DESIGN AND ANALYSIS OF DOUBLE-PIPE HEAT EXCHANGER USING BOTH HELICAL AND ROTATING INNER PIPE

by

Tarkan KOCA[a*] ***and Aydin CITLAK***[b]

[a] Mechanical Engineering Department, Inonu University, Malatya, Turkey
[b] Mechanical Engineering Department, Firat University, Elazı, Turkey

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In this study, the effects of rotating straight and helical inner tubes is experimentally discussed to determine heat transfer and pressure losses in rotating tubes and improve heat transfer. The outer tube remains stationary and the inner tube is rotated at different speeds in the work. In the experiments for straight and helical tubes, the flow regime is turbulent. According to the results, Nusselt number, pressure loss, and efficiency of heat exchanger were gauged. In addition, empirical formulas were obtained for each pipe type. It is observed that as the rotation speed of the pipe increases, the heat transfer rate increases. The pipe that provides the best increase in heat transfer is the five helixes tubes. At five helixes tubes; after the number of revolutions per minute exceeds 300, the increase in heat transfer rate has almost halt. At five helixes tubes and at 300 rpm speed when the flow of cold water through the annular gap with the fluid passing through the inner tube is equal, the heat transfer increases by 124.10% compared to straight tube, 23.47% compared to two helixes tubes, 7.92% compared to three helixes tubes, and 1.65% compared to four helixes tubes. Maximum effectiveness was obtained while rotating with 300 rpm in five helixes pipes.

Key words: *double-pipe heat exchanger, helical tubes, rotationally inner pipe, effectiveness, experimental, thermal science*

Introduction

One of the most important and the most common processes of engineering applications is the heat exchange between two or more fluids at different temperatures. Significant progress has been made in recent years, particularly on the application of heat exchangers, to make heat transfer more efficient. In practice, heat exchangers can be found in thermal power plants, chemical industries, heating, air conditioning, cooling installations, vehicles, electronic devices, alternative energy applications, and heat storage. Variations of current and heat transfer characteristics in thermal systems can be determined depending on whether the flow type is external flow or internal flow, whether the flow field is laminar or turbulent, the flow is developing or fully developed in the flow direction, and the thermal boundary conditions in the system. In order to increase the heat transfer surface, grooves are opened on the surface, curved surfaces are formed, rough elements and fins are used. In particular, turbulators, which are widely used, increase the heat transfer because they increase the heat transfer area, settle easily into the flow, break the boundary-layer and re-form, extend the flow path, increase turbulence and create rotational and secondary flow. Today, especially in industrial type boilers and heating boilers,

* Corresponding author, e-mail: tarkan.koca@inonu.edu.tr

the use of turbulators (turbulence generators) to increase heat transfer and thus boiler efficiency has become widespread. For example, by using spiral fins, the current is turned on one side and the heat transfer surface area is increased on the other hand. By placing bent strips, propellers, fins, winding wires in the flow medium the generated rotational flows increase the heat transfer considerably. There are two types of rotational flow [1]. These are rotational flow through the pipe and rotational flow only at the pipe inlet. The amount of heat transfer in the rotational flow along the pipe is high and the pressure losses increase. Because the fluid is constantly rotating and pumping force is formed. For the rotational movement at the pipe inlet, the heat transfer increases slightly, but the pressure losses are not high compared to the heat transfer. Because the fluid is given only the rotation effect at the inlet and is released along the pipe. A plurality of data has been published for different types nested tube heat exchangers. The heat transfer enhancement characteristics of concentric tube heat exchanger could be fully utilized in the industrial applications only if accurate correlations are available for Nusselt number and frictional pressure drop with all the necessary details [2]. While the heat transfer parameters are available in many studies on concentric tube heat exchangers lack of data is a kind of obstacle in the use of concentric tube heat exchanger. There are numerous studies in the literature on turbulators and rotational flows. Avval and Damangir [3] presented an optimization program for calculating the amount of gasket part and the optimum baffle spacing for all available body-to-pipe heat exchangers. In fact, the baffle plates are used to improve the heat transfer performance in body-to-pipe heat exchangers with opposite flow on the tubes. Smithberg and Landis [4] investigated the heat transfer and friction losses by the use of twisted type rotating generators placed in the tube while in forced convection. The researchers found that the velocity field was helical and that the vortex was formed in the nucleus of the flow. Migay and Golubev [5] investigated heat transfer and friction losses in the event that there are rotating flow generators in the pipe and the inlet flow is turbulent. According to the researchers, heat transfer increases in rotational flows, but the coefficient of friction increases more than heat transfer. Chen and Wu [6] experimentally studied a double pipe heat exchanger. The overall heat transfer coefficient of the modified tube is 8.4 times higher. Yang *et al*. [7] performed numerical study to investigate the effect on the overall heat transfer of a helical heat exchanger. Algifri *et al*. [8] investigated heat transfer in a turbulent flow damped in a circular pipe and produced empirical formulas for the determination of heat transfer coefficients in different types of rotary flow generators. Majidi *et al*. [9] experimentally studied attaching a copper fin in the annulus region of a helical heat exchanger. An enhancement in the overall heat transfer coefficient was observed. In order to improve the heat transfer in concentric type heat exchangers. Salem *et al*. [10] experimentally studied a double pipe heat exchanger. They concluded that causes enhancement in the heat transfer rate. Alam and Ghoshdastidar [11] examined the heat transfer along a fin inserted pipe. The flow is smooth and laminar and constant heat flux is applied. In addition, finite difference method was used. Variation of heat conduction coefficient and viscosity with temperature was considered. As a result of the study, it was determined that there was a significant heat transfer improvement. Ho *et al*. [12] have conducted studies to improve the performance of solar air heaters operating according to the multi-pass reverse flow principle. The absorber plate and insulation plate are arranged horizontally and vertically. Open channels are divided into four channels. The study was carried out numerically and it was found that the added plates increased the heat transfer. Jassim [13] implemented an experimental analysis of spiral coil. Vicente *et al*. [14] studied on heat transfer and pressure drop for laminar flow. Dizaji *et al*. [15] and Xu *et al*. [16] performed experimental studies on heat transfer and pressure drop in double- pipe heat exchangers. Kumar *et al*. [17] and Sivaprakash *et al*. [18] researched on effectiveness of the heat exchanger. Hashemian *et al*. [19] a structural modification the double pipe heat exchanger was done. They used conical tubes. Nusselt number in 63%, 54% increase in the rate of heat transfer was performed. Wen *et al*. [20] conducted a numerical investigation and they increased the overall heat transfer coefficient by 82.8-86.1% in the new design. Jassim and Ahmed [21] experimentally studied a double pipe heat exchanger. Hajmaideen and Somu [22] to increase the efficiency of double-pipe heat exchanger was done concave and convex corrugated type of outer and inner tubes.

This study is focused on experimental investigation of heat exchangers with rotational helical and straight inner pipe. In this study, a new nested tube heat exchanger has been manufactured in order to increase heat transfer in heat exchangers and studies have been conducted on it. One of the aims of the article was to experimentally validate Dittus-Boelter correlations [2] by comparison experimentally obtained heat transfer coefficients with appropriate predictions. Experiments were carried out by-passing hot water in the inner pipe and cold water through the outer pipe. The difference from other studies; the outer tube remains stationary and the inner tube is rotated at different speeds. Straight and helical inner pipes were rotated to increase heat transfer. The inner tube was rotated at seven different speeds in each experiment. While the outer pipe is kept constant; the experiments were carried out while the inner tube was not rotating, by rotating at 100, 150, 200, 300, 400, 500, and 750 rpm, respectively. Both straight flow and reverse flow experiments were performed for each pipe. In the experiment, the ratio of the flow rate of hot water to the flow rate of cold water was accepted as the capacity ratio. In this experiment, four different capacity ratios were applied: 0.25, 0.5, 0.75, and 1. The cold water flow rate was taken as six different values and the hot water flow rate was adjusted in all experiments depending on the capacity ratio. Heat transfer and pressure losses were detected in the rotating flat and helical pipes. The flow is turbulent in all experiments. The aim of this study is to maximize the amount of heat transfer.

Materials and methods

The test apparatus manufactured consists of two nested pipes. Both inner and outer pipes were selected as straight pipes. Schematic diagram of experimental set-up is shown in fig. 1.

Information about elements and properties used in the experiment set is shown in tab. 1.

Test-rig calibration and data validation are necessary for results reliability prior to the start with the experimental. The experimental validation, the system was examined with water for turbulent cases at Reynolds number ranging 30.000-70.000. The prepared test set is shown in figs. 2 and 3. Information about the elements used in the experiment set is given below.

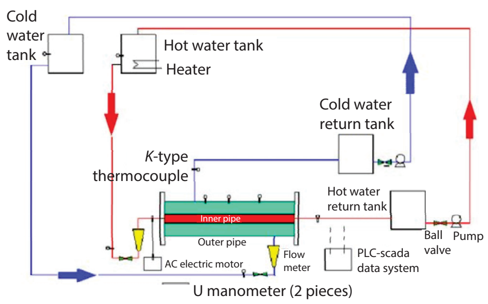

Figure 1. Schematic diagram of experimental set-up

Outer pipe is straight and fixed pipe and made of steel. Its roughness is very low and both ends are fixed with flanges and gaskets. The cold water entering the outer pipe from the bottom takes some heat and goes out from the top. Inner tube is made of copper for better heat transfer. The inner tube used in the test set is shown in fig. 3.

The electric motor used in the experiment set is a three-phase, AC-standard electric motor with star-delta connection, maximum 1000 revolutions per minute, 1.1 kW power, 50 Hz

Table 1. Materials in the experimental set-up

Component		Specification-value
Outer tube	1 piece	Length 650 mm, outer diameter 80 mm, inner diameter 66 mm, steel
Straight inner tube	1 piece	Outer diameter 19 mm, inner diameter 16 mm, copper
Helical inner tubes	4 pieces	Helix step 5.5 mm, helix diameter 55 mm, 2 helixes, 3 helixes, 4 helixes, 5 helixes
Electric engine	1 piece	AC, 1000 rpm, 1.1 kW, 50 Hz
Speed controller	1 piece	0.12-3 kW, 0-1200 rpm
Pumps	2 pieces	0,33 kW, Pump, 35 Lpm, 280 rpm
Rotameters	2 pieces	50 Lpm
Manometer	2 pieces	Hg- U manometer
Scanner	1piece	16 inputs, 16 bit resolution
Power supply	1 piece	220 V
Tanks	4 pieces	200 lt
Heater	4 pieces	Tubular thermostat, 1800 W
Valves	8 pieces	Ball valve
Control panel	1 piece	
Thermal couples	4 pieces	Sheath: Stainless steel, Temperature range: 0 ~ 800 °C, *T*-type, Sensitivity 0.2 °C

Figure 2. Views of the experimental set-up from different perspectives

Figure 3. Different types of inner tubes used in the experiment set

operating frequency and 0.68 power factor. The belt conveys the movement to the rotating pipe by means of the belt pulley assembly. In addition, speed control is made with the variator. Accurate measurement is possible between 0 and 1200 rpm. The speed can be adjusted manually by means of a potentiometer. Two pumps were used in the experiment and their function is to ensure the transfer of the water in the tanks where the return water accumulates to the higher tanks. The rotameter used can measure a maximum flow rate of 50 Lpm. The flow rates are adjusted by means of valves. The scanner is used to display information received at different points up to 16 entries and to transfer information the computer. It has high read accuracy at 16 bit resolution. Thermo elements require special copper-constantan (Cu-Const) cable. The *U*-type mercury manometers were used for pressure measurements. Two manometers are installed at the hot and cold water inlets and outlets. Measurements were made with the help of a ruler. Water tanks are made of 2 mm sheet metal with a volume of 200 L. Two of them were used as cold water supply and return tanks, while the other two were used as hot water supply and return tanks. They are left open to take advantage of the open air pressure. There are four 1800 W tubular heating elements in the hot water supply tank to heat the water. Two of the water tanks are approximately 8 m high. In this way, desired head and flow rates are provided. The flow measurement was done with the help of rotameters. Another important point as much as the results obtained in the experimental study is the accuracy of the measured values. For this purpose, error analysis was performed. The error values of the measurements were determined from the error values and calibration studies recommended by the manufacturer of the measuring instruments. The errors in the flow measurement vary depending on the precise rotameter and the reader used during the experiments. The errors caused by flow measurement during the experiments were determined as ±0.08 Lpm. An electric motor compatible with the variator was used to rotate the inner tube. Speed control is performed with optical tachometer and it is seen that the amount of error is negligible. Pressure sockets are placed at the inlet and outlet of inner pipes and outer pipes and *U*-type mercury manometers are placed between them. For each experiment pressure losses were measured and recorded by manometers. In pressure loss measurement, error caused by pressure gauge accuracy is negligible. Temperature measurement, which is one of the most important measurements of the experiment, was carried out with the help of thermocouples. In addition, heat insulation is made to the outer pipe to minimize heat leakage. Errors caused by temperature measurements at various points in the system during the experiments can be listed as thermocouple couples error, digital thermometer error, error due to fasteners and connection points can be classified as error caused by ambient temperature. The total error in the temperature measurement was determined to be ±0.380 °C. According to the values obtained from the measurement errors, the maximum error value in the amount of heat transfer by using the basic heat equation is 5.06 10^{-4} J/s.

Governing equations

The amount of heat transferred:

$$Q = \dot{m}C_p(T_{\text{out}} - T_{\text{in}}) \tag{1}$$

The log mean temperature difference method is employed to calculate the overall heat transfer coefficient based on outer surface area:

$$U_0 = \frac{Q}{A_0\left[T_{\text{in}} - T_{\text{out}} / \ln\left(\dfrac{T_{\text{in}}}{T_{\text{out}}}\right)\right]} \tag{2}$$

The Nusselt number:

$$\mathrm{Nu} = \frac{hd}{k} \tag{3}$$

The effectiveness of a heat exchanger is defined as the ratio of the heat transferred to maximum heat transfer:

$$\varepsilon = \frac{Q_{\max}}{Q} \tag{4}$$

For the hot and cold fluids, the Reynolds numbers:

$$\mathrm{Re} = \frac{Vd_{\mathrm{H}}}{\nu} \tag{5}$$

Experiment

Five types of internal pipes are used. These are helical and straight pipes. In the experiments, the inner pipes are straight, 2, 3, 4, and 5 helixes pipes. In the experiment set, the outer tube is stationary and fixed. The inner tube is rotated at different speeds. Four different capacity ratios were applied in the experiment. The ratio of the flow rate of hot water to the flow rate of cold water is accepted as the capacity ratio and our ratio is 0.25, 0.5, 0.75, and 1. In all experiments, the flow rate of cold water was kept constant and the flow rate of hot water was changed to ensure capacity rates. Straight and helical inner pipes were connected to the system and experiments were done for these inner pipes. The water filled into the lower tanks was pumped into the 8 m high tanks via electrical pumps. Hot water supply tank and cold water supply tank with resistance inside are open top. After waiting for a while, the water in the hot water supply tank was heated. Water is heated by four thermostat resistance. In all experiments, it has been paid attention the hot water inlet temperature is within certain ranges. For this reason, it was possible for the water temperature to reach the desired range after a little waiting after each experiment. Hot water and cold water flow adjustment valves were opened and water was supplied to the system. In counter-flow experiments, firstly equal flow rates were taken and operations were performed. The flow rates are equal to 20, 22, 24, 26, 28, and 30 Lpm for hot and cold water and thus the capacity ratio is ensured to be 1. Thus, heat exchange in equal amounts of water was investigated. The experiments were carried out by keeping the inner pipe stationary at each selected flow and then doing the same measurements while rotating it at certain rpm. The applied speeds are 0, 100, 150, 200, 300, 500, and 750 rpm. After the experiments with equal selection of the fluid capacities, the experiments were continued with a capacity ratio of 0.75. Measurements were made at seven different speeds for the selected flow rates. The experiments were continued by adjusting the ratio of the hot water flow rate to the cold water flow rate of 0.5. Finally, the capacity ratio of 0.25 was selected and the flow rates were adjusted accordingly. In these experiments conducted for four different capacity ratios for straight and helical pipes, six different hot and cold water flow values were selected for each capacity ratio and the inner pipe was rotated at seven different speeds at each selected flow rate. During these operations, hot water inlet and outlet temperatures and cold water inlet and outlet temperatures were measured and recorded. Outer pipe surface temperatures were determined from three different points. In addition, inlet and outlet pressure difference of hot and cold water pressure differences were measured with U manometers.

Results and discussions

In the experiments, as the speed of rotation increases, heat transfer increases. However, the increase in heat transfer in the helical inner tube decreases exponentially after 300 rpm. 5 helixes pipes, when the flow of cold water through the annular gap with the fluid passing through the inner pipe is equal, the heat transfer increases by 124.10% compared to straight and pressure loss increased by 156.50%. The following tables show the increase and increase percentage of heat transfer, the increase amount and increase percentage of pressure loss, and the increase in efficiency, which occur in measurements between 0 rpm and 750 rpm according to capacity ratios. Table 2 shows the values average heat transfer, for hot water at different capacity rates and at different rotation speeds and at different inner pipes. As can be seen from the tables, as the speed of rotation increases, the increase in heat transfer increases. However, the increase in heat transfer in the helical inner tube decreases exponentially after 300 rpm.

Table 2. Average heat transfer, for hot water at different inner pipes

Capacity ratio	Rotation speed [rpm]	Straight tube	2 helixes tubes	3 helixes tubes	4 helixes tubes	5 helixes tubes
$C = 1$	0	11874.8	19556.54	26580.09	29061.67	33656.26
	100	12319.14	19643.43	27024.22	29940.91	34807.68
	150	12459.77	19725.82	27553.92	30450.39	35381.79
	200	13397.42	20376.53	28467.31	31225.34	35585.51
	300	13844.89	20989.58	29588.34	32185.39	37005.31
	500	14842.91	21521.07	30047.09	31772	36121.89
	750	14937.01	20738.68	29572.4	31634.2	35851.17
$C = 0.75$	0	11909.81	15790.63	21056.19	23846.51	29497.23
	100	12293.75	16396.4	21628.16	24385.66	30809.13
	150	12488.7	16797.15	22152.37	24942.85	31274.32
	200	12787.92	17331.24	22307.7	25237.33	31764.23
	300	13013.1	17622.2	22839.38	25987.24	32644.68
	500	13357.46	17692.05	23206.99	25398.53	32064.93
	750	13454.87	17609.72	22431.87	25218.18	31847.29
$C = 0.5$	0	10279.26	11638.62	15966.47	19639.78	23386.23
	100	10618.97	12115.05	18120.04	19889.94	23538.59
	150	11093.61	12544.51	19171.8	20168.71	23513.6
	200	11416.11	12841.05	19647.1	20656.74	23752.74
	300	12106.19	13723.99	20092.41	21406.77	24468.95
	500	12310.42	13658.21	19871.86	21002.96	23769.2
	750	12408.4	13625.03	19936.37	20493.01	23680.41
$C = 0.25$	0	6388.403	7634.307	11415.49	11768.64	12222.04
	100	6702.608	7940.659	11489.64	12135.46	12406.46
	150	7097.723	8002.693	11651.58	12341.27	12579.19
	200	7290.376	8151.915	11855.43	12475.72	12623.01
	300	7661.93	8708.236	12115.2	12874.3	13145.63
	500	7846.55	8676.655	12146.85	12762.19	12906.78
	750	7910.022	8466.393	11526.16	12482.98	12556.76

Table 3 shows the values the amount and percentage of heat transfer of five helixes tubes compared to other tubes, for hot water at 300 rpm rotation speeds. As can be seen from the tables, as the capacity rates increases, the increase in heat transfer increases.

Table 3. Increment amounts and percentage in heat transfer for hot water from 0-300 rpm in 5 helixes tubes

	Compared to straight tube		Compared to 2 helixes		Compared to 3 helixes		Compared to 4 helixes	
Capacity ratio	Increase [Js^{-1}]	[%]	Increase [Js^{-1}]	[%]	Increase [Js^{-1}]	[%]	Increase [Js^{-1}]	[%]
C = 1.00	23160.43	167.29	16015.74	76.30	7416.97	25.07	4819.93	14.98
C = 0.75	19631.58	150.86	15022.48	85.25	9805.29	42.93	6657.44	25.62
C = 0.50	12362.75	102.12	10744.96	78.29	4376.54	21.78	3062.18	14.30
C = 0.25	5483.70	71.57	4437.39	50.96	1030.43	8.51	271.33	2.11

Table 4 shows the values the amount and percentage of pressure loss of five helixes tubes compared to other tubes, for hot water at 300 rpm rotation speeds. As can be seen from the tables, as the capacity rates increases, the increase in pressure loss increases.

Table 4. Increment amounts and percentage in pressure loss for hot water from 0-300 rpm in 5 helixes tubes

	Compared to straight tube		Compared to 2 helixes		Compared to 3 helixes		Compared to 4 helixes	
Capacity ratio	Increase [kPa]	[%]	Increase [kPa]	[%]	Increase (kPa)	[%]	Increase [kPa]	[%]
C = 1.00	7.34	109.63	4.00	39.91	3.60	34.54	3.09	28.25
C = 0.75	3.51	59.62	2.27	31.78	1.36	16.85	1.09	13.10
C = 0.50	0.67	15.00	0.27	5.50	0.56	12.20	1.09	27.07
C = 0.25	1.65	49.66	1.47	42.04	1.51	43.87	0.71	16.75

Figure 4 shows the values Nusselt number-Reynolds number average heat transfer, for hot water at different capacity rates (1, 0.75, 0.5, and 0.25) and at different rotation speeds and at straight inner tubes. A comparison of Nusselt number evaluated from the experimental for straight tubes with the theoretical value obtained from Dittus-Boelter correlations is shown in fig. 4. The Nusselt number obtained from the experimental results are larger than that of theoretical values obtained from Dittus-Boelter correlation.

Figure 5 shows the values Nusselt number-Reynolds number average heat transfer, for hot water at different capacity rates (1, 0.75, 0.5, and 0.25) and at different rotation speeds and at 2 helixes tubes. A comparison of Nusselt number evaluated from the experimental for 2 helixes tubes with the theoretical value obtained from Dittus-Boelter correlations is shown in fig. 5

Figure 6 shows the values Nusselt number-Reynolds number average heat transfer, for hot water at different capacity rates (1, 0.75, 0.5, and 0.25) and at different rotation speeds and at 3 helixes tubes. Further A comparison of Nusselt number for 3 helixes tubes with the theoretical value obtained from Dittus-Boelter correlation is shown in fig. 6.

Figure 7 shows the values Nusselt number-Reynolds number average heat transfer, for hot water at different capacity rates (1, 0.75, 0.5, and 0.25) and at different rotation speeds and at 4 helixes tubes. Also Dittus-Boelter correlation graphic is shown in fig. 7.

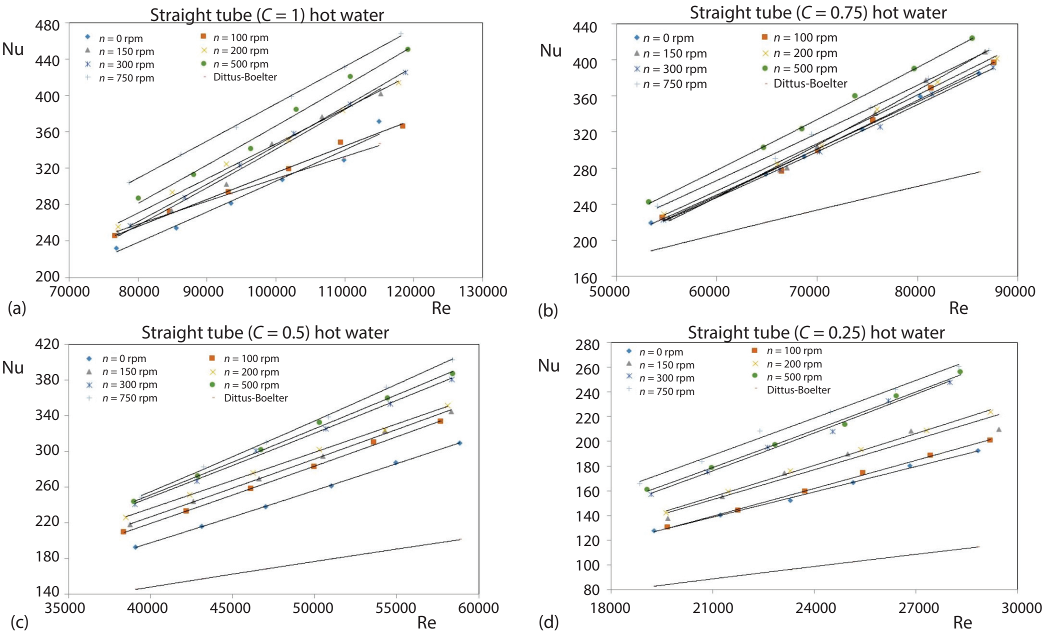

Figure 4. The Nu-Re change for *C* in straight tube for hot water; (a) *C* = 1, (b) *C* = 0.75, (c) *C* = 0.5, and (d) *C* = 0.25

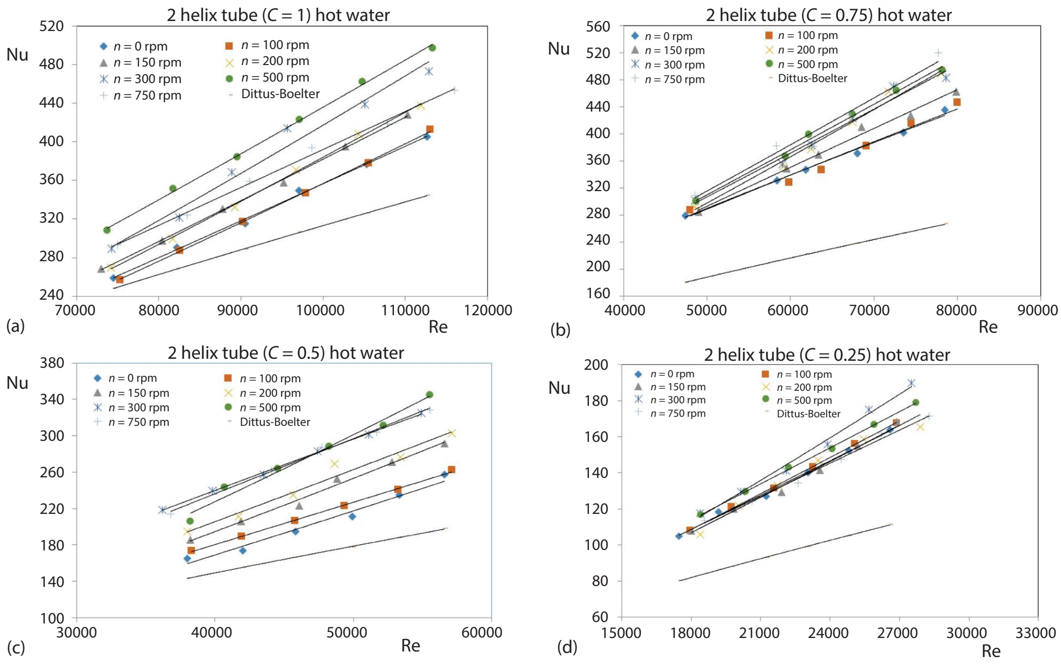

Figure 5. The Nu-Re change for *C* in 2 helixes tube for hot water; (a) *C* = 1, (b) *C* = 0.75, (c) *C* = 0.5, and (d) *C* = 0.25

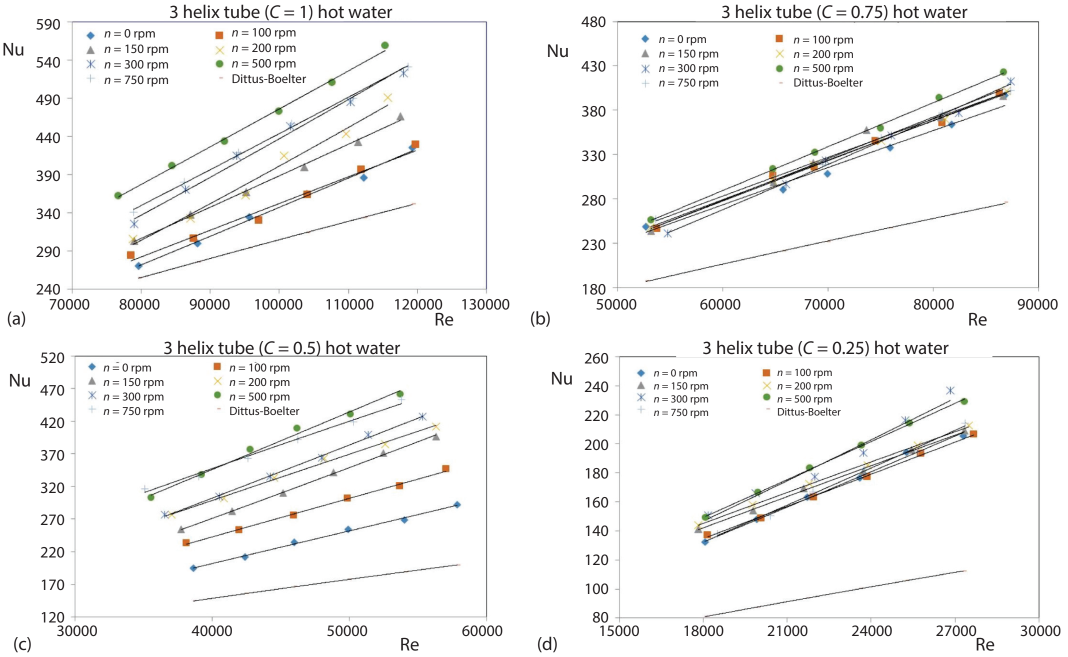

Figure 6. The Nu-Re change for *C* in 3 helixes tube for hot water; (a) *C* = 1, (b) *C* =0.75, (c) *C* =0.5, and (d) *C* =0.25

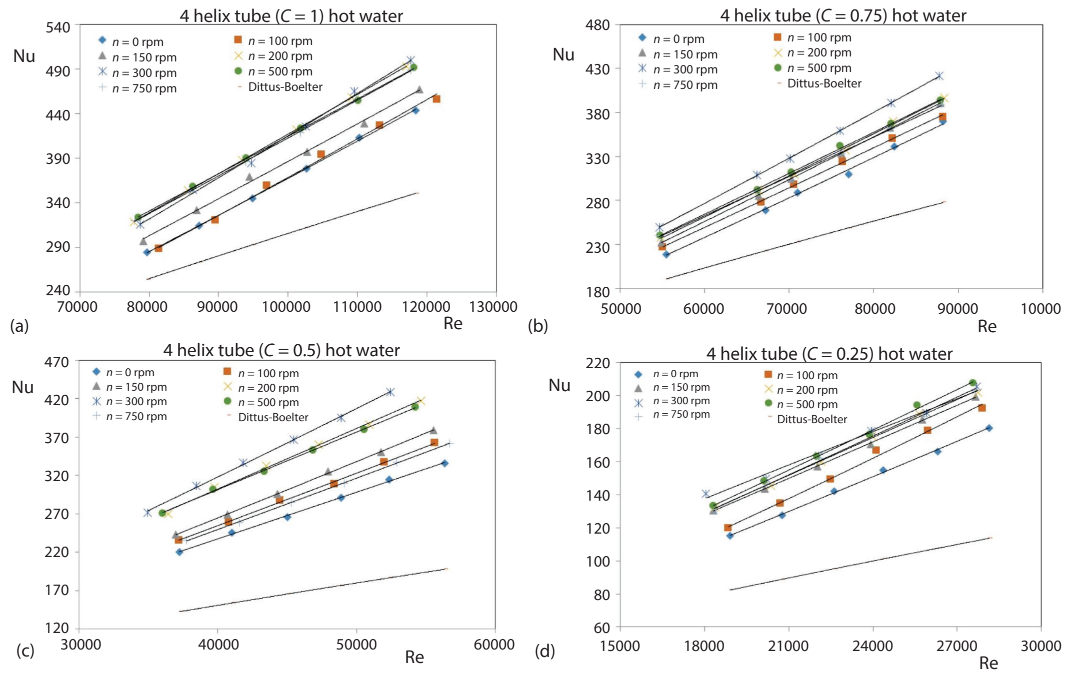

Figure 7. The Nu-Re change for *C* in 4 helixes tube for hot water; (a) *C* = 1, (b) *C* = 0.75, (c) *C* = 0.5, and (d) *C* = 0.25

Figure 8 shows the values Nusselt number-Reynolds number average heat transfer, for hot water at different capacity rates (1, 0.75, 0.5, and 0.25) and at different rotation speeds and at 5 helixes tubes.

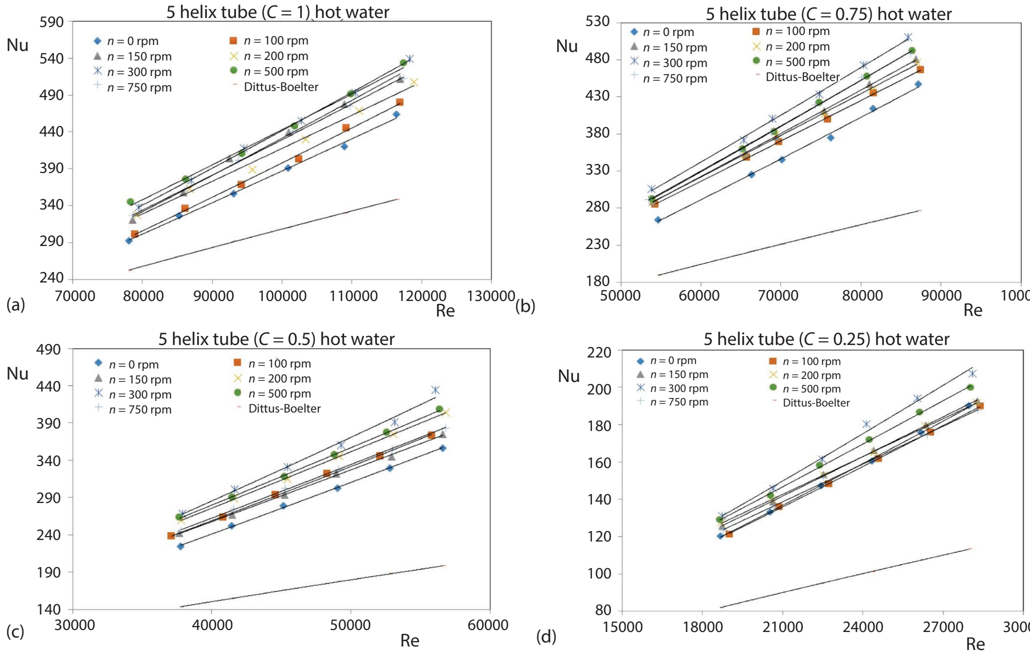

Figure 8. The Nu-Re change for *C* in 5 helixes tube for hot water; (a) *C* = 1, (b) *C* = 0.75, (c) *C* = 0.5, and (d) *C* = 0.25

The graphics obtained from the experimental results for straight and helixes pipes are very close to the correlation Dittus-Boelter. All graphics shows that as Reynolds number increases the Nusselt number increases both for plain tube in tube and for helixes inner tube heat exchanger.

Obtained empirical formulas

Depending on Reynolds number, Prandtl number and rotation speed: empirical formulas of Nusselt number and efficiency are given. Empirical formulas were obtained for the inner five helixes tubes with varying rotational speeds starting from 0 rpm. However, the empirical formula for five helixes tube tube at variable rotational speeds is valid up to 300 rpm.

Nusselt number

The Nusselt number obtained from the experimental results was very close to that of theoretically obtained Nusselt number. The empirical formulas obtained from the experimental results are close to the correlation given by Dittus and Boelter. According to Dittus and Boelter, Nusselt number is given in eq. (3) for $l/d \geq 10$ in the range of Re > 10000, 0.7 < Pr < 160 in the form of fully developed turbulent flow in the tube. According to Dittus and Boelter, n number is $n = 0.4$ for heated fluid and $n = 0.3$ for cooled fluid.

$$\mathrm{Nu} = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{n} \tag{6}$$

Dittus-Boelter correlations are recommended to be used in calculation of heat transfer coefficient in heat exchanger with both helical and rotational inner tube. It is possible to use for channels with different cross-sections using equivalent diameter. Heat transfer in passages between tubes is enhanced by strong local turbulence resulting from helix. Empirical formulas of Nusselt number and efficiency are given.

While $C = 1$:

$$n = 0 \text{ rpm}, \ \mathrm{Nu}_h = 0.0008459 \times \mathrm{Re}^{1.1145} \times \mathrm{Pr}^{0.3} \times \left(\frac{h}{l}\right)^{0.04} \tag{7}$$

$$0 < n < 300 \text{ rpm}, \ \mathrm{Nu}_h = 0.00596 \times \mathrm{Re}^{1.164} \times \mathrm{Pr}^{0.3} \left(\frac{n}{n_{\max}}\right)^{0.25} \times \left(\frac{h}{l}\right)^{0.04} \tag{8}$$

While $C = 0.75$:

$$n = 0 \text{ rpm}, \ \mathrm{Nu}_h = 0.001101 \times \mathrm{Re}^{1.1231} \times \mathrm{Pr}^{0.3} \times \left(\frac{h}{l}\right)^{0.04} \tag{9}$$

$$0 < n < 300 \text{ rpm}, \ \mathrm{Nu}_h = 0.0016 \times \mathrm{Re}^{1.101} \times \mathrm{Pr}^{0.3} \left(\frac{n}{n_{\max}}\right)^{0.25} \times \left(\frac{h}{l}\right)^{0.04} \tag{10}$$

While $C = 0.5$:

$$n = 0 \text{ rpm}, \ \mathrm{Nu}_h = 0.001256 \times \mathrm{Re}^{1.1323} \times \mathrm{Pr}^{0.3} \times \left(\frac{h}{l}\right)^{0.04} \tag{11}$$

$$0 < n < 300 \text{ rpm}, \ \mathrm{Nu}_h = 0.000923 \times \mathrm{Re}^{1.1743} \times \mathrm{Pr}^{0.3} \left(\frac{n}{n_{\max}}\right)^{0.25} \times \left(\frac{h}{l}\right)^{0.04} \tag{12}$$

While $C = 0.25$:

$$n = 0 \text{ rpm}, \ \mathrm{Nu}_h = 0.0014199 \times \mathrm{Re}^{1.1343} \times \mathrm{Pr}^{0.3} \times \left(\frac{h}{l}\right)^{0.04} \tag{13}$$

$$0 < n < 300 \text{ rpm}, \ \mathrm{Nu}_h = 0.001086 \times \mathrm{Re}^{1.172} \times \mathrm{Pr}^{0.3} \left(\frac{n}{n_{\max}}\right)^{0.25} \times \left(\frac{h}{l}\right)^{0.04} \tag{14}$$

Efficiency

Empirical formulas of efficiency are given below.

While $C = 1$:

$$n = 0 \text{ rpm}, \ \varepsilon = 0.05845 \times \mathrm{Re}^{0.1158} \times \mathrm{Pr}^{0.3} \times \left(\frac{h}{l}\right)^{0.04} \tag{15}$$

$$0 < n < 300 \text{ rpm}, \ \varepsilon = 0.04875 \times \mathrm{Re}^{0.1424} \times \mathrm{Pr}^{0.3} \left(\frac{n}{n_{\max}}\right)^{0.25} \times \left(\frac{h}{l}\right)^{0.04} \tag{16}$$

While C = 0.75:

$$n = 0 \text{ rpm}, \ \varepsilon = 0.08251 \times \mathrm{Re}^{0.0945} \times \mathrm{Pr}^{0.3} \times \left(\frac{h}{l}\right)^{0.04} \tag{17}$$

$$0 < n < 300 \text{ rpm}, \ \varepsilon = 0.07926 \times \text{Re}^{0.1052} \times \text{Pr}^{0.3} \left(\frac{n}{n_{\max}}\right)^{0.25} \times \left(\frac{h}{l}\right)^{0.04} \tag{18}$$

While C = 0.5:

$$n = 0 \text{ rpm}, \ \varepsilon = 0.06097 \times \text{Re}^{0.113} \times \text{Pr}^{0.3} \times \left(\frac{h}{l}\right)^{0.04} \tag{19}$$

$$0 < n < 300 \text{ rpm}, \ \varepsilon = 0.06366 \times \text{Re}^{0.1293} \times \text{Pr}^{0.3} \left(\frac{n}{n_{\max}}\right)^{0.25} \times \left(\frac{h}{l}\right)^{0.04} \tag{20}$$

While $C = 0.25$:

$$n = 0 \text{ rpm}, \ \varepsilon = 0.04761 \times \text{Re}^{0.1457} \times \text{Pr}^{0.3} \times \left(\frac{h}{l}\right)^{0.04} \tag{21}$$

$$0 < n < 300 \text{ rpm}, \ \varepsilon = 0.06598 \times \text{Re}^{0.1219} \times \text{Pr}^{0.3} \left(\frac{n}{n_{\max}}\right)^{0.25} \times \left(\frac{h}{l}\right)^{0.04} \tag{22}$$

Conclusions

The results obtained in this experimental study were presented in the results and discussion section as graphs, tables and empirical formulas. Based on these obtained values, the results and recommendations are as follows.

- As the number of rotations for the all inner tubes increases, heat transfer increases.
- After the number of revolutions per minute exceeds 500, the increase in heat transfer has almost come to a halt in straight tube.
- There is very little rise in heat transfer at 750 rpm.
- After the number of revolutions per minute exceeds 300, the increase in heat transfer has almost come to a halt in five helixes tubes. In other words, the maximum rotation speed should be selected as 300 rpm. At 300 rpm, when the flow of cold water through the annular gap with the fluid passing through the inner tube is equal, the heat transfer increases by 29.9% compared to 0 rpm. In contrast, the pressure loss increased by 9.47%.
- It is observed that as the helix number of the tube increases, the heat transfer rate increases. The tube that provides the best increase in heat transfer is the five helixes tubes. In 5 helixes tubes, it has been observed that pressure drop is high while increasing heat transfer.
- At 5 helixes tubes, when the flow of cold water through the annular gap with the fluid passing through the inner tube is equal, the heat transfer increases by 124.10% compared to straight tube, 23.47% compared to 2 helixes tubes, 7.92% compared to 3 helixes tubes and 1.65% compared to 4 helixes tubes.
- At 5 helixes tubes, when the flow of cold water through the annular gap with the fluid passing through the inner tube is rates 0.5, the heat transfer increases by 111.59% compared to straight tube, 46.27% compared to 2 helixes tubes, 43.68% compared to 3 helixes tubes, 23.77% compared to 4 helixes tubes. In contrast, the pressure loss increased.
- When the flow rate of hot water is reduced according to certain capacity ratios, the heat transfer rises but the pressure losses are increasing at the same rate.
- Different capacity ratios affect heat transfer and pressure losses. The ideal result is found that the inner tube rotates at 300 rpm at all different capacity ratios. When these values are

taken into consideration, an internal 5 helixes-tube heat exchanger to be manufactured will have an optimal rotation speed of 300 rpm.

The most important result achieved in this experiment is when the inner tube is rotating at high rpms, the increase in heat transfer is very small because of the turbulent flow inside the inner tube causes flow instability. Thus an unstable flow occurs. Due to the rotation effect, the internal flow tends to change its characteristics from the turbulent flow to the laminar flow or to flow instability in the tube, heat transfer cannot rise with high rates. However, as the boundary-layer in the annular space is distorted by the rotating effect and reoccurs, heat transfer increases.

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Nomenclature

A – area, [m^2]
C_p – specific heat, [Jkg^{-1}K^{-}1]
d – tube diameter, [m]
d_H – hidraulic diameter
h – convective heat transfer coefficient, [Wm^{-2}K^{-1}]
k – thermal conductivity, [Wm^{-1}K^{-1}]
l – pipe length, [m]
$\dot{m}$ – mass-flow, [kgs^{-1}]
Nu – Nusselt number
Pr – Prandtl number
Q – heat transfer rate, [W]
Re – Reynolds number
T – temperature, [K]
U – overall heat transfer coefficient, [Wm^{-2}K^{-1}]
V – velocity

Greek Symbols

ε – effectiveness
ν – kinematic viscosity, [m^{2}s^{-1}]

Subscripts

out – outer
in – inner
max – maximum

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