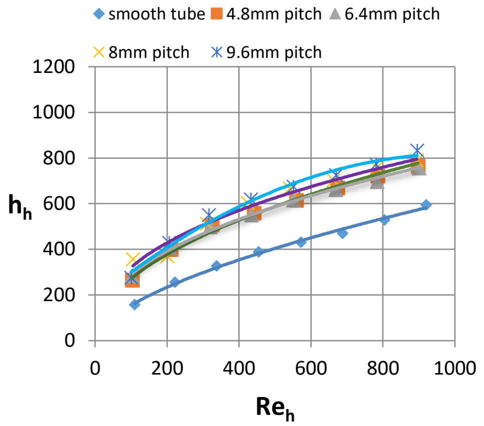
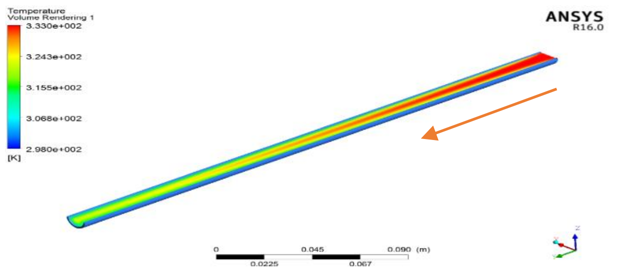
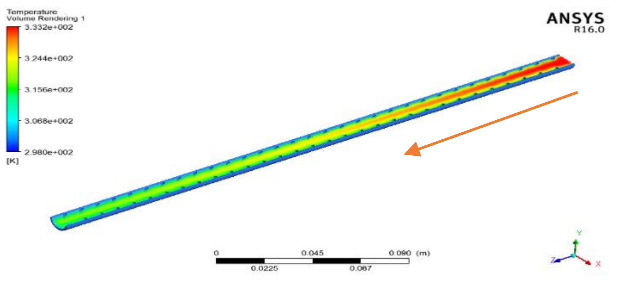


Fig. 8. The relation between Reynolds number and heat transfer coefficient for tube side (experimental results).



(a)



(b)

Fig. 9. Temperature volume rendering, (a) smooth tube and (b) tube with insert.

5.2. The Relationship between Number of Transfer Units and Effectiveness

Fig. 10 Shows the relationship between the number of heat transfer units (NTU) and effectiveness of the heat exchanger (Ɛ). The results showed that the highest average number of transfer units and effectiveness was in the case of the tube bundle fitted with 8.0 mm pitched helical inserts, and the highest value of (NTU) and (Ɛ) obtained was 0.76 and 0.42 respectively. On the other hand, the highest values of (NTU) and (Ɛ) in the case of the heat exchanger fitted with smooth tube bundle were, 0.5, and 0.32 respectively

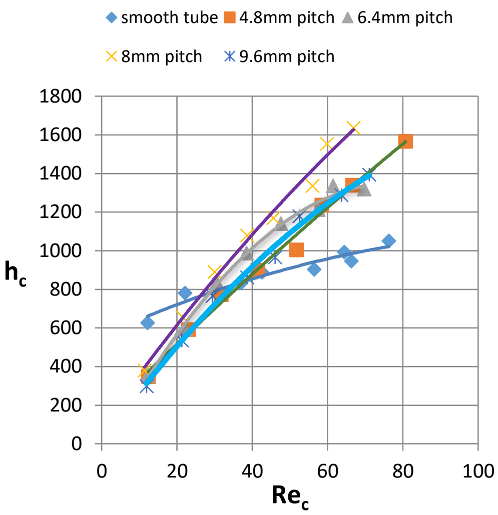


Fig. 10. The relation between Reynolds number and heat transfer coefficient for shell side

5.3. Pressure Drop

For the current range of Reynolds number (100-920), a sensible pressure drop could not be measured experimentally due to the short length of the heat exchanger, so a CFD analysis was use to find the pressure drop inside one tube of the tube bundle. The analysis was done for two cases, the first was for a smooth tube, and the second was for a tube fitted with 9.6 mm pitched helical insert. From Fig. 11 we see that the highest pressure drop in the case of tube inserts was 18 Pa., while in the case of the smooth the highest pressure drop was about 4 Pa. Fig. 12 demonstrates the difference in velocity vectors between the two studied cases.

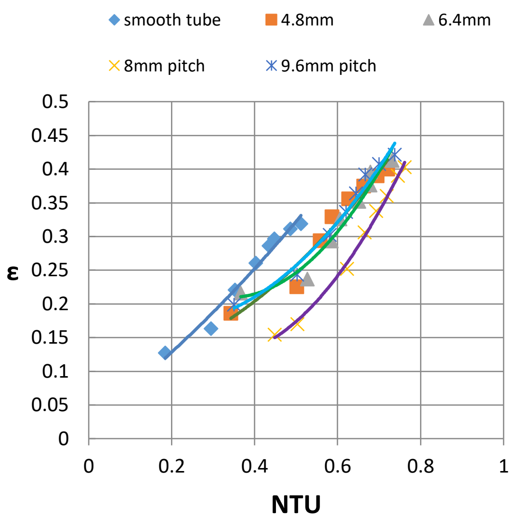


Fig. 11. The relation between number of transfer units and effectiveness.

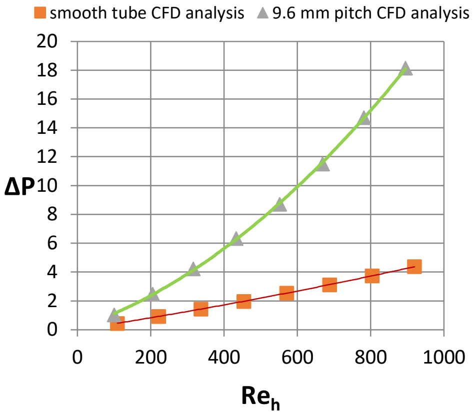


Fig. 12. The relation between Reynolds number and pressure drop.

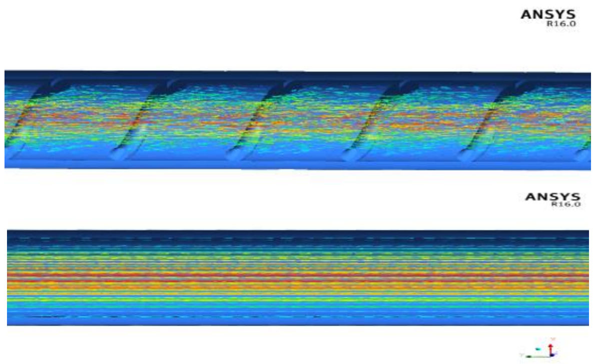


Fig. 13. Velocity vectors for both a tube with insert and a smooth tube.

6. CONCLUSIONS

From the discussion of the experimental and analytical results the fallowing was concluded:

1. An increase in convection heat transfer coefficient for tube
2. bundle side reaching up to 54%, in the case of the tube bundle fitted with 9.6 mm inserts.
3. A rise in the thermal effectiveness of the heat exchanger as a result of the addition of helical inserts inside the tube bundle, and the highest value obtained was in case of 9.6 mm pitched helical inserts.
4. Helical coiled wire inserts caused an increase in pressure drop for tube bundle side of the exchanger by 260%, compared to a smooth tube bundle, this is due the obstructions of flow inside the tube bundle and the swirl effect forced by the inserts which in turn decreases flow velocity and increase pressure.
5. Increasing the helical wire pitch size led to an increase in convection heat transfer coefficient within the study range. The increase in pitch size led to an increase in turbulences along the flow and deformation of the boundary layer, which in turn increase heat transfer rates between hot fluid inside the tube bundle and the tubes inner wall.

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