



PRACTICAL STUDY TO IMPROVEMENT THE PERFORMANCE OF HEAT EXCHANGER USING PASSIVE TECHNIQUES

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Abstract:

The aim of this study to improve the performance of heat exchanger by using the medium integral fins on the cross flow heat exchanger practically, so, two the heat exchangers from copper were manufactures with eight passes for comparison under different boundary conditions. The water flow rate flow inner the tubes with (2, 3, 4, 5, 6) L/min with inlet temperatures (50, 60, 70) oC, as for the air flow rate were passed outer the tubes by speeds (1, 1.7, 2.3) m/sec. The results show that the medium integral finned tube gives more improvement the heat transfer than the smooth tube about 203.97% and 205.1% was enhancement factor.

Motivation: The aim is improvement the heat transfer coefficient for cross flow heat exchanger by using medium integral finned tube. Study the effect of various water stream rate, air speed and inlet water temperature on heat transfer coefficient for them, Finding 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 medium integral finned tubes as function of (Pra) and (Rea).

Keywords: Heat Transfer; Improvement the Performance; Medium Integral Finned Tubes; Cross Flow Heat Exchangers; Reynolds Number; Heat Transfer Coefficient.

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1. Introduction

A heat exchanger is a device used to exchange heat between two liquids, without mixing them. The heat transfer is of great interest to the development of heat exchangers in order to obtain high thermal effectiveness and low manufacturing cost. The integral finned tube heat exchangers have been widely applied. To improve the effectiveness of the exchangers, the passive techniques used to increase the heat transfer rate. The integral finned tubes are tubes that can be formed by their outside surface by using the ring fins to improve condensation in the layout 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 presented by the fins that also increases the area for heat transfer [1]. The medium integral finned tubes have fin height more than pitch. tube length up to 20 m, outgoing diameter between (19.05 - 31.75) mm, tube wall thickness 2.1 mm, fins per inch 11, 16 and 19 [2]. This type of integral finned tube are using for the warming of pressurized gases and bath treatment of the condensation water and cooling of hydrocarbons, [3]. **Mohsin Jani**

et. al. (2019) [4], 3D experimental and CFD study to the cross flow heat exchangers, employing the fluids of working (water, oil and with nanofluid SiO₂) with the vary concentration Nano fluid and their influence on the heat transfer coefficient. The outcomes appeared that heat transfer and heat transfer coefficient were higher in the tube with low fins than the smooth tube. The growing was 72.05% for oil and 104.1% for water in the finned tube. **Farah and Zena (2019) [5]**, a practical study of the effect of fins on improving the heat transfer of a cross flow heat exchanger with integrated fins 6.2 mm height. The air flows from the outside at a speed of (1, 1.7 and 2.3) m/s. The hot water flows inside it (2, 3, 4, 5, 6) L / min with inlet temperatures of (50, 60, 70). The effect of these variables on heat transfer and optimization were the high integral finned tube the best in heat transfer about (329.9%) and the improvement factor (291%). **Chen et. al. (2014) [6]**, the qualities of the heat transfer and pressure drop in finned tube banks through a trial open high-temperature wind. The impacts of the dimensions of the fins (width , height , pitch) and air speeds (6, 8, 10, 12 and 15) m/s on fin efficiency as well as the convective heat transfer coefficient. The result show that as the air speed, fin height and fin width increment, fin efficiency diminishes. Convective heat transfer coefficient is corresponding to fin pitch, yet conversely relative to fin height and fin width. The heat transfer limit is identified with fin efficiency, convective heat transfer coefficient and finned proportion. Pressure drop increments with the increasing the fin height and width. The relationships of the fin efficiency, Nusselt's number and Euler Number are produced in light of the empirical information. **Kumar et. al. (2005) [7]**, an exploratory examination has been done to the condensation of R-134 vapors more than five single horizontal round integral fin tubes of 472 fpm, 417 mm length and diverse fin heights of 0.45, 1.14, 1.47, 1.92 and 2.40 mm. The roundabout fins are rectangular in style and the fin thickness of all tubes is 0.70 mm. The tube with the fin height of 0.45 mm has given the most maximum improvement factor (*EF*), of the request of 3.18 in compare with that anticipated by the Nusselt's demonstrate for a plain tube. **Ayad Mezher et. al. (2011) [8]**, Examine the heat transfer qualities for cross stream air cooled single aluminum tube multi passes (smooth and integral low finned tube) and the impact of the indispensable low fins (trapezoidal or rectangular) in upgrade the heat and concentrate all factors which have impact on heat transfer phenomena. The speeds of air over the test area are (1, 2 and 3) m/sec, the water stream rate is (5l/min) and the temperatures of the entry water to the test tube are (50, 60, 70, 80) °C. The main test area has a smooth aluminum tube of eight corridors with internal diameter 17mm and external distance across 19mm. The second test segment has an essential low finned aluminum tube of eight corridors with internal diameter 17mm, root diameter 19mm and external diameter at the tip of fin 22 mm. The corridor has a length 251mm inside the conduit with 125 fins. The fin's height is 1.5 mm with a thickness of 1mm and pitch 2mm. The test show that the air side heat transfer coefficient of the smooth tube was lower than that of the low finned tube and the improvement proportion or (N_{uaf}/N_{uas}) was (1.86 to 2.38) for eight passes. The experimental connections for the air side spoke to by Nusselt's number. The heat load of the low finned tube is higher than that of the smooth tube, by (1.8 to 2.13) times. Enhance of the outside heat transfer coefficient by increasing the air speed.

2. Materials and Methods

2.1. Theoretical Equations

The energy balance in heat exchanger total heat transfer rate in heat exchanger. [9]

$$Q = m_w C_{ph}(t_{h1} - t_{h2}) = m_a C_{pc}(t_{c2} - T_{c1}) \quad 1$$

$$Q = UA\theta_m \quad 2$$

The overall heat transfer coefficient

$$U = \frac{1}{h_o} + \frac{1}{K} + \frac{1}{h_i} \quad 3$$

Log mean temperature difference (LMTD)

$$LMTD = \frac{\theta_1 - \theta_2}{\ln(\theta_1/\theta_2)} \quad 4$$

Calculations heat transfer coefficient for smooth tube, [10]

For smooth tube

$$h_o = \frac{1}{\frac{1}{U_o} - \frac{d_o \ln(d_o/d_i)}{2k} - \frac{d_o}{h_i d_i}} \quad 5$$

For integral finned tubes

$A_{of} = A_{os} = \pi d_o L$, Clean surfaces, [11, 12].

From eq. (3) we get h_o

$$h_o = \frac{1}{\frac{1}{U_i} - \frac{d_i \ln(d_r/d_i)}{2K} - \frac{1}{h_i}} \quad 6$$

Reynolds number, Prandtl number and Nusselt's number for air side:

$$Re_a = \frac{U_{air} d_h}{v_a} \quad 7$$

$$Pr_a = \frac{\mu_a C_{pa}}{K_a} \quad 8$$

$$Nu_a = \frac{h_o d_o}{K_a} \quad 9$$

Reynolds number, Prandtl number and Nusselt's number for water

$$Re_w = \frac{u_w d_i}{v_w} \quad 10$$

$$Pr_w = \frac{\mu_w C_{pw}}{K_w} \quad 11$$

$$Nu_w = \frac{h_i \times d_i}{K_w} \quad 12$$

To turbulent flow by Dittus and Boelter, [13]:

$$Nu_w = 0.023 Re_w^{0.8} Pr_w^n \quad 13$$

(0.6 < Pr < 100)

The actual heat transfer for the counter flow exchanger can be calculated from equ.1

$$Q_{max} = C_{min}(t_{h1} - t_{c1}) \quad 14$$

For cross flow C_{max} mixed and C_{min} unmixed or vice versa

$$\varepsilon = \left(\frac{1}{C}\right) \{1 - \exp[-C(1 - e^{-NTU})]\} \quad 15$$

$$\varepsilon = 1 - \exp \left\{ - \left(\frac{1}{C}\right) [1 - \exp - (NTU * C)] \right\} \quad 16$$

Where, the heat capacity ratio is,

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

The number of transfer units (NTU)

$$NTU = \frac{U_o * A_{os}}{C_{min}} \quad 17$$

Enhancement factor:[14]

$$E.F \% = \frac{h_{o,fin} - h_{o,smooth}}{h_{o,smooth}} \times 100 \quad 18$$

2.2. Experimental Apparatus

A laboratory device involved design and manufacturing the laboratory apparatus and test models, which consist of a test section made of Pyrex glass with dimensions (250×500×1200) mm. The test tubes with a length of 2 meters, description of the smooth and medium integral finned tube with 384fpm shown in table (1). 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), while the medium integral finned tube was shown in figure (2).

The hot 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. The cold air

supply unit in this unit is passed the cold air through the following parts the centrifugal blower, the diffuser from galvanized steel and the air duct. Measuring devices are utilized to take readings from the laboratory apparatus hot-wire anemometer, Thermometer, flow meter, the pressure meter recorder model (ps-9302) with the rang (1-400) bar and 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. Thermal imager utilizes to measure the surface temperature of the test tubes.

Table 1: Description of the copper tubes

Tubes	di	do	dr	H _f	T _f	T _w	S _f	A _o /A _i
Smooth	19	24	-	-	-	2.5	-	1.263
MIF	19	28.5	21	3.75	1	1	1.6	5.0



Figure 2: Photos of the medium integral tube

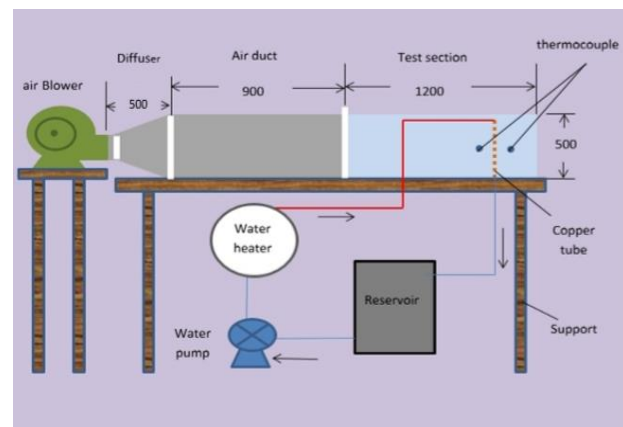


Figure 1: Schematic diagram of the cold air cycle and dimensions in (mm)

2.3. 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,[15].Calculation error were overall heat transfer coefficient (2.27 to -2.38)%, heat transfer (11.65-15.19)%, air side Reynolds number (0.174-0.17)%, air side heat transfer coefficient (2.95-3.14)% and (Nu_a) (3.027 to-3.223)%.

3. Results and Discussions

Figure (3) offer the relation between the (h_i) with volumetric flow rates at inlet water temperatures and air velocity various for smooth tube and medium integral finned tube. (h_i) 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.

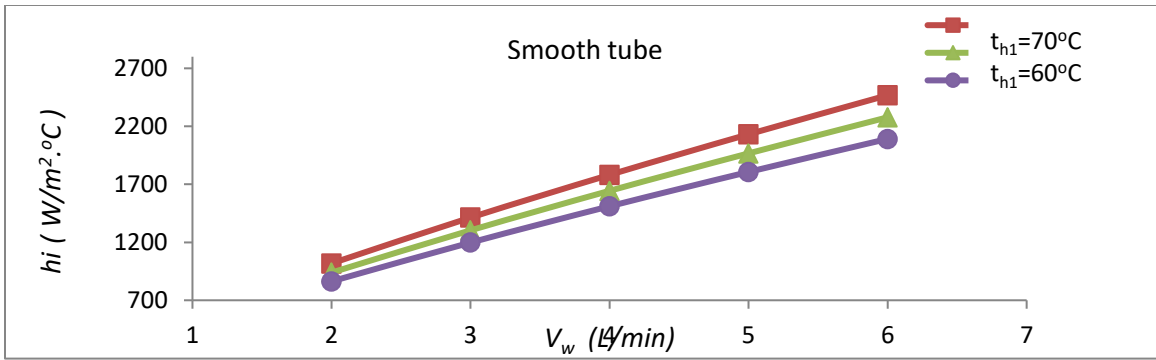


Figure 3: Inner side heat transfer coefficient with various water flow rate

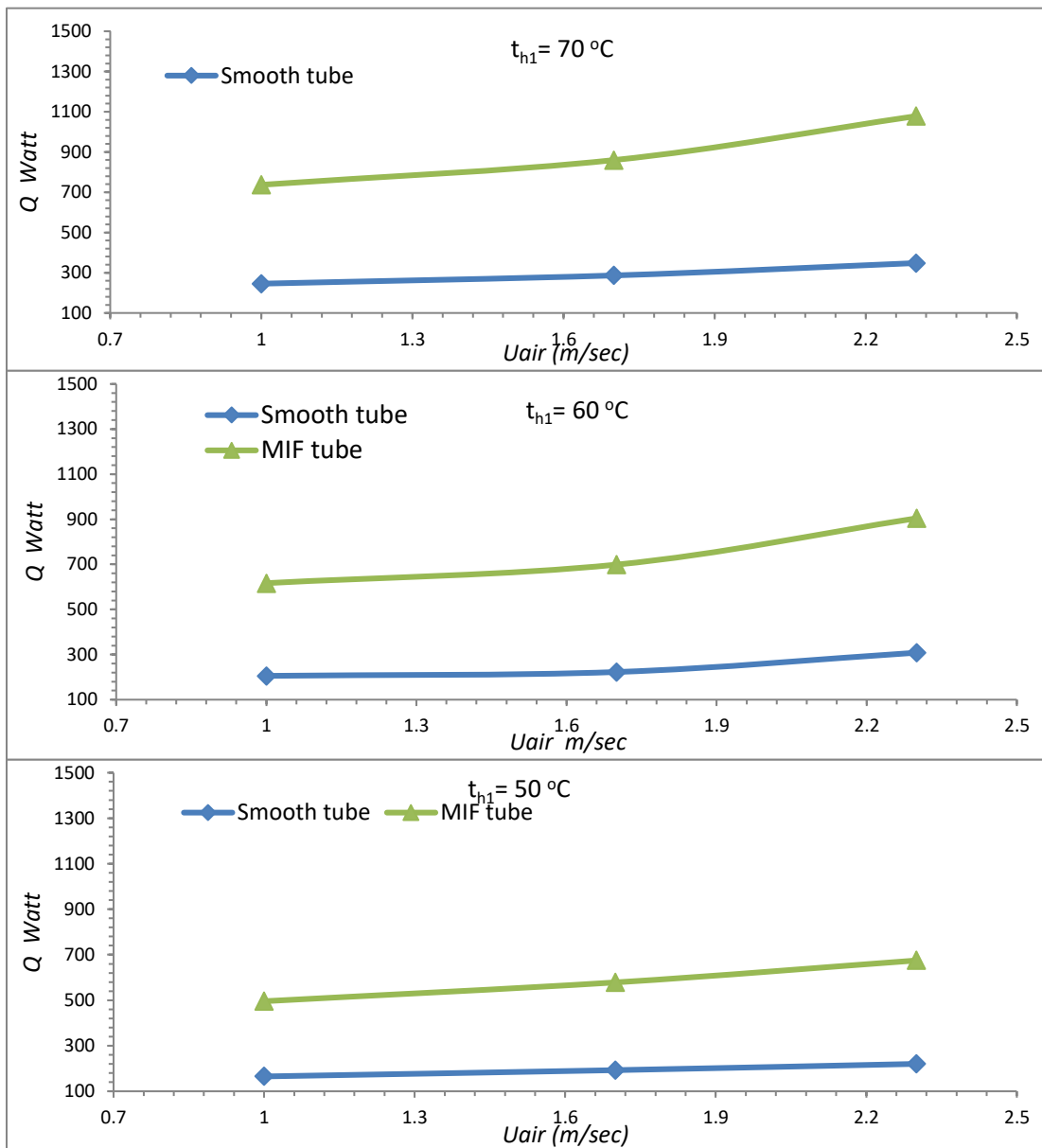


Figure 4: The heat transfer rate with the air speed and various inlet water temperatures for two models

Figure (4) clear the relation between the heat transfer rate with inlet water temperature and air velocity for the two heat exchangers (Smooth and Med). 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 tubes 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. **Figures (5)** 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).

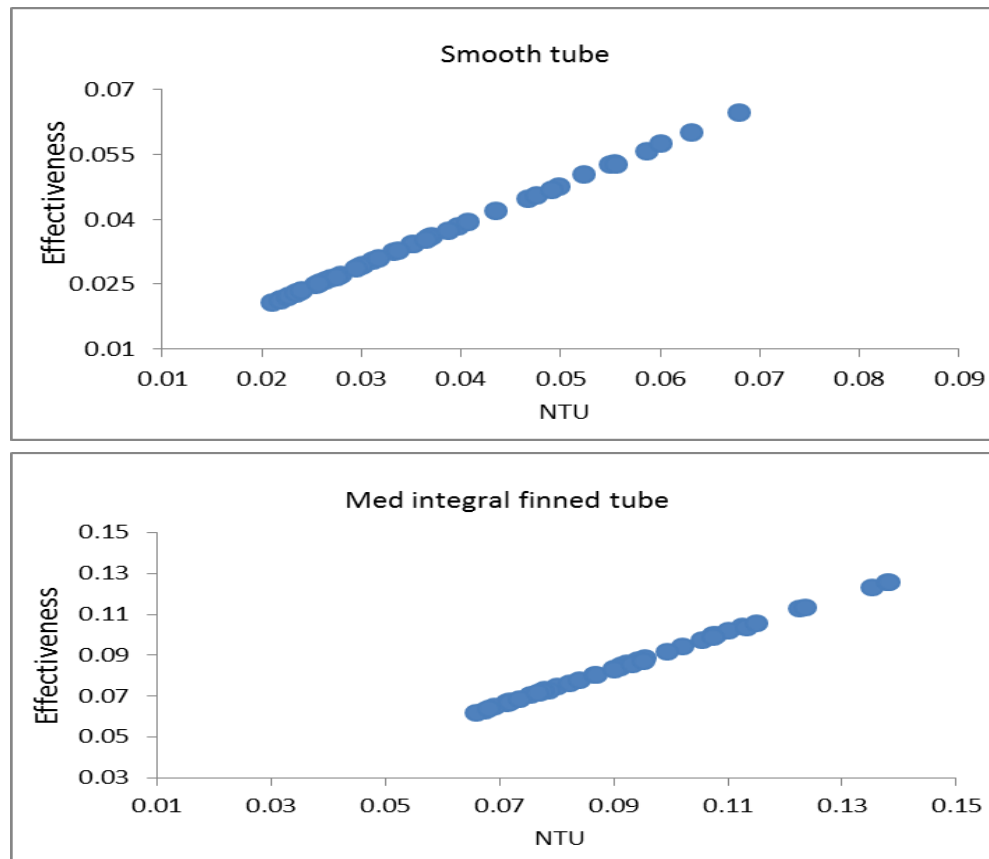


Figure 5: Effectiveness against the (NTU)

Figures (6), explain that the air side temperature difference (Δt_c) with air speeds and (t_{h1}). (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 (7)** show the effect of the different air speed and inlet water temperatures on the (h_o) for two test models. The values of (h_o) increase with air speed at the same inlet temperature water as a result of increasing the (Re_a). **Figure (8)** show comparison of the (U_o) versus the various velocities (U_{air}). The (U_o) increases with increasing air velocity at constant inlet temperature for all 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.

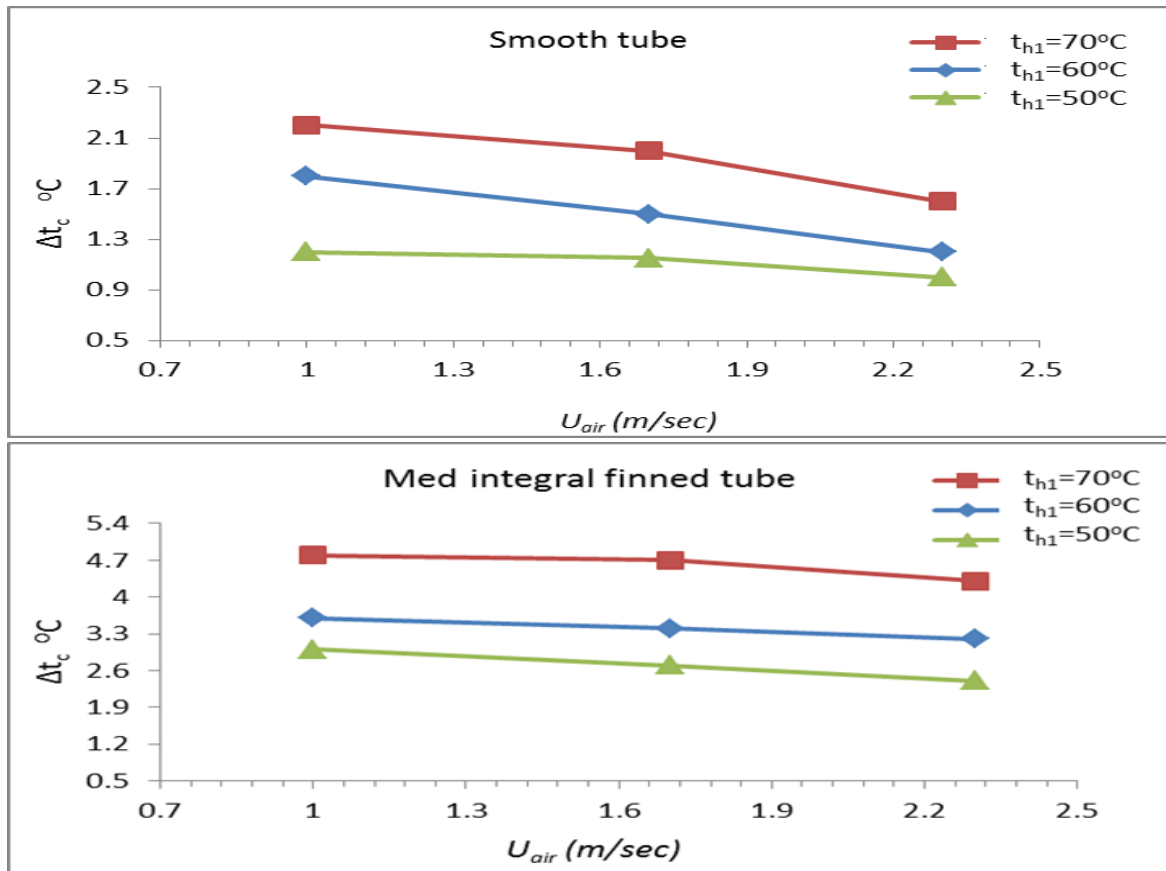


Figure 6: Air side temperature difference with air speed for various inlet water temperature

3.1. Comparison These Results for Heat Exchangers with Study Farah And Zena [5]

The comparison is made under the covered boundary conditions in the air speed (1, 1.7, 2.3) m / s, and the rate of volume flow of water (2 L/min) and the water inlet temperature (70°C) of the high and medium integral finned tubes as shown in Figure 9. From this figure concludes that the behavior of (Q). from the comparison found that the behavior of the heat is similar in the three heat exchangers and that the heat transfer increases with the increase the air speed. MIF is have middle heat transfer rate relative to HIF in study [5] and the smooth tube. the fins height have effect large on the heat transfer due to the increase of surface area of heat loss, where the rate of improvement in the HIF relative to the smooth tube (329.9%) and the rate of improvement in MIF (203.97%). This leads to the preference of the high fin exchanger (mm) on the other models.

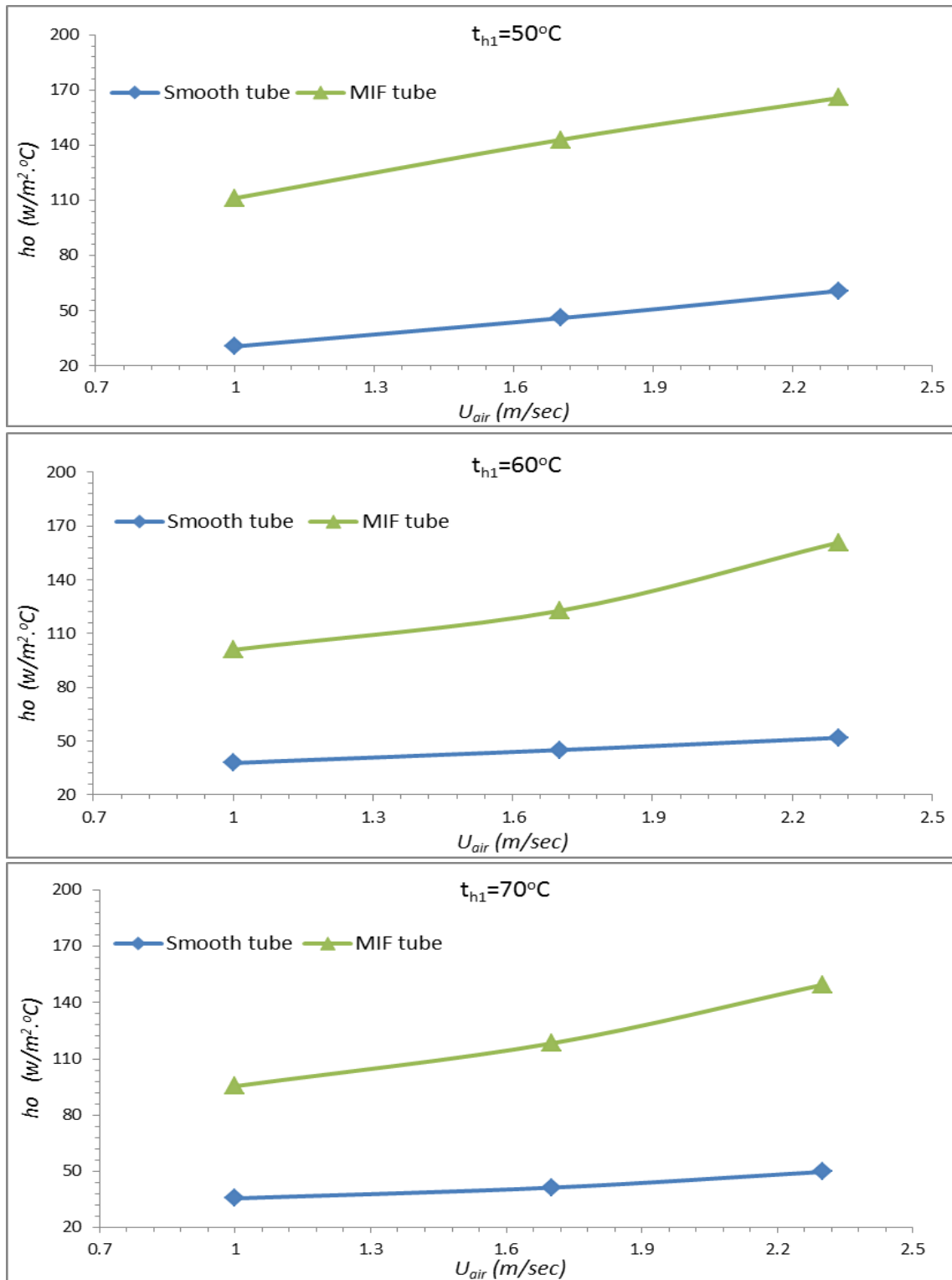


Figure 7: Comparison the air side heat transfer coefficient against air speeds and various inlet water temperature for two models

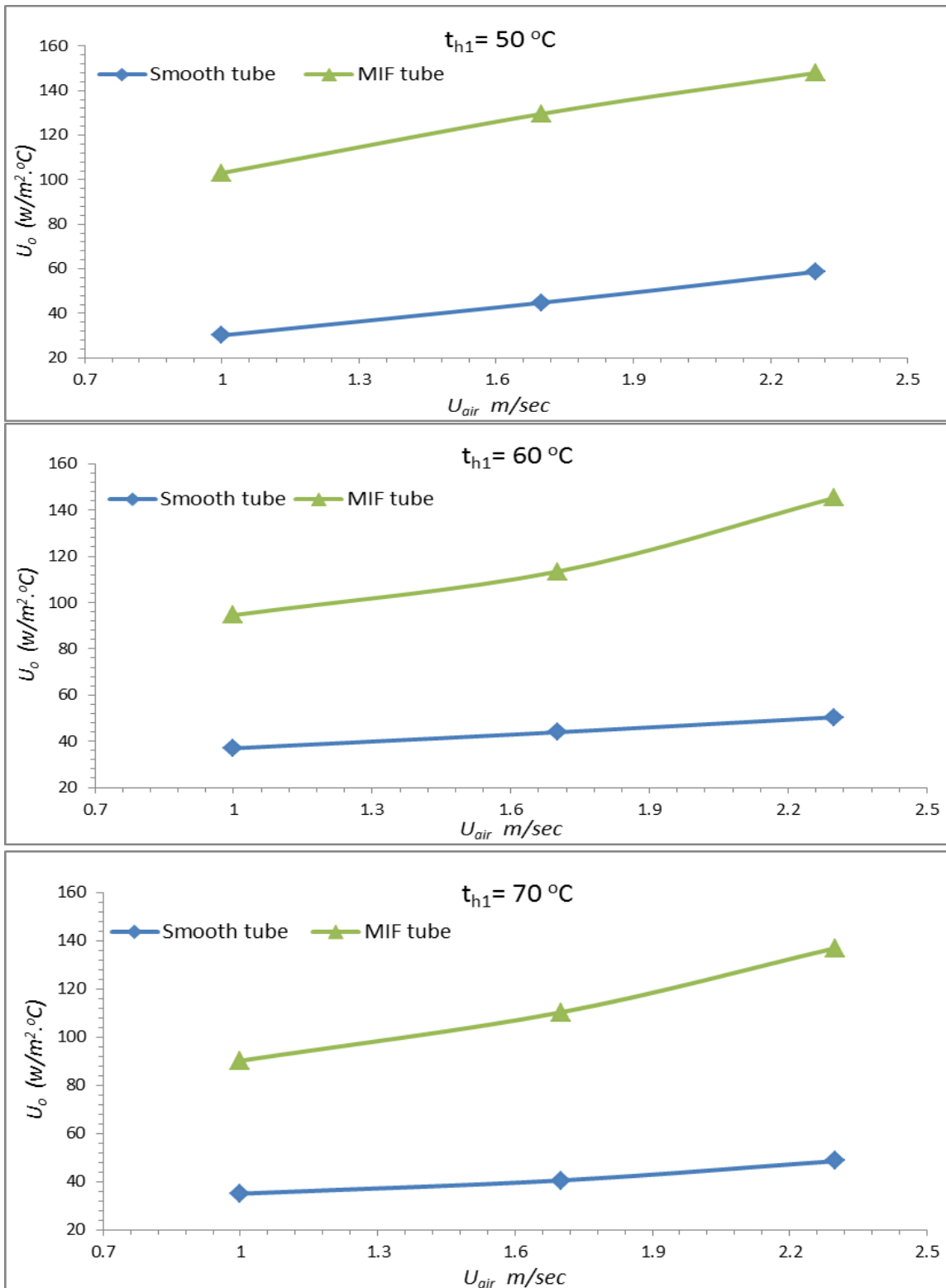


Figure 8: The air side overall heat transfer coefficient with air velocities and various inlet water temperature for two models

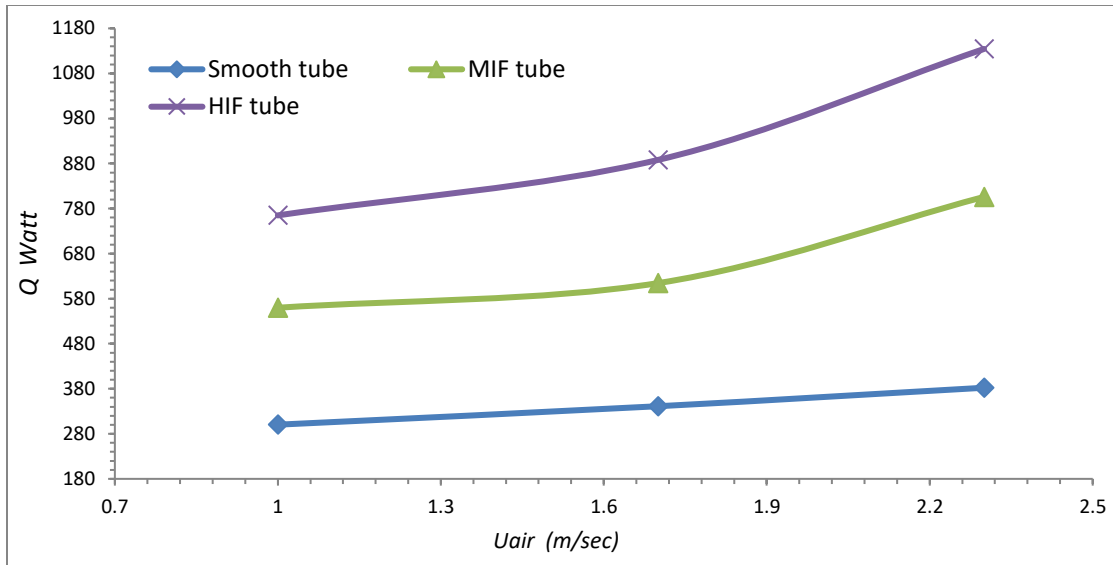


Figure 9: comparison the heat transfer rate for two models with study [5]

4. Conclusions

- The heat transfer coefficient (h_i) increase with water flow rate (V_w) and inlet water temperature (t_{h1}).
- The heat transfer coefficient (h_o) increased with (U_{air}) and decreased with the inlet water temperature. The enhancement factor for MIF (205.1%)
- The heat transfer rate increase with air speed (U_{air}) and inlet water temperature (t_{h1}). The heat rate of medium integral finned tube higher than smooth tube with enhancement factor $E.F\%$ (203.97%).
- The effectiveness (ϵ) was increased with increasing the number of the transfer units (NTU), to smooth tube and MIF (0.073 and 0.125) respectively.
- The overall heat transfer coefficient of the air side (U_o) directly proportional with (U_{air}) and inversely with the temperature (t_{h1}).
- The medium integral finned tube was more improvement than smooth tube due to presence the fins.
- 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 (19) and table (2) for medium integral finned tube.

$$Nu_a = 0.60697 Re_a^{0.435137} Pr_a^{1/3}$$

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Table 2: Empirical correlations for integral finned tube

t_{h1} °C	(20496.54 < Re_a < 48394.26) Air velocity range (1, 1.7, 2.3) m/sec
$V_w=6L/min$	
70	$Nu_a = 0.2449 Re_a^{0.4577} Pr_a^{1/3} (A_o/A_i)^{0.7661}$

60	$Nu_a = 0.1166 Re_a^{0.5317} Pr_a^{1/3} (A_o/A_i)^{0.7921}$
50	$Nu_a = 0.2279 Re_a^{0.4798} Pr_a^{1/3} (A_o/A_i)^{0.7577}$
V _w =5L/min	
70	$Nu_a = 0.5883 Re_a^{0.3790} Pr_a^{1/3} (A_o/A_i)^{0.7051}$
60	$Nu_a = 0.3773 Re_a^{0.4258} Pr_a^{1/3} (A_o/A_i)^{0.7177}$
50	$Nu_a = 0.4339 Re_a^{0.4214} Pr_a^{1/3} (A_o/A_i)^{0.7264}$
V _w =4L/min	
70	$Nu_a = 0.0740 Re_a^{0.5849} Pr_a^{1/3} (A_o/A_i)^{0.6263}$
60	$Nu_a = 0.0949 Re_a^{0.5638} Pr_a^{1/3} (A_o/A_i)^{0.6431}$
50	$Nu_a = 0.1069 Re_a^{0.5687} Pr_a^{1/3} (A_o/A_i)^{0.610}$
V _w =3L/min	
70	$Nu_a = 0.0417 Re_a^{0.6250} Pr_a^{1/3} (A_o/A_i)^{0.6806}$
60	$Nu_a = 0.0451 Re_a^{0.6276} Pr_a^{1/3} (A_o/A_i)^{0.6536}$
50	$Nu_a = 0.0434 Re_a^{0.6430} Pr_a^{1/3} (A_o/A_i)^{0.6462}$
V _w =2L/min	
70	$Nu_a = 0.1641 Re_a^{0.5009} Pr_a^{1/3} (A_o/A_i)^{0.6493}$
60	$Nu_a = 0.2481 Re_a^{0.4680} Pr_a^{1/3} (A_o/A_i)^{0.6539}$
50	$Nu_a = 0.4128 Re_a^{0.4123} Pr_a^{1/3} (A_o/A_i)^{0.7543}$

Symbols

A_i : inner surface area of tube (m²), A_o : outer surface area of tube (m²), C_p ., Specific heat of the fluid (kJ/kg.° C) d : tube diameter (m), h_i : inner side heat transfer coefficient (W/m².°C), H_f : fin height (mm), h_o : air side heat transfer coefficient (W/m².°C), K : thermal conductivity of tube material (W/m.° C), K_w : thermal conductivity of water (W/m.° C), L : length of tube (m), m : mass flow rate (kg/s), N_f : number of fins (fpm), Nu_a : air side Nusselt's number, Nu_w : inner side Nusselt's number, NTU: number of the heat transfer units, Q : heat transfer rate (Watt), Re_a : air side Reynolds number, S_f : fin space, 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_w : wall thickness (mm), U_i : inner side overall heat transfer coefficient (W/m².°C), U_o : air side overall heat transfer coefficient (W/m².°C), U_{air} : velocity of air (m/s), ϵ : the exchanger effectiveness, θ : temperature difference (°C), MIF: medium integral finned tube, HIF: high integral finned tube.

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