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Improvement of Heat Transfer Coefficient for A Cross Flow Heat Exchanger by Using the High Integral Finned Tube

Farah Abdulzahra Taher **Zena Khalefa Kadhim**

Mechanical Engineering Department, University of Wasit, Wasit, Iraq

Ftahar@uowasit.edu.iq drzena@uowasit.edu.iq

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Abstract:

This study investigated experimentally of the heat transfer and the effect of fins in improving heat transfer for the cross flow heat exchangers with eight passes (smooth and high integral finned tubes) and that are cooled by air. Two models of heat exchangers are designed and manufactured for testing from pure copper metal. They were designed for comparison under different test conditions. The water flow rate in the inner tubes with (2, 3, 4, 5, 6) L/min with inlet temperatures of (50, 60, 70)ºC. The air was passed outer the tubes with speeds (1, 1.7 and 2.3) m/s. This study resulted that the high integral finned tube was more improvement the heat transfer than the smooth tube. The enhancement of (Q) was (329.9%) for high integral finned tube and The enhancment factor (291%).

Keywords: Heat transfer**,** Cross flow heat exchangers**,** High integral finned tube, Heat Transfer Coefficient, Improvement, Pure Copper.

Symbols

Symbols		Unit
Ai	inner surface area of tube	m^2
Ao	outer surface area of tube	m^2
C_p	Specific heat of the fluid	kJ/kg.°C
hi	inner side heat transfer coefficient	W/m^2.°C
H_f	fin height	Mm
h_o	Air side heat transfer coefficient	W/m^2.°C
K	Thermal conductivity of tube material	W/m.°C
K_w	Thermal conductivity of water	W/m.°C
m	mass flow rate	kg/s
N_f	Number of fins	Fpm
Nu_a	Air side Nesselt number	-
NTU	Number of the heat transfer units	-
Q	heat transfer rate	Watt
Re_a	Air side Reynolds number	-
S_f	Fin space	Mm
T	Temperature	°C
t_h	water temperature	°C
T_f	fin thickness	Mm
Δt_c	Air side temperature difference	°C
t_m	Mean temperature	°C
t_s	Surface temperature	°C
T_w	wall thickness	Mm
Ui	Inner side overall heat transfer coefficient	W/m^2.°C
U_{air}	Velocity of air	m/s
θ	temperature difference	°C
ε	The exchanger effectiveness	-
ρ_a	density of air	Kg/m^3
ν_a	kinetic viscosity of water	m^2/s
HIF	High integral finned tube	

1. Introduction

The heat exchanger is a device, used to exchange heat between two liquids, without mixing between them .The increasing demand for a heat exchanger with good specifications in heat transfer because their Necessary in applications, such as, automobiles, ventilating, electronic chip cooling, heating, refrigeration, ecological building, air conditioning systems and power generation, the air conditioning unit improvement reliant on the execution of its constituent devices as heat exchangers, compressors and fans, as well as this improvement will decrease of electrical power consumption [1-3]. The integral finned tubes are tubes that can be formed from the tube outside surface by using the ring fins to improve condensation in the design of surface condensers in the refrigeration and steam turbine. Main reason why integral finned tubes are enhanced over smooth tubes is because of the added surface area by the fins that increases the area for heat transfer as given by figure [4]. Pressure meter, pressure sensors, Thermocouples and digital reader were calibrated have been calibrated in [Central Organization for Standard and Quality Control] as in table (1)

Measuring device	Error ratio %
Pressure meter, pressure sensors	0 - 0.6
Thermocouples and digital reader	0.31 – 0.38
Flow meter	1 - 2

The literature survey will be divided into the following:

Chaudhari et. al. [2], studied the influence of the round fin from aluminum on heat transfer coefficient and pressure drop on finned tube and without finned tube heat exchanger (HE) is exploratory and numerical examination by The heat transfer coefficient for the coolant water by utilization of round about finned tube and without fin tube exchanger with force convection, From the test of finned tube heat exchanger, the result show that the heat transfer coefficient (14.07 W/m^2K) , as well as the overall heat transfer rate of finned tube is more marked than without finned tube heat exchanger and also heat transfer rate is increases. **Kumar et. al. [5]**, presented a trial examination to expand heat transfer rate by means of raising heat transfer coefficient to working liquid as unadulterated vapor steam, R134a and R12.The tubes and fins produced using copper. The examination center for smooth tubes and impact of various integral finned tube, for example, [circular integral finned tubes, and somewhat spine roundabout integral finned tubes and spine integral finned tubes]. The outcome demonstrates that the spine integral finned tubes gave greatest heat transfer coefficient. **Dasgupta [6],** The heat transfer rate through an air to the deionized water cross stream serpentine small scale channel heat exchanger through air cooling in the single stage, under working conditions were kept up inside the scope of air and deionized water side Reynolds number ($283 \leq Rea \leq 1384$) and ($105 \leq Re_w \leq 159$) separately. holding the steady entry temperatures of the two liquids, water temperature (38 ±0.5)°C and air temperature (9 ±0.5)°C. The result show that the heat transfer rate from hot air to the cool de-ionized water under 4%, air side (Re_a) marked effect on heat transfer great than water (Re_w) ,air side (Nu_a) increases with air (Re_a) increases, the efficiency(ε) and (NTU) increase with (Rea) non directly and were observed that to be higher at decreases (Re_w) for a specific (Re_a), on the base that is the prevalent component for (ε) and (NTU). **Zhang et. al. [7],** A practical study to perform heat transfer of exchanger a helically baffled single tube heat. The test consists of a smooth tube and five petal-shaped fin tubes (PF). from the work concluded that Nusselt number of the five tubes (PF) increased than a smooth tube, as well as increased with the increase of fins and a decrease of fin pitch by 233% from the smooth tube. These show that the use of (PF) is better than the smooth tube. **Mohsin Jani et. al. [8],** a three dimensional numerical and experimental examination to the cross flow heat exchangers (Smooth and Low integral finned tube), Number of passes are eight, of utilizing the difference working fluids as water, oil without and with nanofluid (SiO_2) from where their effect on the heat transfer, heat transfer coefficient and effect of the concentration Nano fluid on them. The results show that the heat transfer, heat transfer coefficient and the enhancement in general higher in low integral finned tube than the smooth tube. The enhancement was 72.05 % for oil and 104.1% for water. **Chen et. al. [9]**, A practical study of the heat transfer characteristics of the 3-D finned tube bundle of different shapes from Carbon steel and their comparison with previous studies. The heat exchanger of the 3-D finned tube bundle has the highest heat transfer and lower drop pressure.

2. The Aim of the study

This study aims to improve the heat transfer coefficient for cross flow heat exchanger by using high integral finned tube.

3. Scope of the study

Manufacture heat exchangers cross flow (smooth tube and high integral finned tube). Study the effect of various water stream rate, air speed and inlet water temperature on heat transfer coefficient for them. Find the cases, which enhanced heat transfer for various ranges of air and water as well as inlet temperatures and the speed at the entrance. Develop the empirical correlations for (Nua) of smooth and high integral finned tubes as function of (Pra) and (Rea).

4. Theoritical equations

The energy balance in heat exchanger total heat transfer rate in heat exchanger. [10]

$$Q = \dot{m}_w C_{ph}(t_{h1} - t_{h2}) = \dot{m}_a C_{pc}(t_{c2} - T_{c1}) \tag{1}$$

$$Q = UA\theta_m \tag{2}$$

The overall heat transfer coefficient [11]

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{r_i}{k}\ln({}^{r_o}/_{r_i}) + ({}^{r_o}/_{r_i})\frac{1}{h_o}} \tag{3}$$

$$U_o = \frac{1}{({}^{r_o}/_{r_i})\frac{1}{h_i} + \frac{r_o}{k}\ln({}^{r_o}/_{r_i}) + \frac{1}{h_o}} \tag{4}$$

Where,

$A_i = 2\pi r_i KL$, $A_o = 2\pi r_o KL$

Log mean temperature difference (LMTD) [11]

$$LMTD = \frac{\theta_1 - \theta_2}{\ln\left({}^{\theta_1}/_{\theta_2}\right)} \tag{5}$$

Correction factor [12and 13]

$$F = \frac{\sqrt{(R^2+1)}\,\ln\left[\frac{(1-a)}{(1-Ra)}\right]}{(R-1)\ln\left[\frac{2-a\left\{R+1-\sqrt{(R^2+1)}\right\}}{2-a\left\{R+1+\sqrt{(R^2+1)}\right\}}\right]} \tag{6}$$

Where,

$$R = \frac{t_{h1} - t_{h2}}{t_{c2} - t_{c1}} = \frac{\text{Tube side temperature difference}}{\text{Air side temperature difference}} \tag{7}$$

$$a = \frac{t_{c2} - t_{c1}}{t_{h1} - t_{c1}} = \frac{\text{Air side temperature difference}}{\text{maximum possible temperature difference}} \tag{8}$$

$$\theta_m = F\ LMTD \tag{9}$$

Calculations heat transfer coefficient for smooth tube, [14]

For smooth tube

$$h_o = \frac{1}{\frac{1}{U_o} - \frac{d_o \ln\left({}^{d_o}/_{d_i}\right)}{2k} - \frac{d_o}{h_i d_i}} \tag{10}$$

For integral finned tubes

$A_{of} = A_{os} = \pi\, d_o L$, Clean surfaces, [15 and 16].

From eq.(3) we get h_o

$$h_o = \frac{1}{\frac{1}{U_i} - \frac{d_i \ln\left({}^{d_r}/_{d_i}\right)}{2K} - \frac{1}{h_i}} \tag{11}$$

Reynolds number for air side and for water[17]

$$Re_a = \frac{U_{air} d_h}{v_a} \tag{12}$$

$$Re_w = \frac{u_w\, d_i}{v_w} \tag{13}$$

Where, $d_h = \frac{4\times A_C}{P}$, $A_c = W \times H$, $P = (W + H) \times 2$

Prandtle number for air side and inside of tube:

$$Pr_a = \frac{\mu_a\, Cp_a}{K_a} \quad \text{and } Pr_w = \frac{\mu_w\, C_{pw}}{K_w} \tag{14}$$

Nusslte's number for air and water side:

$$Nu_a = \frac{h_o\, d_o}{K_a} \quad \text{and} \quad Nu_w = \frac{h_i \times d_i}{K_w} \tag{15}$$

To turbulent flow by Dittus and Boelter, [18]:

$$Nu_w = 0.023 {Re_w}^{0.8} {Pr_w}^{n} \tag{16}$$

($0.6 < Pr < 100$) , For cooling process n= 0.3

The actual heat transfer for the counter flow exchanger [11]

$$Q_{act} = \overset{\circ}{m}_w cp_h (t_{h1} - t_{h2}) = \overset{\circ}{m}_a cp_c (t_{c1} - t_{c2}) \tag{17}$$

$$Q_{max} = C_{min}(t_{h1} - t_{c1}) \tag{18}$$

$$C_{min} = \overset{\circ}{m} C_p \quad \text{[Minimum heat capacity of cold or hot fluid]} \tag{19}$$

For cross flow C_{max} mixed and C_{min} unmixed

$$\varepsilon = \left(\frac{1}{C}\right)\{1 - \exp[-C(1 - e^{-\mathrm{NTU}})]\} \tag{20}$$

For cross flow C_{max} unmixed, C_{min} mixed

$$\varepsilon = 1 - \exp\{-\left(\frac{1}{C}\right)[1 - exp - (\mathrm{NTU} * C)]\} \tag{21}$$

The heat capacity ratio is,

$$C = \frac{C_{min}}{C_{max}} \tag{22}$$

The number of transfer units (NTU)

$$\mathrm{NTU} = \frac{U_o * A_{os}}{C_{min}} \tag{23}$$

Enhancement factor:[19]

$$E.F\ \% = \frac{h_{o,fin} - h_{o,smooth}}{h_{o,smooth}} \times 100 \tag{24}$$

5. Experimental apparatus

A laboratory device that meets the requirements of the study, which consists of four main parts, which is designed and manufactured with the measuring equipment listed below as figure (1).

1- The test section, which made of Pyrex glass with dimensions (250×500×1200) mm.

2- Water supply unit is consisted of the following part. The pump of hot water, the water heater and the water reservoir is (250x250x400) mm made from galvanized steel as in figure (2) with measurements. The test tubes with a length of 2 meters, description of the smooth and high integral finned tube shown in table (2). The copper tubes that were joined by arrangement eight passes, the one passes are gone by horizontally through the test section with length (250) mm. The distance from slots center to other slots center (55mm) and diameter is equal to (24mm). Schematic diagram for laboratory device in figure (1 and 2), while the high integral finned tube was shown in figure (3), the fins explain in figure (4).

3- Air supply unit is passed the cold air through the following parts the centrifugal blower, the diffuser from galvanized steel and the air duct.

Table (2) description of the copper tubes

Tubes	di	do	Dr	H_f	T_f	T_w	S_f	A_o/A_i	N_f
Smooth	19	24	-	-	-	2.5	-	1.263	-
High integral	19	33.4	21	6.2	1	1	1.6	8.17	384

4- Instrument Measuring Devices are utilized to take readings from the laboratory apparatus Thermometer, hot wire anemometer, flow meter, thermometer type (tm-947sd) involve four channel with the thermocouple type (k) (-100 to 1300) °C is utilized to measure the temperature of entry and exit for water and air, the pressure meter type (ps-9302) with the domain (1-400) bar and thermal imager to measure the surface temperature of the test tubes.

5. Calculate error in experimental readings

When taking experimental readings from the laboratory apparatus, there is an error rate that comes from the accuracy of the work of the measuring instruments and to achieve the accuracy required for the measured parameters, the experimental measured repeated times three or more in order to find the uncertainty error by using the method of Klein,[20]. Calculation error were heat transfer (11.65518-15.1978)%, Overall heat transfer coefficient (2.279578-2.38847)%, Air side Reynold's number (0.174529-0.17514)% and air side heat transfer coefficient (2.950562-3.1356)%.

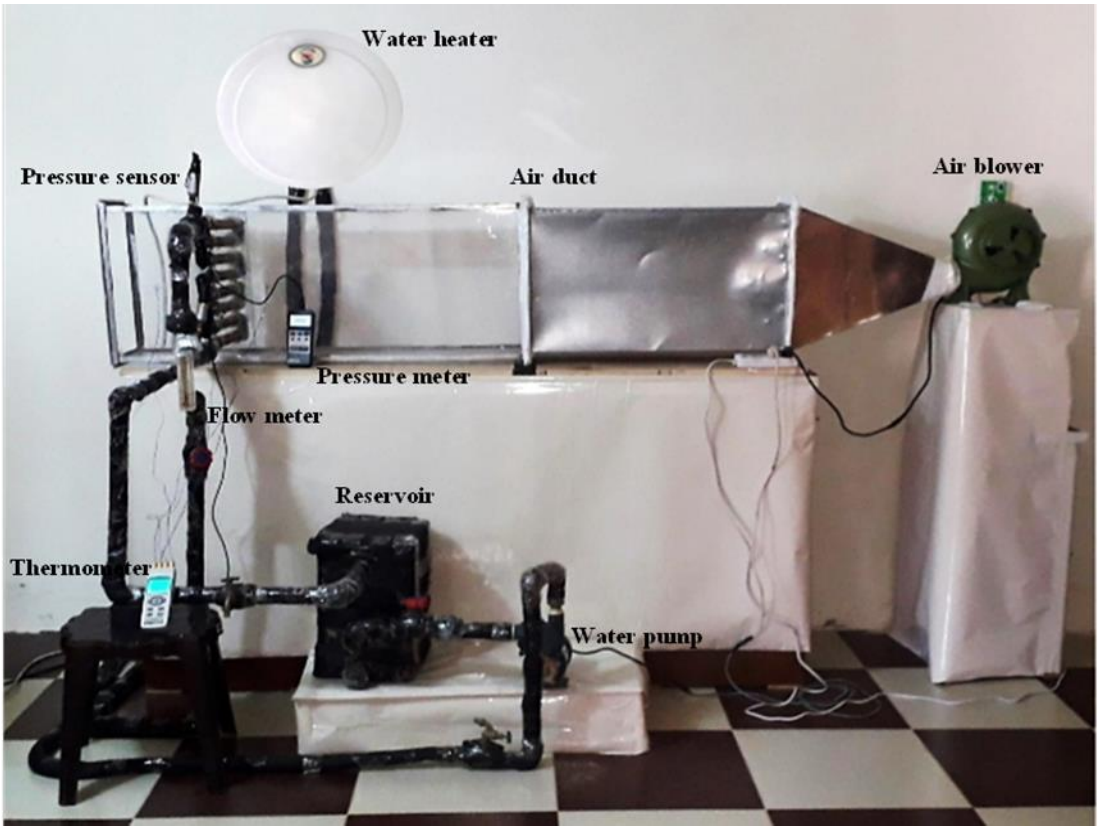

Fig. (1) Photo of the laboratory apparatus

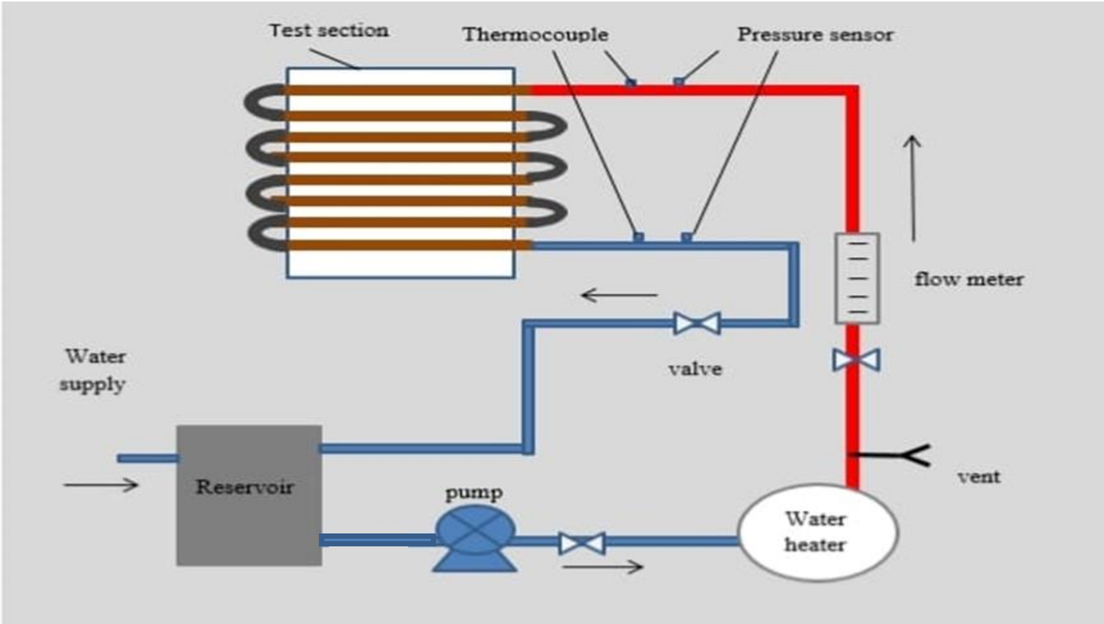

Fig. (2) Schematic diagram of hot water cycle

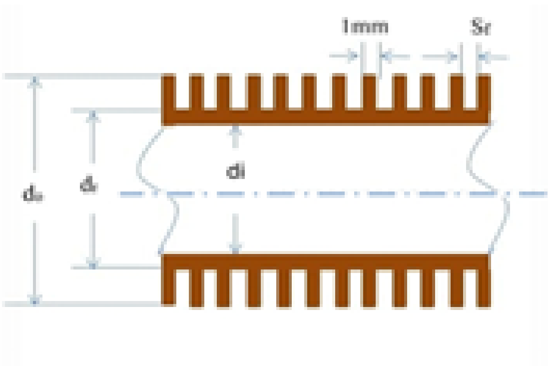

Fig. (3) Photos of the high integral finned tube

Fig. (4) Dimension of integral finned tube

6. Results and discussions

Figure (5) show the relation between the heat transfer rate with inlet water temperature and air velocity for the two heat exchangers (Smooth and High). The heat transfer rate was observed its increases with increasing the temperature of the water entering and air velocity due to high cold air speed occurred higher disturbance outside the tube that increases the water side temperature difference (Δt_h), led to rises the surface temperature and the heat capacity of water within a little value. The maximum the heat transfer rate for Smooth tube and high finned were (1474.56 and 347.95) Watt respectively. **Figure (6) and table (3)** offer the relation between the (*hi*) with volumetric flow rates at inlet water temperatures and air velocity various for smooth tube and high integral finned tube. This figure depicts the (*hi*) have the same trends which was increased with increasing the temperature of the water entering and volumetric flow rate for water at the same air speed as a result of higher disturbance for water and the effect of less thermal conductivity of water (k_w) increases with higher temperature of water. At the same boundary conditions the inside heat transfer coefficient of water for the High integral finned tube was less than the heat transfer coefficient of water for the smooth tube, that means (*hi*) of water decreases whenever increases H_f and is not improvement to water side (*hi*) to increase fins height.

Table (3) heat transfer coefficient of water for the smooth tube and high finned tube at air speed 2.3 m/sec

t_{h1}	Smooth	HIF	V_w L/min
70	2467.94	2451.62	6
70	2131.46	2117.37	5
70	1781.39	1769.31	4
70	1413.13	1402.57	3
70	1017.65	1007.58	2
60	2277.57	2263	6
60	1966.77	1954.21	5
60	1643.81	1632.64	4
60	1303.09	1294.8	3
60	938.5	929.82	2
50	2091.31	2081.5	6
50	1806.71	1798.23	5
50	1510.36	1501.68	4
50	1197.55	1190.94	3
50	862.86	856.67	2

Figures (7 and 8) carried out the effectiveness of heat exchangers with (NTU), for two cases, the effectiveness (ε) was increased with increasing the number of the transfer units (NTU), that increasing due to rise air side overall heat transfer coefficient (U_o). **Figures (9 and10),** explain that the air side temperature difference (Δt_c) with air speeds and volumetric flow rate for water. From these figures were observed that (Δt_c) decreased with increasing air speed at the same inlet water temperature and increased with increasing inlet temperature water, due to higher heat transfer rate. **Figure (11)** show the effect of the different (U_{air}) and inlet water temperatures on the (h_o) for two test models .These figures show the behavior of air heat transfer coefficient for two tested tubes which same phenomena but with different values for the (h_o).The value of (h_o) increasing with air speed at the same inlet temperature water as a result of increasing the (Re_a). At the same air speed and water flow rate with different inlet temperature was noticed that from the figures increasing (h_o) with decreasing inlet water temperature, due to the (h_o) is a function of (U_o and hi). As well as when decreases inlet water temperature therefore (hi) decreases and (U_o) increases. The (h_o) for finned tube was bigger than the heat transfer coefficient (h_o) for Smooth tube because of the fins that increase the surface area for tubes and thus increase heat transfer. The highest value of (h_o) equal 215.89W/m^2.°C in the finned model. **Figure (12) s**how the comparison of the (U_o) versus the various velocities (U_{air}) for two models. It has been observed that (U_o) increases with increasing air velocity at constant inlet temperature for models which examined, and to increasing the surface area as result as rise of high fins works to regulate the flow direction and the turbulence of air flow. The overall heat transfer coefficient which calculated based to the outer surface area proportion directly with the inner overall heat transfer coefficient(Ui). The (U_o) was decreased with increasing inlet temperature of water due to the increasing the Log mean temperature difference at the same air speed.

7. Conclusions

1. The heat transfer rate (Q) proportional with the inlet water temperature and the air speed for the two models. The heat transfer rates of high integral finned tube were bigger than of the smooth tube The maximum enhancement of (Q) at the volumetric flow rate of water 6L/min (329.9%) for high integral finned tube respectively to above the smooth tube.
2. The heat transfer coefficient of water (hi) proportional with the inlet water temperature and the volumetric flow rate (V_w) for the two heat exchangers models. The integral finned tubes have the heat transfer coefficient of water side lower than (hi) for the smooth tube.
3. The effectiveness (ε) was increased with increasing the number of the transfer units (NTU), to smooth tube and HIF (0.073 and 0.18) respectively.
4. The air side temperature difference is immediately proportional with the inlet water temperature and inversely proportional to the air speed.

5. The air side temperature difference is immediately proportional with the inlet water temperature and inversely proportional to the air speed.
6. The heat transfer coefficient (h_o) is immediately proportional with (U_{air}) and inversely with the inlet water temperature. The enhancment factor for HIF (291%).
7. The overall heat transfer coefficient of the air side (U_o) immediately proportional with (U_{air}) and inversely with the temperature (t_{h1}).
8. In this study the high integral finned tube was more empovement than smooth tube due to preseance the fins.
9. The experimental correlation obtained in the present study under the approved working conditions (inlet temperature, the water volumetric flow rate and the speed of air flow) for the case of smooth tube shown in equation (41) and table (4) for high integral finned tube.

$$Nu_a = 0.60697\, Re_a{}^{0.435137}\, Pr_a^{1/3} \qquad (25)$$

Table (4) empirical correlations for high integral finned tube

t_{h1} °C	C	m	n	R^2	($20496.54 < Re_a < 48394.26$) Air velocity range (1, 1.7, 2.3) m/sec
V_w =6L/min					
70	0.2449	0.4577	0.7661	0.945	$Nu_a = 0.2449\, Re_a^{0.4577}\, Pr_a^{1/3}\, (A_o/A_i)^{0.7661}$
60	0.1166	0.5317	0.7921	0.959	$Nu_a = 0.1166\, Re_a^{0.5317}\, Pr_a^{1/3}\, (A_o/A_i)^{0.7921}$
50	0.2279	0.4798	0.7577	0.95	$Nu_a = 0.2279\, Re_a^{0.4798}\, Pr_a^{1/3}\, (A_o/A_i)^{0.7577}$
V_w =5L/min					
70	0.5883	0.3790	0.7051	0.931	$Nu_a = 0.5883\, Re_a^{0.3790}\, Pr_a^{1/3}\, (A_o/A_i)^{0.7051}$
60	0.3773	0.4258	0.7177	0.927	$Nu_a = 0.3773\, Re_a^{0.4258}\, Pr_a^{1/3}\, (A_o/A_i)^{0.7177}$
50	0.4339	0.4214	0.7264	0.926	$Nu_a = 0.4339\, Re_a^{0.4214}\, Pr_a^{1/3}\, (A_o/A_i)^{0.7264}$
V_w =4L/min					
70	0.0740	0.5849	0.6263	0.931	$Nu_a = 0.0740\, Re_a^{0.5849}\, Pr_a^{1/3}\, (A_o/A_i)^{0.6263}$
60	0.0949	0.5638	0.6431	0.942	$Nu_a = 0.0949\, Re_a^{0.5638}\, Pr_a^{1/3}\, (A_o/A_i)^{0.6431}$
50	0.1069	0.5687	0.6100	0.957	$Nu_a = 0.1069\, Re_a^{0.5687}\, Pr_a^{1/3}\, (A_o/A_i)^{0.610}$
V_w =3L/min					
70	0.0417	0.6250	0.6806	0.951	$Nu_a = 0.0417\, Re_a^{0.6250}\, Pr_a^{1/3}\, (A_o/A_i)^{0.6806}$
60	0.0451	0.6276	0.6536	0.951	$Nu_a = 0.0451\, Re_a^{0.6276}\, Pr_a^{1/3}\, (A_o/A_i)^{0.6536}$
50	0.0434	0.6430	0.6462	0.934	$Nu_a = 0.0434\, Re_a^{0.6430}\, Pr_a^{1/3}\, (A_o/A_i)^{0.6462}$
V_w =2L/min					
70	0.1641	0.5009	0.6493	0.965	$Nu_a = 0.1641\, Re_a^{0.5009}\, Pr_a^{1/3}\, (A_o/A_i)^{0.6493}$
60	0.2481	0.4680	0.6539	0.972	$Nu_a = 0.2481\, Re_a^{0.4680}\, Pr_a^{1/3}\, (A_o/A_i)^{0.6539}$
50	0.4128	0.4123	0.7543	0.974	$Nu_a = 0.4128\, Re_a^{0.4123}\, Pr_a^{1/3}\, (A_o/A_i)^{0.7543}$

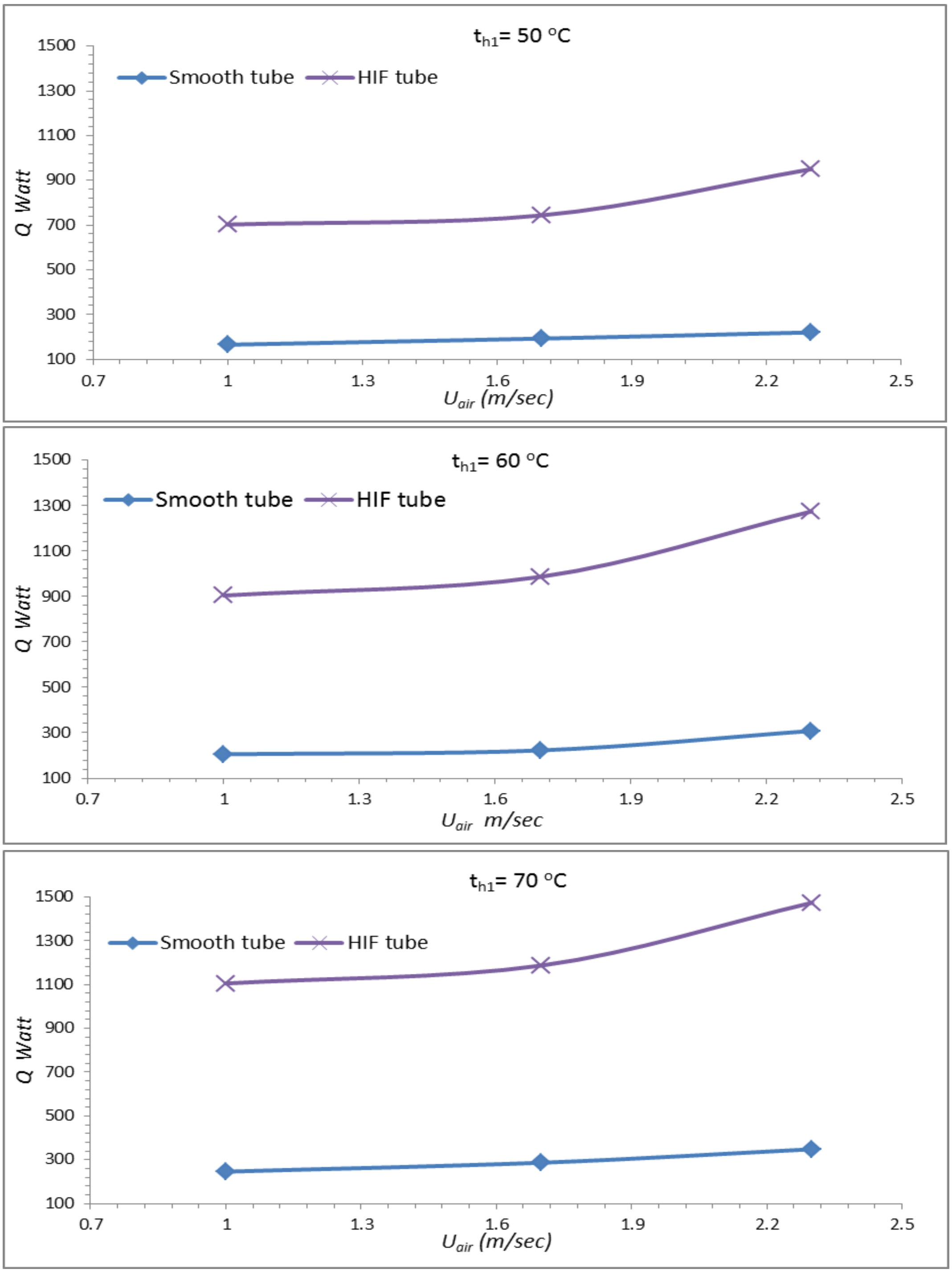

Fig. (5) The heat transfer rate with the air speed and various inlet water temperature for two models

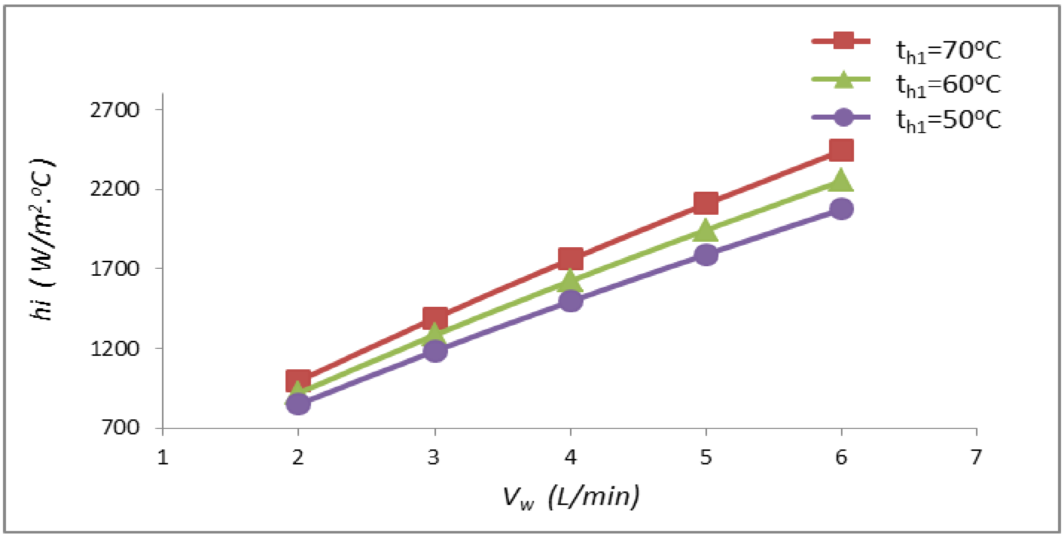

Fig. (6) Inner side heat transfer coefficient with various water flow rate

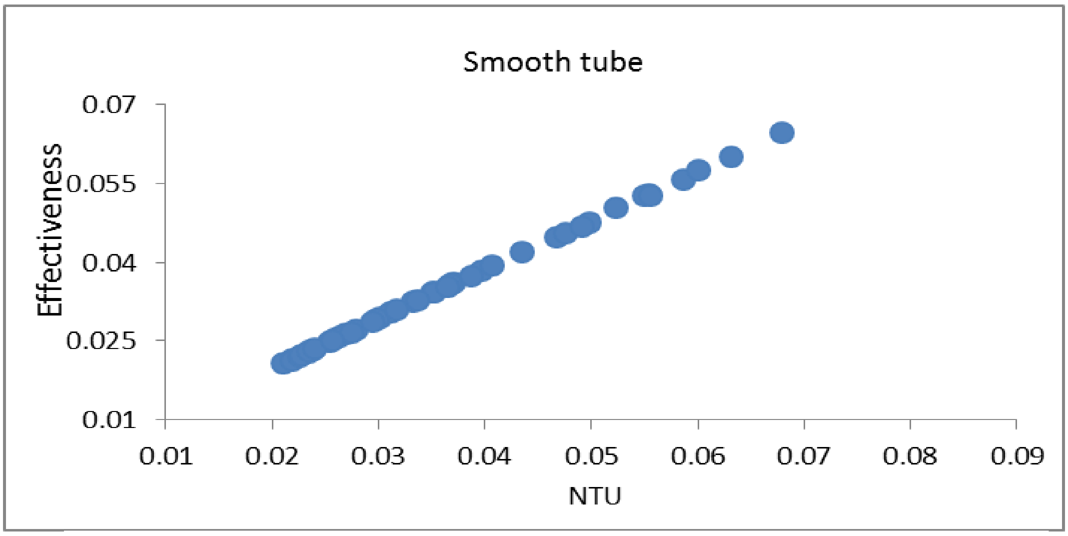

Fig. (7) Effectiveness for against the (NTU)

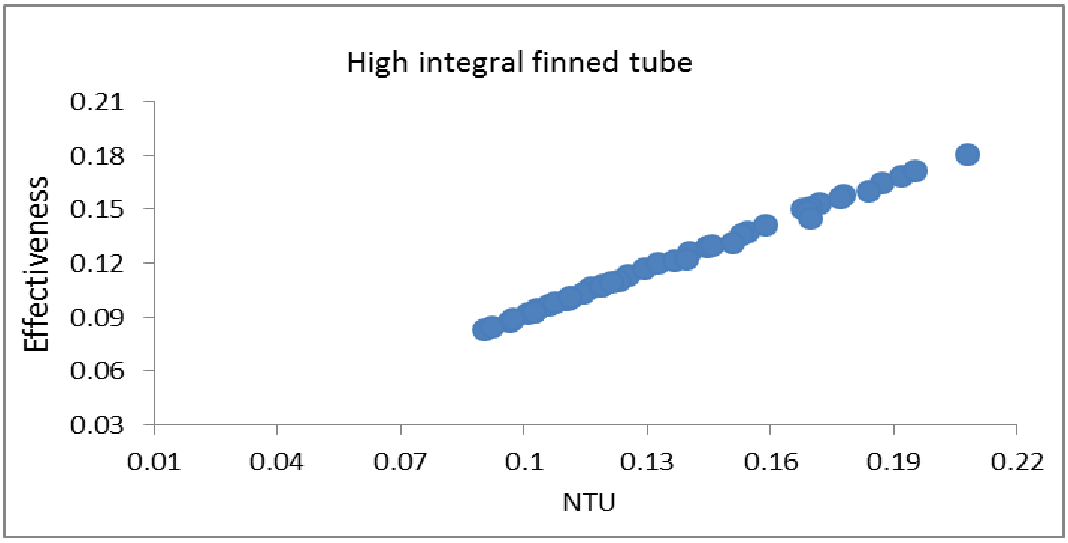

Fig. (8) Effectiveness against the (NTU)

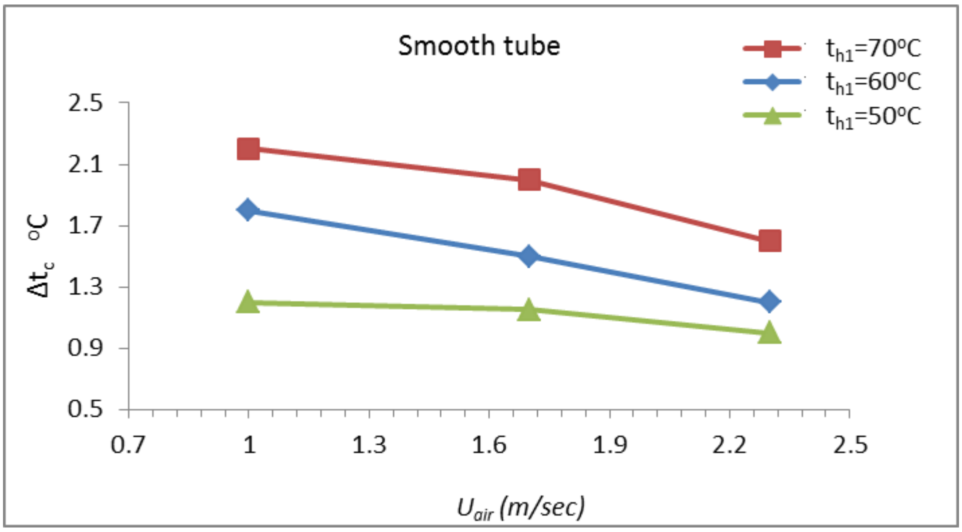

Fig. (9) Air side temperature difference with air speed for various inlet water temperatures

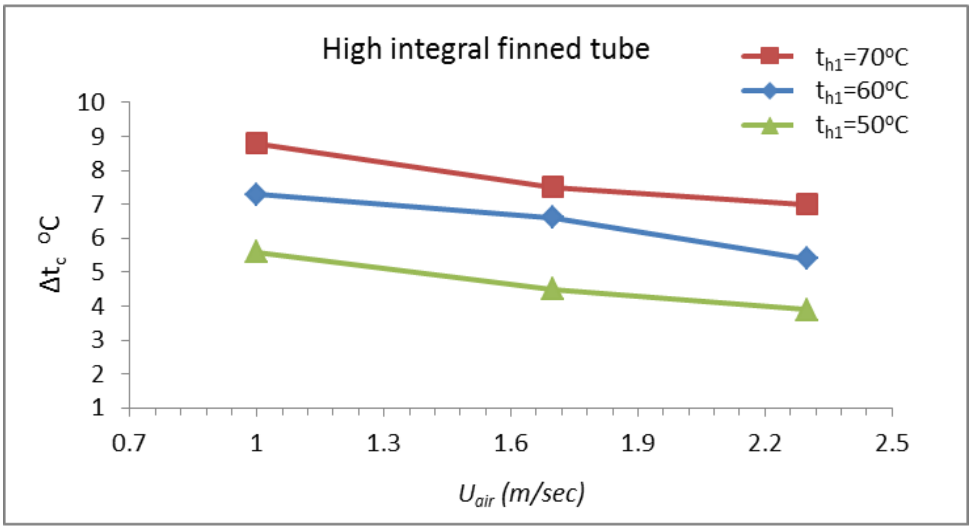

Fig. (10) Air side temperature difference with air speed for various inlet water temperatures

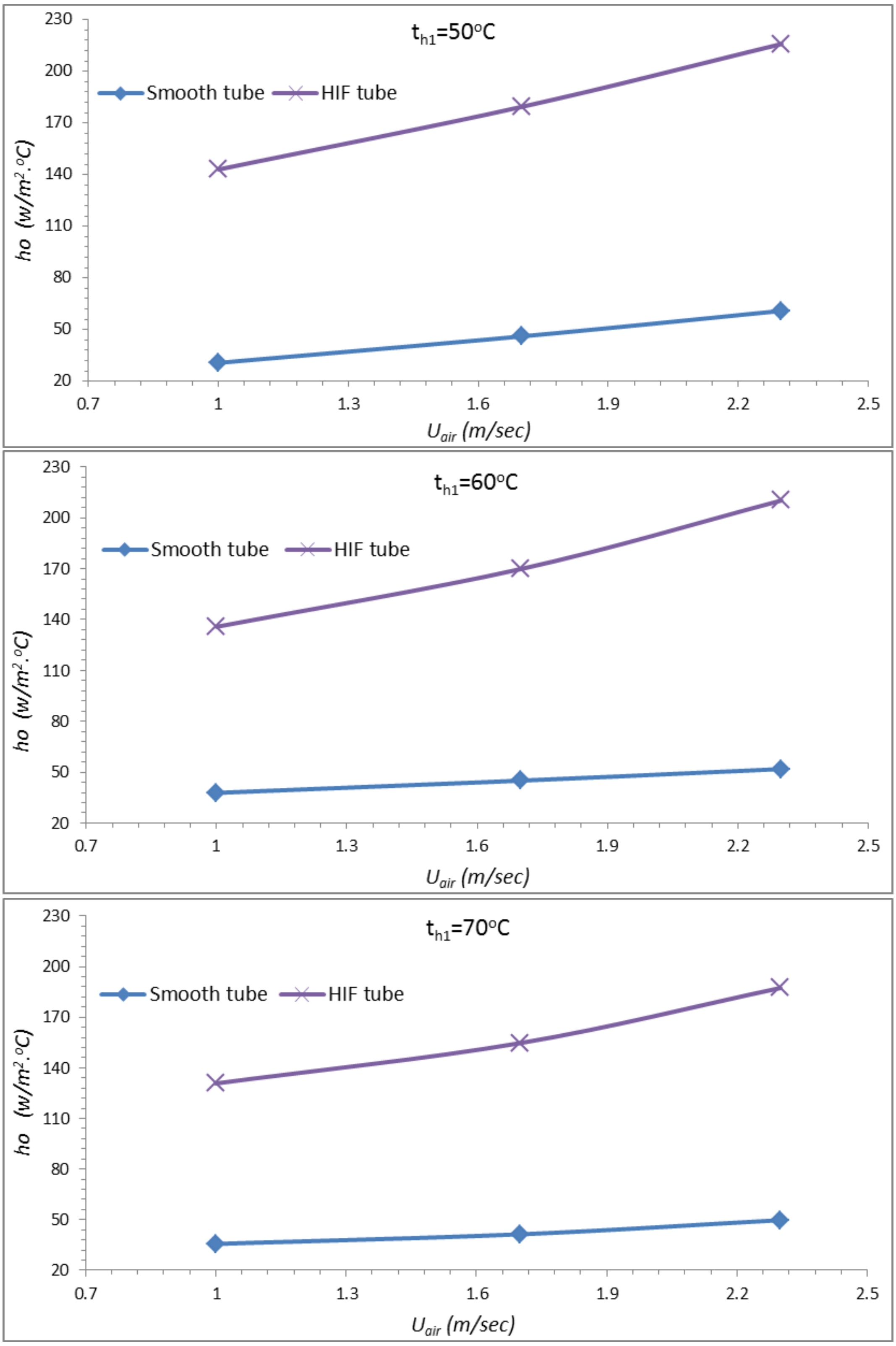

Fig. (11) Comparison the air side heat transfer coefficient against air speeds and various inlet water temperature for two models

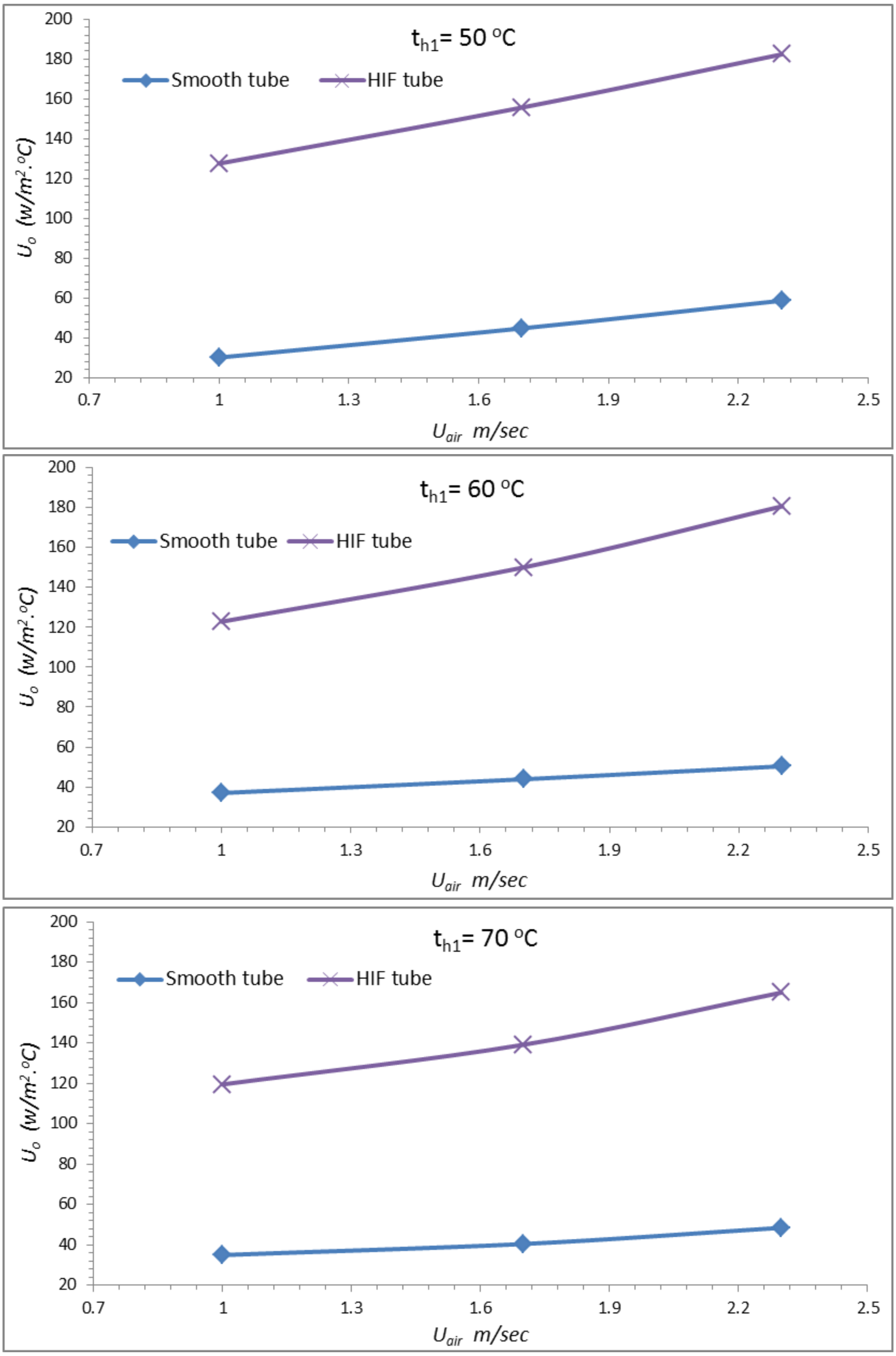

Fig. (12) The air side overall heat transfer coefficient with air velocities and various inlet water temperature for two models

Conflicts of Interest

The author declares that they have no conflicts of interest.

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تحسين معامل انتقال الحرارة لمبادل متعاكس الجريان باستخدام انبوب ذي زعانف متكاملة مرتفعة

فرح عبد الزهرة طاهر 112 **زينة خليفة كاظم**

قسم الهندسة الميكانيكية، جامعة واسط، واسط، العراق

Ftahar@uowasit.edu.iq drzena@uowasit.edu.iq

الخلاصة

يقدم هذه البحث دراسة عملية لانتقال الحرارة لمبادلات حرارية متقاطعة الجريان مبردة بالهواء ذو ثمان ممرات (انبوب أملس وذي زعانف متكاملة مرتفعة) وتأثير ارتفاع الزعانف في تحسين انتقال الحرارة. تم تصميم وتصنيع المبادلات من معدن النحاس النقي تركب هذه المبادلات داخل مقطع الاختبار. مقطع الاختبار مصنوع من البايركس بأبعاد (١٢٠٠، ٢٥٠، ٥٠٠) ملم طول، عرض وارتفاع على التوالي. المسافة من مركز انبوب واخر هي (55) ملم. القطر الداخلي وجذر الزعنفة للأنابيب هو (١٩، ٢١) ملم لكلا المبادلين. المبادل الاول ذو انبوب أملس بقطر خارجي (٢٤) ملم، المبادل الثاني ذو زعانف متكاملة مرتفعة بأبعاد (٣٣,٤، ٦,٢، ٢,٦، ١) ملم و (٣٨٤) زعنفة. كل ممر بطول (٢٥٠) ملم داخل مجرى الهواء. ظروف للعمل المعتمدة لسرع الهواء خلال مقطع الاختبار (١، ١,٧، ٢,٣) متر/ثا، معدلات تدفق الماء (٢-٦) لتر/دقيقة، درجة حرارة دخول الماء (٥٠، ٦٠، ٧٠) درجة مئوية. النتائج العملىة اظهرت ان معامل انتقال الحرارة في جانب الهواء عند استخدام الانابيب المزعنفة تكون اعلى من الانبوب الاملس وأفضل تحسين عند الانبوب المزعنف المرتفع. نسبة التحسين في انتقال الحرارة عند استخدام الانبوب المزعنف المرتفع (329.9%) والتحسين في معامل انتقال الحرارة لجانب الهواء (291%). تم ايجاد معادلات تجريبية لعدد نسلت في جانب الهواء لنماذج المبادلات.

الكلمات الدالة: – انتقال الحرارة، المبادلات الحرارية متعاكسة الجريان، انبوب ذو زعانف عالية متكامله، معامل انتقال الحرارة، التحسين، النحاس النقي.