Experimental Investigation to Enhance the Heat Transfer in Heat Exchanger by Made Groove in Outer Surface of the Inner Tube

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Abstract:

This study deals with experimental work implementing to recover the benefit by changing the shape of the tube in heat exchanger (HE) and improving the heat transfer using water as the working fluid. The experimental tests were carried out in build and design a complete test system for counter flow heat exchanger. The tested system consisting of a copper tube with (1m) length (17.05) mm inner diameter (19.05) mm outer diameter, fixed concentric within the outer tube was made of a material PVC. With an “inner diameter (ID) (43 mm) and outer diameter (OD) (50 mm)” isolated from the outside by using insulating material to reduce heat loss. The modify tube was manufacture containing transverse grooves with the depth equivalent to the half thickness of the copper tube. The distance between the grooves on the outer surface of the copper tube is take as a ratio between (0.5, 1) from the outer tube diameter. The laboratory experiment use the hot water at a flow rate ranging between (1-5) LPM, passes in the inner copper tube. As well as the cooling water with the mass flow rate ranging between (3-7) LPM. Three temperatures were the hot fluid are the adoption of (40, 50 and 60) ºC and (25) ºC the cold fluid. The experiment result showed that the improvement for temperature difference ranging from (14.94 % to 43.2 %) for both corrugated tubes with respect to smooth tube.

دراسة عملية لتحسين انتقال الحرارة في مبادل حراري عن طريق عمل اخاديد في الوجه الخارجي للأنبوب الداخلي

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الخلاصة:

تتناول هذه الدراسة العملية تحسين معامل انتقال الحرارة في المبادل الحراري نوع قشرة وانبوب وذلك عن طريق تغيير شكل الانبوب الداخلي وذلك باستخدام الماء كسائل للعمل. اجريت الاختبارات التجريبية عن طيق تصميم وبناء نظام اختبار كامل لمبادل حراري متعاكس الجريان. تتكون منظومة الاختبار من انبوب نحاسي ذو قطر خارجي (17.05) مم و قطره الداخلي (19.05) مم، مثبت مركزيا داخل قشرة مصنوعة من مادة البي في سي

قطره الداخلي (43) مم وقطره الخارجي (50) مم معزولة عن الخارج باستخدام المواد العازلة للحد من فقدان الحرارة. تم صنع انبوب معدل وذلك عن طريق عمل اخاديد عرضية بعمق يعادل نصف سمك الانبوب النحاسي. المسافة بين الاخاديد اخذت كنسبة بين قطر الانبوب الخارجي الى البعد بين مركز اخدود الى الاخر وهذه النسب هي (1،0.5). تم عمل التجارب المختبرية باستخدام الماء الساخن الذي يمر داخل الانبوب النحاسي الداخلي وبمعدل تدفق يتراوح بين (1- 5) لتر\ثانية. و معدل تدفق الماء البارد الذي يمر خلال القشرة بمعدل (3- 7) لتر\ثانية. كانت درجات الحرارة للماء الحار خلال العمل هي (40، 50 و 60) م° و (25) م° للماء البارد. وأظهرت النتائج التجريبية أن التحسن في الفرق في درجة الحرارة يتراوح بين (14.94٪ إلى 43.2٪) لكل من الأنابيب المعدلة مقارنتا بالأنبوب الاملس.

Introduction

"A heat exchanger is a device that is used to transfer heat between two or more fluids that are at different temperatures, it's a component that allows the transfer of heat from one fluid (liquid or gas) to another fluid. Heat exchangers are essential elements in a wide range of systems, including the human body, automobiles, computers, power plants, and comfort heating/cooling equipment" [1].

Generally one of wide utilized sort of heat exchanger is the shell-and-tube heat exchanger, the enhancement of heat transfer rate which is the principle target of this work Increasing heat exchanger performance usually means transferring more duty or operating the exchanger at a closer temperature approach. This can be accomplished without a dramatic increase in surface area [2]. The present work deals with finding the effect of change the outer surface geometry of tube on heat transfer in the heat exchanger with different corrugated ratio and comparing those results with smooth tube.

B. Chandra sekhar et al., 2014, [3] The fundamental basis for this statistic is shell and tube technology is a cost effective, proven solution for a wide variety of heat transfer requirements. There are limitations associated with the technology which include inefficient usage of shell side pressure drop, dead or low flow zones around the baffles where fouling and corrosion can occur, and flow induced tube vibration, which can ultimately result in equipment failure.

K. Boomsma et al., 2003, [4] study the experiments performed with water that were scaled to estimate the heat exchangers' performance, when used with a 50% water-ethylene glycol solution. Open-cell metal foams with an average cell diameter of 2.3 mm were manufactured from [6101-T6] aluminum alloy and were compressed and fashioned into compact heat exchangers measuring [40 mm × 40 mm × 2 mm high], possessing a surface area to volume ratio on the order of [10,000 m2/m3]. They were placed into a forced convection arrangement using water as the coolant. Heat fluxes measured from the heater-foam interface ranged up to [688 kW m−2], which corresponded to Nu up to 134 when calculated based on the heater-foam interface area of [1600 mm2] and a Darcian coolant flow velocity of approximately 1.4 m/s. The compressed open-cell aluminum foam heat exchangers generated thermal resistances that were 2-3 times lower than the best.

N. Sahiti et al., 2005, [5] demonstrated that the proposed technique for exchange upgrades is a great deal more powerful than present strategies, since it brings about an expansion in exchange region (like fins) furthermore an increment in the heat exchange coefficient. Significant upgrades were exhibited in that work by utilizing little round and hollow sticks on surfaces of heat exchangers.

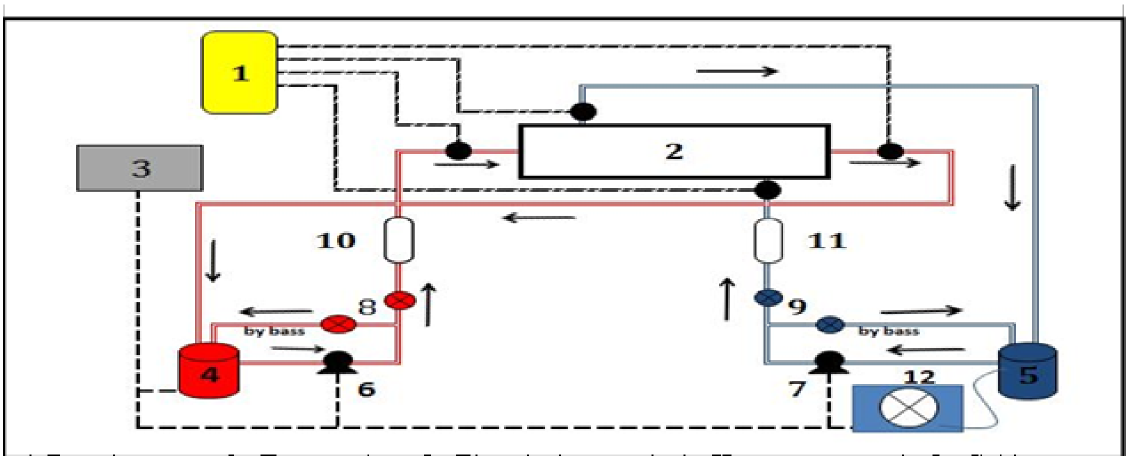
Wang et al., **2009,** [6] presented experimental work done on STHE by blocking the gaps between the baffle and the shell by using the sealers. Results of closing the small gaps in the shell side increase the (h) by [18.2-25.5%], the overall heat transfer coefficient (U) increase by [15.6-19.7%] and the exergy efficiency increased by [12.9-14.1%]. Increase losses in pressure [44.6-48.8%], but rise in pump power wanted can by neglect compared with the rise of heat flux (Q) .

Chinaruk et al., 2009, [7] investigated the impacts of the pitch and contort proportion on normal heat exchange coefficient and the weight misfortune when they are resolved in a round tube with the completely created stream for the (Re) in the scope of 12,000 to 44,000, two dimpled tubes with various space proportions of dimpled surfaces (PR = 0.7 and 1.0) . Three curved tapes with diverse bend proportions (y/w = 3, 5, and 7) are utilized, tentatively utilizing air as working liquid. The exploratory results uncover that both heat exchange coefficient and grating component in the dimpled tube fitted with the contorted tape, are higher than those in the dimple tube acting alone and plain tube. It is additionally found that the exchange coefficient and contact calculate joined gadgets increment as the space proportion and contort proportion (y/w) diminish.

Qasim et al., 2013, [8] investigated experimentally the heat exchanger with different wire coils insert in the tube with different pitches for Reynolds number ranging between [5000 to 40000] inside the tube and counter flow arrangement to the working fluid (water in two side). The results show that the heat transfers as well as friction factor increase with increasing the intensity of coils or decreasing the coiling pitches and enhancement by wire coils is more effective at low values of (Re) than high values. For (Re) range, a maximum increase in Nu of 2.43 is obtained, corresponding to an increase in friction factor of 4.75.

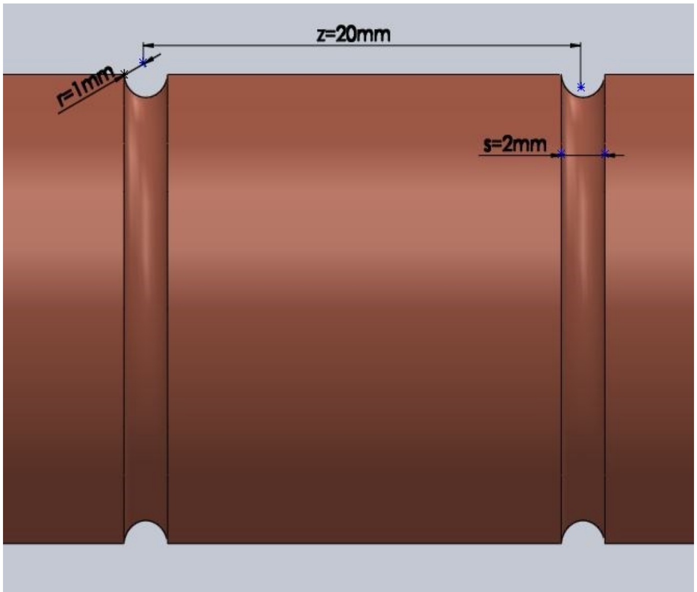
Experimental method

The shell and tube heat exchanger shown in figure (1a) is used in experiment work. Water from the hot vessel is first heated then flow into the inner tube and is cooled then by cooling water, the cold water inlet in the shell side. The water flow on the heat exchanger is counter flow. The test section in the first time used is made from smooth copper tube with dimensions of 1000mm length, 17.05mm ID and 19.05 OD, Second change tube by the corrugated tube with ratio of grooves on the outer surface is (z/d=1) and third corrugated the copper tube with ratio (z/d=0.5) [z/d = distance from center to center of groove / outer tube diameter], (z) can see in figure (1b). Shell made from P.V.C. with 1000mm length, 43mm ID and 50mm OD. Two flow meter s used to measure the flow rate of hot and cold water, four thermocouples are used to measure temperatures in the inlet and outlet of the heat exchanger for hot and cold water and digital manometer are used to measure the pressure drop. Hot water flow rate used are from (1-5) LPM and its temperature (40, 50, 60°C). Cold water flow rate are from (3-7) LPM at (25°C).



1-Data logger , 2- Test section, 3- Electrical control, 4- Hot water vessel, 5- Cold water vessel, 6- Hot water pump, 7- Cold water pump, 8- Hot water valves, 9- Cold water valves, 10- Hot water flow meter, 11- Cold water flow meter, 12- Cooling unit, 13- Thermocouples

(a) Schematic clarifies equipment's of the system



(z) (b) Sketch shows value of

Fig. (1)

Experimental procedure

a. All the flow meter and thermocouples are calibrated.

b. The hot water is used at temperatures (40°C), heated by used electrical heater and water flow rate (1-5) LPM.

c. Cold water temperature (25°C) cooling water by used cooling system and flow rate (3-7) LPM.
d. The hot water inner to the copper tube, and the cold water inner to the shell side.
e. Temperature of water in both side hot and cold are recorded after temperature of both the fluid attains a steady state.
f. The procedure was repeated for all hot and cold flow rate and for all hot water temperature.

Calculation equation

The heat dissipation for both side hot (inner tube) and cold (shell) can express as:

$$Q = \dot{m} * C_p * \Delta T = U * A_s * LMTD * F \qquad (1)$$

The correction factor can be taken (1). This is because of the counter flow arrangement and has one pipe and one shell within the present heat exchanger.

Non-dimensional Numbers

1- Reynolds number

For both sides dissipation as:

$$Re = \frac{u\rho d}{\mu} \qquad (2)$$

$$u = \frac{\dot{m}}{\rho A_c} \qquad (3)$$

For hot side:

$$A_{c,ip} = \frac{\pi}{4} {d_i}^2 \qquad (4)$$

For cold side (smooth):

$$A_{cs} = \frac{\pi}{4}\left({D_i}^2 - {d_o}^2\right) \qquad (5)$$

General equation for hydraulic diameter finds by following equations.

$$D_h = \frac{4A_{cs}}{P}, \quad P = \pi D_i \qquad (6)$$

For smooth tube

$$D_h = d_i \qquad (7)$$

For annular

$$D_h = D_i - d_o \qquad (8)$$

For cold side (corrugated), find two outer diameters in the pipe, $d_{o1} = 19.05$ mm, $d_{o2} = 18.55$ mm

Then use mean outer diameter, $d_o = d_{om} = \frac{d_{o1}+d_{o2}}{2}$ (9)

2- Nusselt Number

$$Nu = \frac{hd}{k_f} \qquad (10)$$

For turbulent flow in tube [9]

$$Nu = 0.023 Re^{0.8} Pr^{n} \tag{11}$$

Where, [n = 0.3 for cooling, n = 0.4 for heating]

With condition, [$Re \geq 10000$, $0.6 \leq Pr \leq 160$, $\frac{L}{D} \geq 10$]

3- Prandtl Number

Prandtl number can be obtained from [9]:

$$Pr = \frac{\mu \, Cp}{k} \tag{12}$$

4- Overall Heat Transfer Coefficient (U)

The heat exchanger is system contains two flowing fluids separated by a solid wall. The overall heat transfer coefficient of this mechanism can be expressed [10]:

$$\frac{1}{UA} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi k_t L} + \frac{1}{h_o A_o} \tag{13}$$

Because of the high value of thermal conductivity of the copper tube, the thermal resistance (second term on the left side) can be neglected. Equation (12) used to find value of (h) because (U) find from equation (2)

For smooth pipe:

$$A_i = \pi d_i L \quad \text{and} \quad A_o = \pi d_o L \tag{14}$$

For corrugated pipe [11]:

$$A_{oc} = A_{os,s} + N A_c \tag{15}$$

$$A_{oss} = \pi d_o (L - N * S_{corr.}) \tag{16}$$

N: number of grooves

Area of corrugated (groove) $A_c = 121.04 * 10^{-6} m^2$

5- Inner Side Heat Transfer Coefficient (hi)

The heat transfer coefficient of water in the inner side (hot water) calculated by the Newton's law of cooling [9]:

$$h_i = \frac{Q_h}{A_i \Delta T_h} \tag{17}$$

6- Outer Side Heat Transfer Coefficient (ho)

To calculate the cold side heat transfer coefficient for smooth and corrugated tubes can be calculated by using the following relation [9].

$$h_o = \frac{1}{\frac{1}{U_o} + \frac{A_o}{h_i A_i}} \tag{18}$$

7- Pressure drop

The value of pressure drop can be found by used the equation [10]:

$$\Delta P = f \frac{L}{d} \rho \frac{u_m^2}{2g_c} \tag{19}$$

Darcy friction factor find by equation
Laminar flow [11]:

$$f = \frac{64}{Re_d} \tag{20}$$

For smooth pipe turbulent flow Blasius's equation

$$f = \frac{0.316}{Re^{0.25}} \qquad 2000 \leq Re \leq 100000 \tag{21}$$

Results and discussions

Figures (2, 3 and 4) clarifies the relation between change hot water flow rate for smooth and two corrugated tubes (z/d=1, 0.5) for different hot temperature (40, 50 and 60°C). Show increase in temperature difference by changing the tube geometry and see an increase in temperature difference by increase cold water mass flow rate bbecause the water that contains heat change by cold water. And the enhanced range of temperature difference for corrugated tube are (14.94%, 43.2%) for corrugated tube (z/d=1, 0.5) respectively respect to smooth tube. The relation between change of cold water mass flow rate and heat dissipation in the hot side for smooth and two corrugated tubes (z/d=1,0.5), are shown in figures (5,6 and 7). The heat dissipation increase as cold water mass flow rate increase due to the fact that new water inter to the shell is cold and the water that having the heat is change, as well as the surface area for corrugated leads to increased heat exchange area. The enhancement percentage are (15.29%, 45.65%) for corrugated (z/d=1, 0.5) respectively with respect to smooth tube for tube side. Figures (8, 9 and 10) explain the relation between overall heat transfer coefficient (U) in tube side and increase in hot water mass flow rate the overall heat transfer coefficient (U) increase in constant cold water mass flow rate. The overall heat transfer coefficient more increasing by change the tube geometry and be higher in the corrugated tube (z/d=0.5), increase by (22.36%, 47.89%) by change the tube geometry from smooth to corrugated (z/d=1) and (z/d=0.5) respectively. The change of cold water mass flow rate leads to enhancement overall heat transfer coefficient and the percentage of increase in the overall heat transfer coefficient ranging between (54.54%, 55.72%) in tube side. This because the new cold water inter to the shell instead of the water have hot and the increase in cold mass flow rate lead to change the water in the shell and the groove in the outer surface of the pipe made disruption in the movement of water. The variation of hot water mass flow rate in the tube side with the Nusselt number at different hot temperature depicts in table (1). The Nusselt number increases by increasing the hot mass flow rate and change the hot temperature. This is due to the actuality that Nusselt number is a function of (h). The hot side doesn't have noticeable change because the properties of water don't have large change and the inner tube shape still smooth. Figures (11, 12 and 13) depict the variation of Nusselt number with hot mass flow rate in shell side at different hot temperature. The Nusselt number increase by change the tube geometry from smooth to corrugated (z/d=1) and more in (z/d=0.5). The increase percentage are (37.81%, 39.9%) for corrugated (z/d=1, 0.5) respectively. It could be concluded that the increasing in Nusselt numbers due to increasing in the surface area which due to increases the heat transfer. Figure (14, 15 and 16) shows the effect of hot mass flow rate on the pressure drop for different temperature and different geometry. The pressure drop change inversely with increase in hot mass flow rate and change tube geometry. The maximum value for pressure drop at smooth tube for $\dot{m}_c$ =3 LPM and 7 LPM, but the corrugated tube reduce this value in general for cold mass flow rate 3Lpm but in 7 LPM the corrugated tube z/d=1 beast than z/d=0.5,this due to the turbulent flow growth forth eddy connected with other generated the large swirl working to increase energy losses and reduce the amount of heat transfer as a result of increasing the number of grooves on the surface of the outer tube. The percentage enhancement for corrugated tubes respect to smooth tube is (88.34%, 83.44%) at temperature 40°C.

Conclusions

Important conclusions resulting from this work will be summarized in following points:

1. The corrugated tube enhancement heat transfer in heat exchanger, this clear in hot temperature difference and heat dissipation , the behavior of change show clearly as directly proportional for both.
2. The maximum enhancement percentages in hot temperature difference for two corrugated pipes (z/d= 1, 0.5) are (36.43, 50.31), (42.44, 51.12) and (44.26, 52.12) at temperatures (40, 50, 60°C) respectively.
3. The maximum enhancement percentages in heat dissipation for two corrugated pipes (z/d= 1, 0.5) are (40.4, 52.2), (45.4, 52.69) and (47.66, 52.78) at temperatures (40, 50, 60°C) respectively.
4. The behavior of the hot temperature difference inversely with increase hot water mass flow rates, but the heat dissipation directly proportionally.
5. The behavior of ΔT_c is increase directly proportional with $\dot{m}_h$ and inversely with $\dot{m}_c$.

Table (1) Nusselt Number in Tube-side

$\dot{m}_c$	$\dot{m}_h$	Nu_h at 40°C			Nu_h at 50°C			Nu_h at 60°C		
L/M	L/M	Smooth	z/d=1	z/d=0.5	smooth	z/d=1	z/d=0.5	Smooth	z/d=1	z/d=0.5
3	1	14.73	14.75	14.8	15.75	15.81	15.86	16.76	16.86	16.98
3	2	25.67	25.75	25.78	27.49	27.54	27.67	29.31	29.33	29.65
3	3	35.54	35.62	35.69	38.09	38.17	38.37	40.63	40.69	41.07
3	4	44.79	44.88	44.97	48.07	48.12	48.38	51.25	51.33	51.82
3	5	53.65	53.76	53.78	57.63	57.63	57.9	61.46	61.38	62.03
7	1	14.68	14.71	14.74	15.64	15.76	15.81	16.65	16.82	16.89
7	2	25.59	25.76	25.72	27.31	27.5	27.621	29.1	29.36	29.46
7	3	35.47	35.55	35.65	37.87	38.14	39.05	40.37	40.69	40.87
7	4	44.7	44.79	44.91	47.8	48.09	49.23	50.95	51.31	51.54
7	5	53.52	53.63	53.65	57.21	57.53	57.66	61.04	61.42	61.7

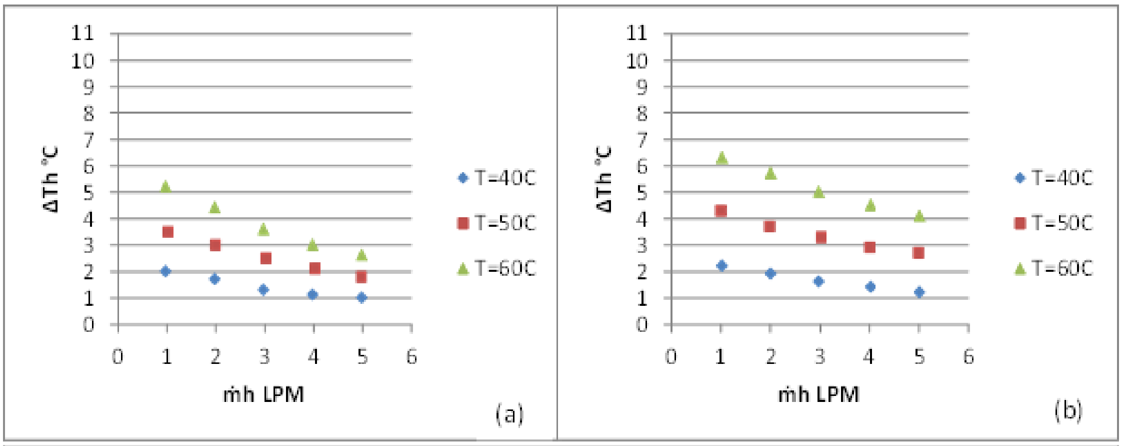


Figure (2) Effect of hot mass flow rate on temperature difference in smooth tube at different temperatures (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

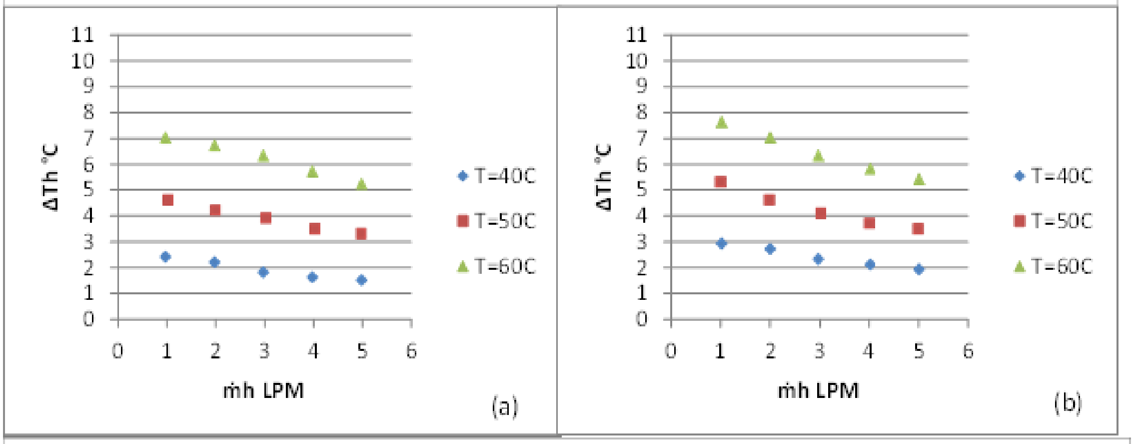


Figure (3) Effect of hot mass flow rate on temperature difference in corrugated tube (z/d=1) at different temperatures (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

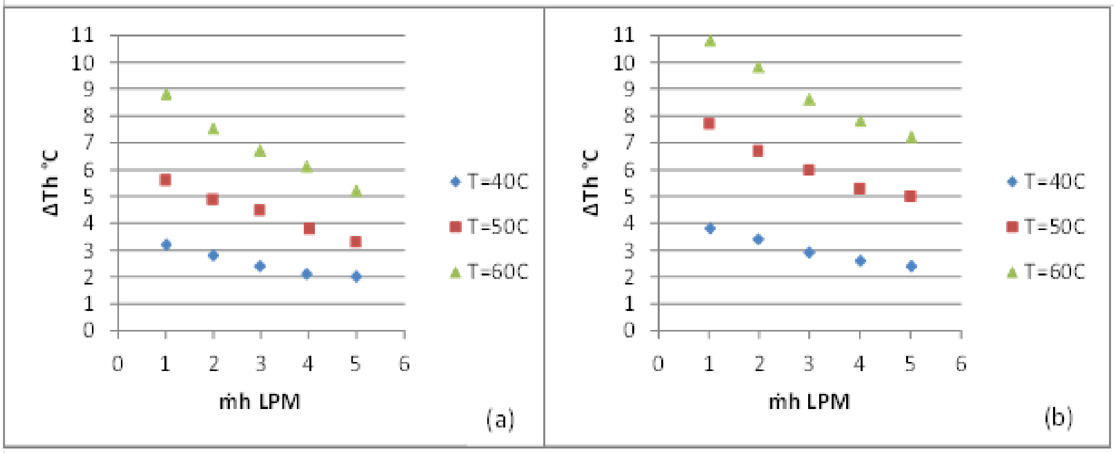


Figure (4) Effect of hot mass flow rate on temperature difference in corrugated tube (z/d=0.5) at different temperatures (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

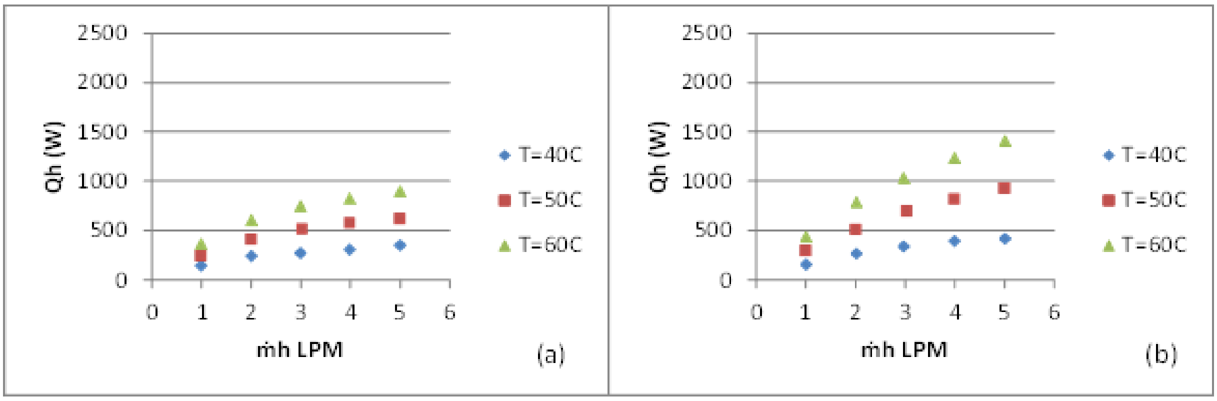


Figure (5) Effect of hot mass flow rate on heat dissipation in smooth tube at different temperatures (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

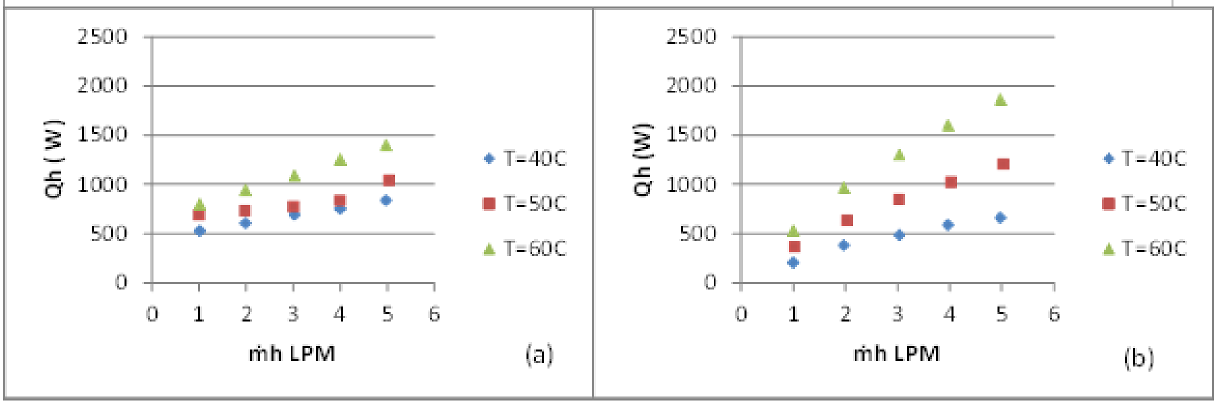


Figure (6) Effect of hot mass flow rate on heat dissipation in corrugated tube (z/d=1) at different temperatures (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

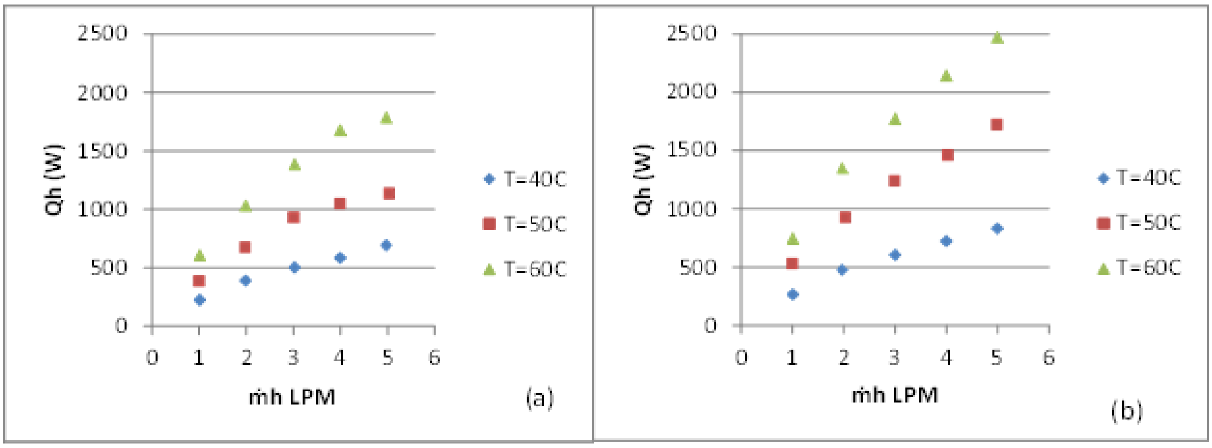


Figure (7) Effect of hot mass flow rate on heat dissipation in corrugated tube (z/d=0.5) at different temperatures (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

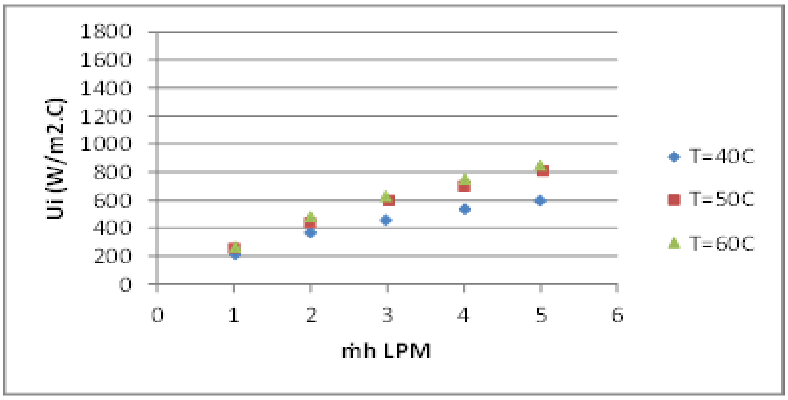


Figure (8) Effect of hot mass flow rate on overall heat transfer coefficient in smooth tube at $\dot{m}_c$=7LPM in different temperatures

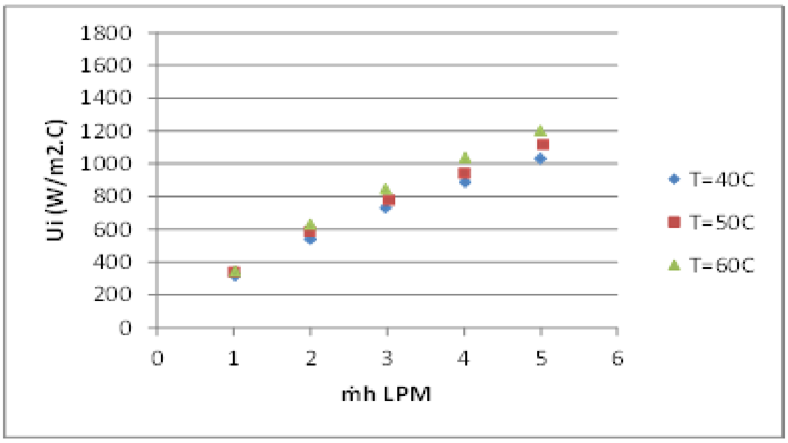


Figure (9) Effect of hot mass flow rate on overall heat transfer coefficient in corrugated tube (z/d=1) at $\dot{m}_c$=7LPM in different temperatures

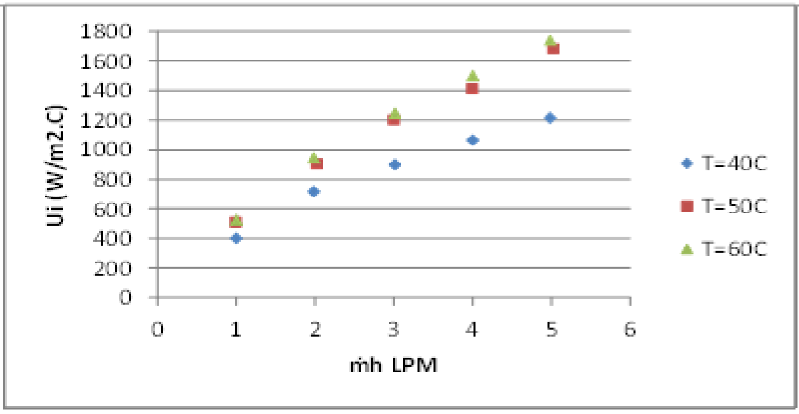


Figure (10) Effect of hot mass flow rate on overall heat transfer coefficient in corrugated tube (z/d=0.5) at $\dot{m}_c$=7LPM in different temperatures

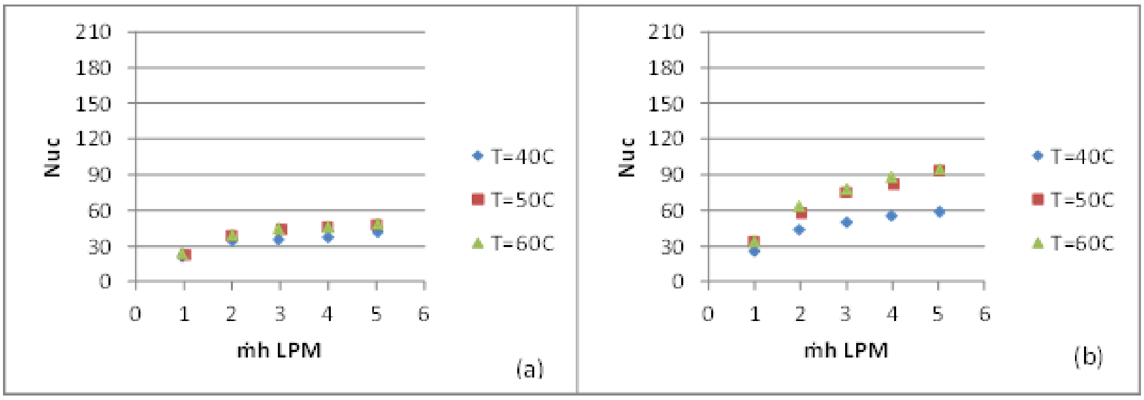


Figure (11) Effect of hot mass flow rate on Nusselt Number in smooth tube at (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

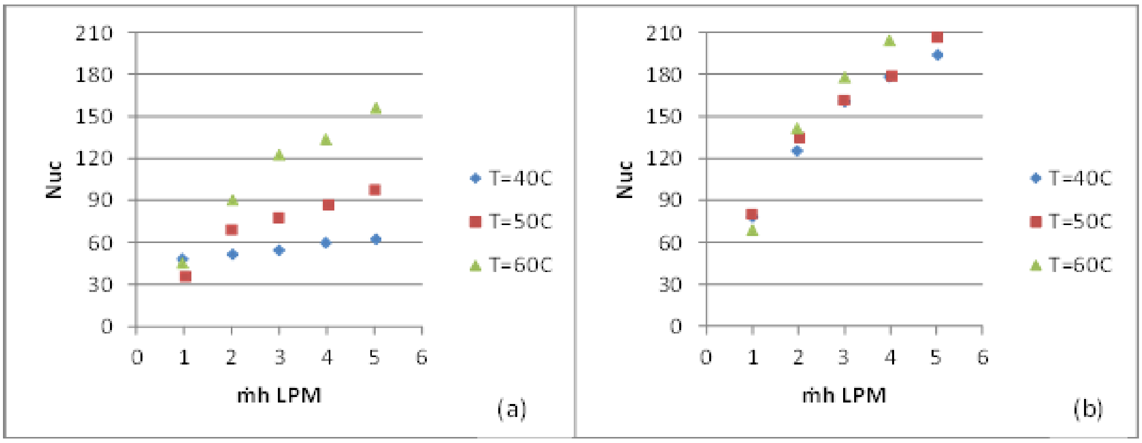


Figure (12) Effect of hot mass flow rate on Nusselt Number in corrugated tube (z/d=1) at (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

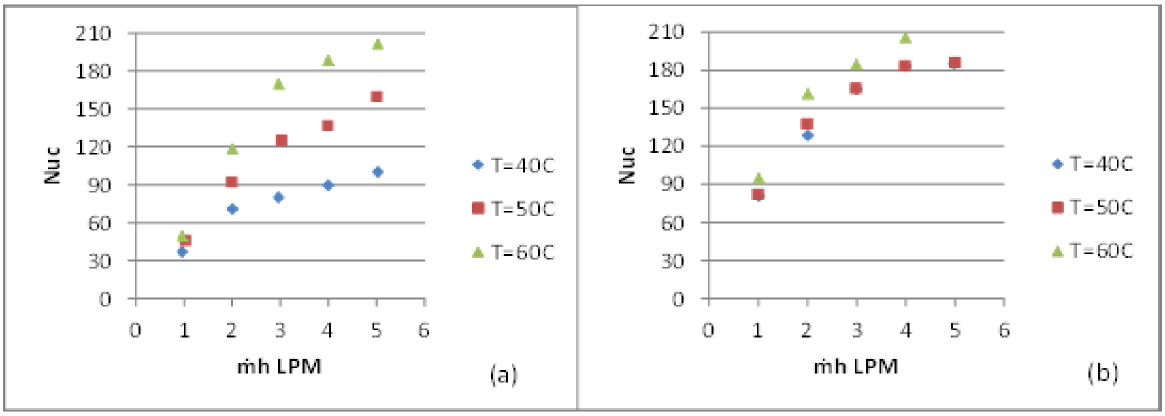


Figure (13) Effect of hot mass flow rate on Nusselt Number in corrugated tube (z/d=0.5) at (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

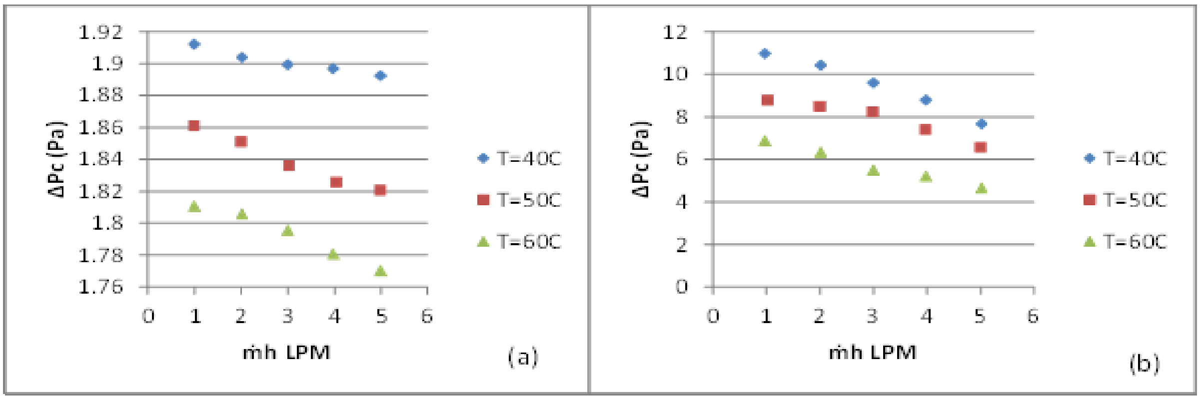


Figure (14) Effect of hot mass flow rate on pressure drop in smooth tube at (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

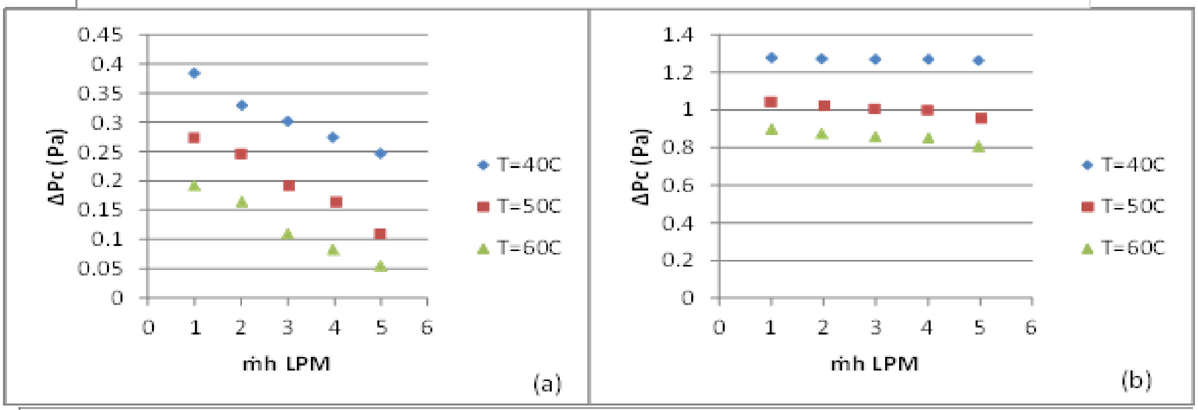


Figure (15) Effect of hot mass flow rate on pressure drop in corrugated tube (z/d=1) at (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

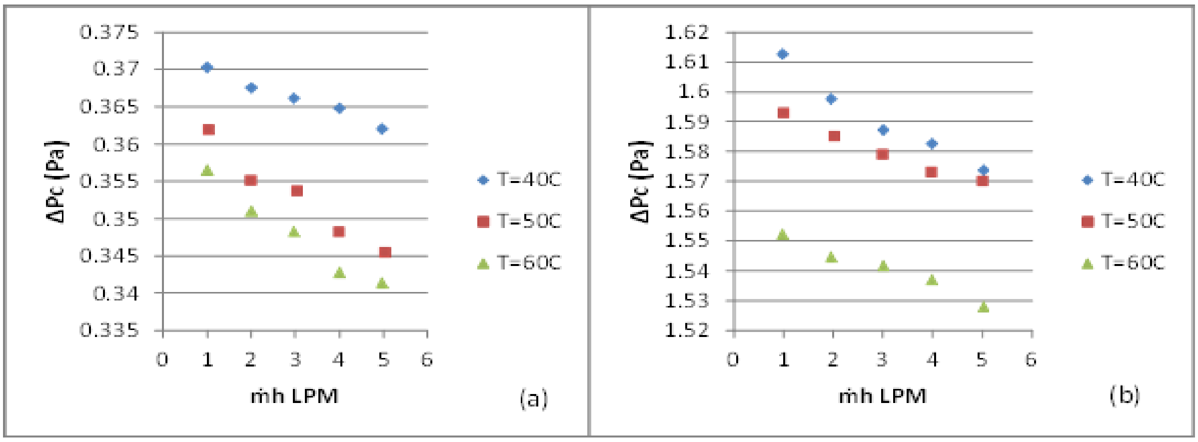


Figure (16) Effect of hot mass flow rate on pressure drop in corrugated tube (z/d=0.5) at (a) $\dot{m}_c$=3LPM (b) $\dot{m}_c$=7LPM

Nomenclature					
Symbol	Description	Units	Symbol	Description	Units
A	Area	m^2	M	Dynamic viscosity	kg/m. sec
As	Surface area	m^2	*P*	Density	kg/m^3
Ac	Cross section area	m^2	ΔP	Pressure drop	Pa
Cp	Specific Heat	J/kg. °C	Q	Heat dissipation	W
D	Diameter of shell	M	z/d	Corrugated ratio	…
De	Equivalent diameter	M	Pr	Prandtl number	…
Dh	Hydraulic diameter	M	Π	PI equal to 3.145926	…
D	Diameter of inner tube	M	L	Length	M
F	Correction factor	…	N	Number of grove	…
f	Fraction factor	…	Nu	Nusselt number	…
H	Heat transfer coefficient	$W/m^2.°C$	*P*	Perimeter	M
K	Thermal conductivity	W/m. °C	*Pe*	Equivalent perimeter	M
Re	Reynold number	…	Ds	Diameter of smooth	M
S	Width of corrugated	…	Dc	Diameter of corrugated	M
T	Temperature	°C	U	Velocity	m/sec
U	Overall heat transfer coefficient	$W/m^2.°C$			

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