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Forced convection heat transfer for air flow across flat tubes heat exchanger

"إنتقال الحرارة بالحمل الجبرى لهواء يمر على أنابيب مسطحة لمبادل حرارى"

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خلاصة البحث:

فى هذا البحث تم عمل دراسة نظرية و معملية لإنتقال الحرارة بالحمل الجبرى وفقد الضغط لهواء يمر على أنابيب مسطحة لمبادل حراري. وقد تمت دراسة تأثير ميل الأنابيب على معدل إنتقال الحرارة والفقد فى الضغط نظريا وعمليا. فى الجزء النظرى تمت دراسة ثلاث حالات لميل الأنابيب : الحالة الأولى بجعل مسارات الهواء بين الأنابيب مفرق- مجمع ، الحالة الثانية بجعل الأنابيب موازية لبعضها البعض ومائلة للأمام على إتجاه مدخل السريان. بينما الحالة الثالثة بجعل الأنابيب موازية لبعضها ومائلة للخلف عكس إتجاه مدخل السريان. فى الحالتين الثانية والثالثة تمت دراسة تأثير وجود الزعانف حول الأنابيب من الخارج. وقد تم حل المعادلات الحاكمة للسريان بإستخدام برنامج (FLUENT 6.1). بعد حل هذه المعادلات أمكن الحصول على توزيع كل من السرعة ودرجة الحرارة حول الأنابيب المسطحة. وبالتالي تم حساب معامل إنتقال الحرارة بالحمل الجبرى وفقد الضغط للهواء المار حول الأنابيب المسطحة.

تم تصميم وتنفيذ دائرة إختبار تتكون من نفق هوائى يتم سحب الهواء من الوسط المحيط ليمر على مقطع الإختبار وهو عبارة عن مبادل حرارى مكون من عمود واحد من الأنابيب المسطحة. مقطع الاختبار يتم تسخينه بطريقتين منفصلتين؛ الطريقة الأولى عن طريق طرد الحرارة من وحدة تبريد تستخدم وسيط تبريد R406A أما الطريقة الثانية فكانت عن طريق إستخدام بخار الماء المشبع. تم تغييرسرعة الهواء خلال التجارب وكذلك درجة حرارة السطح الخارجى لمقطع الاختبار.

وقد أظهرت النتائج النظرية و المعملية فى كلتا الحالتين الأولى والثانية أن زاوية الميل 4° على إتجاه السريان هى أفضل زاوية أعطت أكبر تحسين فى معامل إنتقال الحرارة بالحمل. هذه الزيادة فى معامل إنتقال الحرارة بالحمل ناحية الهواء في حالة عدم وجود زعانف كانت حوالى ٤٦.٩% للحالة الأولى وحوالى ٤٦% للحالة الثانية بينما زاد الفرق فى الضغط بنسبة ١١٢% للحالة الأولى وبنسبة ٩٥% للحالة الثانية. أما فى حالة وجود زعانف فكانت الزيادة فى معامل انتقال الحرارة للحالتين الأولى والثانية حوالى ٢٠% أما فى الحالة الثالثة فلم تحقق تحسن ملحوظ. وقد لوحظ توافق بين النتائج النظرية والمعملية. كما تم استنتاج صيغ تجريبية لابعدية لحساب رقم نوسلت من النتائج المعملية كدالة فى رقم رينولدز عند زاوية الميل المثلى للأنابيب المسطحة (4°). وأيضا تم عمل تقييم عام للأداء أخذا فى الإعتبارالزيادة فى رقم نوسلت مع الزيادة فى فقد الضغط ووجد أنه فى حالة ميل أنابيب المكثف للأمام فى إتجاه السريان أو إستخدام الأنابيب المفرقة المجمعة يكون هناك تحسن عام أما إذا كانت أنابيب المكثف مائلة للخلف فليس هناك تحسن ملحوظ.

Abstract

Forced convection heat transfer for air side flow across flat tubes heat exchanger is investigated numerically and experimentally. The effect of inclination of flat tubes on the amount of heat transfer is investigated. Fluent 6.1 software program is used to solve this problem. Three proposed cases for inclination of tubes are studied. The first case is to make convergent and divergent channels for air flow (Case 1). The other two cases are tilting of flat tubes in forward (Case 2) and backward (Case 3). Velocity and temperature distribution around the flat tubes for the studied cases are obtained. In turn, convection heat transfer coefficient and pressure drop for air side around flat tubes are calculated.

An experimental apparatus is designed and constructed. Air is drawn through the wind tunnel from the surroundings to flow across one column of the tested tubes with different velocities. Two separated heating fluid units are used in the experimental work to perform different wall temperatures. The first unit is a refrigeration circuit, using R406A, as a working fluid. The air cooled condenser for this unit is considered the test section for this work and fixed inside the wind tunnel. The second unit is an open circuit, using steam as working fluid where steam is condensed inside the tested tubes. Three proposed cases for inclination of tubes in the theoretical work are manufactured and tested in the experimental work. The apparatus is equipped with the required sensors.

The experimental and numerical results showed that the optimum angle of inclination for the proposed first and second cases is 4°. For convergent divergent construction of one row coil without fins (case 1), the obtained enhancement of convection heat transfer coefficient is about 46.9 %

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meanwhile in pressure drop increased by about 112%. The second proposed construction of tilting forward all tubes in parallel with respect to horizontal, the enhancement in convection heat transfer coefficient is about 46 % against an increase in pressure drop by about 95%. There is an acceptable agreement between the numerical and experimental results. Empirical correlations to calculate average Nusselt number from the experimental work as a function of Reynolds number at optimum inclination angle of 4° are obtained. It is found that in case of convergent-divergent channels (inclination one tube toward clockwise direction and the next in counter-clockwise direction), and tilting all tubes in the forward direction considerable improvements are obtained but in case of tilting all tubes in the backward direction there is no improvement.

Key Words: Flat tube – air cooled condenser – enhancing heat transfer

1. INTRODUCTION

Air-cooled finned-tube condensers are widely used in radiators of cars, refrigeration and air-conditioning applications ...etc. A micro-channel flat tubes heat exchanger is one of the potential alternatives for replacing the conventional finned tube heat exchangers. These kinds of heat exchangers are made of flat tubes 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. Aoki et al. (1989) performed an experimental study on heat transfer characteristics of different louver fin arrays such as louvers angles, louver, fin pitches, and reported that the heat transfer coefficients at low air velocity decreased with increasing fin pitch. Also, they found that, heat transfer coefficient increased with louver angle, reaching a maximum at an angle (28-30 deg.) and decreased again. Rugh et al. (1992) investigated heat transfer coefficients and friction losses for high-density louvered fin and flat tube heat exchangers. They reported that a louvered fin heat exchanger produced a 25 % increase in heat transfer and 110% increase in pressure drop relative to plate fin. 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. (1997). The flat tube design offers higher thermal performance and lower pressure drop than the finned-round tube heat exchangers (Webb and Wu, 2002). 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 heat exchanger. The heat capacity of a parallel-flow heat exchanger (PFHE) is 150–200 % larger than that of the conventional heat exchanger (Chang et al., 2002). This high heat capacity of the PFHE can meet the requirements of compactness and lightness. Kim and Bullard (2002) studied experimentally the air side heat transfer and pressure drop characteristics for multi-louvered fin and flat tube heat exchanger. The louvered angle is changed from (12-29 deg.), fin pitches (1, 1.2, and 1.4 mm) and flow depth (16, 20, and 24 mm), Reynolds number (100-600) and constant tube side water flow rate of 0.32 m^3/min. The air side heat transfer coefficient and pressure drop for heat exchangers with different configurations were reported in terms of Colburn J.-factor and fanning friction factor f, as a function of Reynolds number based on louvered pitch. The effect of tube profile change from round to flat shape on condensation heat transfer inside tube has been investigated experimentally by Wilson et al. (2003). They found that a considerable enhancement of condensation heat transfer coefficient inside tube and an increase in pressure drop as the tube profile is a flat shape. Kim (2005) found theoretically that the fin pitch should be greater than 7 mm to avoid boundary layer interruption between the fins. Also, their experimental and theoretical studies showed that, the optimum design of flat plate finned-tube heat exchangers with large fin pitches of 0.4 mm. It is found that the air side heat transfer coefficient decreases with a reduction of fin pitch, and an increase of the number of rows.

Malapure et al., (2007) investigated numerically the fluid flow characteristics over louvered fins and flat tube in compact heat exchanger. They found that at low Reynolds number the flow is fin directed and at higher Reynolds number the flow is louvered directed. Also, both Stanton number and friction factor decrease with the increase fin pitch. Although, the PFHE has the above mentioned good thermal performance, there is still a lot of potentials for improving the air side convective heat transfer.

So, the present work is mainly concentrated convection heat transfer from air side with flat tube condensers by inclination of its flat tubes, to make convergent -divergent channels for air flow (case 1). This can be achieved by inclination of one tube toward clockwise direction and the next in counter-clockwise direction by angles ranging from zero 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 forward direction (counter-clockwise), case 2 or backward direction (clockwise direction), case 3 by the same angles range (from zero up to 16 deg.) but all tubes are kept in parallel with each other, as illustrated in Fig.(2). These three cases are analyzed numerically and experimentally in the present study and compared with parallel flat tubes which are used in PFHE, to obtain the optimum inclination angle (β_{opt}).

2- MATHEMATICAL MODEL

For steady state, turbulent flow, and two-dimensional, the governing equations are presented, [4] below as:

Continuity equation

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

Momentum equations:

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

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

Energy equation:

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

Turbulence energy equation:

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

Turbulence dissipation rate equation:

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

$$\mu\phi=2\left[\left(\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial y}\right)^2\right]+\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2 \quad \text{......} \quad (7)$$

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

$$\nu_T=C_\mu\frac{k^2}{\varepsilon} \quad \& \quad \varepsilon=C_D\frac{k^2}{l} \quad \text{......} \quad (8$$

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

The k-ε model constants C_μ, C_1, C_2, σ_t, σ_k, and σ_ε values are presented in table 1.

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

Cμ	C₁	C₁	σₜ	σₖ	σ_ε
0.09	1.44	1.92	1	1.3	0.9

Because of the symmetry of the tube bank geometry, only a portion of the domain needs to be modeled. 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 given in Fig.(3) and 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 : $T=T_w$, $u=0$

The generated meshes for the studied cases are prepared first by GAMBIT software. Then modeled as a 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 the air side. Due to symmetry of the tube bank, only a portion of the geometry was modeled in FLUENT6.1 as shown in Fig.(3).

The domain is discretized into a finite set of cells. General transport equations for 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 is conducted to investigate the influence of inclination angle (β) 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 in solving the studied problem;

1- Air flow is steady, two dimensional and turbulent.
2- Air face velocity, V_f=0.5, to 7.5 m/s.
3- Condenser wall temperature =323 K.
4- Ambient air temperature=308 K
5- Flat tube condenser configurations:
   Tube height (b) =1.8 mm,
   Tube width (L) = 18 mm,
   Tubes transverse pitch (H) = 10.4 mm.

The 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) has been studied first, which is called parallel flow heat exchanger (PFHE). Then, the proposed modifications sequences are presented as: construction of convergent-divergent channels for air flow (Case 1). Tilting of all tubes in parallel to each other by angles up to 16 deg. with respect to horizontal either forward (Case 2) or backward (Case 3), as illustrated in Fig. (2).

3. EXPERIMENTAL APPARATUS

The present experimental apparatus has been designed and constructed to investigate the air side convection heat transfer coefficient and pressure drop over the condenser flat tubes. The schematic diagram of the experimental apparatus is shown in Fig.(4).

Air is drawn to the wind tunnel (1) from the surroundings to flow across the test section (3) through a flexible connection (4) by using air blower (5) with different volume flow rates up to 1700 m^3 /hr by using the gate plate (7). Wind tunnel dimensions are 51 cm wide, 30 cm height and 3m long. To insure good distribution for air over the test section, the mesh screen (2) is fixed in the entrance of wind tunnel.

Two separated heating fluid units are used in this work. The first unit is usual refrigeration circuit which using R406A as working fluid (valves no. 27, 28 are closed while valves no. 25, 26 are opened). The air cooled condenser for this unit (3) is considered the test section for this work and is fixed well inside the wind tunnel. The heat input to the refrigeration system is provided from evaporator (14) which loaded by ambient air. The compressor (15) is choosen 2 kW to accommodate the required loads in condenser within the test range. The second unit is using dry and saturated steam as a working fluid (valves no. 25, 26 are closed while valves 27, 28 are opened). Dry and saturated steam is supplied from boiler (16) which is heated by an electrical heating element (17) to flow through an insulated water separator (19) before entering the flat tube condenser (test section) to ensure that the steam is dry and saturated. The manual valves (21, 22, 23) are used to purge out the separated water, make-up water, and drain of boiler water, respectively. The boiler is thermaly insulated by a glass wool insulation (18) to decrease the heat loss to the surroundings to a negligable value. A pressure control valve (20) is fitted to the boiler to control the steam pressure to the desired value "1.5 bar". Steam is condensed inside the tested flat tubes, then collected into graduated glass vessel (24).

The test section is constructed to be removable and replaceable to allow for testing different cases. Three tested cases for flat tube condensers with different inclination angles (from 0 up to16 deg) are investigated. The three tested cases of condensers are of one-column flat tubes construction.

- The first case is fabricated by tilting the first flat tube forward and the next backward and so on to achieve the convergent-divergent shape, as shown in Fig.(2).
- The second case is parallel flat tubes tilted forward as a whole.
- The third case is parallel flat tubes tilted backward as a whole.

The second and third cases are tested with and without fins as shown in Fig.(5). The fin type is louvered fins which are practically in use with flat tube condensers.

A summary of the three tested cases, test runs, and working fluids are given in table (2).

Table(2) :Summary of the tested cases.

Case	Description	Working Fluid
1	Convergent-divergent flat tubes.	Steam.
2	Parallel flat tubes with fins tilted forward	R-406A.
	Parallel flat tubes without fins tilted forward.	R-406A and steam.
3	Parallel flat tubes with fins tilted backward	R-406A
	Parallel flat tubes without fins tilted backward.	R-406A and steam.

4. MEASUREMENTS TECHNIQUE

Each run is carried out for certain fixed values of the problem parameters (air face velocity, inclination angle for certain case and surface temperature of flat tubes). Before starting a new run, the apparatus is adjusted to prevent leakage of air. The experimental procedure for each run can be described as; switch on the motor of fan, adjust the air face velocity to obtain the desired flow rate for air, switch on the refrigeration unit or heating steam unit and adjust the surface temperature to a certain value. The air is flowing inside the tunnel till steady state condition is reached. The steady state condition is achieved after about 60-90 minute. The air face velocity is measured by using Pitot tube (6) which has a pressure difference gave a corresponding resolution of 0.1 m/s (in ten grid points across the duct). The temperatures of air at up-stream and down-stream are measured by using calibrated thermocouples (11) (copper-constantan) which has a resolution of 0.1 °C and accuracy of ±1.5% of reading. The pressures of refrigerant are measured using calibrated bourdon tube pressure gauges. The refrigerant temperature at inlet and outlet of condenser is measured by two calibrated thermocouples. All thermocouples are connected to a temperature recorder (12). The refrigerant flow rate is measured using rotameter (13) placed at outlet of the condenser. It has an accuracy of ±0.5 % of full scale. While for steam, the condensate volume flow rate is measured and estimated by dividing the collected volume of condensate by a graduated glass vessel (24) through a certain time.

The air side pressure difference at inlet and outlet of the test section is measured by a water U-tube manometer through a pressure taps (8) which has an accuracy of ±1 mm H_2O.

In order to obtain a measure of the reliability of the experimental data, an uncertainty analysis is performed for the parameters of interest. The root-mean-square random error propagation analysis is carried out in the standard fashion using uncertainties of the basic independent variables. The maximum uncertainties are less than 6.1% for convection heat transfer coefficient and 4.9% for pressure drop.

5. DATA REDUCTION

The average convection heat transfer coefficient and pressure drop for air which flows over flat tubes can be calculated as follows:

1. The average air face velocity V_f through the wind tunnel is obtained through the integration of local velocity downstream of the test section as:

$$V_f = \frac{\int_{i=1}^{n} V_{f,i}\, dA_{c,i}}{A_c} \tag{9}$$

In terms of average air face velocity (V_f), test section face area (A_f), and air density (ρ_{air}), the air flow rate is calculated as;

$$\dot{m}_{air} = \rho_{air} V_f A_f \tag{10}$$

2. The rate of heat transfer required for calculating air side heat transfer coefficient can be expressed as;

$$\dot{Q} = \frac{\dot{Q}_{air} + \dot{Q}_h}{2} \tag{11}$$

Where: $\dot{Q}_{air}$ and $\dot{Q}_h$ are heat transfer rates of air and refrigerant or steam sides, respectively.

For air side;

$$\dot{Q}_{air} = \dot{m}_{air}\, Cp_{air}\left(T_{air,o} - T_{air,i}\right) \quad (12)$$

For refrigerant or steam side;

$$\dot{Q}_h = \dot{m}_h\left(i_i - i_o\right) \quad (13)$$

3. Effectiveness and number of transfer unit, *NTU* method can be used for obtaining air-side heat transfer coefficient. The overall heat transfer coefficient for test section is calculated as;

$$UA = \dot{m}_{air}\, Cp_{air}\, NTU \quad (14)$$

The *NTU* is obtained from the following relation for condensation:

$$\mathbf{NTU} = \ln(1-\varepsilon) \quad (15)$$

Where ε is the effectiveness which is defined as;

$$\varepsilon = \dot{Q} / \dot{Q}_{max} \quad (16)$$

The maximum rate of heat transfer is given by:

$$\dot{Q}_{max} = \dot{m}_{air}\, Cp_{air}\left(T_{h,i} - T_{air,i}\right) \quad (17)$$

4. For heating fluid side, the condensation heat transfer coefficient (h_{TP}) can be calculated from Shah correlation, [3, 10, 11].

$$h_{TP} = h_L\left\{(1-x)^{0.8} + \frac{3.8\, x^{0.76}\,(1-x)^{0.04}}{\text{[illegible]}}\right\} \quad (18)$$

Where h_{TP} is the two-phase flow heat transfer coefficient, x is the vapor quality, h_L is the liquid heat transfer coefficient, P is the fluid pressure, P_{cr} is the critical pressure, and h_L is calculated as:

$$h_L = 0.023\left(\frac{k_L}{D_{hyd}}\right)\mathrm{Re}_L^{0.8}\,\mathrm{Pr}_L^{0.4} \quad (19)$$

5. Now, the air side heat transfer coefficient is obtained from the following relation:

$$\mathrm{UA} = \left(\frac{1}{\eta_a A_o h_{air}} + \frac{1}{h_h A_i}\right)^{-1} \quad (20)$$

Where η_a is the overall effectiveness of the finned surface, given by;

$$\eta_a = 1 - \frac{A_f}{A_o}\left(1-\eta_f\right) \quad (21)$$

The louvered air side fins are assumed as plan with rectangular profile whose efficiency is given by the following equation:

$$\eta_f = \frac{\tanh(mL_f)}{mL_f} \quad (22)$$

Where: L_f is the fin length and m is defined as;

$$m = \sqrt{2h_{air} / k\,\delta_f} \quad (23)$$

6. The friction factor can be determined by measuring the pressure drop across the tubes and the average velocity of the air as:

$$f = \frac{(\Delta P / L)D_h}{\rho V_f^2 / 2} \quad (24)$$

Where:

L is the tube length in direction of flow, m.

7. The dimensionless Nusselt number and Reynolds number of air side is defined as:

$$Nu = \frac{h_a\, D_{hyd}}{k_a} \quad (25)$$

$$\mathrm{Re} = \frac{\rho_a V_f D_{hyd}}{\nu_a} \quad (26)$$

6. RESULTS AND DISCUSSIONS

The performance of flat tube air cooled condensers with the proposed three cases of tube inclination, are studied and compared with parallel horizontal flat tubes at the same operating conditions. The results have been compared in terms of two important parameters heat transfer coefficient and pressure drop.

6.1 Flat and circular Tube Performance

Figure (6) shows the average heat transfer coefficient and pressure drop for both flat and circular tubes, respectively. It is found that the average heat transfer coefficient for flat tube is higher than that of circular tube by about 38% but there is 300% increase in pressure drop.

6.2 Velocity and Temperature Contours

Contour lines for velocity and temperature in axial direction are shown in Figs.(7) and (8) for convergent divergent tubes (case 1), tilted forward (case 2), tilted backward (case 3), and parallel horizontal flat tubes. It is found from Fig.(7) that for the case of convergent divergent tubes, the velocity in the axial direction increases in the convergent passage and decreases in the divergent passage. It is observed from Fig.(8) that there is an increase in fluid temperature around the hot flat tubes for the proposed inclined cases compared to that of parallel horizontal flat tubes.

6.3 Heat transfer and pressure drop for flat tubes without-fins

6.3.1 Case 1 (Convergent-divergent)

The effect of inclination angle (β) for both numerical and experimental results on convection heat transfer, and pressure drop for air side over convergent divergent flat tubes without fins (case1), is illustrated in Fig.(9). The heating fluid is steam and the air face velocity is 4.18 m/s. The numerical results are included individually in these figures. It is observed that, there is an acceptable agreement between the numerical and experimental results. For the convergent divergent tubes the increase in ΔP is small in the first part up to $\beta = 8^o$. Also, it is found that there is a peak value for the average convection heat transfer coefficient at inclination angle, $\beta = 4^o$ and 12^o. But there is a higher value for ΔP in the second part of the curve which is not preferable practically. Therefore, the optimum value of inclination angle for case 1 is found at β=4°.

6.3.2 Case 2 (Tilted forward)

Figure(10) shows the experimental and numerical results for the average convective heat transfer coefficient for air side and pressure drop across the flat tubes tilted forward versus the inclination angle for case 2 without fins. Steam is used as a heating fluid and face air velocity is 4.18 m/s. The same trend like case 1 is obtained for h and ΔP versus β.

6.3.3 Comparison between the studied two cases

Figure (11) shows the comparison between the experimental values of convection heat transfer coefficient for case 1 and case 2 without fins. Also, pressure drop for the studied two cases was presented in Fig.(12). It is clear that, pressure drop for two cases is identical and take smaller values at lower values of β. Therefore, at the first peak for h the optimum inclination angle was obtained ($\beta_{opt} = 4^o$).

6.4 Louvered fin performance

Figure (13) presents the temperature and velocity contours of the louvered fin at 3 m/s for the studied cases. From the contours we can get the detail information about the air flow, the position of eddy and the distribution of the flow boundary layer. The boundary layer around the louvers is thinner and the flow is nearly aligned with the louvers. So, the temperature of air increases along the flow direction and a significant temperature difference is maintained between air and fins. It can be seen that low pressure zone is formed near the louvers due to formation of boundary layer. The air which flows through the louvers strikes on the flat plate and is turned. This flow direction causes high pressure zone in the middle position. It is clear that louvers act to interrupt the air flow and create a series of thin boundary layers which have lower thermal resistance.

6.5 Heat transfer and pressure drop for case 2 and case 3 with louvered fins:

The experimental results of heat transfer coefficient and pressure drop variation at different inclination angles for finned flat tubes tilted forward (case 2) tested with R406A at certain value of air face velocity is shown in Fig.(14). The numerical results for tilted forward flat tubes without-fins and tilted forward flat tubes with louver fins are included in the figures for comparison purpose. It is observed that, there is an acceptable agreement between the numerical and experimental results and a remarkable increase in the value of average convection heat transfer coefficient about 20% at inclination angle, $\beta = 4^o$ is obtained.

By the same way the results for tilted backward tubes (Case 3) is shown in Fig. (15). It is observed from the figure, the effect of inclination angle on values of h and ΔP was small. Therefore, there is no remarkable change for this case (tilting backward) compared with the horizontal one.

6.6 Verification of the present numerical results

To verify the obtained numerical results, a comparison with the previous experimental works is shown in figures (16) and (17). The effect of inclination angle on the performance of aluminum brazed heat exchanger was investigated experimentally by Kim et al., (2002). Figure (16) shows a comparison between present experimental and numerical results for convective heat transfer coefficient of parallel flat tubes at β=0, case (1) at β=14° and experimental results for Kim et al 2002 at β=0, 14°. It is clear that the comparison is in acceptable agreement.

As shown in Fig.(17), Nusselt number (Nu) is plotted against β, for both the present numerical results and the experimental results for case (1) and the results obtained by Arid et al., (1986). It is observed that, good agreement is observed only for β < 10°. But for large values of β, the difference between the present results and the previous experimental results is noticeable. Also, experimental results obtained by Arid et al., 1986 showed that, the optimum value of β was about 5.5° which is fairly agree with the present results (at β -4°).

6.7 Empirical correlations

The present experimental results for heat and fluid flow characteristics on a uniform heat flux tube condenser of different cases are reported in forms of Nusselt number and friction factor as:

$$Nu = C_1 \left(\mathrm{Re}_{max}\right)^{m1} \tag{27}$$

$$f = C_2 \left(\mathrm{Re}_{max}\right)^{m2} \tag{28}$$

The validity of these equations are 500<Re<4500.

The coefficients of equations 27 and 28 are given in table (3) and the maximum deviations of these correlations are ±5% as shown in Fig. (18).

Table (3): coefficients of equations (27, 28).

Case	*Nu*		*f*	
	c1	*m1*	*c2*	*m2*
Circular tube	7.6752	0.1842	7.1363	-0.2761
Parallel tube	0.2167	0.3666	7.6379	-0.2404
Case (1) Con-div passage	0.2106	0.6851	1.6754	-0.2322
Case (2) Tilted forward	0.2171	0.68	1.594	- 0.24

6.8 Performance Evaluation

A useful comparison between different types of tested condensers, which are used is made by comparing heat transfer coefficients at equal pumping power, since this is relevant to the operation cost.

The relationship between friction factor and Reynolds number for the same pumping power can be expressed as, [14]:

$$\left(\frac{f\,A\,\mathrm{Re}^3}{D^3_{hyd}}\right)_a = \left(\frac{f\,A\,\mathrm{Re}^3}{D^3_{hyd}}\right)_r \tag{29}$$

Where a and r denotes to augmented cases and reference case, respectively.

For the known values of Re_a and friction factor f_a or the augmented case, the equivalent Re_e for the reference case is calculated using eqn.(29) and the corresponding Nu_e is calculated and the enhancement ratio (R) which is defined as the ratio of Nusselt number of the condenser for different cases to the plain tube case which can be written as:

$$R = Nu_a / Nu_e \tag{30}$$

It is found that in case of inclination one tube toward clockwise direction and the next in counter-clockwise direction (convergent-divergent channels for air flow, case 1) and tilting all tubes in the forward direction (counter-clockwise, case 2) a considerable improvements are obtained as shown in Fig.(19).

7. CONCLUSIONS

Experimental and numerical studies are done to obtain the optimum inclination angle for flat tubes, which are used with 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. For one row coil which is used in car air conditioning, the obtained enhancement in convection heat transfer coefficient is about 46.9 % against increase in pressure drop by about 112% for case 1 (convergent-divergent tubes) comparing to equivalent parallel flat tubes at the same operating conditions.

The second proposed construction of tilting forward (counter clockwise) all tubes in parallel by 4 deg with respect to horizontal is recommended. This leads to enhancement in convection heat transfer coefficient by about 46 % against increase in pressure drop by about 95% for case 2 comparing to equivalent parallel horizontal flat tubes at the same operating conditions.

The experimental results for finned flat tubes tilted forward (Case 2) gave an enhancement about 20%, but there is no remarkable change for tilted flat tube backward case(3).

There is an acceptable agreement between the numerical and experimental results. Also, good agreement between the obtained results with the previous work was found. Empirical correlations from the experimental work were obtained.

Nomenclature

a	Tube cross-sectional area, m^2
A	Surface area, m^2
A_o	Total air side surface area, m^2
A_c	Wind tunnel cross-sectional area, m^2
A_f	Test section face area, m^2
Ar	Aspect ratio =H/L, -
b	Height of flat tube, m
C_1	Constant, Eqs. 5 and 6.
C_2	Constant, Eq. 6
C_D	Constant, Eq. 8
Cp	Specific heat, kJ/kg.K
C_μ	Constant, Eq. 8
D_{hyd}	Hydraulic diameter (D_{hyd} =4a/*P*), m
f	Friction factor, -
g	Gravitational acceleration, m/s^2
h	Heat transfer coefficient, $W/m^2.K$
H	Transverse pitch of parallel tubes. m
i	Specific enthalpy, kJ/kg
l	Mixing length, m
L	Tube width, m
L_f	Fin length, m
k	Kinetic energy, m^2/s^2
k	Thermal conductivity, W/m.K
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
NTU	Number of transfer unit, -
m	Parameter, Eq. 23
$\dot{m}$	Mass flow rate, kg/s
p	Pressure, Pa
P	Tube Perimeter, m
$\dot{Q}$	Rate of heat transfer, kW
S_T	Source term of heat, K/s
T	Temperature, K
u	Velocity in x-direction, m/s
U	Overall heat transfer coefficient, $W/m^2.K$
v	Velocity in y-direction, m/s
V_f	Face velocity, m/s
x, y	Co-ordinates, m
x	Dryness fraction

Dimensionless Groups:

Nu	Nusselt number, (Nu=h D_{hyd}/k)
Re	Reynolds number, (Re= $\rho V_f D_{hyd}/\mu$)
Pr	Prandle number, (Pr=$C_p\mu$/k)

Greek Symbols

α	Thermal diffusivity, m^2/s
β	Inclination angle of flat tubes, deg.
ρ	Density, kg/m^3
δ_f	Fin thickness, m
σ	Constant
μ	Viscosity, kg/m.s
ψ	Stream function, m^3s^{-1}/m
ν	Kinematic viscosity, m^2/s
η_f	Fin efficiency, -
η_R	overall performance =Kh/KP
$\mu\phi$	Viscous dissipation term, K/s
ε	Dissipation or effectiveness, -

Subscripts

av	average
cr	critical
f	face
h	heating steam
hyd	hydraulic
i	inlet, inner
k	turbulent
L	liquid
max	maximum
o	out, case of horizontal parallel flat tube
opt	optimum
T	thermal
TP	two-phase
β	case of inclined flat tube

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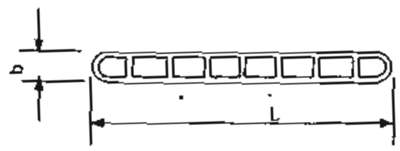


Fig. (1): Micro-channel flat tube.
L= 18 mm and b=1.8 mm.

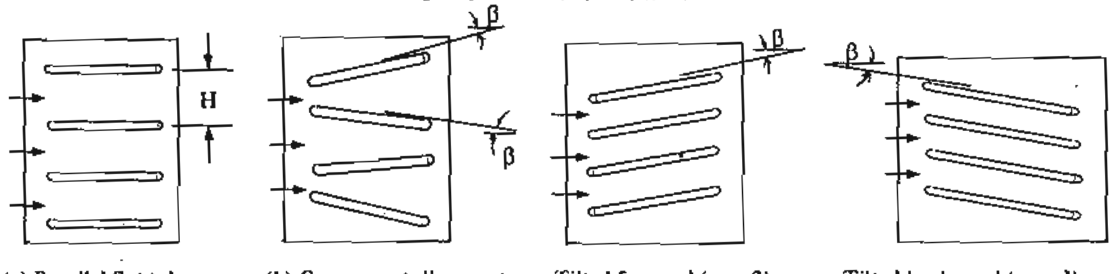


Fig.(2) Layout of flat horizontal tubes for the proposed three cases of modification.

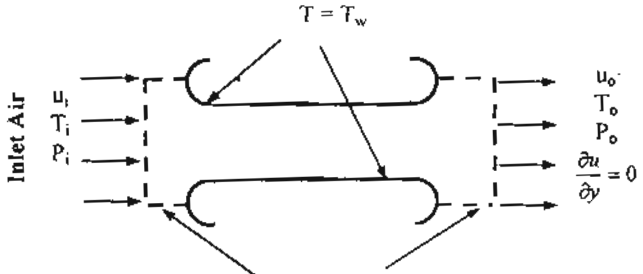
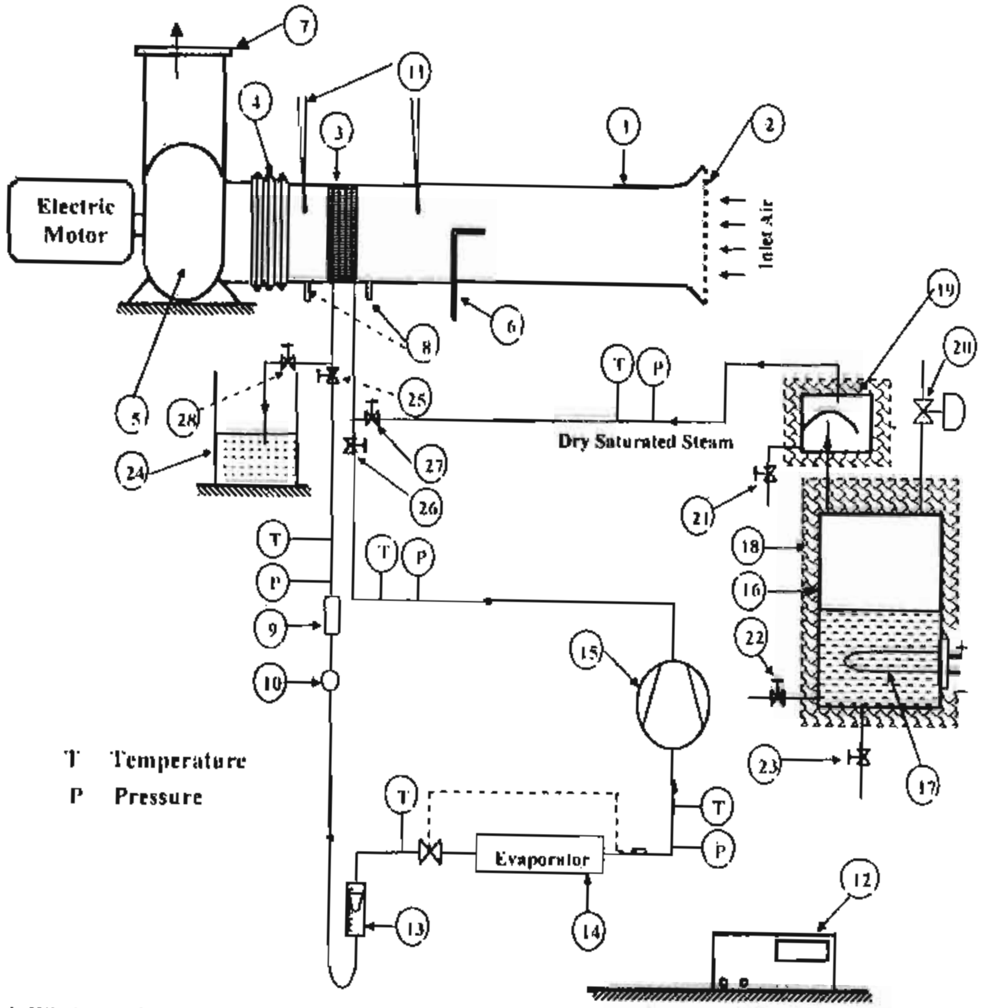
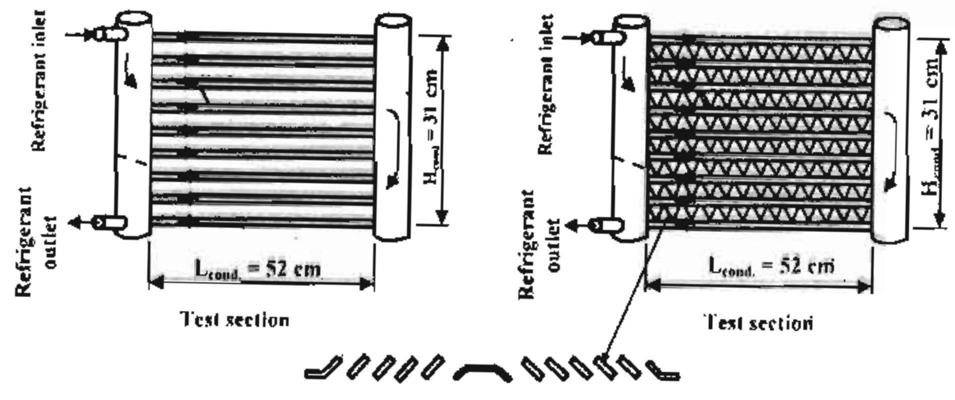


Fig.(3) Symmetry boundary conditions



1. Wind tunnel, 2. Bell mouth inlet and mesh screen, 3. Tested condenser, 4. flexible connection, 5. Air blower, 6. Pitot tube, 7. Gate plate, 8. Pressure taps, 9. Filter dryer, 10. Sight glass, 11. Thermocouples, 12. Temperature recorder, 13. Rotameter, 14. Evaporator, 15. Refrigerant Compressor, 16. Electric boiler, 17. Electrical heating element, 18. Thermal insulation, 19. Insulated water separator, 20. Pressure control valve, 21. Manual relief valve, 22. Make-up water valve, 23. Drain valve, 24. Graduated glass vessel, 25, 26. Manual valves for steam circuit, 27, 28. Manual valves for refrigeration circuit.

Fig.(4) Schematic diagram of the experimental apparatus .



Louvered fins

Figure(5) Details and dimensions of test sections No. 1 and 2, respectively.