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ENHANCEMENT OF HEAT TRANSFER THROUGH FIN- AND-TUBE HEAT EXCHANGER USING CORRUGATED FINS

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ABSTRACT

This work aims to investigate experimentally the effects of corrugation angle, number of fins per inch and velocity on the heat transfer and friction of fin-and-tube heat exchangers. To accomplish this study, six coils (fin-and-tube heat exchangers) with the same geometry were constructed, four coils with 4FPI (fin per inch) but with different corrugation angles ($0^o, 15^o, 30^o$ and 45^o) besides two coils with corrugation angle 15^0 but with different number of fins per inch (6 and 8 FPI). The heat transfer characteristics of the six coils were measured in a wind tunnel. The experimental results indicated that the corrugated fins have important effect on the heat transfer and friction. The present work indicated that the augmentation due to corrugated fins increases with increasing Reynolds number; when the average velocity through heat exchanger increases, both the heat transfer coefficient and the pressure drop increase. If Reynolds number increases, both Colburn factor and friction factor decrease. The corrugation angle 30^ohas the highest ratio of heat transfer Colburn factor to the friction factor (j/f) followed by angle 15^owhile the corrugation angle45^0 has the lowest value of that ratio. The number of fins per inch have an important effect on the heat transfer characteristics and friction. When the number of fins per inch increases, both the heat transfer and friction increase. Correlations between dependent and independent variables were presented. Comparison between the present and previously published results is in good agreement.

KEY WORDS: Heat transfer, Forced Convection; Fin-and-tube heat exchanger; Corrugated fins

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1. INTRODUCTION

Since the first energy crisis that occurred in 1973, there has been great social demand to save energy and resources on all levels. Cross- flow heat exchangers were no exception and an even greater need emerged for more compactness, light weight, high performance and low cost. Cross-flow heat exchangers are widely used in refrigeration and air conditioning equipments besides automotive applications.The need for more efficient cross-flow-finned heat exchangers has led to the development of a variety of unconventional flows through fins passages to enhance the convective heat transfer coefficient; one such passage is the corrugated fins. The most common augmentation technique for plate fin- and- tube heat exchangers is the use of corrugated fins instead of plate fins. The corrugated fins are, in essence, plates that has been fabricated with a periodic waviness in the streamwise direction.

Van den Bulck[1] studied an optimal design of cross flow heat exchangers. The optimal distribution of the transfer surface area for maximum heat exchanger effectiveness and constant total surface area was determined. It was found that the distribution of the transfer surface aligned along the diagonal of the cross flow exchanger gave the best performance; equal to that of a counter flow device. For constant surface area, the exchanger effectiveness might be increased by 0.07 for balanced flow rates.

Idem et al [2] studied the heat transfer characteristics of a finned-tube heat exchanger. The performance of an air-to-water copper finned-tube cross flow heat exchanger was considered. The correlations between Colburn factor-friction factor and Reynolds number (Re_D) were as follows:

$$j = 0.145 Re_D^{-0.484} \quad , \quad f = 0.944 Re_D^{-0.517}$$

Goldstein and Sparrow [3] carried out experiments on the heat transfer characteristics of a corrugated fin and tube heat exchanger. Local and average air-side transfer coefficients were measured for a one-row corrugated fin and tube heat exchanger. The measurements were accomplished via the heat-mass transfer analogy in conjunction with the naphthalene sublimation technique. The average heat transfer coefficients were compared with those for a corresponding plane-walled heat exchanger configuration. The comparison showed that the augmentation due to the corrugated fin surface increasing with Reynolds number.

Kajino and Hiramatsu [4] studied experimentally an enhancement of heat transfer characteristics of automotive radiator by using louver corrugated fins. In the actual radiator, 80-90 percent of the heat rejection comes from the fins and only 10-20 percent from the tubes. In terms of weight proportion, however, the tubes account for 60 percent while the fins 40 percent. It tells us that the reduction in heat transfer surface area of the tubes is the desirable way of making the radiator more compact and lightweight. Naturally, the reduction in number of tubes must be compensated by improvement in fin performance. They indicated that a 20 percent improvement in heat transfer coefficient of fins would make it possible to reduce the number of the tubes by a half.

Giovannoni and Mattarolo [5] carried out many experiments on cross flow heat exchanger coils having continuous corrugated or wavy fins and compared them with geometrically similar heat exchanger coils with straight fins coils. The correlations in terms of heat transfer Colburn -j factor and Reynolds number based on the distance between two rows (S_L) were the following:

(1) For corrugated fins coils with 12 FPI

$$j = 0.308\,(Re_L)^{-0.375} \qquad 2000 \le Re_L \le 20000$$

(2) For straight fins coils with 12 FPI

$$j = 0.426(Re_L)^{-0.43} \qquad 2000 \le Re_L \le 20000$$

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The present work aims to investigate experimentally the effect of corrugation angle, number of fins per inch and velocity on heat transfer characteristics and friction through fin-and-tube heat exchangers. To achieve this study a wind tunnel including a test section was constructed to allow measurements of the flow of air, temperatures and pressure drop through the test section.

The apparatus consists mainly of the following components; air passage, heat exchanger unit and measuring instruments. Figure (1) shows the layout of the apparatus and the associated air supply system.

2.1 Air Passage

The air supplied to the test section was sucked from the laboratory through the air passage which consisted of fan section, inlet convergent part, honey-comb section, main duct and working section. The fan used was double inlet centrifugal fan with forward curved blades. Two gates were constructed on the suction air path to control the flow of the air-streams. The function of the inlet convergent part was to speed up the air velocity into the test section and to improve the stream lines uniformity. Its length was 0.6 m. The inlet to that part had a square cross-section with dimensions 0.4mx0.4m and the outlet cross-section of that part was 0.3mx0.27m. The honey-comb was located inside the main duct before the working section as shown in Fig.(1). The main duct, which supplied air to the test section was a horizontal duct having a cross section with dimensions of 0.3mx0.27m. It consisted of 6 pieces connected to each other by means of flanges and rubber sheets. The lengths of the pieces which followed the inlet convergent part were 0.4, 0.9, 0.9, 0.4, 0.4 and 0.4m respectively.

2.2 Heat Exchanger Unit

Six coils with the same geometry were designed and fabricated, four coils with 4FPI and different corrugation angles ($0^\circ, 15^\circ, 30^\circ$ and 45°) besides two coils with corrugation angle 15° but with different number of fins per inch (6,8FPI). Figures (2,3) show the isometric for the straight and corrugated fin-and-tube heat exchangers. The six coils had the following specifications: each coil had nine tubes made of copper, each tube had 0.29 m length, 0.002 m thickness and 0.0158 m outer diameter. The tube surface was heated electrically by means of nickel-chrome heater which was inserted inside the tube. The space inside each tube was filled with sand to assure uniformity of the heat flux. The space adjacent to the tube edges was filled with low conductive sandy-epoxy mixture to minimize the heat loss through the tube edges, [6].

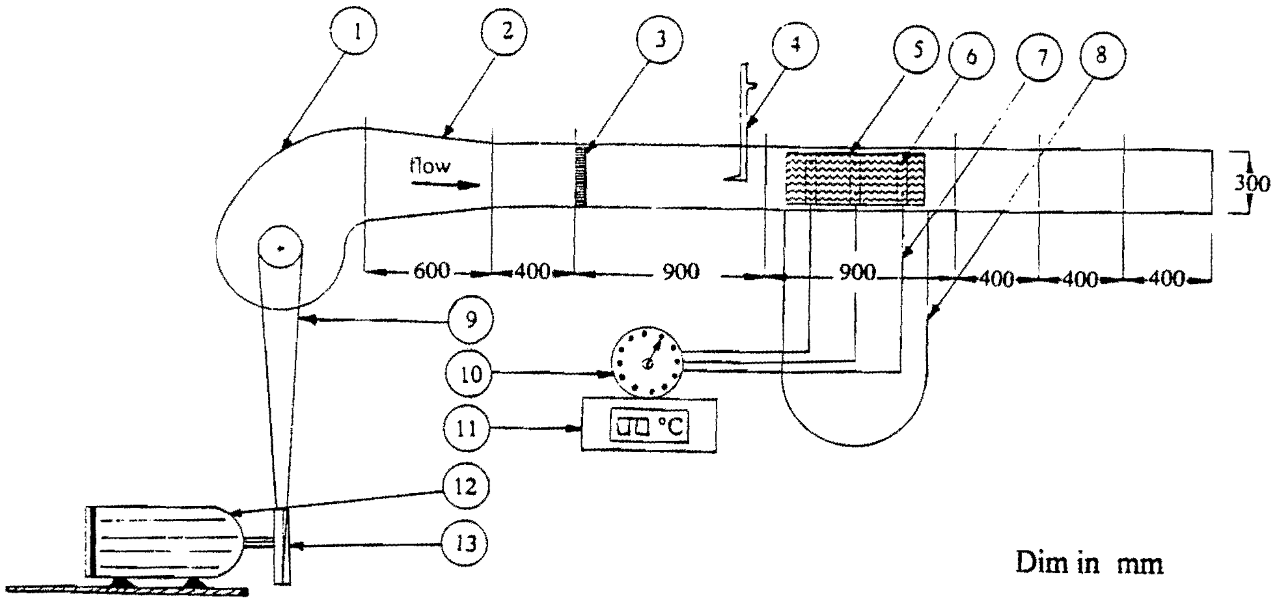

Fig. (1): General layout of the experimental apparatus.

1- Double inlet centrifugal fan	7- Thermocouple
2- Inlet convergent part	8- U-tube manometer
3- Honey-comb	9- V-belt
4- Pitot tube	10- Selector switch
5- Heat exchanger unit	11- Temperature indicator
6- Thermocouple junction	12- A-C motor
	13- Pulley

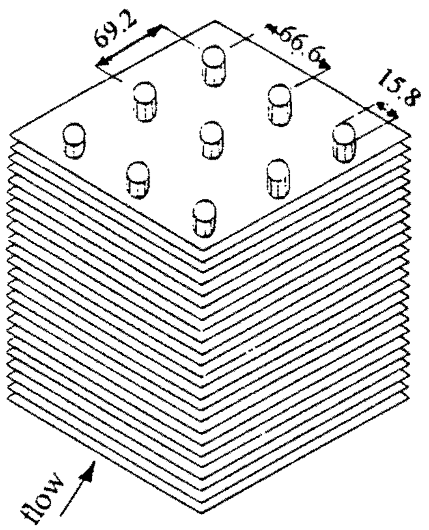

Fig. (2) Isometric of straight fin-and-tube heat exchanger.

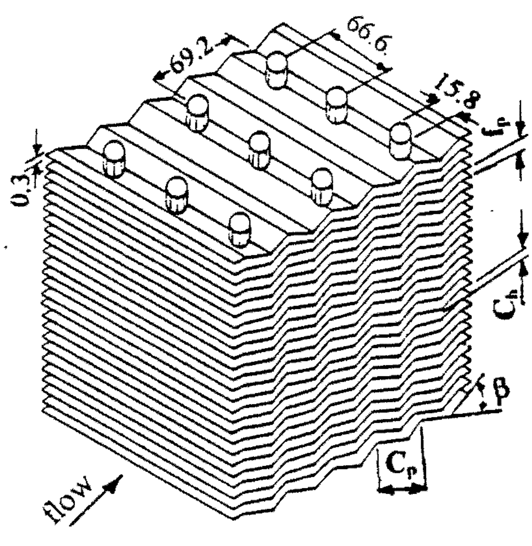

Fig. (3) Isometric of corrugated fin-and-tube heat exchanger.

the heat flux. The space adjacent to the tube edges was filled with low conductive sandy-epoxy mixture to minimize the heat loss through the tube edges, [6].

The heat exchangers tubes were constructed to be three rows, three columns in an in-line arrangement with 0.0692 m longitudinal pitch (S_L) and 0.0666 m transverse pitch (S_t). Two thermocouples of copper constantan, 0.2 mm outer diameter were fixed on the surface of each tube at a distance of 0.095 m from each tube edge to measure the surface temperature of the tube.

The fins were constructed from 0.0003 m aluminum sheets. The corrugated pitch for the fins with angles (15^o, 30^o, 45^o) was 0.746, 0.346 and 0.20 m respectively. The corrugated height C_h was kept constant at 0.01 m. The straight fin dimensions were 0.21mx0.2m.

2.3 Measurements

The measurements of the heat input, velocity and temperature were carried out during the experimental runs. These measurements were processed as follows:

2.3.1 Heat input measurements.

The electric power inputs to the tube's heaters were determined by direct measurement of the power by a wattmeter. The power was adjusted to a certain reading (6500 W, constant heat flux) by controlling the voltage through a voltage regulator.

2.3.2 Velocity measurements

A pitot tube was used to determine the velocity distribution during the calibration procedure of the air passage. The pitot tube had a hemispherical nose of 0.005 m outer diameter. The pitot tube was connected to an inclined U-tube manometer. The duct was divided into discrete sections and velocity was measured at several points in order to calculate the average value. The pitot tube could then be placed at the point of average velocity for extended measurements [7],[8].

2.3.3 Temperature measurements.

The temperatures of the heating tubes of the heat exchangers, inside duct walls, outside insulation duct walls, upstream air and downstream air were measured by copper-constatan thermocouples (0.2 mm outer diameter) through three selector switches and digital indicator with uncertainty of 0.1 percent.

3. RESULTS AND DISCUSSION

The present experimental work is divided into three main parts.

1- Experiments with straight fin-and-tube heat exchanger with 4FPI, $\beta = 0^{o}$

2- Experiments for corrugated fin-and-tube heat exchanger with 4FPI and different corrugation angles ($\beta = 15^{o}, 30^{o}, 45^{o}$).

3- Experiments for corrugated fin-and-tube heat exchanger with $\beta = 15^{o}$ and different number of fins per inch (4,6,8 FPI).

More details can be found in [9].

3.1 Results for the Straight Fin-and-Tube Heat Exchanger

Figure (4) shows the effect of Reynolds number on both heat transfer Colburn factor and friction factor. It is obvious that, when Reynolds number increases, both the heat transfer Colburn factor and friction factor decrease.

3.2 Effect the Angle of Corrugation on the Heat Transfer and Friction.

3.2.1 Effect of Reynolds number on Nusselt number

Figure 5 shows the variation of Nusselt number with Reynolds number at different corrugation angles ($\beta = 15^{o}, 30^{o}, 45^{o}$). This figure shows that the Nusselt number increases with increasing Reynolds number. For a certain value of Reynolds number, Nusselt number increases with increasing the angle of corrugation. This is due to the decrease of logarithmic mean temperature difference and increase of turbulence, with the increase of corrugation angle.

Nusselt number can be correlated as;

$$Nu = 0.127 Re_D^{0.612} \left(C_h / C_p\right)^{0.079} \qquad 280 \le Re_D \le 10700 \quad 15^o \le \beta \le 45^o \quad (1)$$

The maximum deviation of Eq. (1) from the experimental data is 7 percent.

3.2.2 Effect of Reynolds number on Colburn factor

Figure 6 shows the variation of heat transfer Colburn factor versus Reynolds number at different corrugation angles ($\beta = 15^o, 30^o, 45^o$). This figure shows that the heat transfer Colburn factor decreases with increasing Reynolds number, For a certain value of Reynolds number, Colburn factor increases with increasing the angle of corrugation.

Colburn factor can be correlated as;

$$j = 0.149 Re_D^{-0.376} \left(C_h / C_p\right)^{0.146} \qquad 280 \le Re_D \le 10700, \quad 15^o \le \beta \le 45^o \qquad (2)$$

The maximum deviation of Eq. (2) from the experimental data is 4.5 percent.

3.2.3 Effect of Reynolds number on the friction factor

Figure 7 shows the variation of friction factor versus Reynolds number at different corrugation angles ($\beta = 15^o, 30^o, 45^o$). This figure shows that the friction factor decreases with increasing Reynolds number. For a certain value of Reynolds number, the friction factor increases with increasing the angle of corrugation. The friction factor can be correlated as ;

$$f = 0.5 Re_D^{-0.3} \left(C_h / C_p\right)^{0.19} \qquad 280 \le Re_D \le 10700, \quad 15^o \le \beta \le 45^o \qquad (3)$$

The maximum deviation of Eq. (3) from the experimental data is 3.9 percent.

Figure 8 shows the effect of Reynolds number on the ratio of heat transfer Colburn factor to the friction factor (j/f) for fin-and-tube heat exchangers with the angles of corrugation as ($\beta = 0^o, 15^o, 30^o, 45^o$). This figure shows that, for a certain

Reynolds number, the ratio of heat transfer Colburn factor to the friction factor (j/f) for corrugation angle 30^o is higher than that for angle 15^owhich is in turn higher than that for angle 0^o (zero). The ratio for angle 0^o (zero) is higher than that for angle 45^o .This mean that, from the point of view of the ratio of heat transfer to friction, the corrugation angle 30^o has the highest value compared with the angles $15^o, 0^o$ and 45^o respectively. For the same values of Reynods number, Colburn and friction factors can be correlated as;

$$j = 0.36 f^{1.2} \qquad \beta = 0^o \qquad (4)$$

$$j = 1.5 f^{-3.3} (C_h / C_p)^{0.63} \qquad 15^o \le \beta \le 45^o \qquad (5)$$

Figure 9 shows a comparison between Nusselt number of the present correlated values and Nusselt number of the experimental data at the same range of Reynolds number. The experimental points of the corrugated fins $(\beta = 15^o, 30^o, 45^o)$ show a maximum deviation about 20 percent.

3.3 Effect of Number of Fins per Inch on the Heat Transfer and Friction.

3.3.1 Effect of Reynolds number on the Colburn factor

Figure 10 shows the variation of Colburn factor with Reynolds number at corrugation angle $\beta = 15^o$and different number of fins per inch (4,6 and 8). This figure shows that Colburn factor decreases with increasing Reynolds number, at the same Reynolds number, Colburn factor increases with increasing the number of fins per inch.

Colburn factor can be correlated as;

$$j = 0.1 Re_D^{-0.38} (C_h / fp)^{0.273} \qquad 130 \le Re_D \le 10300, (3.17 \le f_p \le 6.35 mm) \qquad (6)$$

The maximum deviation of Eq. (6) from the experimental data is 6.9 percent.

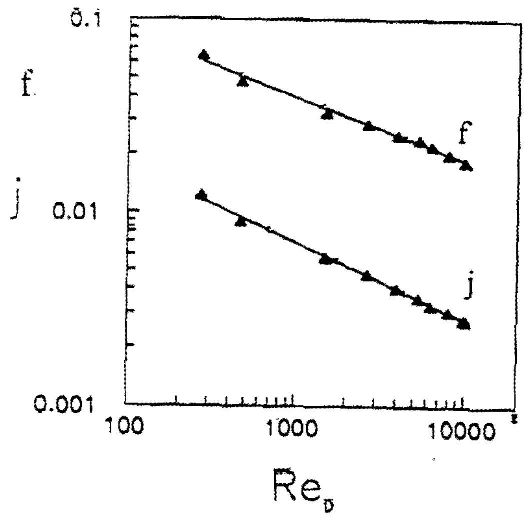

Fig. (4) Variation of Colburn factor and friction factor with Reynolds number for straight fin-and-tube heat exchanger

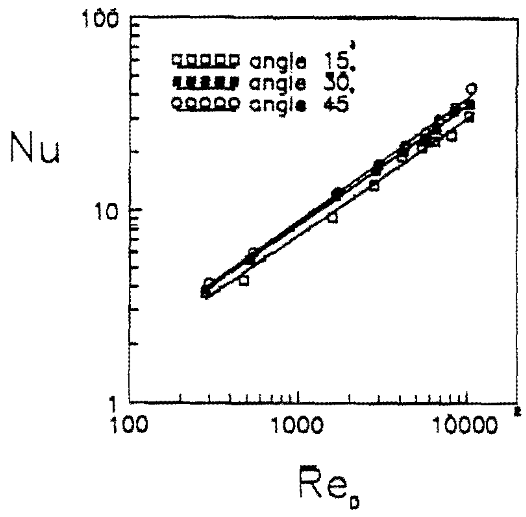

Fig. (5) Variaton of Nusselt number with Reynolds number for corrugated fin-and-tube heat exchanger

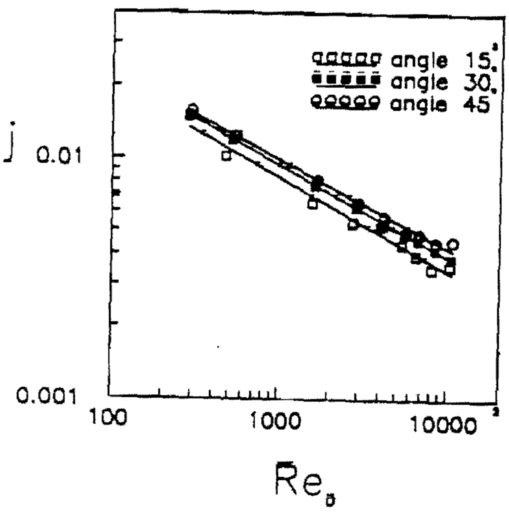

Fig. (6) Variation of Colburn factor with Reynolds number for corrugated fin-and-tube heat exchanger

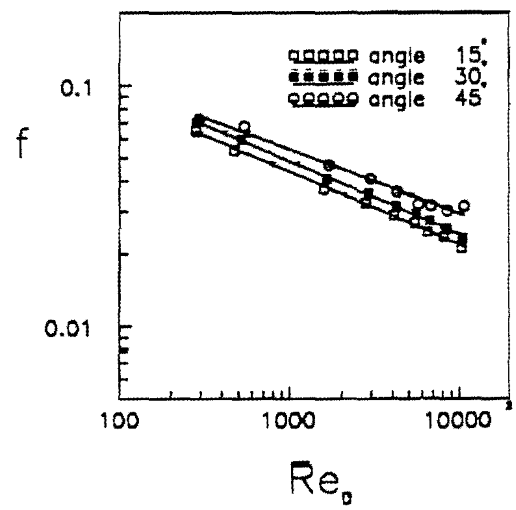

Fig. (7) Variaton of friction factor with Reynolds number for corrugated fin-and-tube heat exchanger

3.3.2 Effect of Reynolds number on the friction factor

Figure 11 shows the variation of the friction factor with Reynolds number at corrugation angle ($\beta = 15^o$) and different number of fins per inch (4,6,8). This figure shows that the friction factor decreases with increasing Reynolds number; and at the same Reynolds number the friction factor increases with increasing the number of fins per inch.

The friction factor can be correlated as;

$$f = 0.31 Re_D^{-0.3} \left(C_h / f_p\right)^{0.21} \qquad 130 \le Re_D \le 10300, \quad (3.17 \le f_p \le 6.35mm) \quad (7)$$

The maximum deviation of Eq. (7) from the experimental data is 3.1 percent.

3.4 Comparison Between Present and Previously Published Work

In the present work, the increase of heat transfer coefficient is between 18.4 and 39.3 percent. This confirms the opinion of Webb [10] who said that corrugated fins gave heat transfer coefficient 30 percent higher than for straight fins.

In the present work, the increase in Colburn factor is between 23.5 and 37.1 percent. This confirms the opinion of Giovannoni and Mattarolo [5] who said that corrugated fins gave heat transfer Colburn factor 20 to 40 percent higher than for straight fins.

3.4.1 Corrugated fin-and-tube heat exchanger with 4FPI, $\beta = (15^o, 30^o, 45^o)$

Figure 12 shows a comparison of Nusselt number between present and previously published work [3] and [11]. This figure shows that Nusselt number of the previously published work is higher than that of present work but they have all the same trend. This can be partly attributed to the non-similarity of the tested configurations.

For Xiao, et. al., [11][S_t = 43.5mm, S_L = 50mm, D = 20.11mm, 4FPI, β=15.5^o]

$$Nu = 0.083 Re_D^{0.702} \left(f_p / C_h\right)^{0.42} \quad 670 \le Re_D \le 7000, \ \beta = 15.5^o$$

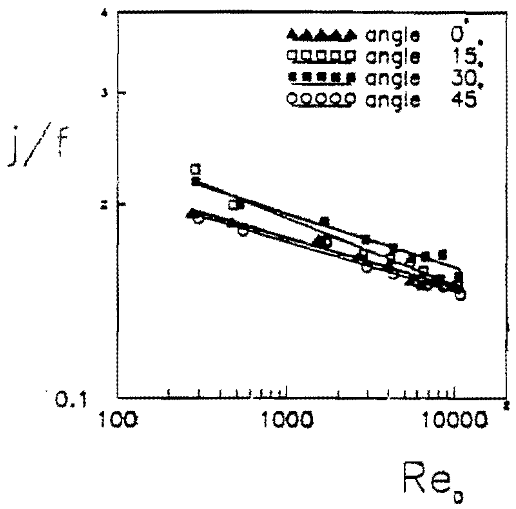

Fig. (8) Variation of j/f factor with Reynolds number for striaght and corrugated fins heat exchangers.

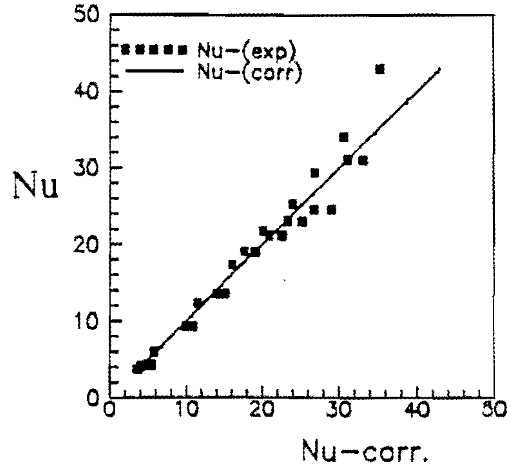

Fig. (9) Comparison between correlated and experimental data of Nusselt number at the same values of Reynolds number

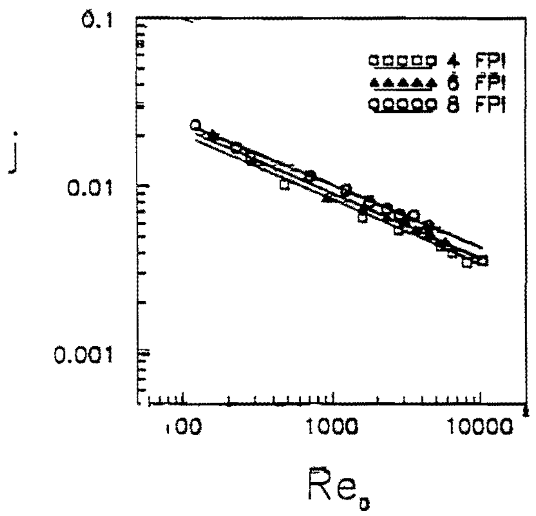

Fig. (10) Variaton of Colburn factor with Reynolds number for corrugated fin-and-tube heat exchanger

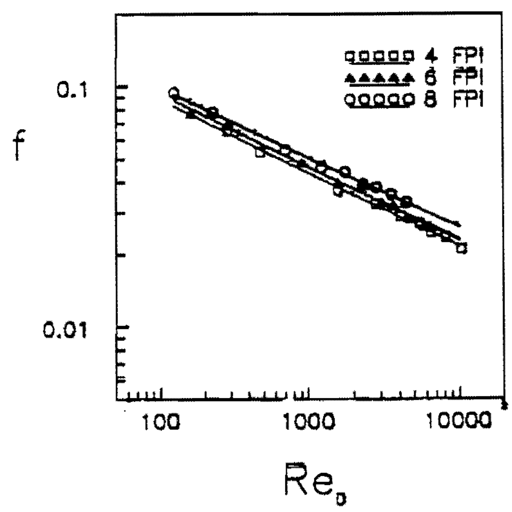

Fig. (11) Variation of friction factor with Reynolds number for corrugated fin-and-tube heat exchanger while in the present work, [S_t = 66mm, S_L = 69.2mm, D = 15.8mm, t = 0.3mm , 4FPI]

$$Nu = 0.13 Re_D^{0.61} (C_h / f_p)^{0.08} \qquad 280 \le Re_D \le 10700, \qquad 15^\circ \le \beta \le 45^\circ$$

3.4.2 Corrugated fin-and-tube heat exchanger with β = 15°, (4,6,8FPI)

Figure 13 shows a comparison of Nusselt number between present and previously published work [11]. This figure shows that Nusselt number of the previously published work is higher than that of present work but they have all the same trend , this can be partly attributed to the non-similarity of the tested configurations.

For Xiao, et. al., [11] [S_t = 43.5mm, S_L = 50mm, D = 20.11mm, (3,4, 5, 6 FPI)]

$$Nu = 0.083 Re_D^{0.702} (f_p / C_h)^{0.42}, \quad 670 \le Re_D \le 7000, \quad \beta = 15.5^\circ$$

while in the present work, [S_t = 66mm, S_L = 69.2mm, D = 15.8mm, β = 15°, (4,6,8FPI)]

$$Nu = 0.07 Re_D^{0.61} (C_h / f_p)^{0.87} \quad 130 \le Re_D \le 10300, (3.17 \le f_p \le 6.35mm)$$

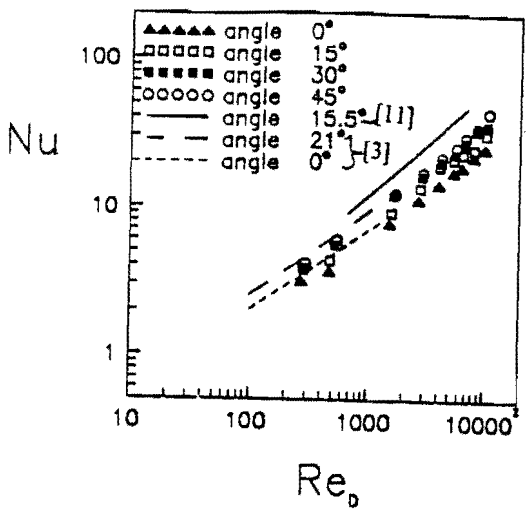

Fig. (12) Comparison of Nusselt number between the present and perviously published work at different corrugation angles.

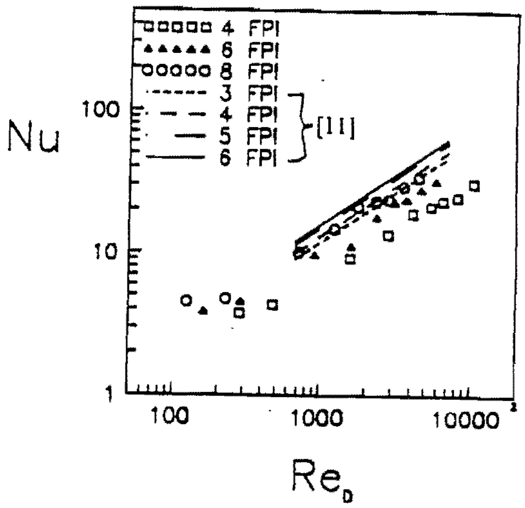

Fig. (13) Comparison of Nusselt number between the present and perviously published work at different number of fins per inch.

4. CONCLUSION

The principal conclusions which had been reached are :

The experimental results indicated that the corrugated fins have an important effect on the heat transfer characteristics and friction. The study indicated that the augmentation due to the effect of corrugated fins increases with increasing Reynolds number, when the average velocity through the heat exchanger increases, the heat transfer coefficient and the pressure drop increase. If Reynolds number increases, both Colburn factor and friction factor decrease.

For the same Reynolds number the Colburn and friction factors of the corrugated surfaces ($\beta = 15^{o}, 30^{o}, 45^{o}$) are higher than those of the straight surface. This is due to the turbulence that is generated by the corrugated surface (acting as a turbulence promoter), thereby increasing the heat transfer coefficient. The complex flow associated with the corrugated nature of the fin profile generates a form drag that increases the pressure drop, seen as increased friction factors.

The corrugation angle 45^{o} has the highest Colburn and friction factor, followed by the angle 30^{o} while the corrugation angle 0^{o} (zero) has the lowest Colburn and friction factors.

The corrugation angle 30^{o} has the highest ratio of heat transfer Colburn factor to the friction factor (j/f) followed by the angle 15^{o} while the corrugation angle 45^{o} has the lowest value of that ratio. So, the problem of heat transfer and friction should be solved together.

The number of fins per inch have an important effect on the heat transfer and friction. When the number of fins per inch increases, at the same rate, the heat transfer and friction increase.

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Nomenclature

A	transfer surface area ,	m^2
A_f	fin surface area ,	m^2
A_{ff}	minimum free-flow area of fin passage ,	m^2
A_{fr}	frontal area,	m^2
C_h	corrugation height,	m
C_p	corrugated fin pitch ,	m
D	tube outer diameter	m
D_h	hydraulic diameter of the flow passage ,$A_{ff}L_m / A$	m
G	maximum mass velocity ,	$kg/m^2.s$
f	friction factor	$f = \frac{A_{ff}\upsilon_i}{A\upsilon_m}\left[\frac{2\Delta P}{G^2\upsilon_i} - (1+\sigma^2)(\frac{\upsilon_o}{\upsilon_i} - 1)\right]$
f_p	fin pitch ,	m

h	heat transfer coefficient ,	$W/m^2.K$
j	Colburn heat transfer factor,	$= Nu / Re_D . Pr^{1/3}$
k	thermal conductivity ,	$W/m.K$
L_m	streamwise length	m
Nu	Nusselt number,	$=h\ D_h / k$
Pr	Prandtl number	
ΔP	pressure drop through heat exchanger core ,	N/m^2
Re_D	Reynolds number based on D_h ,	$=V_m\ D_h / \nu$
Re_L	Reynolds number based on S_L ,	$=V_m\ S_L / \nu$
S_L	longitudinal pitch ,	m
S_t	transverse pitch ,	m
t	fin thickness ,	m
V_m	maximum velocity ,	m/s
υ_i	specific volume of the upstream,	m^3 / kg
υ_m	mean specific volume ,	m^3 / kg
υ_o	specific volume of the downstream,	m^3 / kg

Greek letters

β	corrugation angle ,	deg.
ν	kinematic viscosity of air ,	m^2 / s
σ	the ratio of A_{ff} / A_{fr}	

تحســين انتقــال الحرارة خلال مبادل حــرارى باستخدام زعانــف ممــوجة

يهدف هذا البحث الـى دراســة تأثير زاوية التموج وعدد الزعانف فى البوصة والسرعة على خصائص انتقال الحرارة والاحتكاك للمبادلات الحرارية ذات الأنابيب والزعانف على ست مـبادلات الحرارية لها نفس هندسـة الشكل، فى أربعة منهم أربع زعانف فى البوصة مع اختلاف زاوية التموج (صفر ،١٥، ٣٠، ٤٥ درجه) بالإضافـة الى مبـادلين حراريين بزاوية تموج ١٥ درجه مع (٨،٦) عدد الزعانف فى البوصة وأجريت الإختبـارات داخل نفق هوائى وقيسـت سرعة الهواء بأنبوب بيتوت وأظهرت النتائج انه عند زيادة سرعة سـريان الهواء تزداد قيمة كل من معامل انتقـال الحرارة والإنخفاض فى الضغط و تبين أنـه بزيادة عدد رينولدز تقل قيمة كل من معـامل كلبرن لإنتقـال الحـرارة ومعامل الاحتكاك،كما بينت المقارنـة أن زاوية التموج لها تأثير هام على إنتقال الحرارة والاحتكـاك وأن عـدد الزعانف فى البوصة له تأثير هام على أنتقال الحرارة والاحتكاك وأنه بزيادة عدد الزعانف فى البوصة تزداد قيمة كل مـن معامل أنتقال الحرارة ومعامل الأحتكاك.