# ENHANCING AIR SIDE HEAT TRANSFER COEFFICIENT OF FLAT TUBE AIR COOLED CONDENSERS

# تحسين معامل إنتقال الحرارة لمكثفات مبردة بالهواء ذو أنابيب مفلطحة

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الخلاصة:

تم عمل دراسة نظرية لإنتقال الحرارة بالحمل الجبرى حول أنابيب دقيقة مفلطحة والمستخدمة فى المكثفات المبردة بالهواء. تم دراسة حالتين لميل الأنابيب المفلطحة : الحالة الأولى بجعل ممارات الهواء بين الأنابيب مفرق-مجمع ، بينما الحالة الثانية بجعل الأنابيب موازية لبعضها ومائلة على إتجاه السريان بزاوية معينة. تم عمل نموذج رياضى لماسورة دقيقة مفلطحة مصنوعة من الألومنيوم مبردة بالهواء. وتم إستخدام أحد البرامج المدمجة (CFD) لحل معادلات السريان وكمية الحركة والطاقة. ووجد أن ميل الأنابيب المفلطحة بزاوية معينة يحسن إنتقال الحرارة بالحمل وبالتالى يتحسن الأداء الحرارى للمكثفات المبردة بالهواء. وقد بينت النتائج النظرية فى كلتا الحالتين أن زاوية الميل 4° على إتجاه السريان والمناظرة لنسبة باعية 0.58 هى أفضل زاوية. وهذا يؤدى إلى تحسين معامل إنتقال الحرارة بالحمل ناحية الهواء مابين 1.46 إلى 1.469 بينما يزيد معامل الفرق فى الضغط مابين 2.12 إلى 1.95 للحالة الأولى والثانية على الترتيب.

## Abstract

Forced convection heat transfer for air side over aluminum extruded micro-channel flat tubes, which used in air cooled condensers, was studied theoretically. Mathematical modeling of air flow outside the aluminum flat tubes were carried out to study the proposed inclination angles and evaluate the thermal performance for different operating parameters. A computational fluid dynamic software (**CFD**) is used to solve this problem. Two proposed cases for inclination of the flat tubes are studied. The first case is to make convergent and divergent channels for air flow, while the second case is tilting of all tubes in parallel to each other. Inclination for the flat tubes by a certain angle improves the convection side and in turn improves the overall thermal performance of the air cooled condenser. The theoretical results show that the optimum angle for the proposed two cases was about 4° with corresponding aspect ratio of 0.58. This leads to enhancement the heat transfer coefficient by factor (Kh) of 1.469 and 1.46 against increase in pressure drop factor (KP) of 2.12 and 1.95 for case-1and case-2, respectively.

**Keywords:** CFD – flattened tube heat exchanger – air cooled condenser

**Nomenclature**

| | |
|---|---|
| Ar | Aspect ratio =H/L, - |
| $C_1$ | Constant, Eqs. 5 and 6. |
| $C_2$ | Constant, Eq. 6 |
| $C_D$ | Constant, Eq. 8 |
| $C_\mu$ | Constant, Eq. 7 |
| $D_h$ | Hydraulic diameter, m |
| g | Gravitational acceleration, $m/s^2$ |
| h | Air-side convective heat transfer coefficient, $W/m^2.K$ |
| H | Transverse pitch of parallel tubes. m |
| l | Mixing length, m |
| k | Kinetic energy, $m^2/s^2$ |
| Kh | Enhancement factor of h = $h_\beta/h_o$ |
| Kp | Pressure drop factor= $\Delta P_\beta/\Delta P_o$. |
| L | Width of flat tube, mm |
| Nu | Nusselt number, - |
| P | Pressure, Pa |
| $S_T$ | Source term of heat, K/s |
| T | Temperature, K |
| u | Velocity in x-direction, m/s |
| v | Velocity in y-direction, m/s |
| $u'$ | Fluctuating velocity in x-direction, m/s |
| $v'$ | Fluctuating velocity in y-direction, m/s |

**Greek Symbols**

| | |
|---|---|
| $\alpha$ | Thermal diffusivity, $m^2/s$ |
| $\varepsilon$ | Dissipation rate, $m/s^2$ |
| $\sigma$ | Constant, Eqn.s 4, 5, and 6 |
| $\mu$ | Dynamic viscosity, kg/m.s |
| $\mu\varphi$ | Viscous dissipation rate, $kg/m.s/s^2$ |

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| | |
|---|---|
| $\beta$ | Inclination angle of flat tubes, deg. |
| $\psi$ | Stream function, $m^3s^{-1}/m$ |
| $\nu$ | Kinematic viscosity, $m^2/s$ |
| $\eta$ | overall performance =Kh/KP |

**Subscripts**

| | |
|---|---|
| av | average |
| f | face |
| o | Case of horizontal parallel flat tube |
| opt | Optimum |
| k | turbulent |
| T | thermal |
| $\varepsilon$ | dissipation |
| $\rho$ | Air density, $kg/m^3$ |
| $\mu\phi$ | Viscous dissipation term |

## 1. Introduction

Air-cooled finned-tube condensers are widely used in refrigeration and air-conditioning applications. For the same amount of heat transfer, the operation of air cooled condensers is more economic compared with water cooled condensers [1]. Typically air-cooled condensers are of round tubes and finned type.

A micro-channel flat tubes heat exchanger is one of the potential alternatives for replacing the conventional finned tube heat exchangers. This kind of heat exchangers is made of a flat tube with several independent passages in the cross-section, as shown in Fig. (1) and formed into a serpentine or a parallel flow arrangement. In these heat exchangers, a multitude of corrugated fins with louvers are inserted into the gaps between flat tubes. The flat tube design offers higher thermal performance and lower pressure drop than the finned-round tube heat exchangers [2]. Brazed aluminum heat exchanger is made from micro-channel flat tubes in parallel to each other which is called parallel flow heat exchanger (PFHE). The key advantage of the brazed aluminum design is smaller size and lower weight than finned-round tube condensers. The heat capacity of a parallel-flow heat exchanger (PFHE) is 150–200 % larger than that of the conventional heat exchanger [3]. This high heat capacity of the PFHE can meet the requirements of compactness and lightness. Oval and flat cross-sectional tube for finned tube heat exchangers provides a higher heat transfer performance as compared to those formed with round tube geometry as mentioned by Chang et al. [1]. The effect of tube profile change from round to flat shape on condensation has been investigated experimentally by Wilson et al. [4]. They found a considerable enhancement of condensation heat transfer coefficient inside tube and an increase in pressure drop as the tube profile is flattened. Also, there is a significant reduction in refrigerant charge due to flattened tubes.

The condensation of refrigerants in multi-port micro-channel extruded tubes has been investigated by many authors [5-7]. All of them concluded that the micro-channel flat tube enhance the inside condensation heat transfer many times than conventional round one. In order to enhance the performance of air cooled condensers, it is important to take into consideration both of condensation inside condenser tubes and convection outside, where the enhancement in convection side is the dominant one. So the present work is mainly concentrated on convection heat transfer from air side for flat tube condensers.

Although, the PFHE has the above mentioned good thermal performance, but there is still a lot of potentials for improving the air side convective heat transfer. Therefore, the present study is directed to enhance the convection heat transfer for air side by inclination of its flat tubes, to make convergent and divergent channels for air flow (case 1). This can be achieved by inclination one tube toward clockwise direction and the next in counter-clockwise direction by angles from zero up to 16°. with respect to horizontal direction. Furthermore , without the need of replacing any equipment of production line that producing PFHE, another construction for tilting all tubes (in clockwise or counter-clockwise directions) by the same angles range (from zero to 16 deg.) but all tubes are kept in parallel with each other (case 2). These two cases are analysed in the present study and compared with PFHE, and illustrated in Fig. (2). Finally the effect of aspect ratio (Ar) has been investigated at the optimum inclination angle ($\beta_{opt}$ ).

## 2- MATHEMATICAL MODEL

Air cooled condensers in air conditioner, can be modeled as two-dimensional for heat flow. For steady state, the dimensionless governing equations of continuity, momentum, and energy through tubes can be written as: Because of the symmetry of the tube bank geometry, only a portion of the domain needs to be modeled. The upper and lower sides of computational domain are taken to lie along the centers of two consecutive tubes which are specified as symmetry boundary conditions as

***Continuity equation***

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}=0 \quad \text{(1)}$$

Momentum equations:

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu\nabla^2 u-\rho\left(\frac{\partial \overline{u'^2}}{\partial x}+\frac{\partial \overline{u'v'}}{\partial y}\right) \quad \text{(2)}$$

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial y}+\mu\nabla^2 v-\rho\left(\frac{\partial \overline{u'v'}}{\partial x}+\frac{\partial \overline{v'^2}}{\partial y}\right) \quad \text{(3)}$$

Energy equation:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\left(\alpha+\frac{\upsilon_T}{\sigma_T}\right)\nabla^2 T+\mu\phi+S_r \quad \text{(4)}$$

Turbulence energy equation:

$$u\frac{\partial k}{\partial x}+v\frac{\partial k}{\partial y}=\left(\upsilon+\frac{\upsilon_T}{\sigma_k}\right)\left(\frac{\partial k^2}{\partial x^2}+\frac{\partial k^2}{\partial y^2}\right)+(\upsilon+\upsilon_T)\phi C_1\varepsilon-\beta g\frac{\nu_T}{\sigma_k}\frac{\partial T}{\partial y}-\varepsilon \quad \text{(5)}$$

Turbulence dissipation rate equation:

$$u\frac{\partial \varepsilon}{\partial x}+v\frac{\partial \varepsilon}{\partial y}=\left(\upsilon+\frac{\upsilon_T}{\sigma_\varepsilon}\right)\left(\frac{\partial^2 \varepsilon}{\partial^2 x}+\frac{\partial^2 \varepsilon}{\partial y^2}\right)+\left(\upsilon+\frac{\upsilon_T}{\sigma_\varepsilon}\right)C_1\varepsilon\phi-\left(C_1\beta g\varepsilon\frac{\nu_T}{\sigma_k}\frac{\partial T}{\partial y}-C_2\varepsilon^2\right)/k \quad \text{(6)}$$

$$\mu\phi=2\mu\left[\left(\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial y}\right)^2\right]+\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2$$

The turbulent kinematic viscosity and turbulence dissipation rate are related to turbulent energy and dissipation rate as:

$$\nu_T=C_\mu\frac{k^2}{\varepsilon} \quad \text{(7)}$$

$$\varepsilon=C_D\frac{k^2}{l} \quad \text{(8)}$$

Where $C_D$ is empirical constant , and l is the mixing length.

The k-ε model constants $C_\mu$, $C_1$, $C_2$, $\sigma_t$, $\sigma_k$, and $\sigma_\varepsilon$ values are presented in table 2-1.

Table (2-1): The standard values of k-ε model constants.

| $C_\mu$ | $C_1$ | $C_2$ | $\sigma_t$ | $\sigma_k$ | $\sigma_\varepsilon$ |
|---|---|---|---|---|---|
| 0.09 | 1.44 | 1.92 | 1 | 1.3 | 0.9 |

shown in Fig.(3) which are specified as:

**At inlet** : $T=T_i$ , $u = u_i$ , $P = P_i$

**At exit** : $T=T_{out}$ , $u = u_{out}$ , $P = P_{out}$

**At the tube wall** : $T=T_w$ , $u = 0$

All pre-generated meshes for the studied cases were prepared first by **GAMBIT** software. Then modelled as bank of tubes in cross-flow, and the outside air flow is classified as turbulent and steady. The model is used to predict the flow and temperature fields that result from convection heat transfer for air side. Due to symmetry of the tube bank, only a portion of the geometry was modelled in **FLUENT**. Domain is discretized into a finite set of control volumes or cells. General transport equations for mass, momentum and energy are applied to each cell and discretized. The governing equations are solved for the studied flow field at constant wall temperature case for the flat tubes. The numerical solution was conducted to investigate the influence of inclination angle (β) and aspect ratio (Ar) for different air velocities on the performance of air cooled condensers.

The following values which are applicable to window and split air conditioning systems, are used as input data for solving the studied problem;

1- Air flow is steady, two dimensional and turbulent.

2-Air face velocity ($V_f$)=2.5, 5 and 7.5 m/s.

3- The condenser wall temperature =323 K.

4- Ambient air temperature=308 K

5- The flat tube condenser configurations ;
Tube height (b) =1.8 mm,
Tube width (L) = 18 mm,
Tubes transverse pitch= 10.4 mm.

Flow and heat transfer characteristics are obtained for forced convection of air flow across flat tubes at different operating parameters. By using CFD software, the flat tubes condensers shown in Fig. (2.a) has been studied first, which is called parallel flow heat exchanger (PFHE). Then the proposed modifications in the following sequence; construction of convergent and divergent channels for air flow (**Case 1**), as shown in Fig. (2.b). Tilting of all tubes in parallel to each other by angles up to 16 deg. with respect to horizontal) either forward or backward (**Case 2**), as illustrated in Fig. (2.c).

## 3. RESULTS AND DISCUSSIONS

The performance of air cooled condensers, which used flat tubes with the proposed two cases of inclination, was studied and compared with parallel horizontal flat tubes at the same operating conditions.

### 3.1: Flow and Temperature Contour Field

Contour lines for temperature and velocity in axial direction are shown in Figs. (4) and (5) for parallel horizontal flat tubes, convergent divergent passages (case 1), and tilted one (case 2). It is observed from Fig.(4) that there is an increase in fluid temperature around the hot flat tubes for the proposed two inclined cases compared with parallel horizontal flat tubes. Also, it is found from Fig.(5) for the case of convergent divergent passages, the velocity in the axial direction increases in the convergent passage and decreases in the divergent passage.

### 3.2: Heat transfer and pressure drop

### 3.3.1: Case-1 (Convergent-divergent)

The effect of inclination angle (β) on the convection heat transfer for air side, and pressure drop of flat tube air cooled condenser (case1), is illustrated in figures (6). For the convergent divergent passages the increase in ΔP is small in the first part up to 8° then gradually increases. Also, it is found that there is a peak value for the average convection heat transfer coefficient Nusselt number at inclination angle, $\beta = 4°$ and there is a higher values for both of Nu , and ΔP in the second part of the curve which is not preferable practically.

To obtain the optimum value of β , it is important to collect and draw the values of Kh and Kp in one graph as shown in Fig.(7) at different face velocities. It is clear that, for different values of face velocities ($u_f$), the enhancement heat transfer factor, Kh increases until it reaches a certain value at β=4° and then decreases and reach a minimum value at β=8° after which it begins to significantly increase as β increase from 8° to 16°. On the other hand the pressure drop factor, Kp increases with increasing inclination angle β and the optimum value of inclination angle β is equal to 4 ° in this case.

### 3.3.2: Case-2 (Tilted tube configuration)

Figure(9) shows the average convective heat transfer expressed by the Nusselt number and pressure drop across the tubes versus the inclination angle for case-2. The pressure drop Δp increases to some degree up to 8° and then dramatically increases with β while Nusselt number has two peaks value for the convection heat transfer coefficient Nu at inclination angle, β = 4° and 12°, respectively.

The enhancement factor of convective heat transfer coefficient, Kh and pressure drop factor, Kp are plotted against inclination angle at different face velocity as shown in Fig.(9). It is clear that the enhancement factor of heat transfer is significantly increased up to reach a certain value at β = 4° after which it remains almost constant up to β = 8° . A further increase in β will lead to a great increase in Kh and reaching a maximum value at β = 12° after which it starts to decrease. This decrease in Kh is due to bad contact between inlet air and flat tubes (for β greater than 12 deg.). Also, the pressure drop factor is slightly increase until it reaches a certain value at 8° after which the pressure drop factor has a rapid increase.

### 3-4: Comparison between the studied cases

As shown in Fig.(10), the average Nusselt number in case1 (convergent-divergent tube passage) is higher than case2 (tilted tube). The average Nusselt number, Nu of convergent-divergent tube geometry is about 2.74 time higher than that of the tilted tube condenser. To compare between the two studied cases, the enhancement factor for heat transfer and pressure drop factor are presented in Fig.(11). It is clear that the enhancement factor of heat transfer, Kh and the pressure drop factor in case-2 (tilted tube) is higher than that of case-1 (convergent-divergent tube).

The overall performance (which is defined as η=Kh/Kp) is plotted against β for the studied two cases in Fig. (12) at different values of air face velocities ($u_f$). It is clear that the effectiveness of tilted tube is better than the convergent-divergent tube for the studied face air velocity. Varying $u_f$ from 2.5 up to 5 m/s leads to a considerable change in the overall performance (η). But with increasing $u_f$ from 5 to 7.5 m/s, there is a small change in the performance as the two curves are nearly coincident. So, the value of face velocity of 5 m/s is considered as the maximum limit for operation.

### 3-5: Effect of aspect ratio

The effect of aspect ratio ($A_r$) on the values of $h_{av}$ and Δp was shown in Fig. (13). Easily, the optimum aspect ratio, ($A_r$) is found to be 0.58 which corresponds to H=10.4 mm.

### 3.6: Verification of the present data

Finally, To verify the obtained theoretical results, a comparison with previous researches experimental result are shown in Figs.(14) and (15). For case 1, the convective heat transfer to air flow in converging-diverging tubes were studied experimentally by Ariad et al. [8]. Their study was based on constant wall temperature at different values of β from 0 up to 16 deg., which is similar to the proposed studied cases. They reported that the obtained enhancement comparing to equivalent straight tube at the same mean diameter is Kh=1.45 against KP of 2.2 value. The corresponding values (at 4°, $V_f$=5m/s) for the present proposed case 1 are, Kh=1.469 against KP=2.12 for convergent divergent passages. Also, experimental results obtained by [8] shows that, the optimum value of β was 5.5° which is fairly agreed with the present theoretical results (4°).

As shown in Fig.(14), Nusselt number (Nu) are plotted against β, for both the present theoretical results and the experimental results obtained by [8]. It is observed that, good agreement is observed only for β < 10°. But for large values of β, the difference between the experimental and theoretical results is noticeable.

The effect of inclination angle for case 2 on the performance of aluminium brazed heat exchanger was investigated experimentally by Kim et al. [10]. It is clear from Fig.(15) the comparison between the obtained theoretical results for $h_{av}$ and experimental results obtained by [10] is acceptable agreement. Also, Kim et al. [10] reported that, there is enhancement in $h_{av}$ with increasing $\beta$ up to 12 deg. which agreed with the present results.

## 4. CONCLUSION

A theoretical study was done to obtain the optimum inclination angle for flat tubes which used in air cooled condensers. It is concluded that, using the proposed convergent divergent construction with optimum angle of 4 deg offers the best enhancement in convection heat transfer coefficient and in turn improves the overall thermal performance. For one row coil which is used in car air condition, the enhancement factor is about Kh=1.467 with increase in pressure drop, KP=2.12.

The second proposed construction of tilting the all tubes in parallel by 4 deg with respect to horizontal is recommended also to keep the production line that manufacturing the PFHE without any changing. This leads to enhancement factor of Kh=1.46 with increase in pressure drop of KP=1.9.

### 5. References

[1] Y.P. Chang., R.Tsai and J.W. Hwang, 1997 "Condensing heat transfer characteristics of aluminium flat tubes", Applied Thermal Engineering , Vol. 17, No. 11, pp. 1055-1065.

[2] R.L. Webb. and X.M.Wu, 2002 " Thermal and hydraulic analysis of a brazed aluminium evaporator", Applied Thermal Engineering , Vol. 22, pp. 1369–1390.

[3] K. Chung., K.S. Lee and W.S. Kim , 2002 " Optimization of the design factors for thermal performance of a parallel flow heat exchanger" Int. J of Heat Mass Transfer, Vol. 45, pp. 4773–4780.

[4] M.J. Wilson, T.A. Newell , J.C. Chato and C.A. Ferreira, 2003"Refrigerant charge , pressure drop , and condensation heat transfer in flattened tubes" Int . J. Refrigeration, Vol. 26 pp. 442-451.

[5] J.W. Coleman and S. Garimella , 1988 " Characterization of two phase flow patterns in small diameter round and rectangular tubes" International Journal of Heat and Mass Transfer, Vol. 31, pp. 1758-1770.

[6] E. Bari, J.Y. Noel, G. Comini and G. Cortella, 2002,"Air-cooled condensing systems for home and industrial appliances "Applied thermal engineering, , Vol. 25, pp. 1446-1458.

[7] H.K. Varma and C.P. Gupta ,1995 "Heat transfer during forced convection condensation inside horizontal tube" Int. J. refrigeration ,Vol. 18, pp. 210-214.

[8] F.F. Araid , M.A. Shalaby and M.M. Awad, 1986 " Convective Heat transfer to gas flow in converging diverging tubes" Mansoura university Bulletin,Vol.11, No.1.

[9] L.H.Rabie , A.A.Sultan and Y.E. Abdel Ghaffar , 2001 "Convective Heat transfer and pressure drop in periodically convergent divergent variable area annuli ",IMPEC12.

[10] A. Bensafi, S. Borg and D. Parent, 1997 "CYRANO: a computational model for the detailed design of plate–fin-and tube heat exchangers using pure and mixed refrigerants" Int.J. Refrigeration, Vol. 20 No.3, pp. 218-228.

[11] G.Lazza and U. Merlo, 2001 "An experimental investigation of heat transfer and friction losses of interrupted and wavy fins for fin-and-tube heat exchangers " Int. J. Refrigeration , Vol. 24, pp. 409-416.

[12] S. Sanitjai and R.J. Goldstein ,2004 "Forced convection heat transfer from a circular cylinder in cross flow to air and liquids" International Journal of Heat and Mass Transfer , Vol. 47, pp. 4795–4805.

[13] J.S. Jabardo, G.W. Manami and M.R. Lanella, 2002 "Modelling and experimental evaluation of an automotive air conditioning system with a variable capacity compressor" Int. J. of Refrigeration" Vol. 25, pp. 1157–1172.

[14] X.M. Wu and R.L. Webb, 2002 " Thermal and hydraulic analysis of a brazed aluminum evaporator " Applied thermal engineering, , Vol. 22, pp. 1369-1390.

[15] C.Y Yang. and R.L Webb, 2005 " Condensation of R-12 in small hydraulic diameter extruded aluminium tubes with and without micro-fins" Int. J. of Heat Mass Transfer, 48, pp. 791-800.

[16] M.H. Kim , S. Song and C.W. Bullard , 2002 "Effect of inlet humidity condition on the air side performance of an inclined brazed aluminium evaporator" I. J. of Refrigeration, Vol. 25 pp. 611-620.

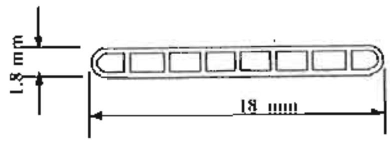



Fig. (1): micro-channel flat tube.

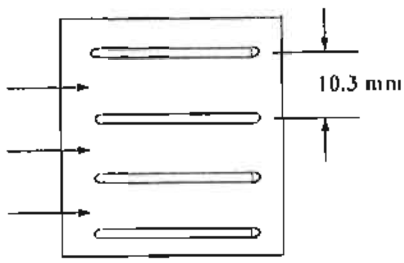



Fig.(2-a) Paralle flow heat exchanger (PFHE

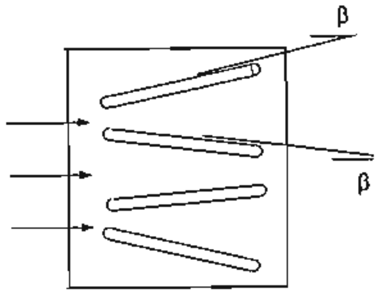
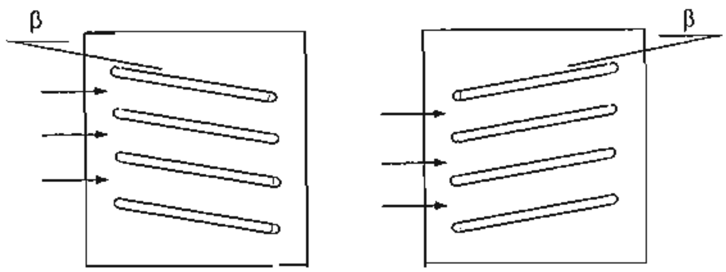



Fig.(2-b) Convergent-divergent tube condenser




Tilted forward Tilted backward

**Fig.(2-c) Case 2 Tilted tube condenser**

Fig.(2) Layout of flat horizontal tube (PFHE) and the proposed two cases of modification.

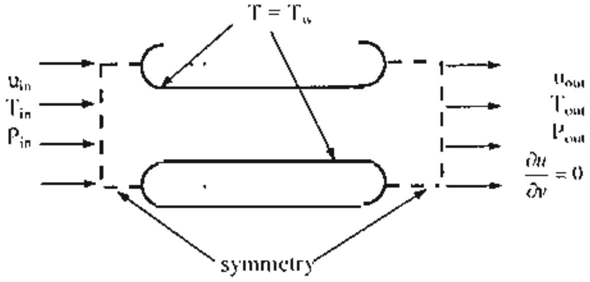
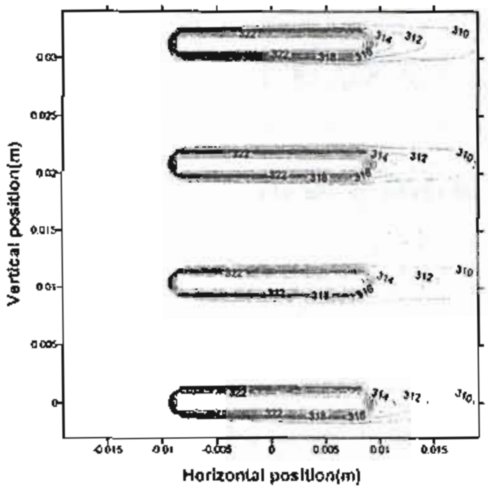
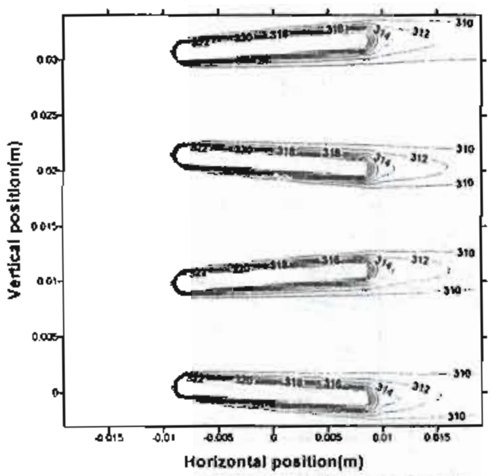


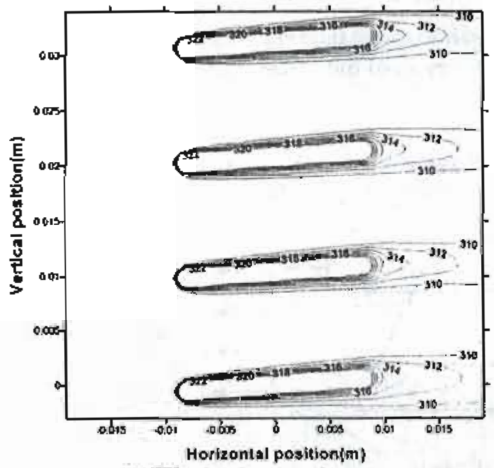

Fig.(3) Symmetry boundary condition




A- Flat tube condenser

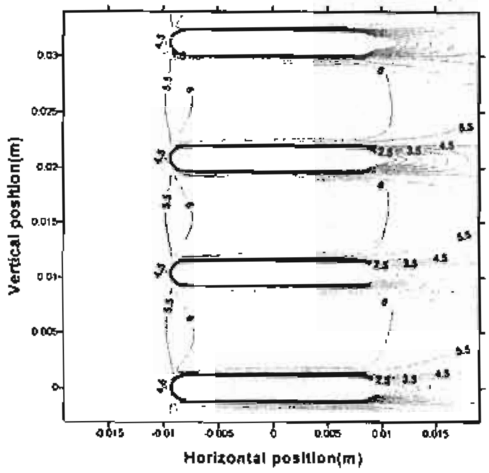



B- Convergent – divergent tubes condenser

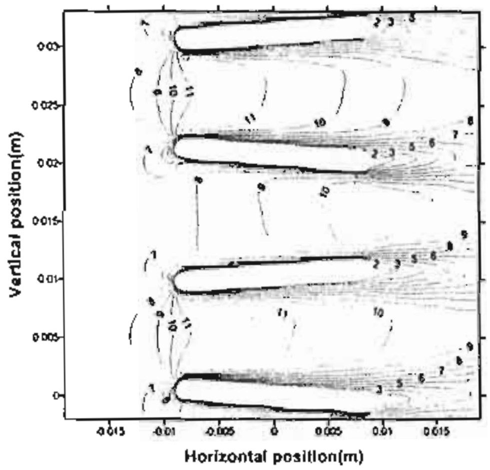



C- Tilted tubes condenser

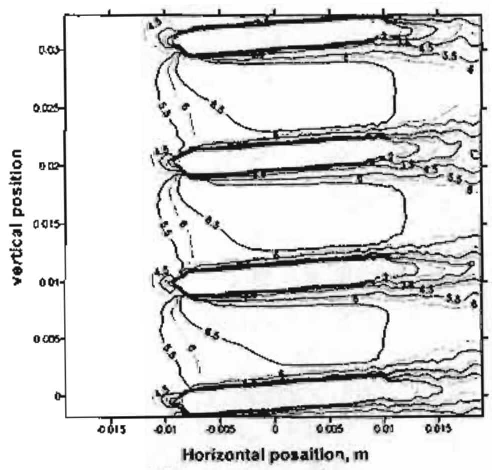
Fig.(4) Temperature contour for the studied two cases compared with horizontal flat tube.




A- Flat tube condenser




B- Convergent – divergent tubes condenser




C- Tilted tubes condenser

Fig.(5) Velocity contour for the studied two cases compared with horizontal flat tube.

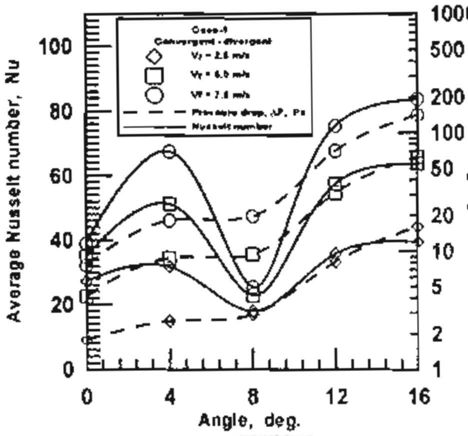



Fig.(6) Air side heat transfer coefficient and pressure drop versus tilting angle in case of convergent-divergent configuration.

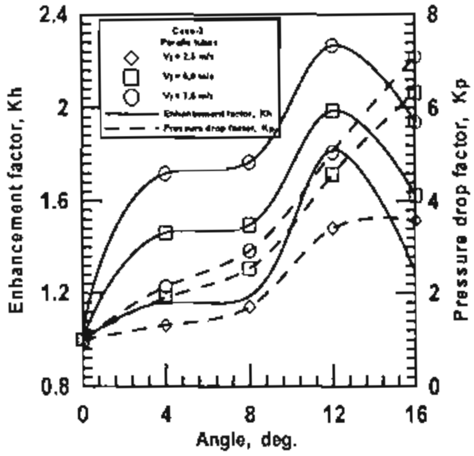



Fig.(9) Variation of enhancing heat transfer factor and pressure drop factor at different velocity.

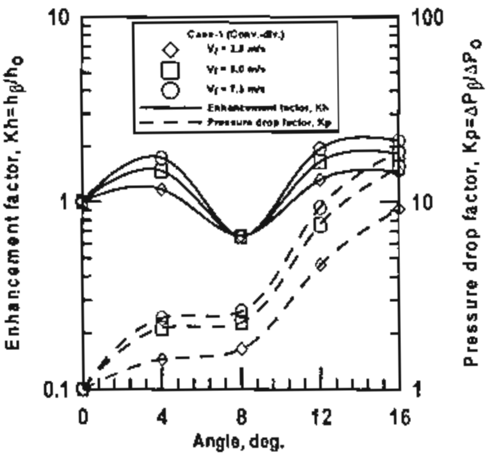



Fig.(7) Heat transfer enhancement and pressure drop factor versus telting angle at different velocity.

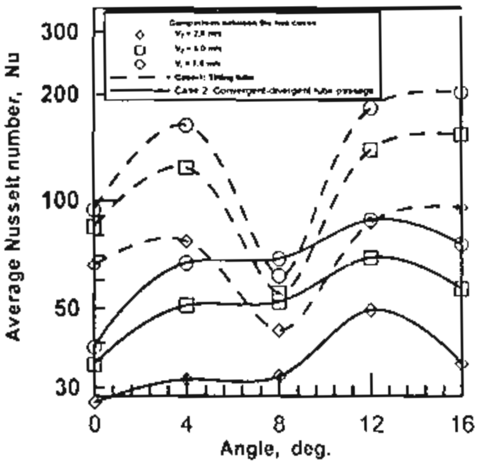



Fig.(10) Average Nusselt number comparison between the proposed two cases.

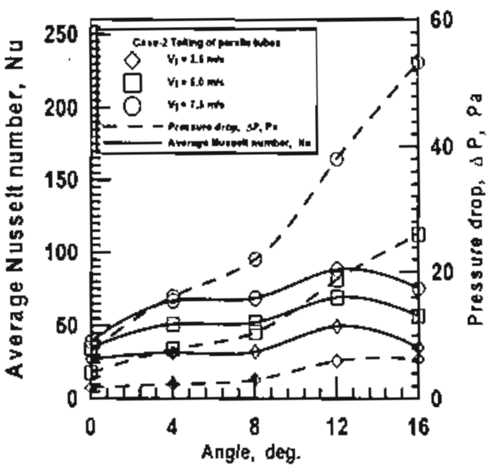



Fig.(8) Average Nusselt number and pressuredrop with tilting angle at different velocity.

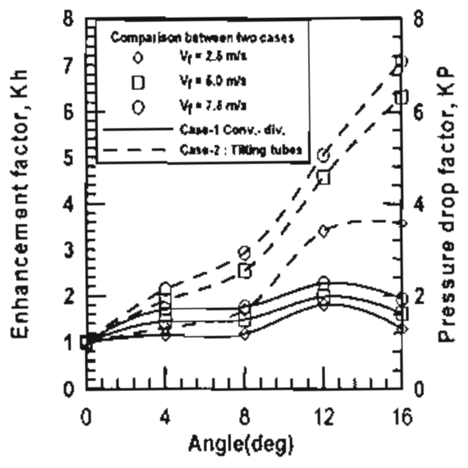



Fig.(11) Comparison between enhancement factor and pressure drop factor of the proposed two cases.

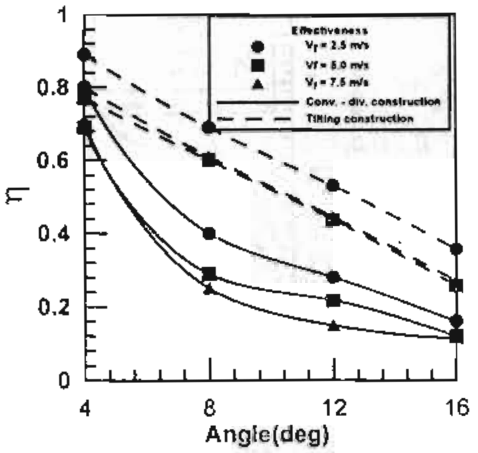



Fig.(12) The effectiveness of condenser versus telting angle.

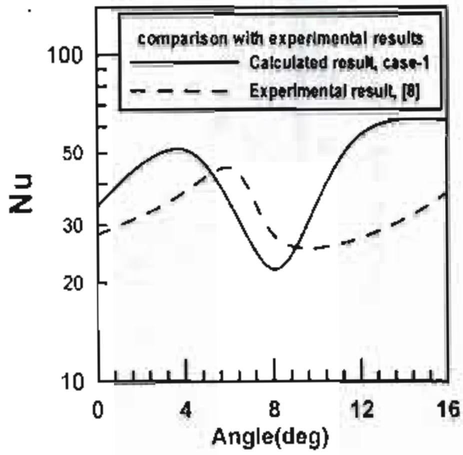



Fig.(14) Comparison of present data with the published experimental data.

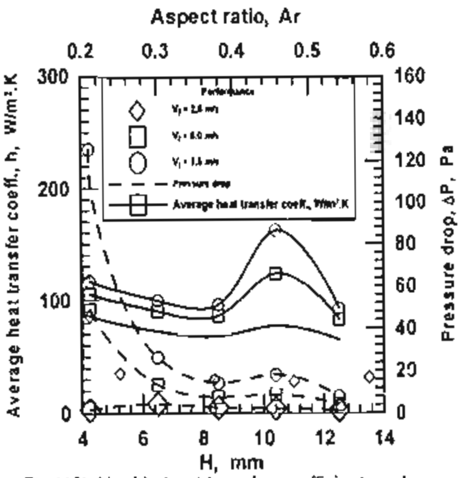



Fig.(13) Air side heat transfer coefficient and pressure drop along the tube.

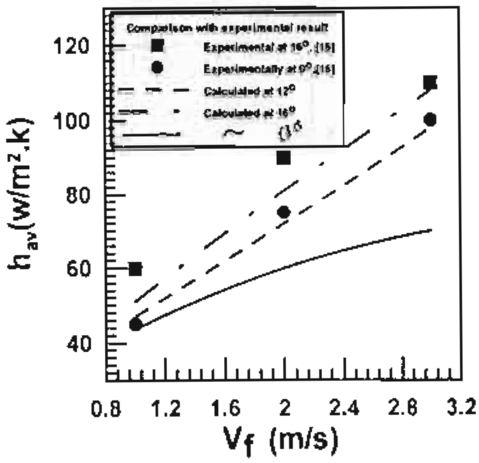



Fig.(15) comparison of the present data with experimental results for case-2.