EXPERIMENTAL INVESTIGATION OF THE PERFORMANCE OF A CROSS FLOW HEAT EXCHANGER WITH BUNDLE OF SEMI - CIRCULAR TUBES

دراسة معملية لخصائص انتقال الحرارة وفاعلية المبادلات الحرارية ذو المواسير النصف دائرية

*E. Z.Ibrahiem, *A.O.Elsyed and *E.S.Sayed Ahmed
Mechanical Dept- Faculty of Eng. Zagazig University- Zagazig Egypt

الخلاصة :

يتضمن البحث دراسة معملية لتوضيح خصائص انتقال الحرارة وفاعلية المبادلات الحرارية ذات المواسير النصف دائرية مع سريان مستعرض والمائعان الخارجي والداخلي المستخدمان هما الهواء والماء الساخن على الترتيب. المواسير المستخدمة موضوعة على شكل حزمة متوازية. وتراوحت قيم رقم رينولدز ما بين ٢ × ١٠^٤ الى ١٦,٥ × ١٠^٤ وتغيرت زاوية الهجوم من صفر إلى ٢٧٠ والخطوة الطولية النسبية من ١,٣٥ إلي ٢,٦٩ بينما الخطوة المستعرضة ظلت ثابتة عند ١,٣٥ . وقد أظهرت النتائج ان معاملا انتقال الحرارة والفاعلية تتغيران سريعا مع كل من زاوية الهجوم ونسبة الخطوة الطونية. وقد تم استنباط معادلتين الأولى معادلة لا بعدية لحساب رقم نوسلت كدالة في كل من رقم رينولدز والخطوة الطولية وزاوية الهجوم والمعادلة الثانية أظهرت ان فاعلية المبادل الحراري دالة في كل من السعة النسبية والخطوة النسبية الطولية وزاوية الهجوم. وقد بينت النتائج ان أقصى قيم لرقم نوسلت والفاعلية تحدث عند زاوية هجوم ٢٧٠ لكل قيم الخطوة الطولية مع ثبات الخطوة المستعرضة.

ABSTRACT:

An experimental investigation has been conducted to clarify heat transfer characteristics and effectiveness of a cross flow heat exchanger with bundle of semi-circular tubes. An ambient air and a hot water have been used as an external and internal fluids respectively. Where the tubes are placed in an in-line arrangement. The Reynolds number ranged from about 2×10^4 to 16.5×10^4. The angle of attack (θ) is varied from 0^o to 270^o and the relative longitudinal pitches (S_l/d) from 1.35 to 2.69,while the relative transfer pitche was kept constant ($S_t/d = 1.35$).

It has been found that the heat transfer in terms of Nusselt number (Nu) and effectiveness (E) varies drastically with the angle of attack (θ) and also with the longitudinal pitches (S_l). Correlations have been obtained for (Nu) as functions of (Re) at different (S_l/d) and (θ) and also for (E) as function of capacity ratio Z, S_l/d, and θ. The results show that the highest values of Nu, E occurred when the angle of attack, (θ) is 270^o for all values of (S_l/d) at constant (S_t/d).

Accepted June 30, 2003.

NOMENCLATURE:

a	: Relative transfer pitch (S_t/d) .	
b	: Relative longitudinal pitch (S_l/d) .	
C	: Heat capacity rate ($\dot{m} c_p$) .	kW/K
c_p	: Specific heat capacity at constant pressure.	kJ/kg K
d	: Diameter of semi- circular tube .	m
h	: Heat transfer coefficient of air, ($q / (t_s - t_{ai})$) .	kW / m^{2}K
k	: Thermal conductivity .	kW / m K
l	: Length of tube .	m
$\dot{m}$	: Mass flow rate .	kg / s
n	: Number of tubes in each row .	
Nu	: Nusselt number of air, (hd/k) .	
q	: Heat flux .	kW / m^2
S_l	: Longitudinal pitch .	m
S_t	: Transfer pitch .	m
t	: Temperature .	K
V_m	: Maximum velocity of air, ($\dot{m}_a / \rho\, n(S_t - d)\, l$) .	m / s
Z	: Heat capacity ratio , (C_{min}/C_{max}) .	
ν	: Kinematic viscosity of air.	m^2/s
ρ	: Density of air.	kg / m^3
θ	: Angle of attack.	
E	: Effectiveness of heat exchanger,[$(t_{ai} - t_{ao}) / (t_{ai} - t_{wi})$] .	
Re	: Reynolds number, ($V_{max}\, d / \nu$) .	

SUBSCRIPTS :

a	: Air .
i	: Inlet .
o	: Outlet .
s	: Surface .
w	: Water .
max	: Maximum .
min	: Minimum .

INTRODUCTION

High performance heat exchanger for saving and making effective use of energy is very important. A mong many types of heat exchangers, those constructed circular pipes have been used in many industries. There are many types of heat exchangers such as parallel flow, counter flow and cross flow. However, the cross flow heat exchangers are widely used in refrigeration and air conditioning.

As far as forced convection heat transfer characteristics of the circular tubes are concerned, only a few investigations are available. Total and local heat transfer analyses were made by Elmar [1]. Cross flow heat exchangers with both staggered and in-line tube arrangements for various ratios of pitch to diameter have been studied by Hewitt et.al [2].

Zhukauskas [3] performed many heat transfer experiments with single tubes and banks of cylinders. Heat transfer around four circular cylinders arranged in in-line tube bank in cross flow of air had been studied by Aiba [4]. Achenbach [5] studied the heat transfer and pressure drop for smooth and rough staggered tube bundle having transfer and longitudinal pitch ratios of 2 and 1.4 respectively. In this study Reynolds number (Re) ranged from 5×10^4 to 7×10^6. In the same way the heat transfer from smooth single cylinders arranged in staggered and in an in-line tube banks with uniform heat flux had been examined by Boughn et.al [6].

Optimal design of cross flow heat exchangers was studied by Bulck [7]. This design showed that the performance of heat exchangers increased by compacting more of the transfer area. Gowell [8] studied a general method for the comparison of compact heat transfer surfaces. This author presented a family of methods for comparing compact heat transfer surface configurations. Chen and Hsieh [9] presented a general procedure for effectiveness for complex assemblies of heat exchangers.

Pavan and Venkatarathnam [10] proposed the methods for the optimum sizing of balanced and unbalanced flow matrix heat exchangers. Heat transfer and flow characteristics around several cylindrical rods arranged in a single row in cross flow, were experimentally obtained by changing the pitch of the cylinders [11]. There are several cases where the contribution of both modes of heat transfer is of the same order. This case is referred to as mixed convection. Some researches did study the problem but limited themselves to either cross flow only [12-13] or assisting flow only [14].

In a previous article [15-16] the author reviewed the data on heat transfer from horizontal heated cylindrical tube of various knurled enhanced surfaces in cross flow. The average Nusselt number due to mixed convection from an isothermal cylinder at arbitrary angles of attack was solved numerically by Bassam and Hijleh [17]. They obtained correlation showing that, the change in the average Nusslet number relative to the case of cross flow was up to (+20%) higher for opposing flow which use one of the major objectives in the design of the enhanced heat exchangers.

An experimental study on heat transfer characteristics from a semi-circular tube in cross flow perpendicular to the curved tube surface have been carried out by Ibrahim and Elsayed [18]. The results illustrated the superior heat transfer characteristics of semi- circular tubes compared to circular tube one.

From that mentioned above, the heat transfer coefficient and effectiveness depend on the angle of attack and Reynolds number. In order to employ semi-circular tubes as a heat transfer surface element of heat exchangers, it is basically important to examine the heat transfer characteristics and effectiveness of semi-circular tubes bank. However, there has been little information about them.

The purpose of the present study is to investigate the behavior of the heat transfer coefficient and the performance of semi-circular tubes bank cross flow heat exchanger with an in-line arrangement, for various longitudinal pitches and angles of attack at constant transfer pitch.

EXPERIMENTAL APPARATUS:

The experiments were conducted in the thermodynamics laboratory of department of mechanical engineering at Zagazig University. The experimental apparatus is an open-circuit airflow wind tunnel system operated in suction mode. Along its path of flow, air from the laboratory space passed through the test section, the air velocity through the tunnel is regulated by a control valve. The overall system is shown in Figure 1.

Three test sections of *125mm* × *125mm* cross section and *420 mm* length were specially designed and fabricated. All test sections were for in-line arrangement. The lower and upper walls of the test sections were fabricated from sheets of compressed wood covered with formica. The sidewalls were made from glass, so that the internal surface and tubes arrays were visible. Strips of rubber material were laid in longitudinal grooves milled into the top and the bottom edges of the side walls to guard against leakage

The assembled test section is held together with the wind tunnel by positioning screws. This construction allows easy removal and insertion of tube bank within the duct. The water boiler was designed to get an enough amount of hot water. The boiler is equiped with inlet valve controlled by a float. The boiler was made from aluminum and insulated to minimize the heat losses to the surroundings. A *0.55kW* water pump was used to deliver the hot water from the boiler to the semi- circular tube bank. The hot water flows through the heat exchanger and then returns again to the boiler. Two collecting boxes are used at the inlet and outlet of the tubes to satisfy the usually distribution of water flow rate. The collecting boxes are insulated via three layers of glass wool. The apparatus is equipied electric heater for the heating of entering air when it is necessary.

The plains of semi- circular tubes bank, which are used during experiments, are illustrated in Fig. 2. The tubes were made from copper of *26mm* outer diameter, *2mm* thickness, and *125mm* length. The tubes are heated by the hot water flowing through the tubes. The tubes in the banks are arrangements in- line. In such arrangements the transfer pitches are kept constant with variable longitudinal pitches. The relative longitudinal pitches (S_l/d) have the values of 1.35, 1.92, and 2.69 while the relative transfer pitch (S_t/d) is kept constant and equals 1.35.

Temperature was measured using copper-constantan thermocouples connected via switching box to a digital thermometer with uncertainty of 0.1%. Thermocouples were located in the inlet and the outlet of the test sections and water header. Also, twenty-four thermocouples were embedded in the walls of experimental tubes.

The air velocity across the tunnel of the heat exchanger was measured using a pitot- static tube connected to a micromanometer. The uncertainty in velocity measurements is estimated to be in the order of 2%. The free stream velocity various from 2.65*m/s* to 22*m/s*. It is important to mention that all measurements were performed at the steady state condition. The laboratory and hence, the inlet air dry bulb temperature was kept around 29 °C.

RESULTS AND DISCUSSIONS:

Figures 3 and 4 show the characteristic of a variation of Nusslet number (Nu) with the Reynolds number (Re). Figure 3 represents the variations of Nu with Re for semi-circular tube bundles at different b for a = 1.35 and $\theta = 0°$. From this figure, it is clear that Nu increases with the increase of Re and/or b. It is very interesting to note that, at $\theta = 0°$and

$Re = 16.5\times 10^4$, the Nu increases by 30% when b changes from 1.35 to 1.92 while it increases by 80% when b increases from 1.35 to 2.69.

Figure 4 shows how the angle of attack (θ) affects the heat transfer coefficient for various relative longitudinal pitches (b) at constant relative transfer pitch, a=1.35. A close examination of this figure show that, for a given Re and b, the increase of the angle of attack, Nu increases when θ changes from 0^{o} to 270^{o}. The percentage of Nu increase depends upon the value of b and θ, it is about 76% for b = 1.35 (Fig.4.a), 87% for b =1.92 (Fig.4.b) and 59% for b = 2.69 (Fig.4.c), all of them are at a = 1.35 and $Re = 16.5\times10^4$.

The increase of the heat transfer coefficient, as a result of the angle of attack variation from,$\theta = 0^{o}$ to $\theta = 270^{o}$ may be due to the growth of the vortices and eddies formation of air around the studied tubes [19]. Thus, from these results, it is clear that Nu depends on Re, b and θ, and could be expressed by the following relationship.

$$Nu = A\,(Re / 10^3)^n \qquad (1)$$

The obtained values for A and n summarized in Table 1, are valid for $2\times10^4 \le Re \le 16.5\times10^4$ and $1.35 \le b \le 2.69$ at any angle of attack.

Exemplified in Figures 5 and 6 are characteristic variations of the effectiveness, E. Figure 5 presents the effectiveness of heat exchanger (E) as a function of capacity ratio, (Z) for different longitudinal pitches (b) at constant angle of attack, $\theta = 0^{o}$ and a = 1.35. The value of E increases with increasing Z or/and b. The percentage of E increase is about 58% at Z = 1 when (b) increases from 1.35 to 2.69.

Figure 6 shows the effect of the angle of attack, (θ) variation on the effectiveness of the heat exchanger (E). As alienated in this figure, E increases with increasing θ at constant a = 1.35 for the given capacity ratio, Z and b. The increase of E depends on the value of b and θ. Also, it is seen from this figure that, when θ changes from 0^{o} to 270^{o} at Z = 1, E increases by about 54% for b =1.35 (Fig. 6.a), by 48% for b =1.92 (Fig. 6.b) and by 32% for b =2.69 (Fig. 6.c).

The effectiveness of the heat exchanger for semi- circular tube bundles and capacity ratio could be expressed by the following correlation:

$$E = B\,Z^{m} \qquad (2)$$

Where B and m in equation (2) are obtained for the applicable ranges for both $0.12 \leq Z \leq 1$ and $1.35 \leq b \leq 2.69$, at any angle of attack θ. The values of B and m are summarized in Table 2.

Figure 7 shows the influence of the angle of attack variation upon Nusslet number (Nu) at different relative longitudinal pitches (b). At $Re = 16.5\times10^4$, and relative transfer pitch a = 1.35, it is clear that, Nu attains its maximum value at $\theta = 270^o$ and reaches a minimum value at $\theta = 0^o$ for all values of longitudinal pitches (b).

The characteristic variations of the effectiveness (E) versus the angle of attack (θ) for different relative longitudinal pitches (b) at $Re = 16.5\times10^4$ are shown in Fig.8. It is noted that, E attains its maximum value at $\theta = 270^o$ and reaches a minimum value at $\theta = 0^o$ for all values of longitudinal pitches (b).

CONCLUDING REMARKS:

Heat transfer characteristics and effectiveness of semi- circular tube bank in in- line arrangement at different longitudinal pitches and angles of attack are clarified experimentally in the present investigation. The Reynolds number range examined is from 2×10^4 to 16.5×10^4. The relative longitudinal pitches changes from 1.35 to 2.69 while the relative transfer pitch was kept constant at a = 1.35. The angle of attack is varied from 0^o to 270^o. The main points obtained are summarized as follows:

1- Nusslet number increases with the increase of Re and/or the relative longitudinal pitch (b). Nu increase is about 80% when b increased from 1.35 to 2.69 at $\theta = 0^o$.

2- The increase of angle of attack from 0^o to 270^o, for the given Re and b, increases Nu. The percentage of Nu increase due to the change of the angle of attack from 0^o to 270^o is about 76% , 87% and 59% for b equal 1.35,1.92 and 2.69 respectively. Nu attained its maximum and minimum values at $\theta = 270^o$ and $\theta = 0^o$ respectively, for all the values of longitudinal pitches. The correlation $Nu = A\ (Re\ /\ 10^3)^n$ could be expressed showing the dependence of Nu upon Re, b and θ.

3- The effectiveness, E, of semi-circular tubes bundle depends, strongly upon θ, b and the heat capacity ratio Z. E increases with the increase of Z and/or b, θ. The highest and lowest values of E are obtained at Z = 1 and 0.12 respectively for all values of b and/or θ. The percentage of E increase, when θ changes from 0^{o} to 270^{o} at $Z = 1$, is about 54%, 48% and 32% for b equal 1.35, 1.92 and 2.69 respectively. Also, E attained its maximum and minim values at $\theta = 270^{o}$ and $\theta = 0^{o}$ respectively for all longitudinal pitches. The correlation $E = B\ Z^{m}$ could be expressed for heat exchanger of semi-circular tubes bundle for $0.12 \leq Z \leq 1$ and $1.35 \leq b \leq 2.69$ at any angle of attack.

REFERANCES :

1- Elmar, Achenbach "Total and local heat transfer from a smooth circular Cylinder in cross- flow at high Reynolds number" Int. J. Heat Transfer, vol. 18 PP. 1387-1396, 1975.

2- G.F. Hewitt, G.L.Shires and T.R. Bott, " Process heat transfer" London, 1994.

3- Zhukauskas, A,"Heat transfer in banks of tubes in cross flow of fluid " Minits, Vilnius, Lithuania, 1968.

4- S. Aiba. " Heat transfer around a tube in in- line tube banks near a plane wall ". Int. J. Heat Transfer. Vol. 112, PP 933-938, 1990.

5- E.Achenbach. " Heat transfer from a staggered tube bundle in cross flow at high Reynolds numbers. Int. J. Heat and Mass transfer. Vol., 32, No 2, PP 271-280, 1989.

6- J.W.Baughn, M.J. Elderkin, and A.A. Mckillop. " Heat transfer from a Single cylinder, cylinders in tandem, and cylinders in the entrance region of a tube bank with a uniform heat flux " Int. J. Heat Transfer. Vol. 108, PP 386-391. 1986.

7- E.Vanden Bulk," Optimum design of cross flow heat exchangers ". J .of Heat Transfer, ASME TRANS, Vol., 113, PP 341-347, 1991.

8- T.A.Gowell, "A general method for the comparison of compact heat transfer surfaces". ASME. J. of Heat Transfer, Vol., 112, pp 288-295, 1990.

9- Jun. Dar. Chen and Shau.Shing Hsieh, " General procedure for effectiveness of complex assembles of heat exchangers “. Int.J.Heat Transfer, Vol, 33,No8, PP 1667- 1674, 1990.

10- K.Pavan Kumar and G.Venkatarathnam " Optimization of matrix heat exchanger geometry “. J.of Heat Transfer, Vol, 1220, August .2000.

11- K.Ichimiya, N.Akino, and T.Kunugi. " A fundamental study of the heat transfer and flow situation around spacers (a single row of several cylinder rods in cross flow). Int.J.Heat Transfer, Vol 33,No 11,pp 2451-2462,1983.

12- H.M.Badr. " A theoretical study of laminar mixed convection from a horizontal cylinder in a cross stream ". Int.J.Heat Mass Transfer, Vol. 26, PP 639-653,1983.

13- M.W.Chang, Ba.Finilayson " Heat transfer in flow past cylinders at R_e< 150 Part I. Calculations for constant fluid properties", Numerical Heat Transfer, 12,pp 179-195,1987.

14- Em.Sparrow.L.Lee " Analysis of mixed convection about a horizontal cylinder" Int.J.Heat Mass Transfer 19,229-231,1976.

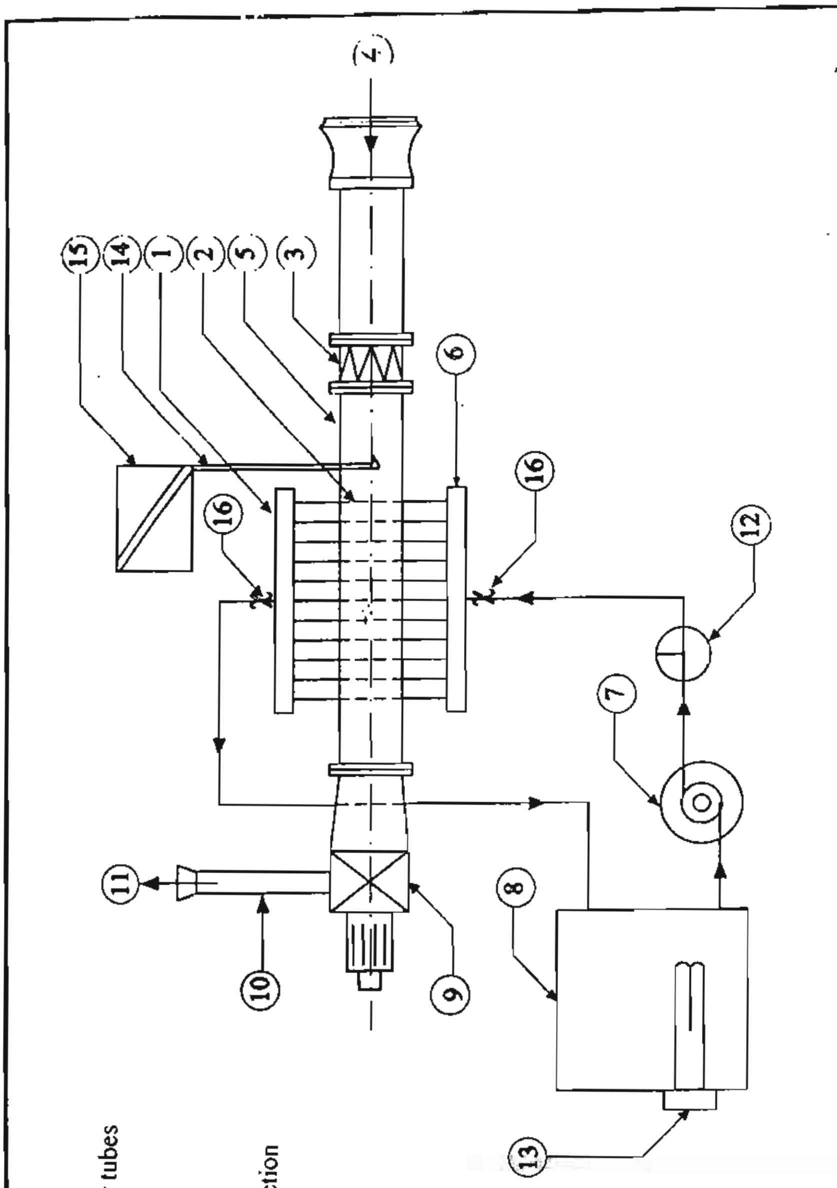
15- Mohamed.A.Aziz." Heat transfer augmentation from horizontal heated tube of Various enhanced surface" Alexandria Engineering Journal, Vol 30,January 1991.

16- Mohamed.A.Aziz " Comparative performance analysis of knuried lenticular tubing heat exchangers in cross flow " The Eighth International Conference For Mechanical Power Engineering Alexandria University, Egypt, April 27-29,1993.

17- Bassam, A.K, Abu.Hijleh " Laminar mixed convection correlations for an isothermal cylinder in cross flow at different angles of attack ". Int.J.Heat and Mass Transfer, 42,pp1383-1388, 1999.

18- Ibrahim.E.Z. and Elsyed.A.O. " Characteristics of heat transfer from a semi-circular pipe in cross flow" Mansoure, Third, Int.Engineering Conference, Egypt, Vol 2, pp 77-83, April, 2000.

19- Rady M.A, Moawed M, Ibrahim E.Z and Elsayed A.O. " An experimental study of fluid flow characteristics around a semi-circular tube" Military Technical College Cairo, Egypt, 10th Int. Conf. on Applied Mechanics and Mechanical Engineering, pp 625-636, May 2002.



1 - Upper header
2 - Experimental copper tubes
3 - Electric heater
4 - Air inlet
5 - Experimental test section
6 - Lower water header
7 - Water pump
8 - Water boiler
9 - Centrifugal fan
10 - Regulating valve
11 - Air outlet
12 - Water flow meter
13 - Electric heater
14 - Pitot tube
15 - Inclined manometer
16 - Water inlet and outlet thermocouples

Fig.1 Experimental test rig.

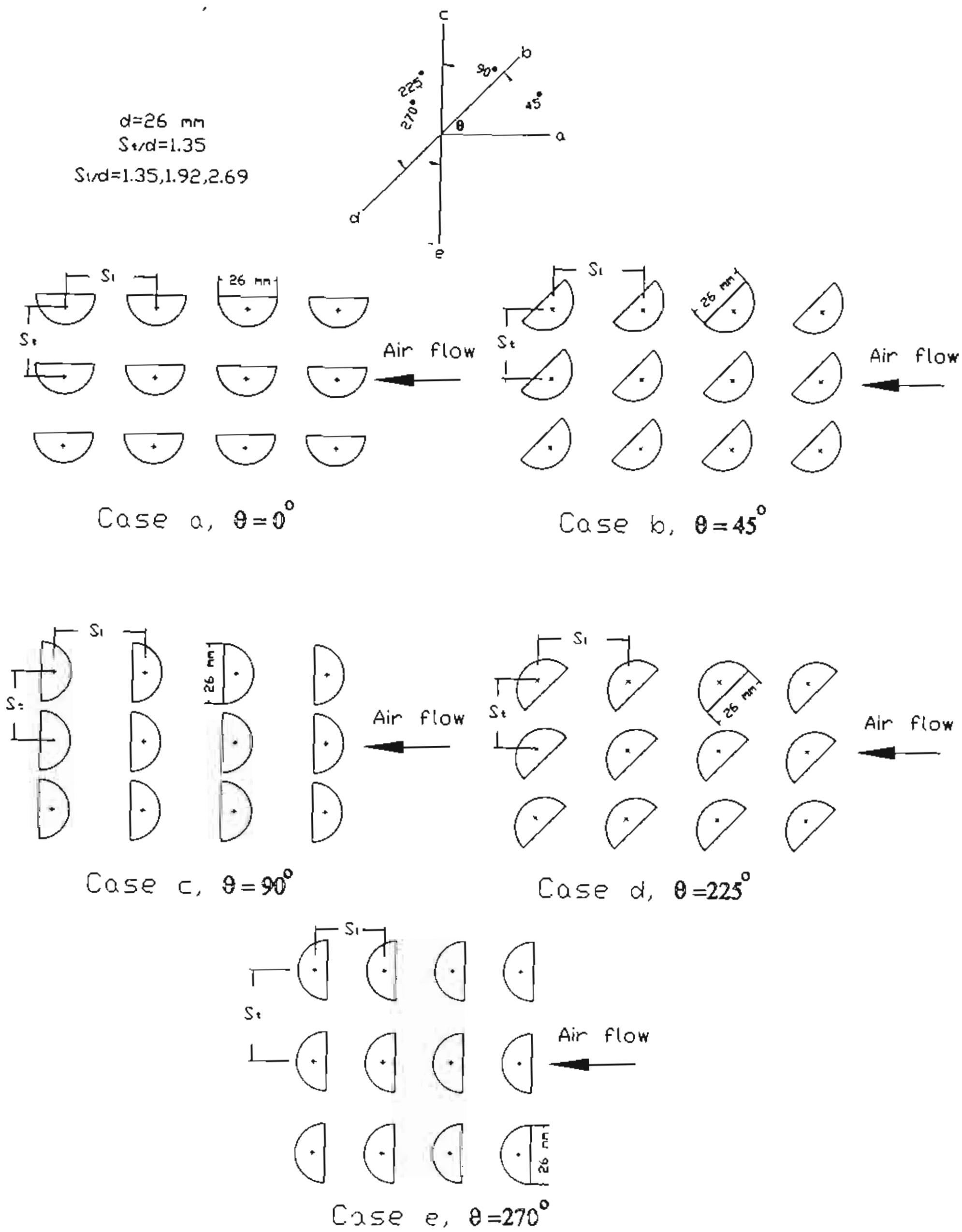


Fig.2. Arrangement of semi-circular tubes

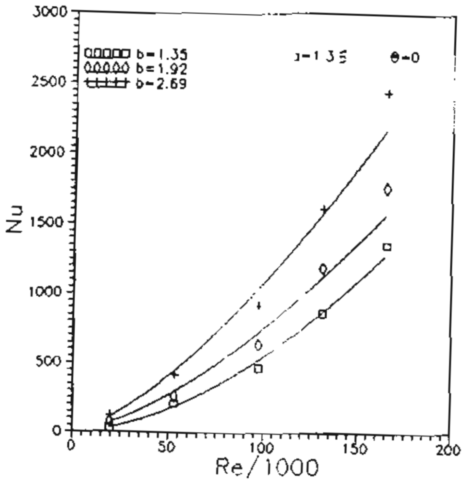
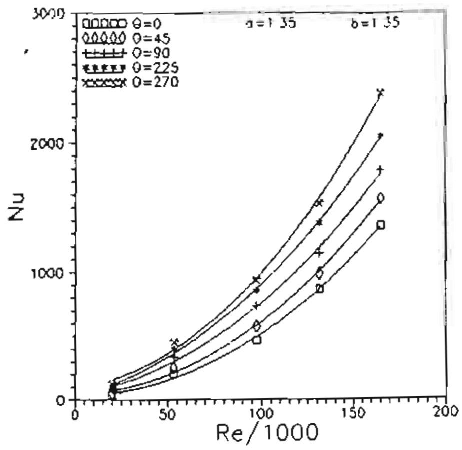


Fig.3. Nusslet number versus Reynolds number at $\theta = 0^{\circ}$

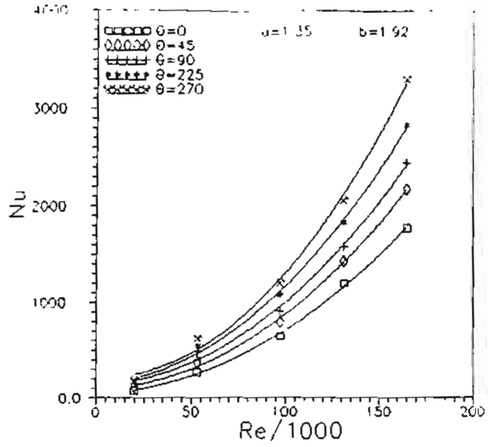
Table 1 constants for the empirical correlation of Eq. (1)

$A = a_1 + a_2 b$		$n = n_1 (b)^{n_2}$	
a_1	$-1.306 + \theta[-1.07 + 1.83\theta - 0.658\theta^2 - 0.07\theta^3]$	n_1	$1.819 + \theta[-0.294 + 0.043\,\theta]$
a_2	$1.138 + \theta[0.425 - 0.123\,\theta + 0.014\,\theta^2]$	n_2	$-0.287 + \theta[-0.01 + 0.192\theta - 0.079\theta^2 + 0.009\theta^3]$

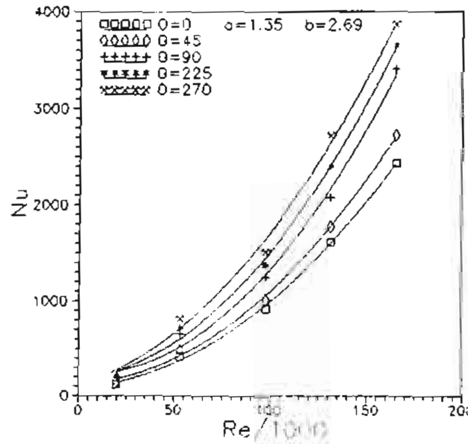
θ: Radian



a



b



c

Fig.4. Nusslet number versus Reynolds number at varies longtidunal pitches.

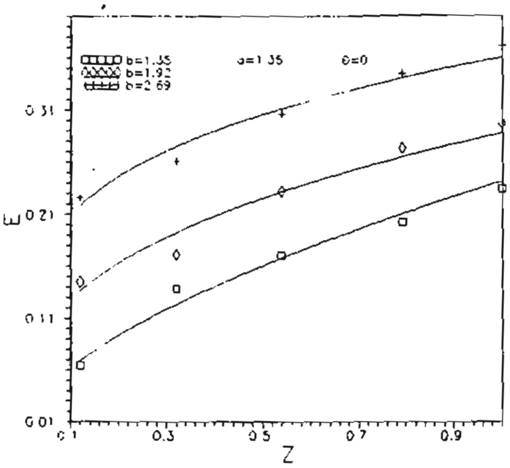


Fig.5. Effectiveness versus capacity ratio at $\theta = 0°$.

Table 2 values of B and m in Eq. (2)

$B = B_1 (b)^{B_2}$		$m = m_1 (b)^{m_2}$	
B_1	$0.202 + 0.0204\theta$	m_1	$0.869 - \theta [0.394 - 0.026\,\theta + 0.052\,\theta^2 - 0.01\,\theta^3]$
B_2	$0.581 + \theta [0.06 - 0.02\theta]$	m_2	$-1.344 + \theta [0.624 - 0.248\,\theta + 0.03\,\theta^2]$

θ: Radian

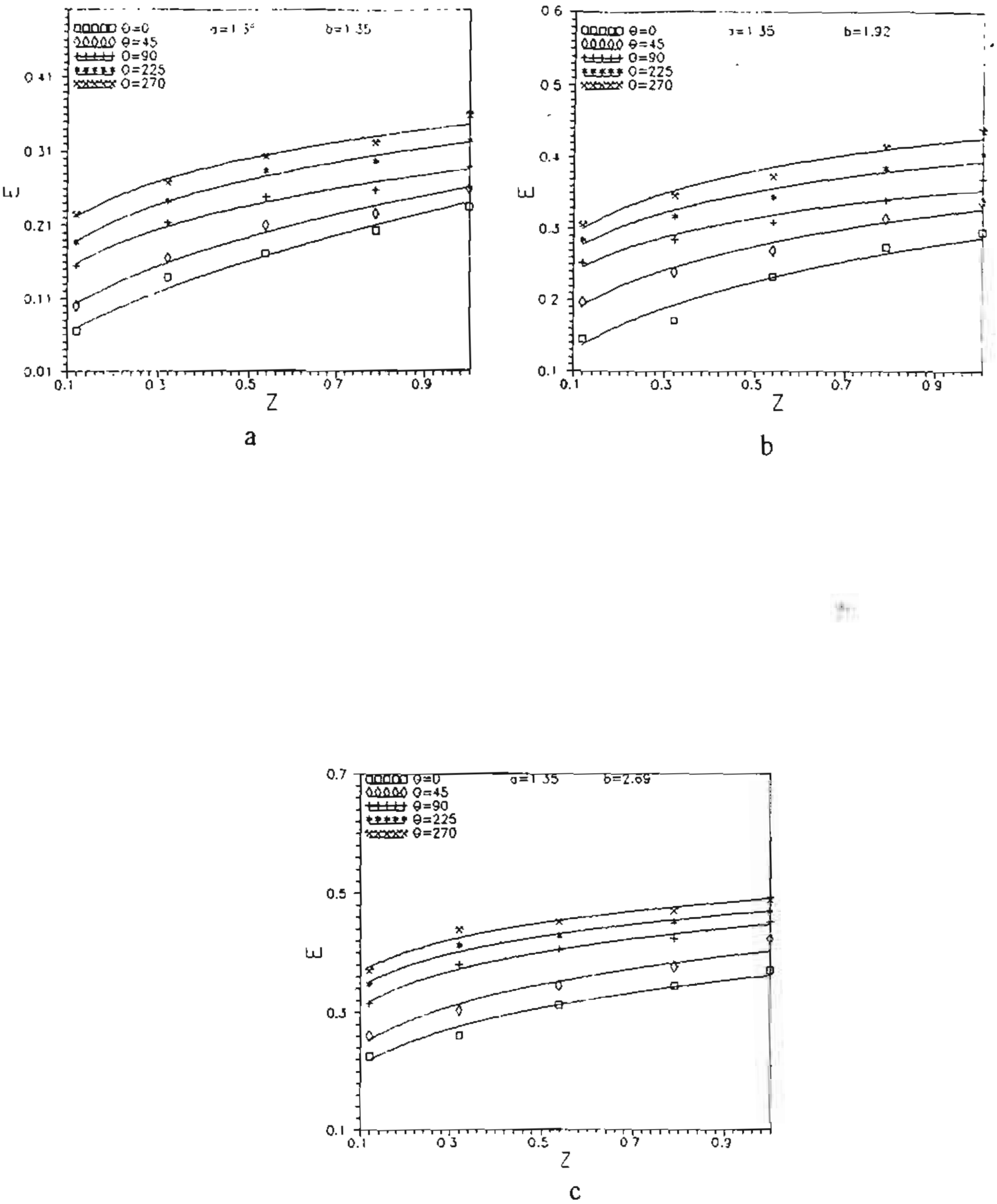


Fig.6. Effectiveness versus capacity ratio at varies longtidunal pitches.

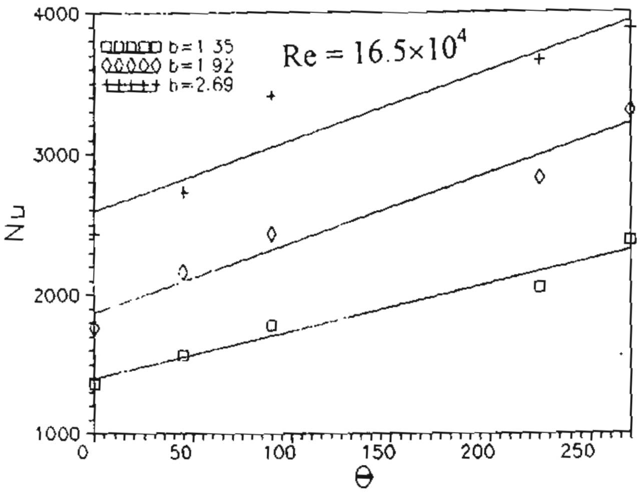


Fig.7. Nusslet number versus angle of attack at Re = 16.5×10^4.

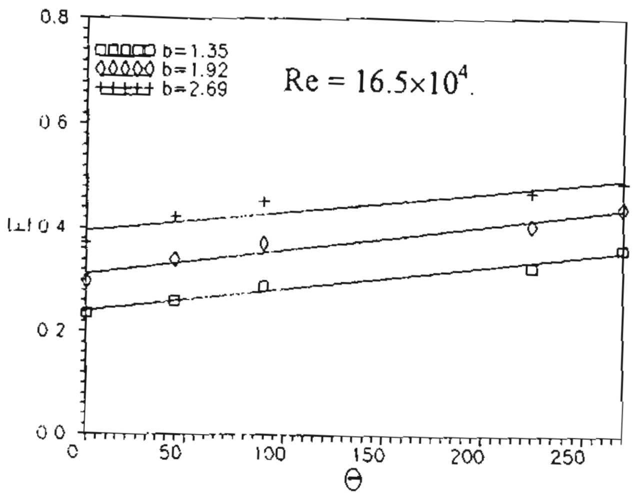


Fig.8. Effectiveness versus angle of attack at Re = 16.5×10^4