Effects of fin per inch on heat transfer and pressure drop of an air cooler with circular and hexagonal fins

Ali Falavand Jozaei[1], Mehdi Mosavi Navaei[2], Alireza Baheri[3],
[1]Department of Mechanics, Ahvaz Branch, Islamic Azad University, Ahvaz, Iran
[2]Arvand Petrochemical Co., Iran
[3]Engineering Group, Dezful Branch, Islamic Azad University, Dezful, Iran
Corresponding author email: falavand@iauahvaz.ac.ir

Abstract: Operation enhancement in an air cooler (heat exchanger) depends on rate of heat transfer, and pressure drop. In this paper for a given heat duty, study of the effects of FPI (fin per inch) and fin type (circular and hexagonal fins) on two parameters mentioned above is considered in an air cooler in Iran, Arvand petrochemical. A program in EES (Engineering Equations Solver) software moreover, Aspen B-JAC and HTFS+ softwares are used for this purpose to solve governing equations. At first the simulated results obtained from this program is compared to the experimental data for two cases of FPI. The effects of FPI from 3 to 15 over heat transfer (Q) to pressure drop ratio (Q/Δp ratio). This ratio is one of the main parameters in design, rating, and simulation heat exchangers. The results show that heat transfer (Q) and pressure drop increase with increasing FPI (fin per inch) steadily, and the Q/Δp ratio increases to FPI=12 (for circular fins about 47% and for hexagonal fins about 69%) and then decreased gradually to FPI=15 (for circular fins about 5% and for hexagonal fins about 8%), and Q/Δp ratio is maximum at FPI=12. The FPI value selction between 8 and 12 obtained as a result to optimum heat transfer to pressure drop ratio. Also by contrast, between circular and hexagonal fins results, the Q/Δp ratio of hexagonal fins more than Q/Δp ratio of circular fins for FPI between 8 and 12 (optimum FPI)
[Falavand Jozaei A, Navaei MM, Baheri A. **Effects of fin per inch on heat transfer and pressure drop of an air cooler with circular and hexagonal fins.** *Life Sci J* 2013;10(1):49-56] (ISSN:1097-8135). http://www.lifesciencesite.com. 8

Keywords: air cooler; fin per inch; heat transfer; pressure drop; circular and hexagonal fins

1. Introduction

Air cooled heat exchangers are widely used in many industrial areas such as chemical process, power generation, petroleum refining, refrigeration, air-conditioning and etc. Air cooled heat exchangers are used under conditions including high pressure and temperature, as well as corrosive fluids and environments. Common applications include gas re-injection, gas lift and pipeline applications, cooling and condensing of hydrocarbon gases, and cooling of machinery oil and heavy hydrocarbons. Air cooled steam condensers are a special type of heat exchanger employed to condense steam at the exhaust end of steam turbines for both power generation and mechanical drive applications.

In the open literature, there are many studies on compact heat exchangers and some of them are focused onhelically finned tubes. The majority of the papers have studied solid fins, like Genic et al. [15]. Only a few papers focus on segmented fins, and so, there are few correlations for heat transfer and pressure drop. One of the most commonly used models was developed by Weierman [16], who developed heat transfer and friction factor correlations for different tube bundles (inline and staggered) with solid and serrated fins. These correlations were modified by ESCOA (Extended Surface Corporation of America) in order to obtain better predictive models (Ganapathy [17]).

The objective of the present paper is the comparative analysis of heat transfer and pressure drop models with experimental data for circular and hexagonal fins and FPI in an air cooler Heat Exchanger in Iran, Arvand petrochemical on an industrial scale. The present paper shows a comparative analysis of heat transfer and pressure drop models for fin type (circular and hexagonal fins) and FPI (fin per inch) in an air cooler on an industrial scale (with geometric parameters and thermo physical properties defined at tables 1, 2). Heat transfer and pressure drop were evaluated with the models of equivalent circular fin methods and sector method for hexagonal fins.

Table 1: Geometric Parameters

Dimension	Value/Type
Tube pitch pace row	37
Tube pitch rows deep	32
Tube OD× wall thk.(mm)	15.88×1
Tube pattern	Staggered
Material(Tube)	copper
Material(Fins)	Aluminum
Fin Type	L-type tension wrapped
Fin outside diameter(mm)	36
Fin thickness(mm)	.4
Tube Length(m)	2.4
Tube passes per bundle	2
Tube rows deep per bundle	4
Number of Tubes per bundle	132

Table 2: Thermo physical properties

Fluid allocation	**Shell Side**		**Tube Side**	
Fluid name	Air		water	
	inlet	outlet	inlet	outlet
Mass flow(kg/h)	51480		68000	
Operating Temperature (°C)	90	40	25	-
Operating Pressure (barg)			3.8	3.5
Altitude above sea level(m)	125			
Minimum ambient temperature(°C)	-3.5		-	
Fouling Factor	-		0.00035 m^2.k/w	

2 .Fin-tube systems for air coolers

The relatively low heat transfer coefficients of flowing cooling air as compared to those of the products to be cooled or condensed may be partly compensated by a surface extension on the air side. This is realized by fin-tube heat exchanger bundles. By finning the tubes, the heat exchange surface may be extended to 10 to 2.5 times the bare tube surface. The surface extension is optimized on the basis of considerations of economy and manufacture. One criterion for the optimization is the specific performance increase in the heat transfer attainable per cost unit C, which first grows with increasing surface extension A/A_i but decreases after having reached the optimum value [1]. The vertex of this function indicates the optimum surface extension, which grows with increasing tube-side heat transfer coefficient. In the optimization coefficient UA/S/C, both A/S and U depend on the surface extension. The parameter A/S, the surface area in contact with air per square meter of face area, is merely a geometric factor and can be determined easily on the basis of the selected fin-tube type. The quantity U, the heat transfer coefficient, however, is very complex. According to Eq. l, it depends on a large number of parameters in addition to the surface extension A/A_i [1]:

$$\frac{1}{UA} = \frac{1}{\eta_F \alpha_0 A} + \frac{1}{\alpha_i A_i} + R_i \qquad (1)$$

Where α_0 = mean value of the locally varying air-side heat transfer coefficient (W/m^2 K). It also depends on the following characteristics: Cooling air velocity, Spacing between the fins, Geometric shape of the fins (round, elliptical, rectangular), Degree of turbulence of the cooling air, artificial turbulence intensifiers on the surface

η_F = fin efficiency, depending on: The fin geometry (height, thickness, cross sectional shape), the fin material (thermal conductivity coefficient), the heat transfer coefficient.

α_i = product-side heat transfer coefficient of the fluid in the tubes (W/m^2 K).

R_i = further heat transfer resistances from the cooling air side up to the product side (m^2K/W), in which the contact resistance between fin and bare tube as well as the tube-side fouling may be essential quantities, whereas the thermal conductivity resistance of the tube wall and the fouling on the air side, which are also contained in R_i, mostly play a minor role.

A = reference area for the overall heat transfer coefficient U. This reference area is arbitrary and may be any real or assumed surface, e.g., the internal tube surface, the external tube surface, the total surface area, or a combination of various surfaces. Consequently, the quantity of the overall heat transfer coefficient U is no standard of valuation. It is only admissible for a comparison of various fin tubes where the reference area A in Eq. 1 is identical.

3 .Thermal rating

The thermal rating first requires a reasonable adaptation of the means to be selected to the specified requirement. This implies a certain experience. Shape, size, finning, and fin-tube material must be suitable for the fluid to be cooled or condensed and must be adapted to its physical properties. The rating is then carried out by the following step-by-step method.

3.1. Estimation of the tube-side heat transfer coefficients

On the basis of the specified task, an initial guess of the expected α_i values may be made by means of tables [1]. The indicated values refer to typical conditions of air coolers and air condensers. When the tube-side conditions such as flow velocity and temperature of the streaming fluids or the temperature difference of condensing or evaporating fluids are specified or known, the tube-side heat transfer coefficients may be determined more exactly.

3.2. Selection of fin tubes

The estimated heat transfer coefficient gives a first idea of the required and expedient surface area ratio

A/A_i of the fin tube to be chosen. The thermal and hydraulic data of the selected fin tube should be available as a function of the cooling air velocity u [1], according to Eq.l.

3.3. Selection of the cooling air velocity u

The cooling air velocity is fixed within rather narrow limits, since the increase of the air-side pressure drop with growing velocity is almost square and due to the low static pressure of 100 to 200 Pa developed by conventional fans. It ranges mostly from 2 to 4 m/s, depending on the face area, the number of tube rows n, and on the admissible and feasible temperature rise of the cooling air.

3.4. Overall heat transfer coefficient U

After the α_i value and the cooling air velocity have been determined, the overall heat transfer coefficient for a selected fin-tube system may be obtained from Eq.l, When an additional product-side fouling r_{foul} needs to be taken into account, the actual service value α_{iserv} is first determined, then:

$$\alpha_{iserv} = \frac{1}{\alpha_i} + r_{foul} \tag{2}$$

The fouling resistance r_{foul} may be taken from pertinent manuals or gotten from experience. Generally, air-side fouling need not be considered for the U value. Although relatively high air-side fouling may adversely affect the air flow through the unit and thus decrease the effective temperature difference, it scarcely influences the U value because of the relatively small air-side heat flux.

3.5. Number of tube rows

The number of tube rows needed depends on both the specific requirement and the efficiency of the selected fin-tube system. Roughly estimated, these two factors can be expressed by a quantity a, which includes the temperature difference between product and cooling air inlet as well as the value U(A/S) of the fin tube[1].

$$a = \frac{t_{prod,in} - t_{air,in}}{UA/S} = \frac{\Delta v_0}{UA/S} \tag{3}$$

$$n_R = C_1 a^{C_2} \tag{4}$$

Where $C_1 = 24$ for fin tubes of the kind, $C_2 = 0.49$

3.6. Product thermal number

One auxiliary term for the further thermal rating is the dimensionless number

$$\varphi_{prod} = \frac{\Delta t_{prod}}{\Delta v_0} = \frac{t_{prod,in} - t_{prod,out}}{t_{prod,in} - air,in} \tag{5}$$

It is used later to determine the effective mean temperature difference (EMTD) and the surface area needed.

3.7. Coolant design number

The dimensionless design number applicable for one tube row is now calculated on the basis of the available results.

$$k = \frac{UA/S}{u\rho c_p} \tag{6}$$

Where c_P is the specific heat of air at constant pressure.

3.8. NTU number

The dimensionless NTU number (number of transfer units) is obtained by the product of the coolant design number and the number of tube rows:

$$NTU = n_R^k = \frac{\Delta t_{air}}{EMTD} \tag{7}$$

For optimum design it normally lies in the range of $0.8 < NTU < 1.5$. This quantity already represents a control value for the available quantities u and n_R.

Coolant thermal number

The dimensionless value $\varphi_{air} = \Delta t_{air}/\Delta v_0$ for various types of flow in air coolers is given in [12, 13]. For three typical flow arrangements with air coolers: cross flow; cross-counter flow return bend; counter flow), φ_{air} is given by Eqs. 8, 9, and 11. Besides the known quantity NTU, the equations also contain the quantity $\tau = \Delta t_{prod}/\Delta t_{air}$, which must first be estimated in order to obtain φ_{air} τ generally lies between 0 and 1, but may also be higher. For isothermal condensation, $\Delta t_{prod} = 0$ and $\tau = 0$, so the same φ_{air} is obtained for all flow types.

Arrangement I: cross flow

On the tube side, one or more passes, side by side:

$$\varphi_{air} = \frac{1-exp\left[-\tau\left(1-e^{-NTU}\right)\right]}{\tau} \tag{8}$$

Arrangement 2: cross-counter-flow return bend

On the tube side two passes in the counter-flow direction to the air flow:

$$\varphi_{air} = \frac{1}{\tau}\left[1 - \frac{1}{1 + \left(1 - \frac{\varphi_0}{2}\right)\left(e^{2\tau\varphi_0} - 1\right)}\right] \tag{9}$$

With

$$\varphi_0 = 1 - e^{NTU/2} \tag{10}$$

Arrangement 3: counter flow

On the tube side four or more passes in the counter flow direction to the air flow:

$$\varphi_{air} = \frac{1 - e^{-(1-\tau)NTU}}{1 - \tau e^{-(1-\tau)NTU}} \tag{11}$$

The flow arrangements are selected according to the product volume to be cooled or condensed and the temperature difference between the product and the cooling air. Arrangement 1 applies when large product volumes are to be cooled or condensed or when high temperature differences are involved. Arrangement 2 applies to liquid coolers with small volumes and low temperature differences. Arrangement 3 applies to high-pressure coolers or coolers where the temperatures of the two flows approach or overlap each other.

3.9. Effective mean temperature difference EMTD

The EMTD is obtained from the definition of NTU according to Eq. 7 as follows:

$$EMTD = \frac{\Delta t_{air}}{NTU} = \frac{\varphi_{air}\Delta v_0}{NTU} \tag{12}$$

In order to simplify the calculation and avoid a trial-and-error procedure, Eqs. 8-11 can be rearranged to get the expression

$$\frac{EMTD}{\Delta v_0} = \frac{\varphi_{air}}{NTU} \tag{13}$$

Which can be solved graphically as a function of Δt_{prod} and NTU.

3.10. Surface area A

The surface area needed is

$$A = \frac{\dot{Q}}{U.EMTD} \tag{14}$$

Where $\dot{Q}$ is the total heat transferred in the exchanger.

3.11. Face area S

The coolant-side face area needed is obtained from the expression

$$S = \frac{A}{\frac{A}{S}.\eta_R} \tag{15}$$

This face area is so apportioned to length and width that reasonable bundle dimensions are obtained.

2 .Analytical method

Fin efficiency equations for dry plain circular fins under the aforementioned assumptions are reported in many handbooks [3, 14]. The analytical solution for a circular fin, which is the same as for an angular sector of circular fin as presented in figure 1, with adiabatic fin tip is given by eq.16, where I_n and K_n are the modified Bessel functions of first and second kind.

$$\eta_f = \frac{Q_{Fin}}{Q_{Max}} = \frac{Q_{Fin}}{h\,A_f\,\theta_n} \tag{16}$$

$$\eta_f = \frac{2r}{m(r_f^2 - r^2)}\left[\frac{K_1(mr)I_1(mr_f) - K_1(mr_f)I_1(mr)}{K_1(mr_f)I_0(mr) + K_0(mr)I_1(mr_f)}\right] \tag{17}$$

Several studies have been performed in order to simplify this circular fin efficiency formulation by avoiding the use of modified Bessel functions. Among all the approximations, the Schmidt approximation [9] is the most widely used one. Hong and Webb [10] propose to slightly modify the Schmidt equation in order to obtain better accuracy (eqs. 18 and 19). In the present study, it is proposed to use a modified φ parameter (eq. 20) in equation (18). With this formula, the error between the analytical solution (eq. 16) and the approximation does not exceed 2% over the practical range of conditions $r_f/r \leq 6$ and $m(r_f - r) \leq 2.5$.

$$\eta_f = \frac{\tanh(mr\emptyset)}{mr\emptyset}\cos(mr\emptyset)\ ; \quad m = \sqrt{\frac{2h}{\lambda_f\delta_f}} \tag{18}$$

$$\emptyset = \left(\frac{r_f}{r} - 1\right)\left(1 + 0.35\ln\left(\frac{r_f}{r}\right)\right) \tag{19}$$

$$\emptyset_m = \left(\frac{r_f}{r} - 1\right)\left[1 + \left(0.3 + \left(\frac{m(r_f - r)}{2.5}\right)^{1.5 - \frac{1}{12}\frac{r_f}{r}}\left(0.26\left(\frac{r_f}{r}\right)^{0.3} - 0.3\right)\right)\ln\left(\frac{r_f}{r}\right)\right] \tag{20}$$

Fin and tube heat exchangers are generally composed of continuous plate fins. The fins are metal sheets pierced through by the tube bank. The tube lay-out is in inline or staggered configuration (fig. 1), with a clear advantage for the staggered lay-out. In order to express the fin efficiency of such continuous plate fins, the fin is divided in unit cells. Considering that all the tubes are at the same temperature, the adiabatic zones of the fins determine the unit cells, as presented in figure 1.

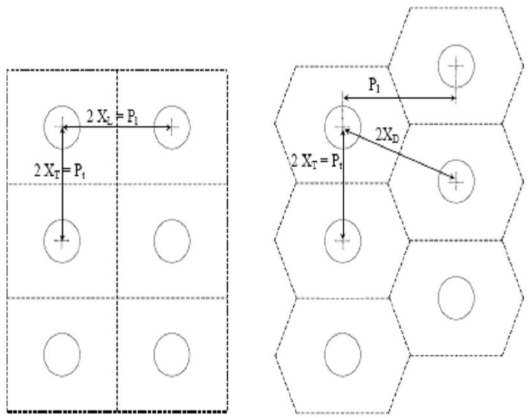

Figure 1: Unit cells for inline and staggered tube layouts with plain plate fins

The considered fin shape is rectangular for the inline configuration and hexagonal for the staggered lay-out. Two methods are used in order to calculate the efficiency of these rectangular or hexagonal fins from the circular fin efficiency with adiabatic fin tip condition. The most accurate method is the sector method. Nevertheless, the equivalent circular fin method is simpler and is more widely used.

4.1 Equivalent circular fin methods

Gardner [8] and Schmidt [9] have shown that in the case of rectangular and hexagonal fins, the fin efficiency could be treated as for a circular fin, by considering an equivalent circular fin radius. For the calculation of the equivalent circular radius, two approaches are possible. The first one consists in considering a circular fin having the same surface area as the rectangular or hexagonal fin. The other method is the Schmidt method in which correlations are developed in order to find an equivalent circular fin having the same fin efficiency as the rectangular fin (eq. 21) or the hexagonal fin (eq. 22).

$$\frac{r_{f,eq}}{r} = 1.28\frac{X_T}{r}\sqrt{\frac{X_L}{X_T} - 0.2} \tag{21}$$

$$\frac{r_{f,eq}}{r} = 1.27\frac{X_T}{r}\sqrt{\frac{X_D}{X_T} - 0.3}\ ;\ 2X_D = \sqrt{P_l^2 + \frac{P_t^2}{4}} = \sqrt{4X_L^2 + X_T^2} \tag{22}$$

4.2 Sector method

The sector method could be characterized as a semi-analytical method. The plain fin surface is divided in several circular sectors generated from the tube center and fitting the fin geometry profile (figures 2 and 3). The inner radius of each sector is equal to the tube radius while the outer radius is equal to the tube center to fin tip distance, corresponding to the considered sector. Doing so, it is possible to approximate every kind of fin profile.

Considering that the conductive thermal flux through each angular sector is purely radial, the rectangular or hexagonal fin efficiency is the surface weighted average of the efficiency of each sector (eq. 23). With the radial flux assumption, the lateral faces of each sector are adiabatic. The sector base is at constant temperature and the sector tip is considered adiabatic. Consequently, the sector efficiency is analytically evaluated from the circular fin efficiency formulas, eq. 17 for the exact solution with Bessel functions, or other approximated equation (Schmidt, Hong and Webb, eqs. 18 to 20).

$$\eta_r = \frac{\sum_{i=1}^{n} \eta_i A_i}{\sum_{i=1}^{n} A_i} \tag{23}$$

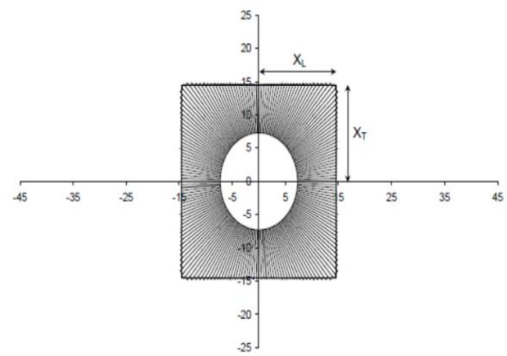

Figure 2: Sector method applied to square fin $X_L/X_T=1$; $X_T/r = 2$

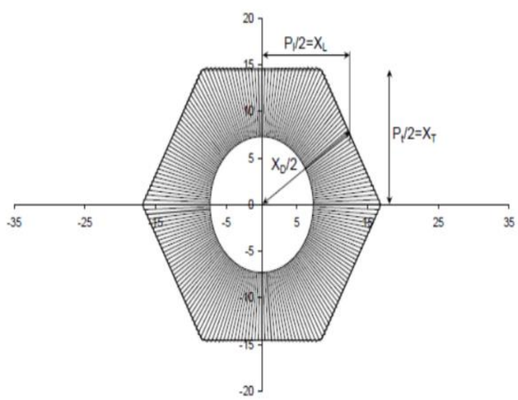

Figure 3: Sector method applied to equilateral hexagonal fin $P_L/P_T = 0.866$; $P_T/2r = 2$

5. Results and discussion

In this paper, the effects of FPI (fin per inch) and fin type (circular and hexagonal fins) on rate of heat transfer, and pressure drop are considered in an air cooler in Iran, Arvand petrochemical with geometric parameters and thermophysical properties (defined at tables 1, 2). A program in EES (Engineering Equations Solver) software moreover, Aspen B-JAC and HTFS[+] software's are used for solve governing equations. For this purpose, according to technical specifications and input fluids conditions the program was run for that air cooler has 10 FPI with circular fins. In this case output temperatures from this program are 2.5 percent less than experimental temperatures. To reconfirm the accuracy of the program, the number of FPI became 12, and again output temperature from the program was compared to experimental data. In this new case, output temperatures from this program are 2 percent less than experimental temperatures. After validation of numerical model, the effects of FPI variations from 3 to 15 was considered on heat transfer, and pressure drop.
From this study we conclude that:

1- Variation of surfuce per unit-finned tube versus FPI is shown in figure 4. It is found that with the increase of FPI, surfuce per unit-finned tube increases, but after increasing FPI more than 12, variation in surfuce per unit-finned tube is gradual (approximately 2 percent).

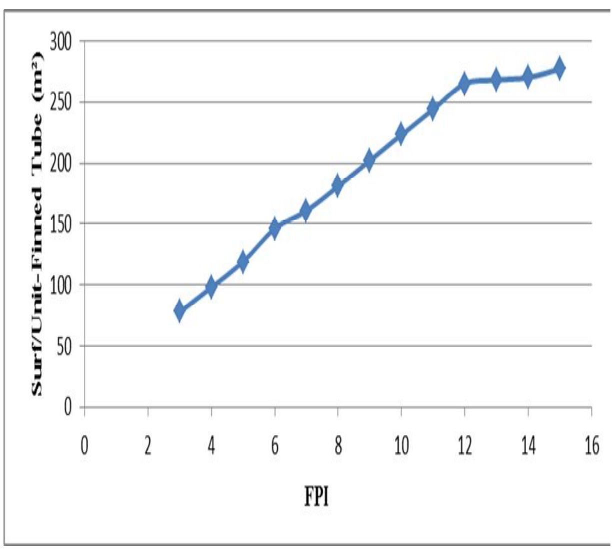

Fig.4. Surf/Unit-Finned Tube versus FPI

2- Fig.5. shows heat exchanged versus FPI. It can be observed in this Fig. The increase of FPI causes the increase of heat transfer over tubes. The increase of FPI from 3 to 12 causes the increase of heat exchanged about 50% (1.5 times). Moreover, increase of FPI from 12 to 15 causes the increase of heat exchanged about only 3%.

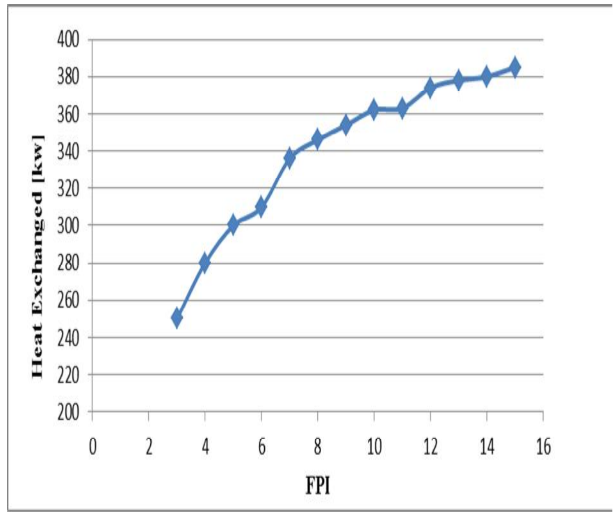

Fig.5. Heat Exchanged versus FPI

3- Pressure drop versus FPI is shown in Fig. 6. According to Fig.6 with the increase of FPI, pressure drop increases. The increase of FPI from 3 to 12 causes only a slight increase of heat exchanged about 1%, but there is a sharp increase between FPI=12 and FPI=15.

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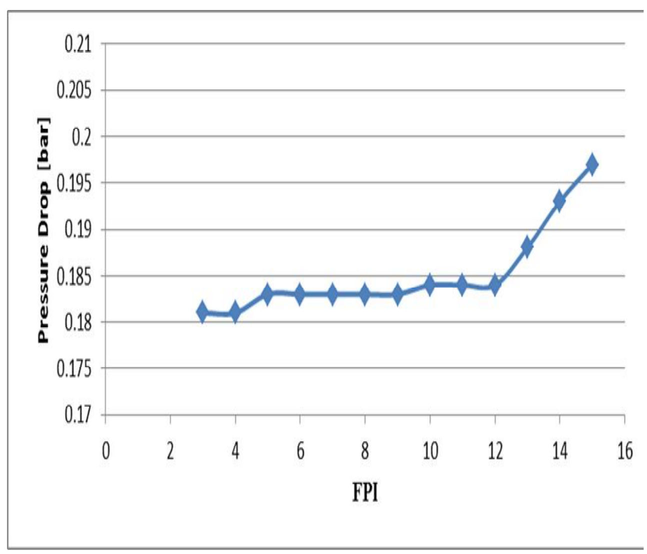

Fig.6. Pressure Drop versus FPI

5- Figure 7 show that variation of heat transfer to pressure drop ratio (Q/Δp ratio) versus FPI for two fin types (circular and hexagonal fins). It can be found that for both circular and hexagonal fins, the Q/Δp ratio increases to FPI=12 and then decreased gradually to FPI=15 and the Q/Δp ratio is maximum at FPI=12. Also by contrast, between circular and hexagonal fins at fig. 7, the Q/Δp ratio of hexagonal fins more than Q/Δp ratio of circular fins for FPI between 8 and 12 (optimum FPI).

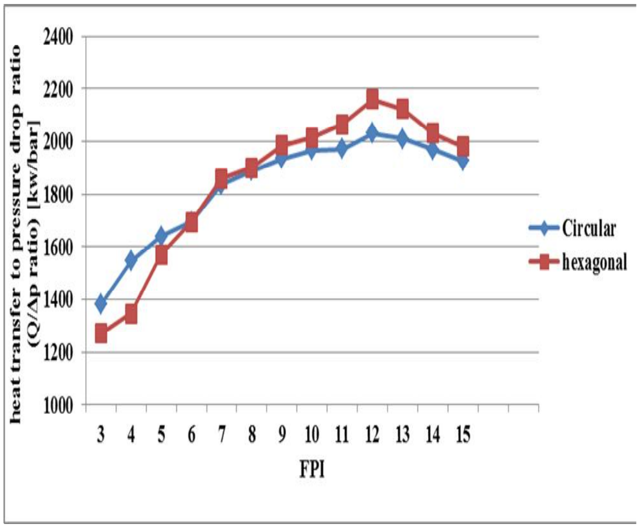

Fig.7. heat transfer to pressure drop ratio (Q/Δp ratio) versus FPI

6. Conclusions

Effects of FPI (fin per inch) on heat transfer and pressure drop of an air cooler petrochemical industry with circular and hexagonal fins is studied. Comparsion of variation of heat transfer and pressure drop in an air cooler, according to FPI (Figures 4 - 7) shows that increase of FPI causes the increase of surfuce per unit-finned tube, heat transfer and pressure steadily. But the Q/Δp ratio increases to FPI=12 (for circular fins about 47% and for hexagonal fins about 69%) and then decreased gradually to FPI=15 for both fins (for circular fins about 5% and for hexagonal fins about 8%), and Q/Δp ratio is maximum at FPI=12. The FPI value selction between 8 and 12 obtained as a result to optimum heat transfer to pressure drop ratio. Also by contrast, between circular and hexagonal fins results, the Q/Δp ratio of hexagonal fins more than Q/Δp ratio of circular fins for FPI between 8 and 12 (optimum FPI)

Acknowledgement

This work is based on a research project found by Islamic Azad University.

Corresponding Author:
Ali Falavand
Ahvaz Branch, Islamic Azad University, Ahvaz, Iran
E-mail: falavand@iauahvaz.ac.ir

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10/22/2012