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Investigation of a Double Tube Heat Exchanger with Dimples

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Abstract - The experimental investigation of the double concentric-tube heat exchanger is presented with inner tube dimples. The internal tube side heat transport enhancement characteristics of double tube heat exchangers composed of a dimpled tube with an elliptic shape that are investigated experimentally in the range of Reynolds number from 2570 to 14405 for inner tube fluid while for outer tube fluid is 1477 to 8103. The purpose is to present a clear view of the thermofluid characteristics of this type of heat exchanger with different key design parameters leading to design optimization. The effects of dimple angles on the inner tube side thermo fluid characteristics are also explored. The effectiveness, number of transfer units, and Nusselt number of the double tube heat exchanger with dimples are higher than that of the double tube heat exchanger without dimples by approximately 87%, 107.8% and 53%, respectively.

Key Words: Double tube, Dimple tube, Cooling performance, Heat exchangers.

1. INTRODUCTION

The need to achieve high thermo-hydraulic performance and compact heat exchangers have pushed a Lot of industries to find new ways to promote heat transfer of double tube heat exchanger [1-7]. Applying dimpled surface is one of the passive techniques for heat transfer enhancement of double tube heat exchanger. In this respect, the dimpled surface has emerged as one way to increase thermo-hydraulic performance. The improved characters create rotating and secondary flows that increase the effectiveness of the heat transfer area of a double tube heat exchanger. It interrupts thermal and velocity boundary layer development with increasing degrees of turbulence close to the rough tube wall. These accompanying increases the convective heat transfer constant (h) with a resultant increase in the pressure drop on the tube.

Sarmadian et al. [8] investigated condensation heat transfer in a helically dimpled tube using refrigerant R-600a as a working fluid. Their results showed that the heat transfer rate was increased up to two times greater than that of a smooth straight tube. Ming Li et al. [9] examined the thermo-hydraulic performance of a dimpled tube in steady-state single-phase (liquid-to-liquid) fluid flow for Reynolds numbers ranging from 500 to 8000 and for a water/glycol. Shuai Xie et al. [10] studied the heat transfer and thermal performance of enhanced tube with cross ellipsoidal dimples. They observed that the cross ellipsoidal dimples induced the transverse and longitudinal dimples, which caused the reattachment and periodic impingement flows that helped improve thermal performance. Aroonrat and Wong wises [11] studied the effect of dimpled depth on heat transfer enhancement, pressure loss and overall performance of dimpled tubes. Their results showed that the dimpled tube with the largest dimpled depth gave the highest heat transfer enhancement and pressure loss up to 83 and 892% over those of the smooth tube. Recently, nanofluids were applied in dimpled tubes for further heat transfer enhancement. In the present study, a thermo-hydraulic performance of a dimpled tube heat exchanger is investigated. The effects of a dimpled tube with dimple angles of 45˚, 60˚ and plain tube are studied.

2. EXPERIMENTAL SETUP AND PROCEDURES

The experimental setup includes a double dimpled tube heat exchanger and three circuits, namely, closed loop chilled water cycle, R-22 vapor compression refrigeration cycle, and closed loop hot water cycle. The double dimpled tube heat exchanger made of copper with a thickness of 1 *mm*, and a length of 1100 *mm* associated with the necessary measuring instruments. The inner tube diameter is 18 *mm*, and the outer tube diameter is 20 *mm*. The geometrical characteristics data of the double dimpled tube heat exchanger are given in Table [1]. Two fluids are being considered which are chilled water in the outer tube and hot water in the inner tube of the heat exchanger. The chilled water system includes a 0.5 m^3 thermally insulated tank, and a cooling system that is controlled by adjusting a temperature controller, the chilled water is generated by the cooling system. The closed loop hot water cycle consists of a 0.56 m^3 thermally insulated tank supplied with 5 electrical heaters with total power of *7.5 kW* and an adjustable temperature controller. Figure [1] shows a photograph of the experimental setup and the double tube heat exchanger. The setup contains two

Table (1) Dimensions of the double tube heat exchanger

Geometrical characteristics	Inner tube	Outer tube
Outer diameter of tube, D_o, (mm)	20	54
Inner diameter of tube, D_i, (mm)	18	52
Length of tube, L, (mm)	1100	1500
Thickness, δ, (mm)	1	1
Tube material	Copper	Copper
Dimpled tube		
Depth of dimple, (mm)	1.5	_
Diameter of an ellipsoidal dimple , (mm^2)	6×3	_
Dimple pitch , P, (mm)	12	_
Angle between dimples, θ, (°)	(45-60)	_
Dimple shape	Elliptic	_

Centrifugal pumps which are used to pump the water of the chilled water and hot water system; the capacity of each pump is *1hp*. Two ball valves are used to control the flow rate of those pumps, while water flow rates are measured using two flow meters with accuracies of *±0.5%* of full scale. The outer surface of the heat exchanger is well insulated, and necessary precautions are taken to prevent leakages from the system [12]. To stabilize the consumed electric power of the electric circuits, voltage stabilizers are conducted on the electric component of the circuits [13]. The temperatures are measured at the inlet and outlet of the test rig by a digital thermometer with J-type thermocouples.

Fig.1 Experimental double tube heat exchanger with instrumentation

3. MEASURING TECHNIQUES

The chilled water and hot water temperatures were adjusted to almost constant values (*20 ºC ± 0.7 ºC*), (and *80 ºC ± 0.7 ºC*). However, the inlet and outlet temperatures were measured using four barked pre-calibrated J-type thermocouples. The chilled and hot water flow rates were measured using two flow meters with a range of (*2* to *18 L/m*). Water pressure drop across the test rig was measured using a U-tube manometer. The data range of the experimental is given in Table (2).

Table [2] Experimental conditions

Variables /Unit	**Range**
Inlet-chilled water temperature ($T_{C,i}$), *ºC*	*20 ºC ± 0.5*
Inlet-hot water temperature ($T_{h,i}$), *ºC*	*80 ºC ± 0.5*
Inlet-inner tube water velocity (v_1), m/s	*0.13-1.18*
Inlet-outer annulus water velocity (v_2), m/s	*0.04-0.37*

4. RESULTS AND DISCUSSION

Heat transfer and flow characteristics of the dimple tube heat exchanger are investigated experimentally. The influence of inner tube fluid velocity, hot water temperature, flow arrangement, and dimple angles were investigated. The major goal is to improve the thermofluid cooling performance of the DTHE. The findings are divided into three sub-sections that detailed and analyzed the comprehensive performance criteria.

4.1 Effect of dimple angle (θ)

Figure (2) shows the relationship between the effectiveness of the double tube heat exchanger against the Reynolds number for the hot water side at different dimple angles (θ) and plain tube at a constant chilled water flow rate of 10 L/m. It can be seen from this figure that the effectiveness increases with the increase of the Reynolds number for all cases. At a constant Reynolds number of $Re_{h,w} = 8395.9$, the effectiveness for the dimple angle (θ =60˚) is higher than the plain tube by approximately 87% for the same condition, and the effectiveness of the heat exchanger for dimple angle of (θ = 60˚) is higher than the dimple angle of (θ =45˚) by approximately of 11.8%. Because the heat exchanger with a dimpled tube has a surface area for heat greater than the surface area for heat transfer in the heat exchanger with plain tube. In addition, it creates eddy currents in the direction of the inner tube working fluid that leads to an improvement in heat transfer coefficients. Figure (3) shows the relationship between the number of transfer units (N.T.U) of the double tube heat exchanger against the Reynolds number for the hot water side at different dimple angles (θ) and plain tube at a constant chilled water flow rate of 10 L/m. It can be seen from this figure that the (N.T.U) increases with the increase of Reynolds number for all cases. At a constant Reynolds number of $Re_{h,w}$ = 8396, the (N.T.U) for the dimple angle (θ = 60˚) is higher than the plain tube by approximately 107.8% for the same condition, (N.T.U) of the heat exchanger for dimple angle of (θ = 60˚) is higher than the dimple angle of (θ =45˚) by approximately of 15.7%. Because the heat exchanger with a dimpled tube has a surface area for heat greater than the surface area for heat transfer in the heat exchanger with a plain tube. In addition, it creates eddy currents in the direction of the inner tube working fluid that leads to an improvement in heat transfer coefficients. Figure (4) shows the relationship between the Nusselt number of the double tube heat exchanger against the Reynolds number for the hot water side at different dimple angles (θ) and plain tube at a constant chilled water flow rate of 10 L/m. It can be seen from this figure that the Nusselt number increases with the increase of the Reynolds number for all cases. At a constant Reynolds number of $Re_{h,w}$ = 8396, the Nusselt number for the dimple angle (θ = 60˚) is higher than the plain tube by approximately 53% for the same condition, Nusselt number of the heat exchanger for dimple angle of (θ = 60˚) is higher than the dimple angle of (θ =45˚) by approximately of 35.5%. Because the heat exchanger with a dimpled tube has a surface area for heat greater than the surface area for heat transfer in the heat exchanger with the plain tube. In addition, it creates eddy currents in the direction of the inner tube working fluid that leads to an improvement in heat transfer coefficients. Figure (5) shows the relationship between the Friction factor of the double tube heat exchanger against the Reynolds number for the hot water side at different dimple angles (θ) and plain tube at a constant chilled water flow rate of 10 L/m.It can be clearly seen from this figure that the friction increases with the increase of the Reynolds number for all cases. At a constant Reynolds number of $Re_{h,w}$ = 8396 , the Friction factor for the dimple angle (θ = 60˚) is found to be higher than the lower inlet plain tube by approximately 140% for the same condition. Because the heat exchanger with a dimpled tube has a surface area for heat greater than the surface area for heat transfer in the heat exchanger plain tube. In addition, it creates eddy currents in the direction of the inhibitor flow, working on an improvement process of heat transfer.

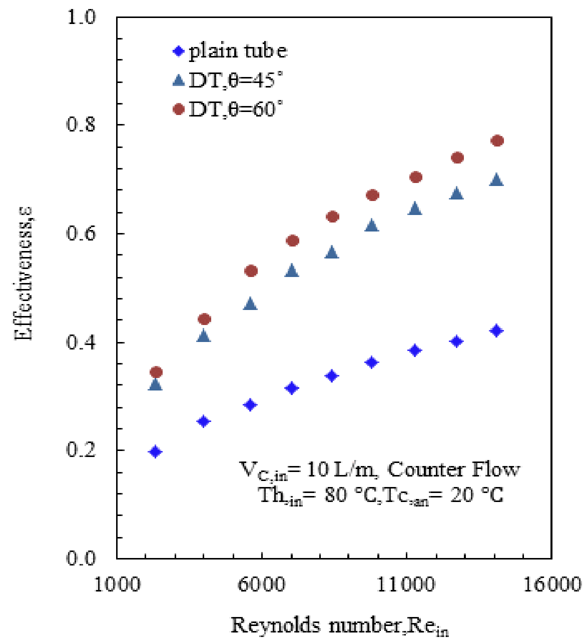

Fig.2 Variations of ε against Re for different dimple angles (θ)

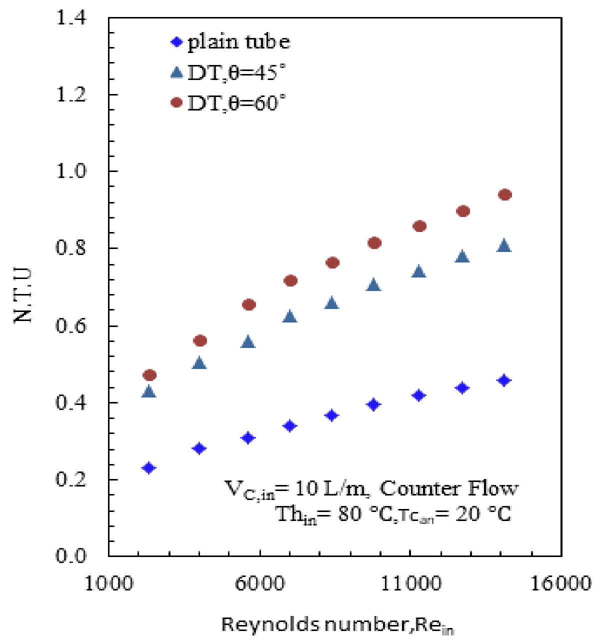

Fig.3 Variations of NTU against Re fordifferent dimple angles (θ)

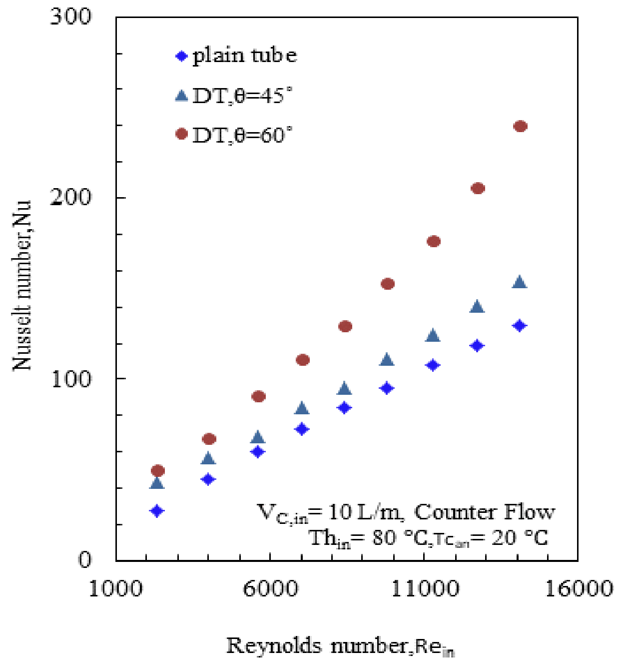

Fig.4 Variations of (Nu) against Re for different dimple angles (θ)

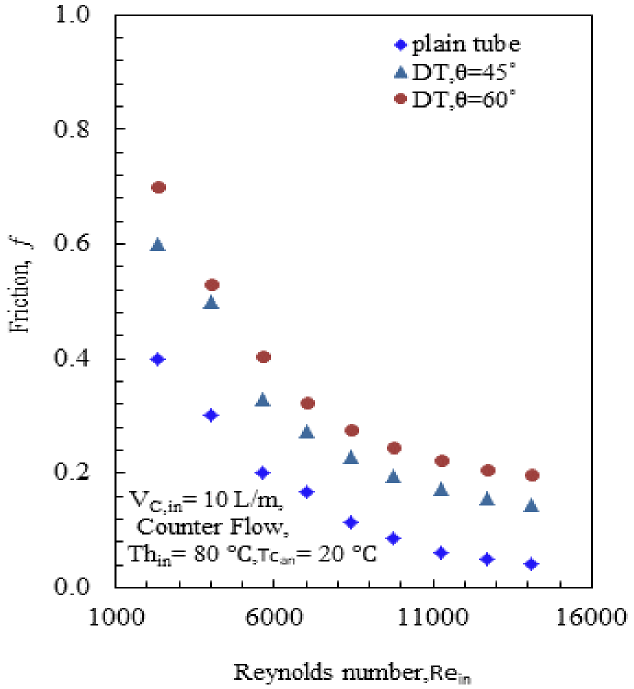

Fig.5 Variations of (ƒ) against Re for different dimple angles (θ)

5. Conclusions

The thermofluid performance of the double concentric-tube heat exchanger is investigated in this study. The study is carried out for Reynolds numbers ranging from 2570 to 14405 for inner tube fluid and 1477 to 8103 for outer tube fluid. The influences of dimple angles are investigated. The number of transfer units, as well as the effectiveness of the heat exchanger, are investigated. The following is a summary of the conclusion:

- The effectiveness, number of transfer units, and Nusselt number of the double tube heat exchanger with dimples is higher than that of the double tube heat exchanger without dimples by approximately 87%, 107.8%and 53%, respectively.
- The double tube heat exchanger with dimples contributes higher effectiveness and more energy-saving than the double tube heat exchanger with plain tubes.

Nomenclature

A	Area (m2)
Cp	Specific heat (kJ.kg^{-1}.K^{-1})
D	Diameter (m)
Ƒ	Friction factor
h	Heat transfer coefficient ($W.m^{-2}.K^{-1}$)
k	Thermal conductivity (W.m^{-1}.K^{-1})
L	Length (m)
m	Mass flow rate (kg.s^{-1})
Nu	Nusselt number
Δp	Pressure drop (N.m^{-2})
Re	Reynolds number (-)
T	Temperature (^{o}C)
U	Overall heat transfer coefficient ($W.m^{-2}.K^{-1}$)

Greek Symbols

δ	Thickness (*m*)
ε	Effectiveness
θ	Dimple angle
μ	Viscosity ($kg.m^{-1}.s^{-1}$)
ρ	Density ($kg.m^{-3}$)

Abbreviations

DT	Dimple tube heat exchanger
NTU	Number of transfer units

Scripts

av	average
c	cold
h	hot
hy	hydraulic
i	inlet/inner
o	outlet/outer
min	minimum
max	maximum

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