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Experimental Heat Transfer Enhancement in a Tubular Heat Exchanger

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ABSTRACT

Heat exchange process is one of the fundamental subjects in chemical and food process industries. This research investigates an experimental study to enhance the heat transfer process in tubular (water-to-water) heat exchanger. Two typical sets of experiments have been conducted using a horizontal double pipe (tubular) heat exchanger. The first set was conducted for the existing apparatus as installed at different fluids flow patterns for parallel and counter flow arrangement. The second experiment had been done after inserting a coiled copper wire in the inner pipe that containing a fluid of hot water. The obtained results from both experiment sets were compared. It is found that the tube insert has tremendously affected the performance of the exchanger and showed a noticeable enhancement. The tube inserts forces the flow to be more turbulent that indicated by the increased value of Reynold's number (*Re*) to reach *3364* in some flow patterns. Consequently, the overall heat transfer coefficient has also enhanced in some cases from 517.31 to 981.93 $W/m^2.^oC$ for parallel flow and from 750.18 to 1105 $W/m^2.^oC$ for countercurrent flow. Furthermore, one of the exchanger effectiveness reading has increased from 25% to 43.9%.

***Key words*:** Tubular heat exchanger, heat transfer enhancement, tube inserts.

Nomenclature

$A_{c,i}$, A_h :	Exchanger's internal tube and hydraulic cross-sectional area, respectively [m^2].
A_i, A_o :	Exchanger's inner tube internal and external heat transfer surface area, respectively [m^2].
Cp_i, Cp_o :	Fluid heat capacity for exchanger's internal and external tube, respectively [kJ/kg.K].
d_i, d_o, d_l, D_i:	Inner, outer diameter for exchanger's internal tube, logarithmic diameter and exchanger's external tube inner diameter, respectively [m].
h_i, h_o :	Fluid heat transfer coefficient for exchanger's internal and external tube, respectively [W/m^2.K].
k_i, k_o :	Fluid thermal conductivity for exchanger's internal and external tube, respectively [W/m.K].
k_w, l, x_w:	Exchanger's internal tube thermal conductivity [W/m.K], exchanger's tube length and exchanger's internal tube wall thickness [m], respectively.
$Nu_{l,i}$, $Nu_{l,o}$, $Nu_{t,i}$, $Nu_{t,o}$:	Nusselt number for internal and external tube's fluid of laminar and turbulent flow, respectively [-].
Pr_i, Pr_o :	Prandtl number for internal and external tube's fluid, respectively [-].
Q_i, Q_o, $Q_{max,i}$:	Heat transfer rate for inner hot fluid side and outer cold fluid side of the exchanger, respectively and maximum heat transfer rate from the inner hot fluid side of the exchanger (W).
Re_i, Re_o :	Reynold's number for internal and external tube's fluid, respectively [-].
T_{h1},T_{h2}, T_{c1}, T_{c2}, ΔT_M:	Inlet and outlet temperature for inner hot side fluid and outer cold side fluid of the exchanger and logarithmic meat temperature difference, respectively [^{o}C].
U_i, U_o :	Overall heat transfer coefficient for inner hot fluid side and outer cold fluid side of the exchanger, respectively [W/m^2.K].
v_i, v_o :	Fluid velocity for exchanger's internal and external tube, respectively [m/s].

μ_i, μ_o :	Fluid viscosity for exchanger's internal and external tube, respectively [kg/m.s].
ρ_i, ρ_o :	Fluid density for exchanger's internal and external tube, respectively [kg/m^3].
$\dot{m}_i, \dot{m}_o$:	Exchanger's internal and external tube's fluid mass flow rate, respectively [kg/s].
$\dot{V}_i, \dot{V}_o$:	Exchanger's internal and external tube's volumetric flow rate, respectively [L/hr].

INTRODUCTION

In many process industries, it is necessary to exchange heat energy for heating, cooling, vaporization or condensation in various fluid streams. Heat exchangers are common equipment used to achieve the heat transfer process [1].

Heat exchanger is a heat-transfer device that used for the transfer of internal thermal energy between two or more fluids available at different temperatures [2].

Heat transfer processes of some fluids usually take place in laminar or transitional flow range, in which heat transfer rate is low. Heat exchangers that work under these flow conditions are usually candidates to undergo enhancement techniques which are effective to improve the thermo-hydraulic behavior in the tube side in single-phase laminar flow where the insert devices stand out. The dominate literature usually mentions some of tube inserts such as; twisted tape, extended surface devices, mesh inserts…*etc*. The main advantage of these types is that they allow an easy installation in an existing smooth tube heat exchanger [3, 4].

Heat Transfer Enhancement and Mechanism

Heat transfer enhancement is a process of improving the performance of a heat transfer system in heat exchangers by increasing the heat transfer coefficient. In the last decades, heat transfer enhancement has been applied widely to heat transfer application such as: refrigeration, automotive, process industry, chemical industry, etc. Different attempts have been made to reduce the size and costs of heat exchangers. Also heat augmentation techniques play a vital role in laminar flow, since the heat transfer coefficient is generally low in plain tube [5, 6].

Heat transfer enhancement makes heat exchangers to operate at small fluid velocity while achieving higher heat transfer coefficient. Moreover, there will be a reduction in pressure drop and hence less operating cost. Such kind of advantages have made heat transfer enhancement technology attractive in heat exchanger application [7].

Regarding the mechanism of heat transfer enhancement, tube inserts could create a combination of some conditions which are favorable for increasing friction flow. For instance, interrupting the fluid boundary layers and consequently increasing the degree of flow turbulence by generating rotational and secondary flow. Moreover, tube inserts will increase the effective heat transfer area in case of excellent contact between the inserted element and tube wall [8].

Several types of exchanger tube inserts were featured and cited in the literature such as; twisted tape inserts, extended surface inserts, mesh inserts, etc. [2]. The selection of tube inserts depends on performance and cost [9].

Wire coil inserts are currently used in several applications such as oil cooling devices, preheaters or fire boilers. Wire coil inserts have shown several advantages compared to other enhancement techniques. For instance, coiled wire inserts are economically cheap, easy to install and remove, preserving the original plain of tube mechanical strength and they are possible to be installing in an existing tube heat exchanger [10].

Based on such advantages: (the current experimental research aims at enhancing the efficiency of heat transfer in a tubular double pipe heat exchanger using coiled metallic wire in the hot fluid tube side).

The tubular double pipe heat exchanger is a simple heat exchange device in which fluids are pumped through two concentric pipes. The fluids can flow either in co-current or countercurrent direction where one fluid is heated while the other is cooled [11]. The existing double pipe heat exchanger considered for this study is just a two concentric pipes installed horizontally. Such exchangers are commonly used in applications involving relatively low flow rates and high temperatures or pressures, for which they are well suited. Moreover, they characterized by their low installation cost and easy maintenance beside operation flexibility [12].

State of the Art for Tube Inserts in Heat Transfer

As sated previously, the thermal performance of heat exchangers can be improved by heat transfer enhancement techniques. The most commonly techniques cited in the literature are either twisted-tape or coiled wire inserts for different application of heat transfer processes [13].

Using twisted tube inserts, *Kang et al* [14] have developed the flooding correlation for a fluted tube. They have examined the effect of twisted tape insert in addition to the angle of inclination on flooding.

Al-Fahed et al [15] have compared the pressure drop and heat transfer coefficients obtained from a plain tube, micro-fin tube, and plain tube with twisted-tape insert.

On the other hand, *Liao* and *Xin* [16] have carried out the study of heat transfer and friction characteristics for four different fluids flowing inside four tubes with three-dimensional internal extended surfaces and twisted tape of copper in segmented form and continuous form.

Using a simple mathematical model, two theoretical studies have been introduced by *Zimparov* for predicting heat transfer coefficient and friction factor in a spirally corrugated tube with a twisted tape insert. The studies were stated for a fully developed turbulent flow. Later, the obtained results from the model were conformed and compared with further experimental study [17, 18].

Simthberg and *Landis* have also studied the friction and forced convection heat transfer characteristic in tubes fitted with twisted tape as swirl generators. They have finally presented a correlation for predicting Nusselt number and friction factor [19]. Moreover, a complete study reporting the prediction of fully developed flow in a tube containing a twisted tape insert has been presented by *Date* [20].

Sivashanmugam and *Suresh* have also studied the laminar heat transfer and friction factor characteristic in a circular tube fitted with full-length helical screw- tapes with different twist ratio, including the increasing and decreasing order of twist ratio sets [21].

Based on energy analysis, *Prasad* and *Shen* have studied the enhancement of heat transfer using different wire-coil inserts but only in turbulent flow region [22].

Using forced convection for the condensation of cooling refrigerant inside horizontal tubes, *Agrawal et al* have experimentally studied the heat transfer enhancement using coiled wire. Three different wire diameter and three different coil pitches were used in full length of the condenser [23]. *Kim et al* have visualized the flow pattern, void fraction and slug rise velocity on counter-current two-phase flow in a vertical round tube with wire-coil inserts [24]. Further study has been conducted by *Rahai* and *Wong* where they experimentally examined the turbulent jets from round tubes with coil inserts [25].

Naphon and *Sriromrulu* presented the single phase heat transfer and pressure drop in the micro-fin tubes with coiled wire insert. The results obtained from micro-fin tubes with coiled wire insert are compared with those from the smooth tube and micro-fin tube without coiled wire insert [13].

In 2005, *Ozceyhan* has numerically studied the conjugate heat transfer and thermal stress in tube with wire coil insert under a uniform wall heat flux [26]. Further, he experimentally studied the effect of coil wire insert on heat transfer enhancement and pressure drop of the horizontal concentric tube for decreasing Reynolds number shown in the study of *Naphon* [27]. *Ravigururajan* and *Bergles* have presented general correlations for friction factor and heat transfer coefficient for single-phase turbulent flow in internally augmented tubes [28].

MATERIALS AND METHODS

The materials and method used for this study is mainly concerned with the heat transfer enhancement techniques by means of tube insert in the double pipe heat exchanger.

The materials to be considered are mainly metallic pure copper wire and tab water flowing in the exchanger. Hot water is flowing in the inner tube as hot fluid and cold at room temperature flowing in the outer tube side as a cold fluid.

The wire is made of pure copper similar to the exchanger inner tube material. The wire thickness is 0.001m and length of 12.00m. The wire is roughly twisted forming a coiled shape of 1.00m length and diameter a bit less than the exchanger inner tube inner diameter (d_i). The coiled wire is shown in Fig. (1).

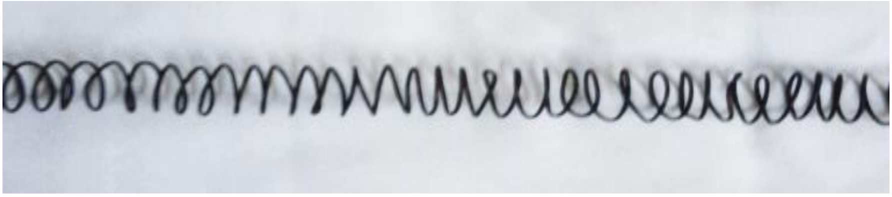


Fig. (1) Coiled Copper Wire

Double Pipe Heat Exchanger Apparatus

The tubular double pipe heat exchanger used for this study is a water-to-water heat exchanger bench, model (HEP-200E), Japan, Ogawa Seiki CO.LTD. It is located in the Unit operation laboratory, Dept. of Chemical Engineering and Chemical Technology, Faculty of Engineering and Technology, University of Gezira, Sudan. A detailed specification of the bench containing double pipe exchanger with all the required accessories could be pointed out as follows:

- Width: 1500mm, depth: 550mm, height: 1700mm and net weight 250kg.
- Electricity requirements: 3 phases' line with a voltage range of (200-220) Volt and frequency range of (50-60) Hz.
- Maximum water supply and drain: 700L/hr.
- Hot water head tank with flow meter of up to 200L/hr and level indicator.
- Cold water head tank with flow meter of up to500L/hr and level indicator.
- Inlet and outlet thermocouples for both hot and cold water sides
- Control panel for switches and temperature indicators.
- Electrically immersed heater: 5kw, 3kw.
- Water circulating pumps.

The in-site picture of heat exchanger apparatus is shown in Fig. (2).

The inner tube material is copper of thermal conductivity k_w = 373.88W/m.K. The geometrical configurations of the two centric tubes are detailed in Table (1) below.

Table (1): Geometrical for inner and outer tube of exchanger.

d_i (m)	d_o (m)	x_w (m)	D_i (m)	$d_h = D_i - d_o$ (m)	l (m)
0.017	0.019	0.001	0.0276	0.0086	1

Fig. (2): Bench of double pipe heat exchanger, model (HEP-200E);
(a) Left side view, (b) Front view.

Methods of Experimental Work and Calculation

Different flow arrangements have been set for parallel and counter flow direction based on the flow type of laminar or turbulent that defined by setting fluid flow rate using the flowmeter for either hot or cold water. Laminar and turbulent flow types were assured using Reynolds number which must be greater than 4000 for turbulent flow and less than 2100 for laminar flow.

Reynold's number is determined based on the fluid properties and exchanger geometrical configuration as in equations (1) and (2) for inner and outer tubes, respectively.

$$Re_i = \frac{d_i . v_i . \rho_i}{\mu_i} \quad (1)$$

$$Re_o = \frac{d_h . v_o . \rho_o}{\mu_o} \quad (2)$$

The fluid velocity v_i and v_o for both inner and outer fluids are calculated as in equations (3) and (4), respectively. They are calculated based on volumetric flow and cross-sectional area for inner tube and hydraulic cross-sectional area for outer tube.

$$v_i = \frac{\dot{V}_i}{A_{c,i}} \quad (3)$$

$$v_o = \frac{\dot{V}_o}{A_h} \quad (4)$$

The cross-sectional areas $A_{c,i}$ and A_h are calculated as in the following equations:

$$A_{c,i} = \frac{\pi}{4}(d_i^2) \quad (5)$$

$$A_h = \frac{\pi}{4}(D_i^2 - d_o^2) \quad (6)$$

The different flow arrangements used for this study are shown in Table (2) below.

Table (2): Fluids flow arrangement for parallel or countercurrent flow

No. of experiment	**Flowrate *(L/h)***		**Type of flow**	
	Hot	cold	Hot	cold
1	30	150	Laminar	Laminar
2	30	450	Laminar	Turbulent
3	90	150	Turbulent	Laminar
4	90	450	Turbulent	Turbulent

For the sake of comparison, two identical experimental sets have been conducted using the aforementioned heat exchanger. The first one conducted before inserting the coiled wire in the exchanger inner tube side, while the other was conducted after applying the tube insert. Pictures captured for the exchanger inner tube while inserting the coiled wire are shown in Fig. (3) below.

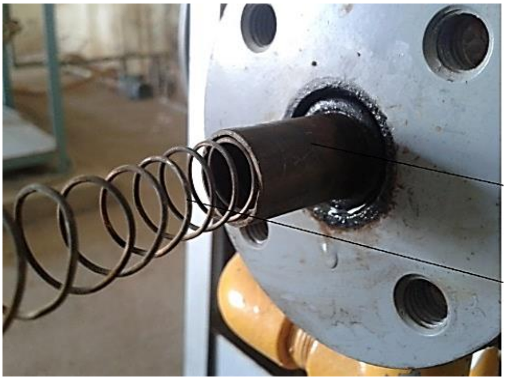
	
Inner tube before wire insertion	Inner tube while inserting coiled wire

Fig. (3): Coiled wire insertion in exchanger inner tube.

The readings of experimental temperature for hot and cold fluids in the exchanger are tabulated in Table (3) for parallel and countercurrent flow pattern, before and after tube insert.

Table (3): Experimental data for inlet and outlet temperature ***before and after*** coiled wire Insertion for parallel and countercurrent flow pattern

Exp. No.	***Flow rate (L/hr)***		***Flow Pattern***	T_{hi} ***(ºC)***		T_{ho} ***(ºC)***		T_{ci} ***(ºC)***		T_{co} ***(ºC)***	
	Hot	*cold*		*before*	*after*	*before*	*after*	*before*	*after*	*before*	*after*
1	30	150	Parallel	56.5	60	44	45	32.5	32	35	35
2	30	450		55	60	41	40	32	31.7	33	33
3	90	150		56	59	51	50	33	33.1	36	38.5
4	90	450		57	59	49	48	32.5	31.8	34	34
1	30	150	Co	57	60	44	45	33	33	35.5	36

2	30	450	56	60	40	40	32.1	32.2	33.1	33.5
3	90	150	56	59	48	49	33	33	36.6	39
4	90	450	56.5	59	49	48	32.5	32.8	34	35

The overall experimental works are briefly explained in the flowchart shown in Fig (4) below.

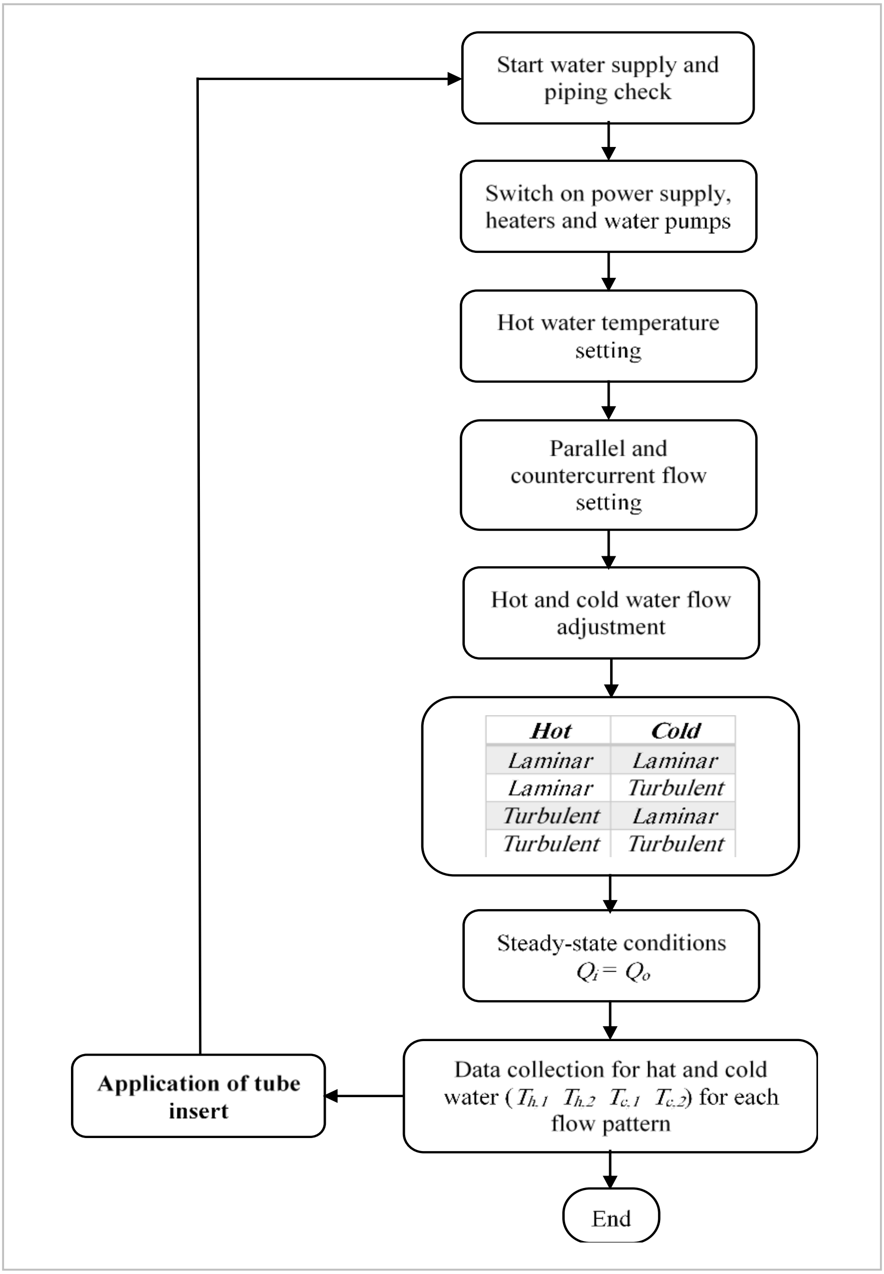


Fig (4): Experimental work flowchart.

The physical properties of hot and cold fluids are required to calculate the non-dimensional numbers like *Nu*, *Pr*, and *Re* and hence the heat transfer coefficients for both sides of the exchanger. Such physical properties were determined according to the average temperature of the fluids (T_{in} and T_{out}) for each experiment set. Table (4), shows the required physical properties of hot and cold fluids for parallel and countercurrent flow patterns.

Table (4): Physical properties of hot and cold fluids in the exchanger

Exp. No.	Flow Pattern	C_P (kJ/kg.K)		ρ (kg/m^3)		$\mu \times 10^4$ (kg/m.s)		k (W/m.K)	
		before	after	before	after	before	after	before	after
Hot fluid(inner tube)									
1	Parallel	4.1902	4.1913	987.3	986.3	5.74	5.49	0.639	0.642
2	Parallel	4.1891	4.1901	988.2	987.4	5.99	5.77	0.636	0.639
3	Parallel	4.1903	4.1913	987.2	986.3	5.71	5.49	0.639	0.642
4	Parallel	4.1896	4.1901	987.8	987.4	5.88	5.77	0.638	0.639
1	Counter	4.1918	4.1923	985.9	985.5	5.38	5.27	0.643	0.644
2	Counter	4.1916	4.1918	986.8	985.9	5.44	5.38	0.642	0.643
3	Counter	4.1916	4.1921	986.1	985.7	5.44	5.33	0.642	0.644
4	Counter	4.1915	4.1918	986.2	985.9	5.46	5.38	0.642	0.643
Cold fluid(outer tube)									
1	Parallel	4.182	4.182	994.1	994.2	7.56	7.59	0.619	0.619
2	Parallel	4.181	4.181	994.6	994.7	7.70	7.71	0.618	0.618
3	Parallel	4.182	4.182	993.9	993.8	7.50	7.48	0.620	0.620
4	Parallel	4.181	4.181	994.6	994.5	7.69	7.66	0.619	0.619
1	Counter	4.182	4.183	993.8	993.3	7.48	7.33	0.620	0.622
2	Counter	4.182	4.181	994.3	994.5	7.61	7.65	0.619	0.618
3	Counter	4.182	4.183	993.7	993.2	7.41	7.31	0.620	0.622
4	Counter	4.182	4.182	994.3	994.1	7.41	7.54	0.619	0.620

The film heat transfer coefficients for both sides of the exchanger, *i.e.*, h_i and h_o, are calculated according to equations (7) and (8) as described below:

$$h_i = \frac{Nu_i . k_i}{d_i} \tag{7}$$

$$h_o = \frac{Nu_o . k_o}{d_o} \tag{8}$$

The dimensionless *Nusselt* number (*Nu*) depends mainly on *Reynold* (*Re*) and *Prandtl* (*Pr*) numbers. For inner and outer tubes of the exchanger, *Nu* is calculated for laminar and turbulent flow using the following equations:

$$Nu_{l,i} = 0.023 Re_i^{0.8} . Pr_i^{0.3} \quad (9)$$
$$Nu_{t,i} = 0.86 Re_i^{0.431} . Pr_i^{0.3} \quad (10)$$
$$Nu_{l,o} = 0.023 Re_o^{0.8} . Pr_0^{0.3} \quad (11)$$
$$Nu_{t,o} = 0.86 Re_o^{0.431} . Pr_o^{0.3} \quad (12)$$

Prandtl number (*Pr*) which is also dimensionless depends on the fluid physical properties, can be determined for both side of the exchanger using equation (13) and (14) below:

$$Pr_i = \frac{Cp_i . \mu_i}{k_i} \quad (13)$$
$$Pr_o = \frac{Cp_o . \mu_o}{k_o} \quad (14)$$

The overall heat transfer coefficients for inner and outer fluids of the exchanger, can be found theoretically based on exchanger's geometrical configuration, thermal conductivity of inner tube wall and film heat transfer coefficients of both sides as in the following equations:

$$U_i = \frac{1}{\frac{d_o}{d_i . h_i} + \frac{x_w . d_o}{k_w . d_l} + \frac{1}{h_o}} \quad (15)$$
$$U_o = \frac{1}{\frac{d_i}{d_o . h_o} + \frac{x_w . d_i}{k_w . d_l} + \frac{1}{h_i}} \quad (16)$$

The logarithmic mean diameter d_l is determined according to equation (17), where h_i and h_o are found using equations (7) and (8) above.

$$d_l = \frac{d_o - d_i}{\ln(\frac{d_o}{d_i})} \quad (17)$$

For the sake of comparison before and after tube insert, the overall heat transfer coefficients (U_i) and (U_o) can also be found based on the direct temperature readings from the experiments using equations (18) and (19) as shown below.

$$U_i = \frac{Q_i}{A_i . \Delta T_M} \quad (18)$$
$$U_o = \frac{Q_o}{A_o . \Delta T_M} \quad (19)$$

The lost and gained amount of heat transfer rate (Q_i) and (Q_o) are calculated using equations (20) and (21), respectively.

$$Q_i = \dot{m}_i . Cp_i (T_{h,1} - T_{h,2}) \tag{20}$$

$$Q_o = \dot{m}_o . Cp_o (T_{c,2} - T_{c,1}) \tag{21}$$

The mass flow rate for hot and cold fluids are determined from volumetric flow and density of such fluids as in equations (22) and (23) below.

$$\dot{m}_i = \dot{V}_i . \rho_i \tag{22}$$

$$\dot{m}_o = \dot{V}_o . \rho_o \tag{23}$$

The heat transfer surface area for both sides of exchanger's inner tube (A_i) and (A_o) are simply found as follows:

$$A_i = \pi . d_i . l \tag{24}$$

$$A_o = \pi . d_o . l \tag{25}$$

Logarithmic mean temperature difference (ΔT_M) for parallel and countercurrent flow are calculated according to equations (26) and (27), respectively.

$$\Delta T_M = \frac{(T_{h,1} - T_{c,1}) - (T_{h,2} - T_{c,2})}{\ln\left[\frac{(T_{h,1} - T_{c,1})}{(T_{h,2} - T_{c,2})}\right]} \tag{26}$$

$$\Delta T_M = \frac{(T_{h,1} - T_{c,2}) - (T_{h,2} - T_{c,1})}{\ln\left[\frac{(T_{h,1} - T_{c,2})}{(T_{h,2} - T_{c,1})}\right]} \tag{27}$$

The effectiveness of the exchanger is calculated before and after tube insert based on the ratio between the actual heat (Q_i) and the maximum expected amount of heat to be released from the hot fluid ($Q_{max,i}$). Equations (28) and (29) expressed the exchanger effectiveness and maximum expected amount of heat to be released, respectively.

$$\varepsilon = \frac{Q_i}{Q_{max,i}} \tag{28}$$

$$Q_{max,i} = \dot{m}_i . Cp_i (T_{h,1} - T_{c,2}) \tag{29}$$

RESULTS AND ANALYSIS

As mentioned earlier, inserting the coiled wire in the internal tube of the exchanger is expected to make the hot fluid more turbulent and hence

increasing the heat transfer coefficient. The idea is proved by the obtained results for Reynold number, film and overall heat transfer coefficient as well as for exchanger effectiveness. Table (5) summarizes the overall results before and after tube insert at different flow patterns (parallel/counter) or flow types (laminar/turbulent) and also according to flow arrangement.
For the sake of comparison before and after tube insert, the obtained results are also shown graphically where there are two plots in each figure, i.e. (a) and (b) for laminar and turbulent flow in hot tube side, respectively.

Reynold's Number for Fluid Flow in Exchanger Hot Tube Side

The insertion of coiled wire in the hot tube side of the exchanger has disturbed the hot fluid flow where the effect has investigated by the degree Reynold's number. Fig. (5) and (6) demonstrate Reynold's number (*Re*) before and after tube insertion for parallel and counter flow in the hot side. It is shown that Reynold's number has increased in all experiment sets after inserting the coiled wire. In Fig. (6), Reynold's number has obviously increased for turbulent flow from 3394 to 3463 and from 3382 to 3431.

Film Heat Transfer Coefficients (h_i and h_o)

The expected effect of tube insert on the internal film heat transfer coefficient (h_i) is very clear as shown in Figures (7) and (8). The film coefficient for the internal tube has increased dramatically for co-current and countercurrent flow. The effect is more noticeable in the countercurrent/turbulent flow. However, the external heat tansfer coefficient (h_o) has not been influenced since the tube insertion was applied only in the inner tube side, as shown in Fig. (9) and (10).

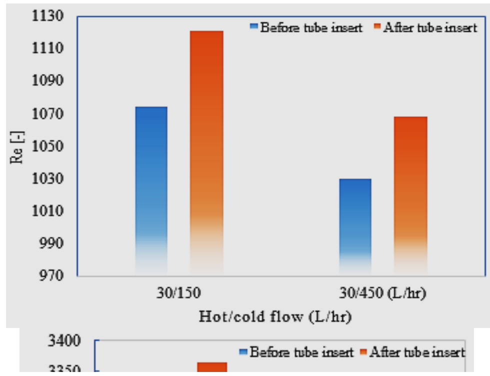
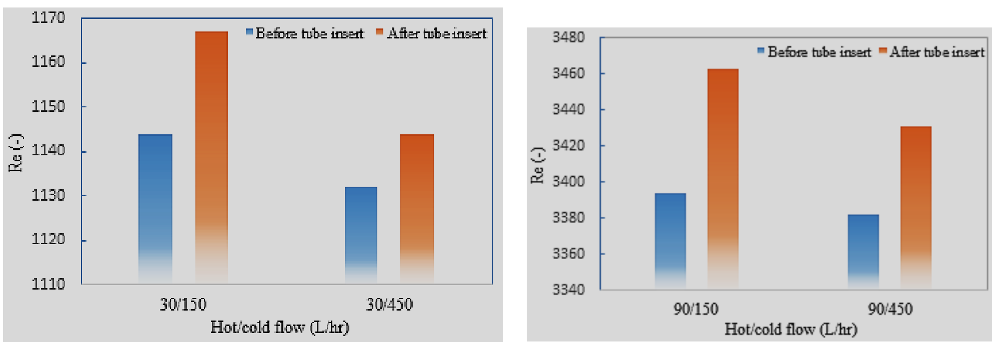


Fig. (5) Reynold's number in the hot side for **parallel flow** before and after tube insert.

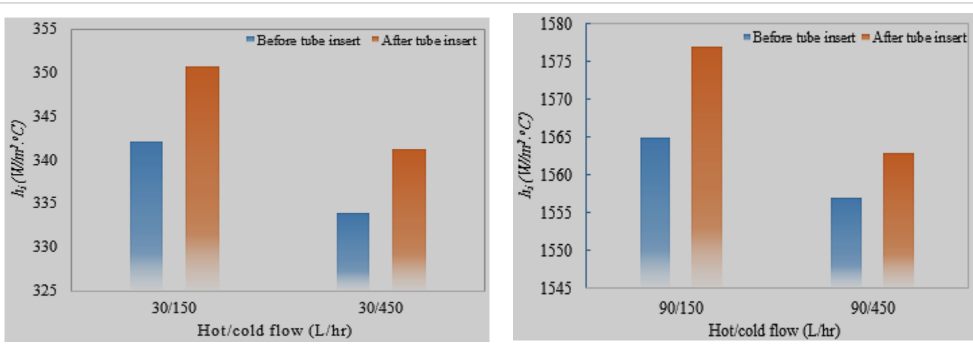
(a) Laminar hot tube flow. (b) Turbulent hot tube flow.



(a) (b)

Fig. (6) Reynold's number in the hot side for **counter flow** before and after tube insert.

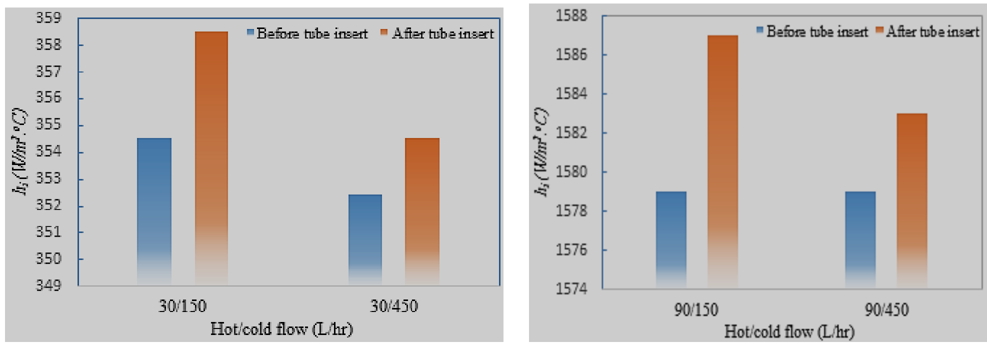
(a) Laminar hot tube flow. (b) Turbulent hot tube flow.



(a) (b)

Fig. (7) Film heat transfer coefficient of the hot side fluid before and after tube insert for **parallel flow**

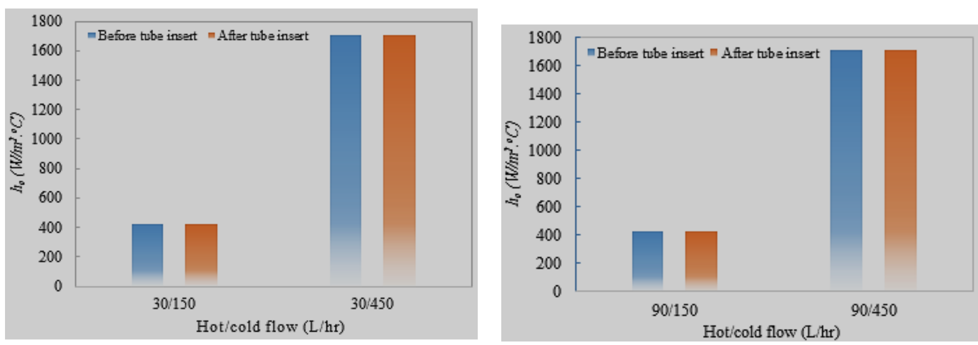
(a) Laminar hot tube flow. (b) Turbulent hot tube flow.



(a) (b)

Fig. (8) Film heat transfer coefficient of the hot side fluid before and after tube insert for **counter flow**

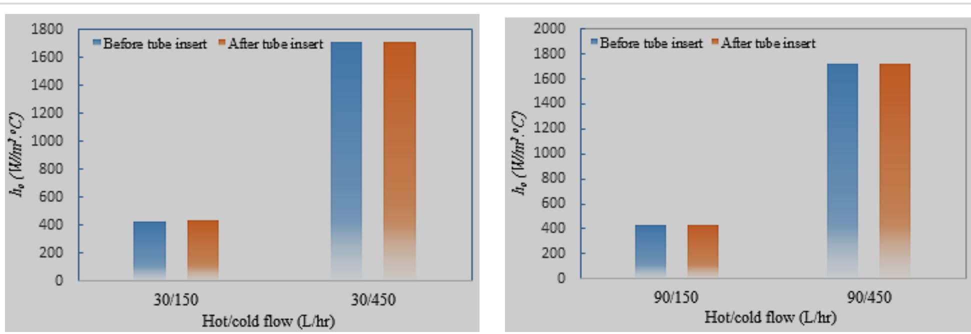
(a) Laminar hot tube flow. (b) Turbulent hot tube flow.



(a) (b)

Fig. (9) Film heat transfer coefficient of the cold fluid before and after tube insert for **parallel flow**

(a) Laminar hot tube flow. (b) Turbulent hot tube flow



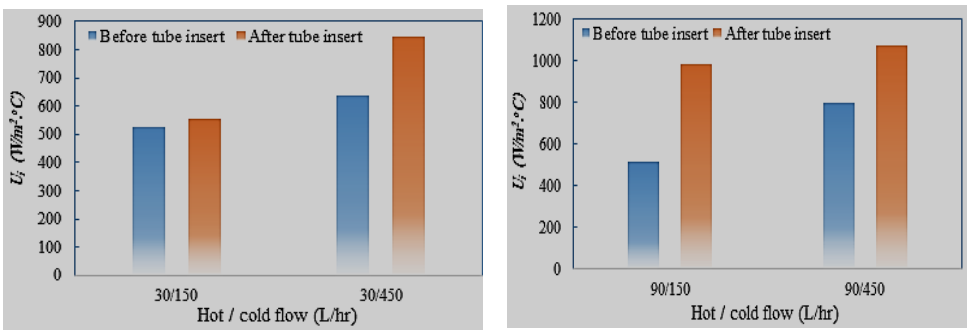
(a) (b)

Fig. (10) Film heat transfer coefficient of the cold fluid before and after tube insert for **counter flow**

(a) Laminar hot tube flow. (b) Turbulent hot tube flow

Overall Heat Transfer Coefficients (U_i and U_o)

. However the results Both overall heat transfer coefficients are directly affected by h_i and h_o shown here are obtained based on the direct effect of wire insertion. In other words, U_i and U_o are calculated based on the difference between fluids inlet and outlet temperatures by means of logarithmic temperature difference (ΔT_M) and heat quantity (Q) as stated earlier in equations (18) and (19). Accordingly, U_i has shown to increase at the different nominated flow patterns for laminar and turbulent flows after applying the tube insert as shown in Fig. (11) and (12). In particular, Fig. (11, b) indicates a noticeable effect of tube insert for parallel flow pattern and turbulent hot fluid flow.



(a) (b)

Fig. (11) Overall heat transfer coefficient of the hot fluid before and after tube insert for **parallel flow**

(a) Laminar hot tube flow. (b) Turbulent hot tube flow

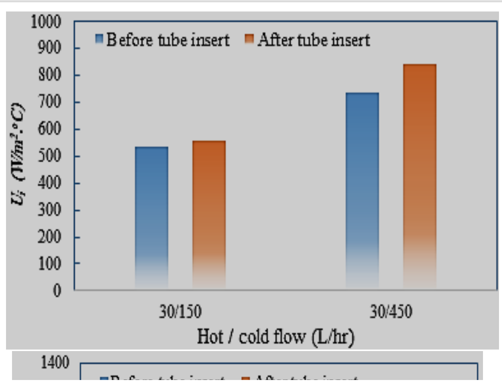
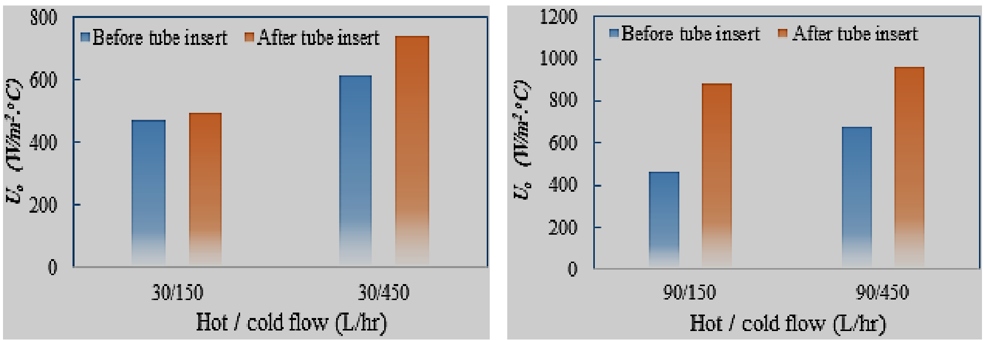


Fig. (12) Overall heat transfer coefficient of the hot fluid before and after tube insert for **Counter flow**

(a) Laminar hot tube flow. (b) Turbulent hot tube flow

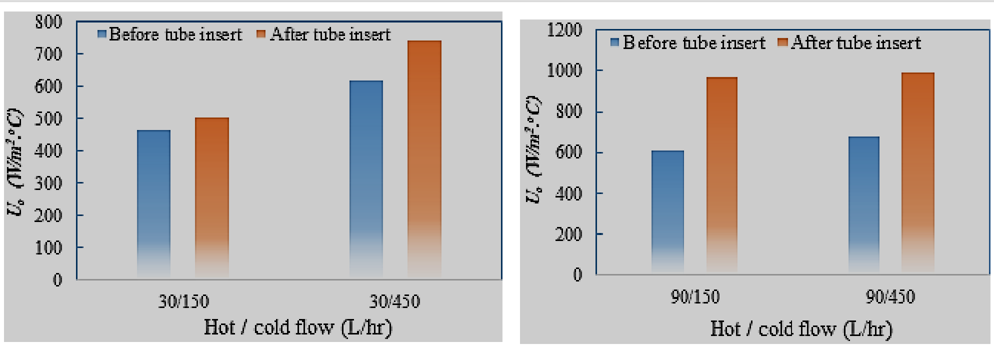
On the other hand, Fig. (13) and (14) illustrate the effect of tube insert on the cold side overall heat transfer coefficient, U_o. Repeatedly, the same effect is shown for U_o although the wire insertion was applied only for the hot side. This could be justified by the effect of h_i on both U_i and U_o. Fig. (13, b and 14, b), the overall coefficient U_o is shown to increase drastically for turbulent cold side flow in either parallel or countercurrent flow pattern.



(a) (b)

Fig. (13) Overall heat transfer coefficient of the cold fluid before and after tube insert for **parallel flow**

(a) Laminar hot tube flow. (b) Turbulent hot tube flow



(a) (b)

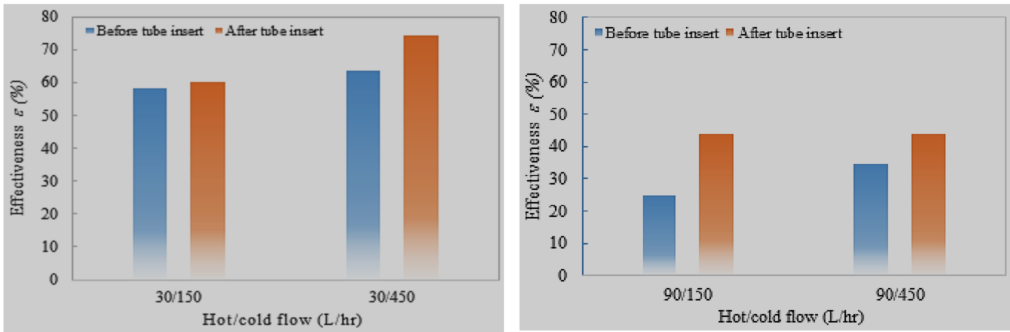
Fig. (14) Overall heat transfer coefficient of the cold fluid before and after tube insert for **counter flow**

(a) Laminar hot tube flow. (b) Turbulent hot tube flow

Heat Exchanger Effectiveness (ε)

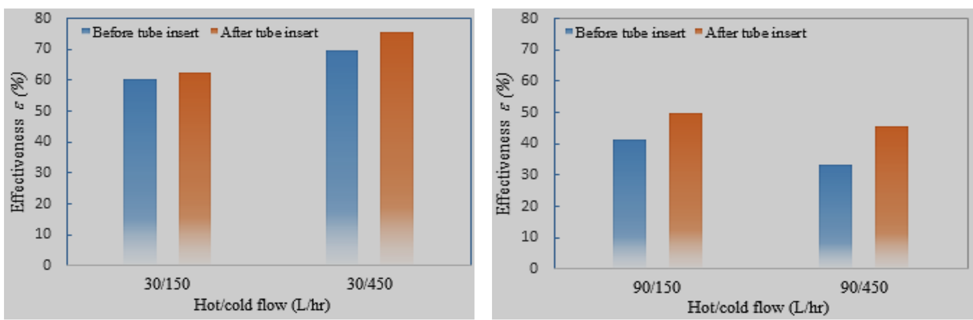
Heat exchanger effectiveness (ε) has been positively influenced by inserting the coiled wire in the internal tube for both laminar and turbulent flow types at parallel and countercurrent flow patterns as indicated in Fig. (15) and (16).

From the figures, it is clear that the efficiency has increased after tube insert application which indicates that the insertion of the coiled wire has enhanced the heat transfer process in the exchanger. The highest effect on the effectiveness is observed clearly in 90/150 *L/hr* flow arrangement (turbulent hot/laminar cold) for parallel flow, where the effectiveness has increased from 25% to 43.9% as shown in Fig. (15, b).



(a) (b)

Fig. (15) Heat exchanger effectiveness before and after tube insert for **parallel flow**
(a) Laminar hot tube flow. (b) Turbulent hot tube flow



(a) (b)

Fig. (16) Heat exchanger effectiveness before and after tube insert for **counter flow**
Laminar hot tube flow. (b) Turbulent hot tube flow

CONCLUSION

Insertion of coiled wire in the inner (hot) tube side of the double pipe heat exchanger has clearly increased the performance of the apparatus. The heat transfer process has significantly enhanced as indicated by the obtained results. Accordingly, the coiled wire insertion can also be applied in the outer (cold) side together with the hot side of the exchanger to obtain the maximum possible performance that can be adopted by industry for energy conservation.

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تعزيز انتقال الحرارة تجريبياً للمبادل الحراري الأنبوبي

ملخص الدراسة

عملية التبادل الحراري هي إحدى العمليات الأساسية في المصانع الكيميائية ومصانع الأغذية. في هذا البحث تمت دراسة تعزيز انتقال الحرارة للمبادل الحراري الانبوبي التجريبي (ماء – ماء). حيث تمت دراسة مجموعتين من التجارب النموذجية باستخدام مبادل حراري أنبوي أُفقي مزدوج. تم إجراء تجارب المجوعة الأولى بمعدلات تدفق مختلفة و نوعي سريان متوازي ومتعاكس. أُجريت تجارب المجموعة الثانية بعد إدخال سلك نحاس حلزوني في الانبوبي الداخلي الذي يحتوي علي الماء الساخن. تمت مقارنه النتائج التي تم الحصول عليها من مجموعتي التجارب، ووُجد أن إدراج سلك النحاس الحلزوني في الأنبوب الداخلي قد أثر كثيراً علي أداء المبادل الحراري وأظهر تحسناً ملحوظاً حيث كان التدفق اكثر اضطراباً، وذلك بزيادة رقم رينولد(*Re*) ليصل الي 3364 لبعض التدفقات. وبالتالي فإن معامل انتقال الحرارة الإجمالي (*U*) قد ازداد في بعض الحالات من 517.31 واط/متر^2.م^o الي 981.93 واط/متر^2.م^o للسريان المتوازي ومن 750.81 واط/متر^2.^{o}م الي 1105واط/متر^2.^{o}م للسريان المتعاكس. بالإضافة الي ذلك فان فعالية المبادل الحراري قد سجلت ارتفاعاً في إحدى القراءات حيث ارتفعت من 25% الي 43.9%.

كلمات مفتاحية: مبادل حراري، تعزيز انتقال الحرارة، معززات انتقال الحرارة في الأنابيب.