Wasit Journal of Engineering Sciences
VOLUME: (6), NO.: (3) 2018

Experimental Investigation of Heat Transfer Enhancement by Using Different Number of Fins in Circular Tube

Dr. Kamil Abdul Hussien

[1]Mechanical Engineering Department, College of Engineering, Wasit University, Iraq
*Corresponding author, drkamel@uowasit.edu.iq

Submitted: 4/6/2017
Accepted: 4/12/2017

Abstract-The present work investigates the enhancement of heat transfer by using different number of circular fins (8, 10, 12, 16, and 20) in double tube counter flow heat exchanger experimentally. The fins are made of copper with dimensions 66 mm OD, 22 mm ID and 1 mm thickness. Each fin has three of 14 mm diameter perforations located at 120o from each to another. The fins are fixed on a straight smooth copper tube of 1 m length, 19.9 mm ID and 22.2 mm OD. The tube is inserted inside the insulated PVC tube of 100 mm ID. The cold water is pumped around the finned copper tube, inside the PVC, at mass flow rates range (0.01019 - 0.0219) kg/s. The Reynold's number of hot water ranges (640 - 1921). The experiment results are obtained using six double tube heat exchanger (1 smooth tube and the other 5 are finned one). The results, illustrated that the heat transfer coefficient proportionally with the number of fin. The results also showed that the enhancement ratio of heat transfer for finned tube is higher than for smooth tube with (9.2, 10.2, 11.1, 12.1 13.1) times for number of fins (8, 10, 12, 16 and 20) respectively.

Keywords: Finned tube heat exchanger, water-water, circular fin, temperature, overall heat transfer coefficient; performance.

مبحث تجريبي لتحسين نقل الحرارة باستخدام عدد مختلف من الزعانف القرصية في أنبوب دائري

كامل عبد الحسين

الخلاصة: الدراسة الحالية لتحسين انتقال الحرارة باستخدام عدد مختلف من الزعانف الدائرية (20،16،12،10،8.) زعنفة لمبادل حراري ثنائي الانابيب متعاكس الجريان تجريبيا. الزعانف مصنعة من النحاس بأبعاد (66)مليمتر قطر خارجي, (22) مليمتر قطر داخلي و (1) مليمتر سمك ومثقبة بثلاث ثقوب دائرية بزاوية (120°) وبقطر (14)مليمتر. الزعانف مثبة على انبوب نحاس املس طوله 1م و 19.9ملم قطر داخلي و22.2ملم قطر خارجي.يثبت الانبوب داخل انبوب معزول من مادة PVC قطره الداخلي 100 ملم.الماء البارد يجهز بمعدل تدفق متغير0.01019 -0.0219 كغم/ثا والماء الساخن ذو ارقام رينولدز تتراوح بين640 الى 1921. بينت النتائج ان نسبة تحسن انتقال الحراره تتناسب مع عدد الزعانف ،حيث ان زيادة نسبة تحسن انتقال الحرارة بين انبوب املس واخر مزعنف بلغت (9.2, 10.2, 11.1, 12.1 13.1)مرة لعدد الزعانف(20،16،12،10،8.) بالتوالي.

I. INTRODUCTION

Recently, finned tube are widely used for the purpose of improvement of heat exchanger efficiency. This improvement was carried out by increasing the extension of surface heat transfer. Such type of fins is very popular in boilers and waste heat recovery. They have also an excellent resistance to wear and fouling problem because of their unique structure [1].

A vast number of numerical and experimental studies have been carried out on the heat transfer efficiency and resistance characteristics of finned tubes. These researches mainly investigates the use of spiral-finned tubes [2, 3], plain finned tubes [4–7] and serrated finned tubes [8, 9]. Only few studies have studied the effectiveness of using H-type finned tube in improving the heat transfer rate.

Furthermore, circular finned tube have also attracted the interest of researches. In these studies, several empirical works have been carried out for the purpose of improving the fin efficiency and compactness as well as the reduction of the pressure losses [10]. The performance of another type of extended surface fins, longitudinal or radial fins, are also investigated [11-14].

The enhancement of performance of a double tube heat exchanger with circular finned was investigated by [15]. Various configurations of fins with different dimensions are used in the experimental work. The results showed that (3.12 to 3.83) enhancement in HT coefficient could be obtained when circular finned tube heat exchanger with three perforated of diameter (14mm) is used.
The current study presents an experimental investigation of the performance of heat exchanger for the case of using different number of circular fins (8, 10, 12, 16, and 20). The results obtained in this work are for Reynold's number range in between 640 and 1921 with (19.9mm) inner diameter of tube.

II. Theoretical Analysis

The present work studies the counter flow of two fluids inside heat exchanger arrangement as shown in Fig (1). The cold water undergoes forced flow through annuli while the hot de-ionized water flow inside inner tube. The insulation of heat exchanger outer surface, steady state condition, and the no phase change have been assumed during the experiment.

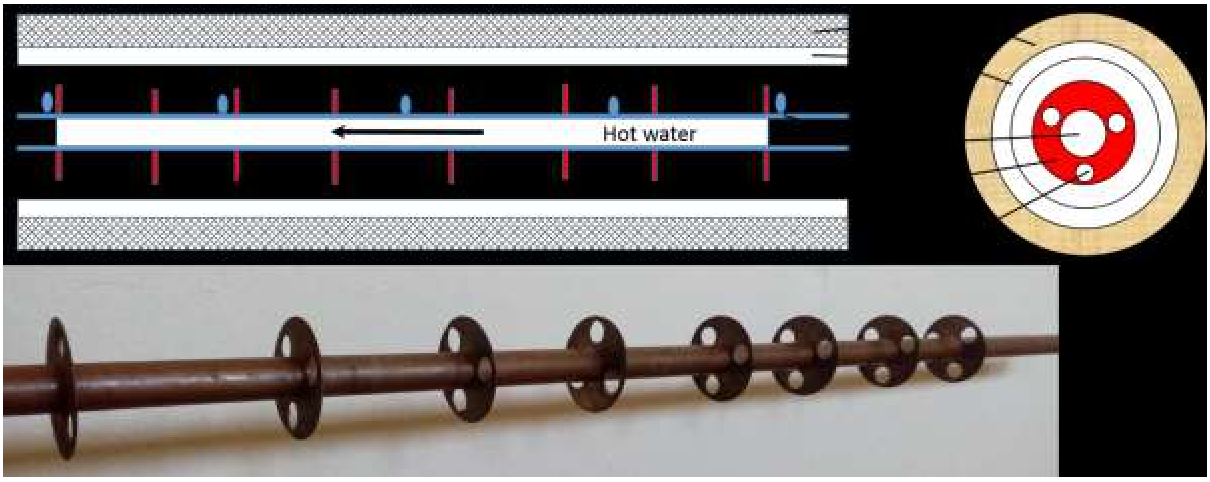

Fig.1. Circular finned tube heat exchanger with three perforated.

Consequently the heat dissipation of both sides can be written as

$$Q\,c = \dot{m}_c\, Cpc\, (T_{co} - T_{ci}) \quad \text{And} \quad Q\,h = \dot{m}_h Cp\,h\, (T_{hi} - T_{ho}) \tag{1}$$

In this study, the heat dissipation is calculated using the temperature deviation on the water side as in equation (1) [16, 17, and 18].
Heat transfer coefficient of hot water side can be calculated as in equation (2):

$$h_i = \frac{Q_h}{A_i(T_m - T_s)} \tag{2}$$

Where: $A_i = \pi d_i L$, $T_m = \frac{T_{hi}+T_{ho}}{2}$, $T_s = \frac{T_{s1}+T_{s2}+\cdots+T_{s5}}{5}$

Then Overall HT coefficient can be calculated as:

$$U_i = \frac{Q_h}{A_i LMTD} \tag{3}$$

Where

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\Delta T_2 / \Delta T_1\right)} \ , \Delta T_1 = T_{hi} - T_{co} \ , \ \Delta T_2 = T_{ho} - T_{ci}$$

The thermal resistance $\frac{\ln\left(r_o / r_i\right)}{2\pi K L}$, can be neglected, annuli HT coefficient can be expressed as:

$$h_o = \frac{1}{\frac{1}{U_0} - \frac{A_o}{h_i A_i}} \tag{4}$$

Where; $A_o = \pi d_o L$, $U_o = \frac{U_i A_i}{A_o}$

Nusselt's number for annuli side can be calculated as

$$Nu_c = \frac{h_o D_e}{K_c} \tag{5}$$

Where:

$D_e = D_{hu} = D_i - d_o$ (laminar flow in smooth annuli)

$D_e = \frac{4A_c}{P_h}$ (laminar flow in finned annuli)

$$A_c = \left[\frac{\pi}{4}(D_i{}^2 - d_o{}^2)L - \frac{\pi}{4}(d_f^2 - d_o^2) * \delta * N_f + \frac{\pi}{4} d_p{}^2 * n_p * \delta * N_f\right]/L$$

$$P_h = \pi d_o + \pi(d_f - d_o)N_f - \pi d_p n_p N_f$$

Reynold's number is calculated according to equation (6):

$$Re = \frac{\rho_c\, u\, D_{hu}}{\mu_c} \tag{6}$$

$D_{hu} = d_i$ (Laminar flow in the inner tube)
$D_{hu} = D_i - d_o$ (Laminar flow in smooth annuli)
$D_{hu} = \frac{4A_c}{p_W}$ (Laminar flow in finned annuli)

$$P_W = \pi(D_i + d_o) + \pi(d_f - d_o) * N_f - \pi * d_p * n_p * N_f$$

The effectiveness and Number of HT units can be calculated as follows:

$$\varepsilon = \frac{Q_{act}}{Q_{max}} \tag{7}$$

$$NTU = \frac{UA}{C_{min}} \tag{8}$$

Enhancement Ratio Factor is given by[19]

$$EF = \frac{h_{of}}{h_{os}} \tag{9}$$

Where: h_{of} HT coefficient for outside of fin tube.
h_{os} HT coefficient for outside of smooth tube.

III. EXPERIMENTAL ISSUES

A. Experimental Test Rig

Figures (2 &3) show the image and diagram of the test device. The test device consists of a water supply system, a test section, a measuring instrument [a temperature recorder, a water flow meter, a digital scale, thermocouples and temperature probes] and supplements.

The test section in this study has two parts: the first part consists of 100mm ID, 110 OD, 1.4m length and 5mm thickness insulated PVC tube. The inside of the tube is insulated by sheet-roll while the outside surface is insulated by glass wool which has (25.4mm) thickness, (0.036 $\frac{W}{m.\mathcal{C}}$) thermal conductivity, to reduce the heat losses to the surrounding. The second part of the test section contains a copper tube with or without circular copper fins. The smooth copper tube is 1m long, 19.9 mm ID, and 22.2 mm OD. Circular fins were made of cooper and each one has three perforations of 14 mm diameter which located at 120º from each to another; the distance between the fins is distributed equally along copper tube. Fins are firstly heated to provide some thermal extension and to increase the inner diameter. Then the tube is inserted and when the fins be on the specified locations, then fins were cooled and full contact is achieved due to shrinkage. Two small Water pumps are used for pumping the cold and hot water in to pipes through the water cycle and test section. All measuring devices, used in the experiment, have been calibrated.

B. Experimental procedure

In order to achieve the goal of this study the following procedure can be applied:

a. The inlet hot water was heated by using electrical heaters up to 60°C and pumped into the copper tube at mass flow rate 0.0055, 0.0082, 0.0109, 0.0137, 0.0164 kg/sec.
b. The inlet cold water was already cooled down by ice to 25°C and pumped at mass flow rate 0.01019, 0.0164, 0.0219 kg/s in the annuli.
c. At the steady state condition, the temperatures of the inlet and outlet of both hot and cold water are recorded for each value of water mass flow rate and heat power.
d. The temperature of a five equal spaced points locations along the outside copper tube surface (as shown in Fig. 1) were also recorded at steady state condition.
e. Pressure drop in both hot water and cold water sides (i.e. inlet and outlet) are also measured at steady state condition by digital manometer.
f. In order to ensure the validity of the experimental results in our lab, each measurement test was repeated three times.

Then the above steps were repeated using different cold water mass flow rates with constant hot water temperatures 60°C.

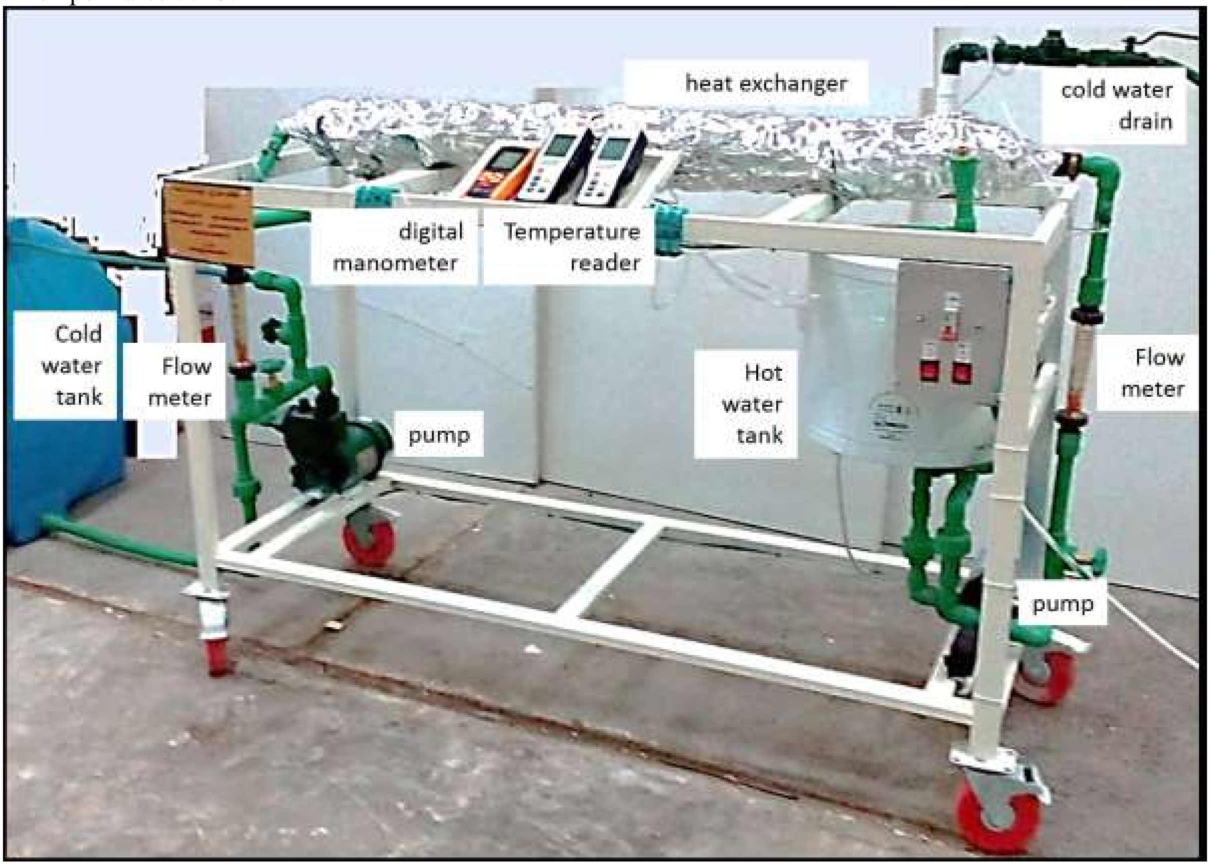

Fig. 2 Photograph of the Experimental Test Rig

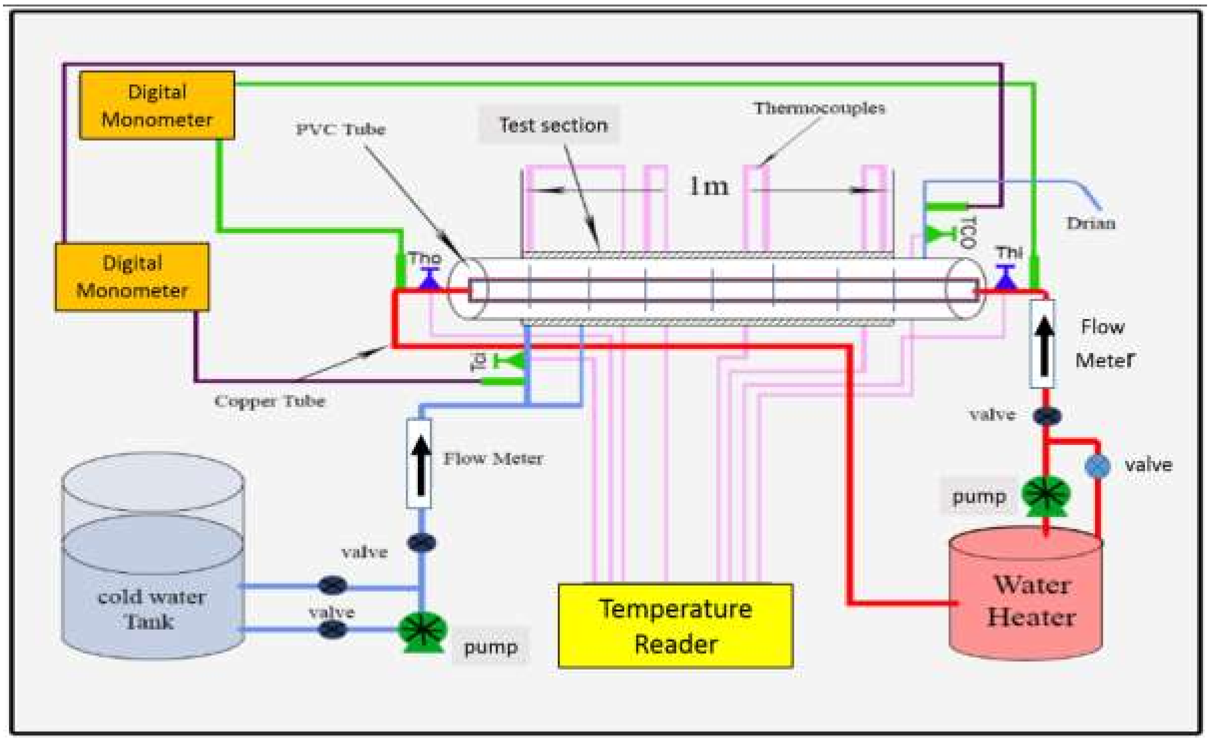

Fig. 3 Schematic diagram for Experimental Test Rig

IV. Results and Discussions

A. Relationship between Reynolds number for Hot Water (Reh) and hot Water Temperature Difference (ΔTh).

Figures (4 to 9) illustrate the variation of the (ΔT_h)with the hot water Reynold's number (Re_h) for smooth tube(ST) and circular finned tube (CFT)with three perforations at different number of fin(Nf) (8,10,12,16and 20) at cold water mass flow (mc) rate equal to (0.0109, 0.0164, 0.0219)kg/s. The results show that when the mc increased the ΔTh decreased because the water recycling in this site found less. The ΔTh of CFT at Nf=8 decreased by (1.15, 1.3) times for Nf(8,10,12,16and 20) respectively at mc 0.0109 kg/s. The ΔTc of CFT higher than for ST by (4.0, 4.2, 4.4, 4.8, 5.2) times for Nf(8,10,12,16and 20) respectively at mc 0.0164 kg/s. The ΔTh of CFT higher than for ST by (4.0, 4.1, 4.2, 4.3, 4.4) times for Nf(8,10,12,16and 20) respectively at mc 0.0219 kg/s. This is because of increasing the surface areas of heat transfer.

B. Effect of Reh on ΔTh at different number of fin (Nf)

The variation of ΔT_h with the Re_h for ST and CFT with three perforations with different Nf (8, 10, 12,16and 20) illustrated in figures (10 to 15). It can be noted that (ΔT_h) for CFT is higher than for ST Because of the surface area of the fins are increasing. The ΔT_h of CFT higher than for ST by (4.8, 5.2, 5.5, 6.0, 6.5) time for Nf(8,10,12,16and 20) respectively at mc 0.0109 kg/s.as well as (3.1, 3.4, 3.5, 3.7, 4.0) times for Nf(8,10,12,16and 20) respectively at mc 0.0164 kg/s.The ΔT_h of CFT is higher than for ST by (3.0, 3.3, 3.4, 3.6, 3.9) times for Nf(8,10,12,16and 20) respectively at mc 0.0219 kg/s. The result show that when the mc increased the ΔT_h is decreased because the time for heat transfer is not long enough.

C. Effect of Reh on hot water heat transfer coefficient

Figures (16) to (18) shown the variation of the hot water heat transfer coefficient (hi)with Re_h for CFT with different Nf (8,10,12,16and 20) at (mc) rate equal to (0.0109, 0.0164, 0.0219)kg/s. It can be observed that (hi) for CFT is higher hen compared to that of ST. This can be interpreted as using the number fins will increase the surface area of heat exchange and then increase the heat transfer process. The hi of CFT higher than for ST by (9.4, 10.6, 11.3, 12.4, 13.1) times for Nf(8,10,12,16and 20) respectively at mc 0.0109 kg/s, (6.3, 6.9, 7.3, 7.9, 8.3) times for Nf(8,10,12,16and 20) respectively at mc 0.0164 kg/s and (6.2, 6.8, 7.2, 7.8, 8.1) times for Nf (8,10,12,16and 20) respectively at mc 0.0219 kg/s.

D. Effect of Reh with hot water side Nusselt's number

Figures (19) to (21) illustrate the variation of the hot water side Nusselt's number (N_{uh}) with Re_h for (CFT) with different Nf (8, 10, 12,16and 20) at (mc) rate (0.0109, 0.0164, 0.0219)kg/s. It can be noted that (N_{uh}) for CFT is higher than for ST because of the Nusselt number is significantly relaying on the thermal boundary condition and the geometry passage in laminar flow. The hi of CFT higher than for ST by (9.5, 10.6, 11.4, 12.5, 13.2) times for Nf(8,10,12,16and 20) respectively at mc 0.0109 kg/s, (6.2, 6.9, 7.3, 7.9, 8.4) times for Nf(8,10,12,16and 20) respectively at mc 0.0164 kg/s and (6.3, 6.8, 7.2, 7.8, 8.1) times for Nf(8,10,12,16and 20) respectively at mc 0.0219 kg/s.

E. Effect of Reh on overall heat transfer coefficient (U)

Figures (22) to (24) illustrate the variation of U with the (Re_h) for (CFT) with different Nf (8, 10, 12,16and 20) at (mc) rate (0.0109, 0.0164, 0.0219) kg/s. It can be noted that (U) for CFT is higher than for ST. This is because U depends on several parameters such as the geometry of heat transfer surface, thermal properties, individual Nu (as a function of relevant parameters), temperature variations, temperature difference variations…etc. The U of CFT higher than for ST by (7.4, 8.7, 9.6, 11.4, 13.7) times for Nf(8,10,12,16and 20) respectively at mc 0.0109 kg/s, (4.3, 4.8, 5.2, 6.0, 6.8) times for Nf(8,10,12,16and 20) respectively at mc 0.0164 kg/s and (4.2, 4.6, 5.1, 5.7, 6.4) times for Nf (8,10,12,16and 20) respectively at mc 0.0219 kg/s.

F. Effect of number of fin on pressure drop (Δp) at deferent cold mass flow

The variation of Nf on pressure drop at different cold mass flow rate (0.0109, 0.0164, 0.0219) kg/s illustrated in fig. (25). It can be noted that water Δp is proportional to Nf and mc. The Δp of CFT higher than for ST by (1.5, 1.9, 2.2, 3.0, 3.7) times for Nf(8,10,12,16and 20) respectively at mc 0.0109 kg/s. The Δp of CFT higher than for ST by (1.6 , 2.0, 2.4, 3.2, 4.1) times for Nf(8,10,12,16and 20) respectively at mc 0.0164kg/s. The Δp of CFT higher than for ST by (1.8, 2.3, 2.7, 3.6, 4.6)times for Nf(8,10,12,16and 20) respectively at mc 0.0219 kg/s. The loss in pressure can be explained as the increase of fin numbers will lead to the increase of fluid frictions with fins.

G. Effect of effectiveness with number of transfer units (NTU)

Figures (26) to (31) shows the change of effectiveness at NTU. From these figures, a direct relationship can be seen. This can be attributed to the dependency of the effectiveness and NUT on U. Effectiveness of CFT at different Nf (8, 10, 12, 16 and 20) is greater than that of ST by (28%, 39%, 45%, 54%, 61%) respectively. The NTU is a compound measure of the heat exchanger size obtained by multiplying the HT surface area by the overall heat transfer coefficient U. Therefore, the effectiveness is improved with the increase of NTU because the number of fins is increased and as a result the heat transfer surface area is increased.

H. Effect of Reh on Enhancement Ratio Factor for different number of fin

Figures (32) to (34) illustrate the variation of Reh on Enhancement Ratio Factor (EF) for (CFT) with different Nf (8, 10, 12,16and 20) at (mc) rate (0.0109, 0.0164, 0.0219) kg/s. It can be noted that EF for CFT is higher than for ST due to the improvement of heat transfer resulting from the extension of heat exchanger surface area. The EF of CFT higher than for ST by (9.2, 10.2, 11.1, 12.1 13.1) times for Nf(8,10,12,16and 20) respectively at mc 0.0109 kg/s, (8.2, 8.8, 9.3, 10.2, 10.8) times for Nf(8,10,12,16and 20) respectively at mc 0.0164 kg/s and (8.1, 8.6, 8.9, 9.2, 9.3) times for Nf(8,10,12,16and 20) respectively at mc 0.0219 kg/s.

V. CONCLUSIONS

The current research study has come up with the following conclusions:

1. The augmentation of heat transfer is significant when the largest number of fins is placed on outer surface of inner tube. This enhancement is seen clearly in heat transfer coefficient indicating (6.2 to 13.1) times than that of smooth tube.
2. The hot water side temperature difference changes directly with number of fins and decreased with increasing the cold water mass flow rate.
3. Pressure drop is directly proportional to number of fins and the cold water mass flow rate.
4. The effectiveness is directly proportional to number of fins increasing (28% to 61%) with number of fins and with

WJES Wasit Journal of Engineering Sciences VOLUME: (6), NO.: (3) 2018

number of transfer units.

5. Enhancement Ratio Factor proportional to number of fins by (8.1 to 13.1) times and decreased with increasing the cold water mass flow rate.

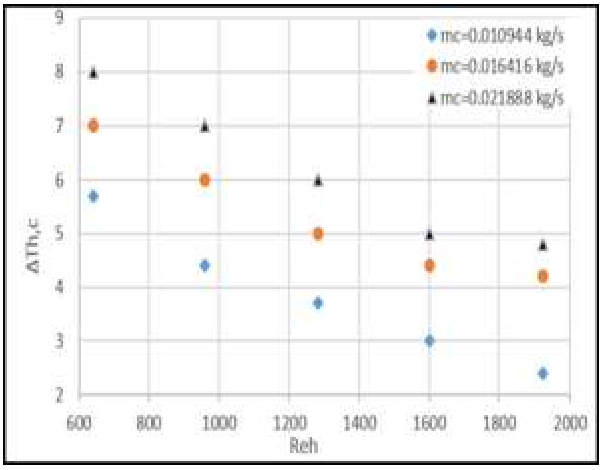

Fig.4. Effect of Reh on ΔT_h from smooth tube at different cold masses flow.

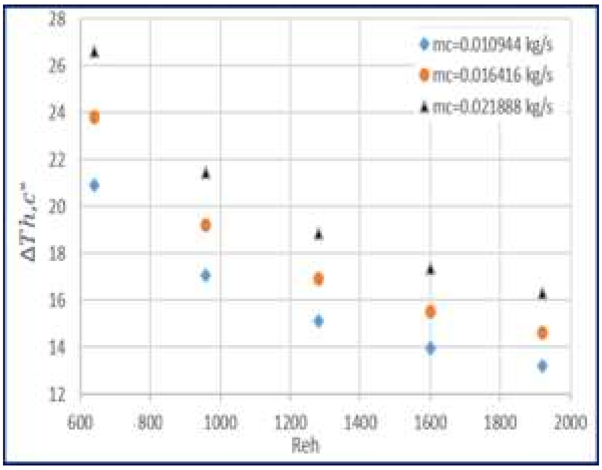

Fig.7 Effect of Reh on ΔT_h for number of =12 at different cold masses flow .

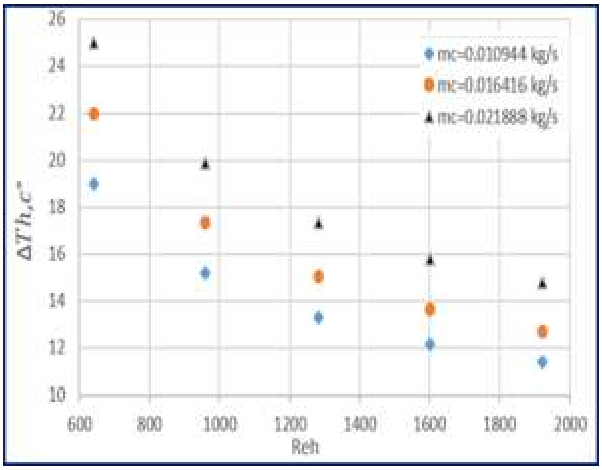

Fig. 5 Effect of Reh on ΔT_h from number of =8 at different cold masses flow.

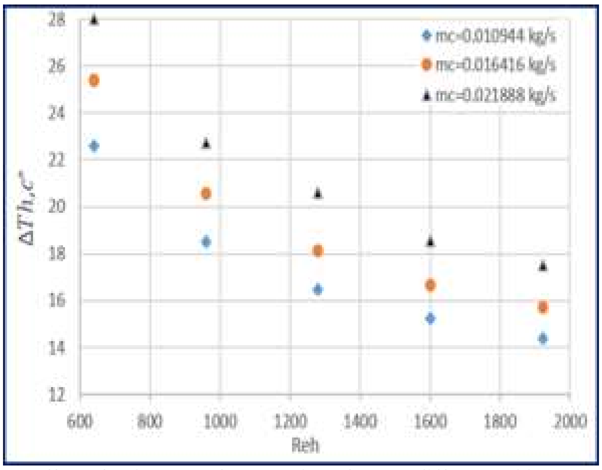

Fig. 8 Effect of Reh on ΔT_h from number of =16 at different cold masses flow.

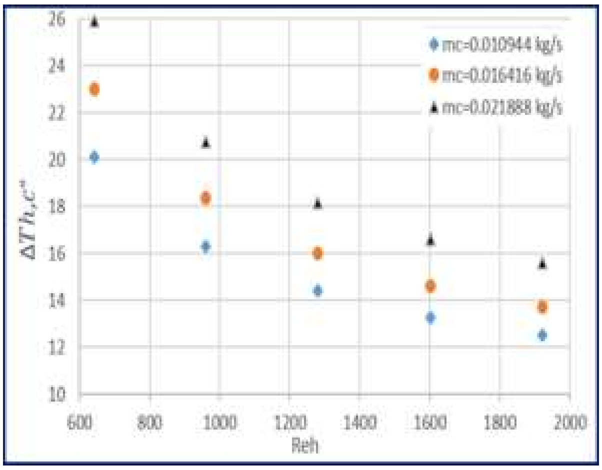

Fig. 6 Effect of Reh on ΔT_h from number of =10 at different cold masses flow.

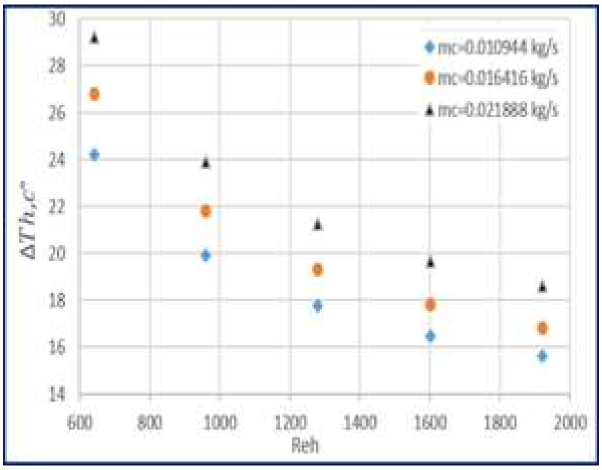

Fig. 9 Effect of Reh on ΔT_h from number of =20 at different cold masses flow.

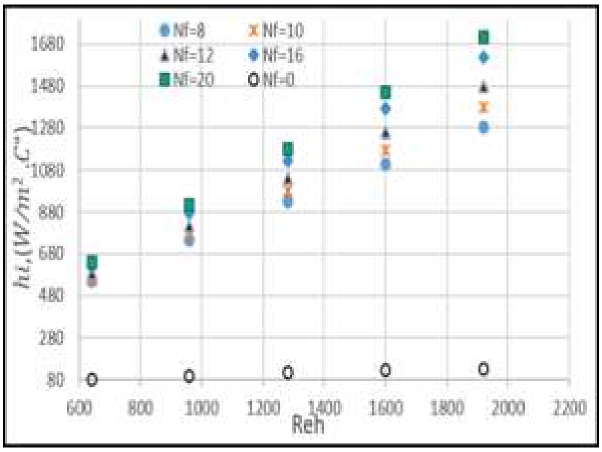

Fig. 16 Effect of Reh on hot water heat transfer coefficient at mc =0.0109 kg/s

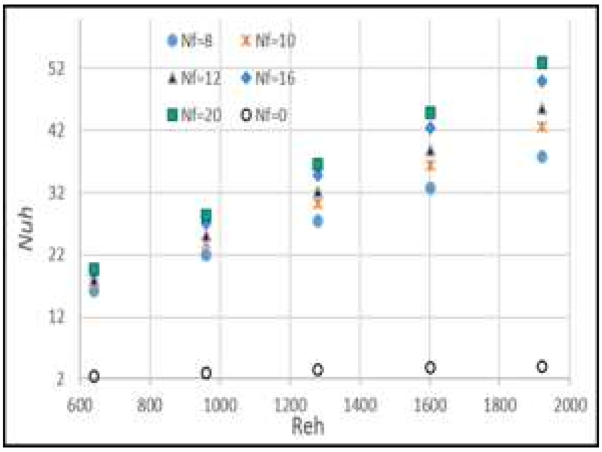

Fig. 19 Variation of Reh with hot water side Nusselt's number at cold mass flow m_c =0.0109 kg/s

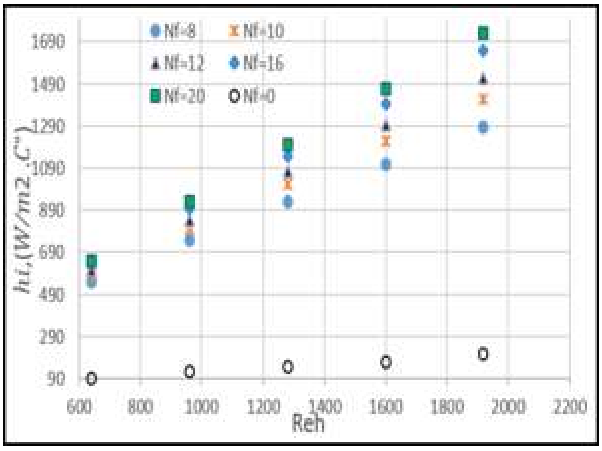

Fig. 17 Effect of Reh on hot water heat transfer coefficient at m_c =0.0164 kg/s

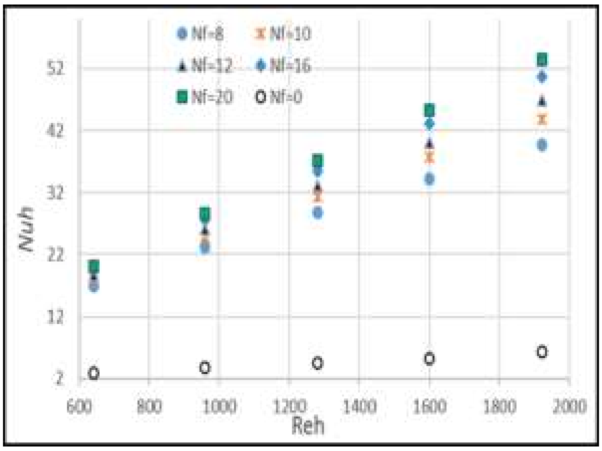

Fig. 20 Variation of Reh with hot water side Nusselts at cold mass flow m_c =0.0164 kg/s

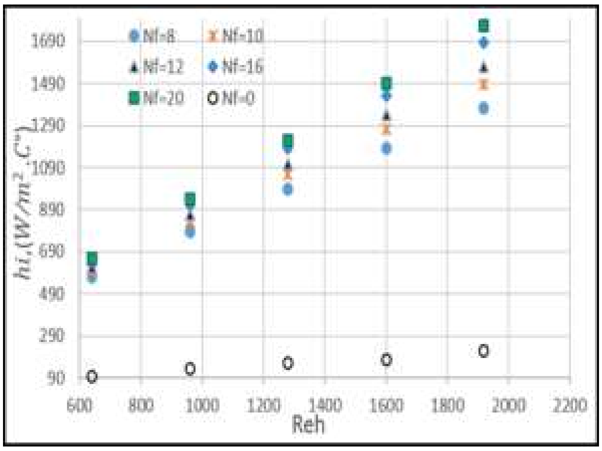

Fig. 18 Effect of Reh on hot water heat transfer coefficient at m_c =0.0219 kg/s

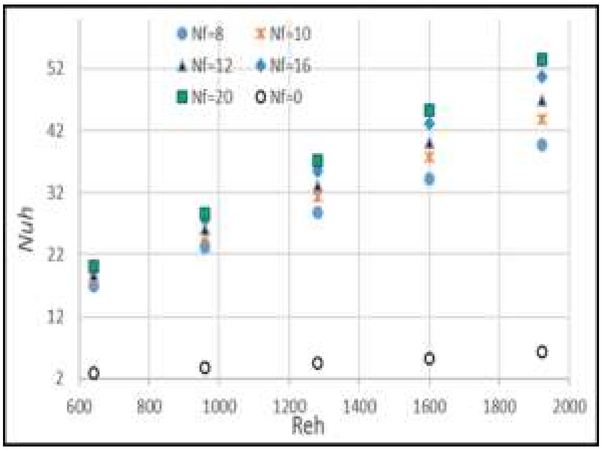

Fig. 21 Variation of Reh with hot water side Nusselts at cold mass flow m_c =0.0219 kg/s

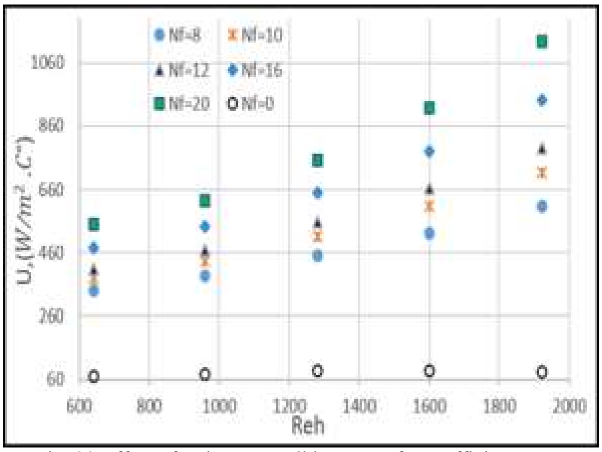

Fig. 22 Effect of Reh on overall heat transfer coefficient at mc =0.0109 kg/s

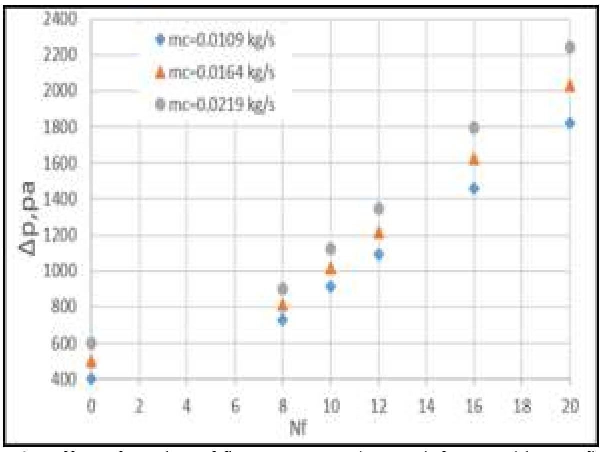

Fig. 25 Effect of number of fin on pressure drop at deferent cold mass flow

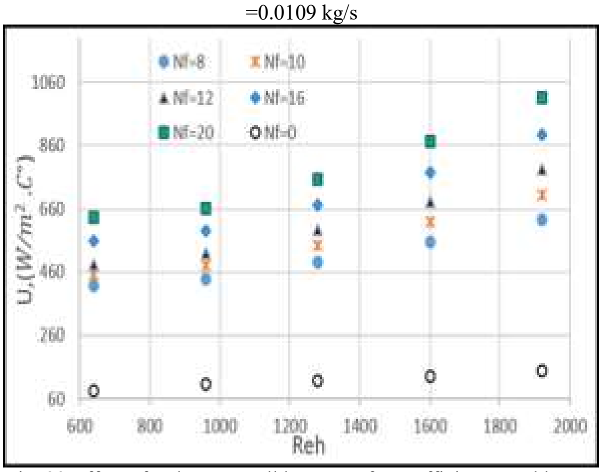

Fig. 23 Effect of Reh on overall heat transfer coefficient at cold mass flow mc =0.0164 kg/s

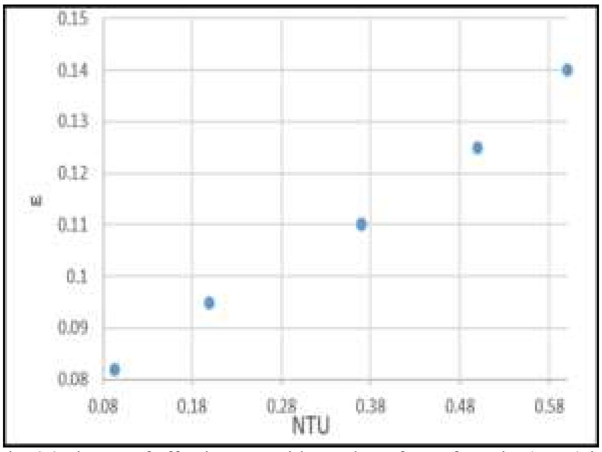

Fig. 26 Change of effectiveness with number of transfer units (NTU) in smooth tube at cold mass flow mc =0.0109 kg/s

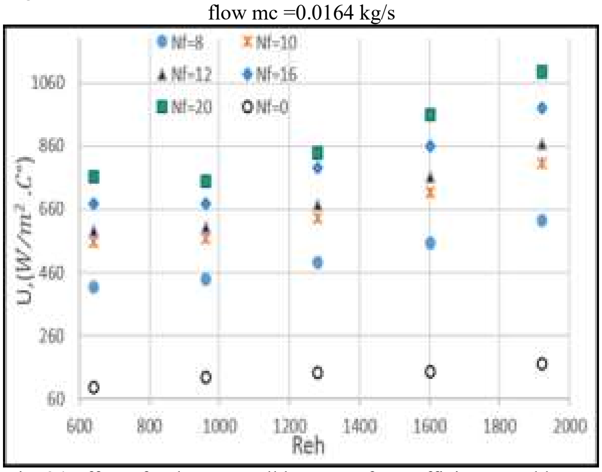

Fig. 24 Effect of Reh on overall heat transfer coefficient at cold mass flow mc =0.0219 kg/s

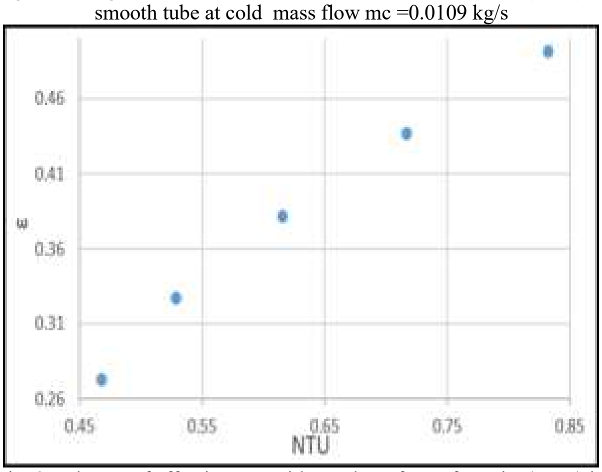

Fig. 27 Change of effectiveness with number of transfer units (NTU) in finned tube with 8 fin at cold mass flow mc =0.0109 kg/s

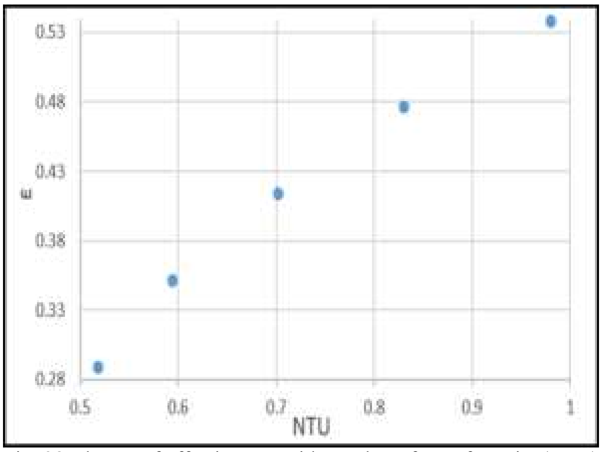

Fig. 28 Change of effectiveness with number of transfer units (NTU) in finned tube with 10 fin at m_c =0.0109 kg/s

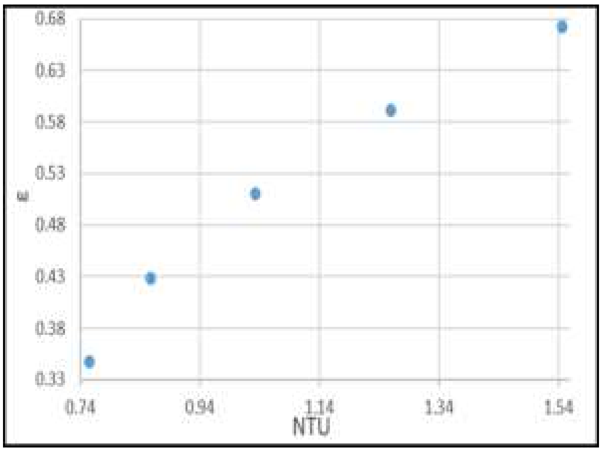

Fig. 31 Change of effectiveness with number of transfer units (NTU) in finned tube with 20 fin at cold mass flow m_c =0.0109 kg/s

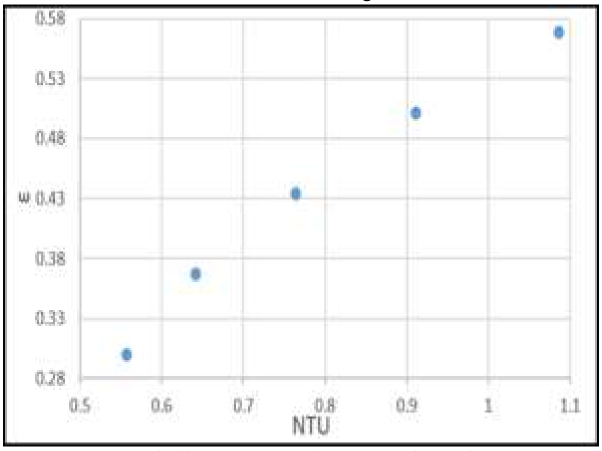

Fig. 29 Change of effectiveness with number of transfer units (NTU) in finned tube with 12 fin at cold mass flow mc =0.0109 kg/s

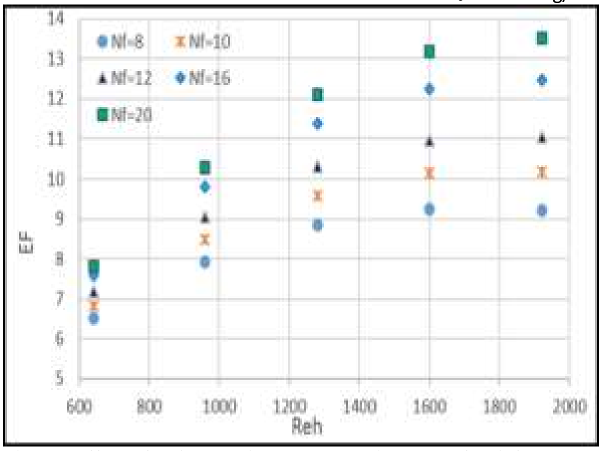

Fig. 32 Effect of Reh on Enhancement Ratio Factor for deferent number of fin at mc =0.0109kg/s

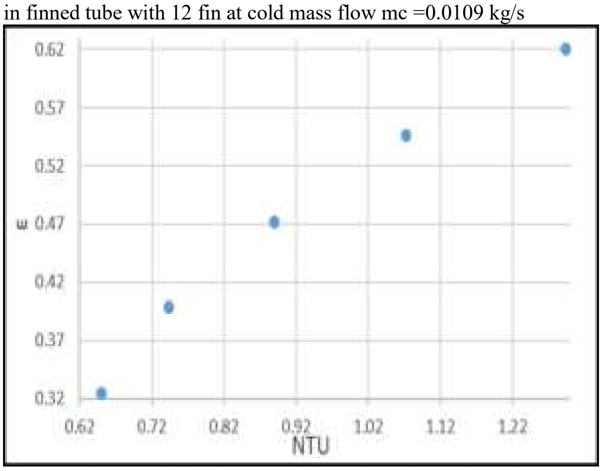

Fig. 30 Change of effectiveness with number of transfer units (NTU) in finned tube with 16 fin at m_c =0.0109 kg/s

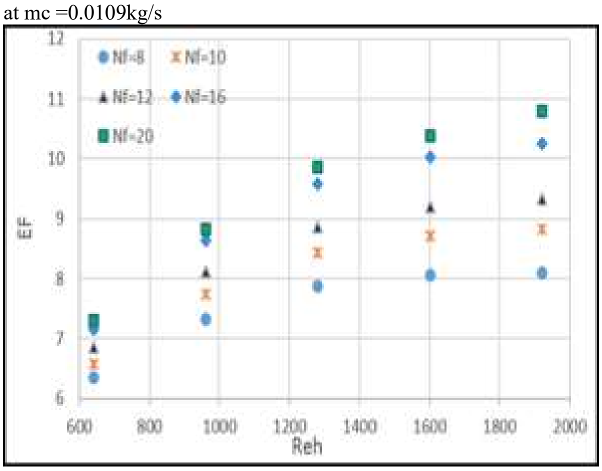

Fig. 33 Effect of Reh on Enhancement Ratio Factor for deferent number of fin at m_c =0.0164 kg/s

Wasit Journal of Engineering Sciences VOLUME: (6), NO. : (3) 2018

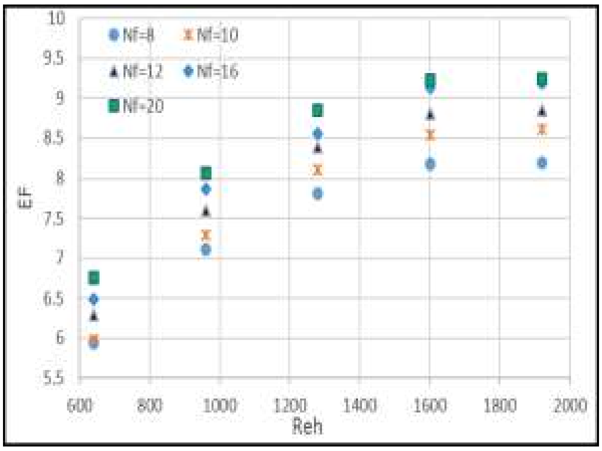

Fig.34 Effect of Reh on Enhancement Ratio Factor for deferent number of fin at cold mass flow m_c =0.0219kg/s

Abbreviations

HT Heat transfer
ST Smooth Tube
CFT Circular Finned Tube
ΔTh hot water side temperature difference
Nf Number of fin
Δp pressure drop
mc cold water mass flow
OD outer diameter
ID inner diameter

Nomenclature

A Area (m^2)
Ac Cross section area of annuli (m^2)
Ao outer surface area of tube (m^2)
Ai inner surface area of tube (m^2)
C Heat capacity W/C°
Cp Specific heat $J/kg.C^{\circ}$
Di inner diameter of annuli (m)
De Equivalent diameter of annuli (m)
D_{hu} Hydraulic diameter of annuli (m)
h Heat transfer coefficient $W/m^2.C^{\circ}$
k Thermal conductivity $W/m.C^{\circ}$
L Length (m)
ṁ Mass flow rate kg/s
ṁc Mass flow rate cold water kg/s
ṁh Mass flow rate hot water kg/s
Nu Nusselt's number

P_h Perimeter (m)
Q Heat dissipation W
Qc Heat dissipation at cold side W
Qh Heat dissipation at hot side W
Re Reynold's number
T Temperature C°
Tm Mean temperature C°
Ts surface temperature C°
Tci Inlet temperature of cold water C°
Tco Outlet temperature of cold water C°
Thi Inlet temperature of hot water C°
Tho Outlet temperature of hot water C°
ΔT Temperature difference
U Overall heat transfer coefficient $W/m^2.C^{\circ}$
V Velocity m/s
Greek symbols
ε Heat exchanger effectiveness
δ Fin thickness (m)
μ Dynamic viscosity $kg/m.s$
ρ Density kg/m^3

REFERENCES

1. Jin, Y.; Tang, G.H.; He, Y.L.; Tao, W.Q." Parametric study and field synergy principle analysis of H-type finned tube bank with 10 rows". Int. J. Heat Mass. Tran. 2013, 60, 241–251.
2. Zhang, J.W.; Zhang, Z. A numerical study on fully developed fluid flow and heat transfer in a spiral finned tube. Chin. J. Chem. Eng. 1999, 7, 56–66. (In Chinese)

3. Li, L.J.; Cui, W.Z.; Liao, Q.; Xin, M.D.; Jen, T.C.; Chen, Q.H. Heat transfer augmentation in 3D internally finned and microfinned helical tube. Int. J. Heat Mass Tran. 2005, 48, 1916–1925.
4. Wang, C.C.; Chi, K.Y. Heat transfer and friction characteristics of plain fin-and-tube heat exchangers, part I: New experimental data. Int. J. Heat Mass Tran. 2000, 43, 2681–2691.
5. Hasan, A.; Siren, K. Performance investigation of plain and finned tube evaporatively cooled heat exchangers. Appl. Therm. Eng. 2003, 23, 325–340.
6. Kim, Y.; Kim, Y. Heat transfer characteristics of flat plate finned-tube heat exchangers with large fin pitch. Int. J. Refrig. 2005, 28, 851–858.
7. Choi, J.M.; Kim, Y.; Lee, M.; Kim, Y. Air side heat transfer coefficients of discrete plate finned-tube heat exchangers with large fin pitch. Appl. Therm. Eng. 2010, 30, 174–180.
8. Lemouedda, A.; Schmid, A.; Franz, E.; Breuer, M.; Delgado, A. Numerical investigations for the optimization of serrated finned-tube heat exchangers. Appl. Therm. Eng. 2011, 31, 1393–1401.
9. P. K. Trivedi, N. B.Vasava "Effect of Variation in Pitch of Tube on Heat Transfer Rate in Automobile Radiator by CED Analysis", International Journal of Engineering and Advanced Technology (IJEAT) ISSN: 2249 – 8958, Volume-1, Issue-6, August 2012.
10. Webb, R. L., "Air-Side Heat Transfer in Finned Tube Heat Exchanger," Heat Transfer Engineering, Vol. 1, No. 3, pp. 33-49, 1980.
11. A. Antony, M. Ganesan. Flow analysis and characteristics comparison of double pipe heat exchanger using enhanced tubes. Journal of Mechanical and Civil Engineering, 2014,7: 16–21.
12. D. Kadam. Performance simulation of fin and tube heat exchanger. International journal of educational science and research, 2012, 2(10).
13. B. Kundu, P. Das. Performance analysis and optimization of straight taper fins with variable heat transfer coefficient. International journal of heat and mass transfer, 2002, 45(24): 4739–4751.
14. M. Mon, U. Gross. Numerical study of fin spacing effects in annular finned tube heat exchanger. International Journal of Heat and Mass Transfer, 2004, 47(8): 1953–1964.
15.Aghareed M.Sfaih," Experimental and Theoritical of Optimum Design of Heat Exchanger". Wasit University M.S. Thesis, 2016**.**

16. Frank P. Incropera, David P. Dewitt, Theodore L. Bergman, Adrienne's S. Lavine, ″Introduction to Heat Transfer″, John Wiley & Sons Inc., 2007.
17. Sadik Kakac, Hongtan Liu, ″Heat Exchangers Selection, Rating and Thermal Design″, Second Edition, Dept. of Mech. Eng. Univ. of Miami, Coral Gables, Florida, 2002.
18. Sahiti N., Durst F., Dewan A., ″Heat Transfer Enhancement by Pin Elements″, International Journal of Heat and Mass Transfer, Vol.48, pp. (4738–4747), 2005.
19. Stevanovic, Gradimir, I., Radojkovic, N., and Vuckovic, G., "*Design of Shell & Tube Heat Exchangers by Using CFD Technique*", Facata Universitatis, Vol.1, No.8, pp.1091-1105, 2001.