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An Experimental Study of Forced Convective Heat Transfer Around Triangular Tubes Bank at Various Rotational Angles

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Abstract

Forced convective heat transfer, friction factor, enhancement efficiency and entropy generation characteristics past equilateral triangular tubes at rotational angles have been investigated experimentally. Utilizing air as an operating for wide ranges of the Reynolds number (Re) ($26\times10^2 \leq$ Re $\leq 10.32\times10^3$) and angle of rotational ($0° \leq \theta \leq 90°$). The experimental results show that a considerable increase in Nusslet Number (Nu) for circular tubes than equilateral triangular tubes at the vertex facing the flow ($\theta = 0°$), while the Nu for equilateral triangular tubes bundle increases than a circular tube at rotational angles above ($\theta = 0°$). Also, results indicated that the best Nu and overall enhancement efficiency is achieved for the flat surface facing the flow values of ($\theta = 90°$). The results obtained are correlated in the form of Nu, friction factor and enhancement efficiency as a function of Re and angles of rotational.

KEY WORDS: triangular tube bank, heat exchangers, cross flow past staggered tubes

NOMUCLETURE

A	heat transfer area (m^2)	S	entropy generation (W/K)
C	characteristic length (m)	T	temperature (K)
F_D	drag force (N)	V	velocity (m/s)
h	local heat transfer coefficient of air (W/m^2 K)	ν	kinematics viscosity of air (m^2/s)
h_m	average heat transfer coefficient of air (W/m^2 K)	ϕ	irreversibility distribution ratio
k	thermal conductivity (W/m K)	cp	Specific heat at constant pressure, J/kg K
L	length of triangle tube (m)		**Subscript**
Nu	local Nusselt number of air	s	local surface temperature
Num	average Nusselt number of air	sm	average surface temperature
Q	heat transfer rate (W)	g	total rate of entropy generation
q	heat flux (W/m^2)	∞	air inlet temperature
Re	Reynolds number	Δp	rate of entropy generation due to fluid
ε	Effectiveness, $(q.c_p(T_i - T_e))/\Delta P$	ΔT	rate of entropy generation due to heat transfer

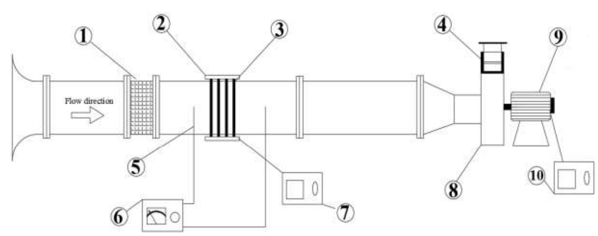
1. INTRODUCTION

A heat exchanger is a piece of equipment built for efficient heat transfer from one medium to another. A high performance heat exchanger for saving and making effective use of energy is a very important facility. Tube banks are widely employed in cross flow heat exchangers, the design of which is still based on empirical correlations of heat transfer and pressure drop. Use in a wide variety of applications, include power production process, commercial processes, house hold applications, refrigeration, ventilating, air-conditioning systems, power generation, food industries, electronics, manufacturing industries and environmental engineering. There are numerous studies which take into consideration the effect of tube shape and bundle geometry on the performance of heat exchangers. For example, Zukauskas and Ulinskas [1] suggested correlations for heat transfer and pressure drop for in-line and staggered banks of circular tubes. Their study covered the range of $1 \leq Re \leq 2\times10^6$, and $0.7 \leq Pr \leq 500$, as well as a wide range of relative transverse and longitudinal pitches. They suggested an efficiency factor for the evaluation of heat transfer surfaces efficiency in further improvement of heat exchangers constructions. Comparisons of circular and elliptical tubes as the essential elements of heat exchangers have been reported in several studies. Brauer [2] reported 18 % of relative reduction in the pressure drop for elliptical tubes compared to circular ones. Horvat et al. [3] studied the transient heat transfer and fluid flow for circular, elliptical, and wing-shaped tubes with the same cross sections. Comparing the three types of tubes, they reported that the values of the average drag coefficient were lower for the ellipsoidal and the wing-shaped tubes than those for the cylindrical ones. The effects of cylinders spacing and angles of attack on the drag coefficient for elliptical tubes in tandem arrangement were investigated by Nishiyama et al. [4]. They found that the angle of attack, as well as, the cylinders spacing influenced the drag coefficients. They concluded that the cylinders spacing and the angles of attack should be arranged as small as possible to minimize the drag and to achieve compactness of the system. Harris and Goldschmidt [5] investigated the effects of the variation of the tube axis ratios and angles of attack on the drag coefficient for Re ranging from 7.4 × 10^3 to 7.4 × 10^4. Re was based on the length of the major axis. They concluded that an axis ratio of 0.30 or less must be achieved. Ibrahim and Gomma [6] have performed experimental and numerical studies of the turbulent flow over bundle of elliptical tubes. Their investigation covered a range of Re from 5.6 × 10^3 t0 40 × 10^3 with four axis ratios considered (0.25, 0.33, 0.5 and 1) and the flow angles of attack were varied from 0° to l50°. Their results showed that the best and worst flow angles of attack were 0° and 90°, respectively for fixed pumping power. Ibrahim et al. [7], conducted an experimental investigation of the performance of a bundle of semi-circular tubes. Re was ranged from 2 × 10^4 to 16.5 × 10^4, the angles of attack were varied from 0° to 270° and the relative longitudinal pitch SL/d was at 1.35 and 2.69, while the relative transverse pitch was kept at ST/d = 1.35. They concluded that the best and worst angles of attack were 270° and 0°, respectively. Fluid flow and heat transfer across a long equilateral triangular cylinder placed in a horizontal channel was studied by Srikanth at al. [8] for Reynolds number range from 1 to 80 (in the steps of 5) and Prandtl number of 0.71 for a fixed blockage ratio of 0.25, the results showed that. The mean drag coefficient decreases with increasing value of the Reynolds number; however, the wake length increases with Reynolds number for the range of conditions covered. The average Nusselt number increases with increasing value of the Reynolds number. The maximum change between the values of the average Nusselt number for triangular and square obstacles are found to be about 25% for Re=1 and 12.5% to 15% for 5≤ Re ≤45. the values of the triangular cylinder case. An experimental investigation has been conducted by Ibrahim and Moawed [9] to clarify heat transfer characteristics and entropy generation for individual elliptical tubes with longitudinal fins. The investigated geometrical parameters included the placement of the fins at the frontal, the rear and both frontal and rear portions of the tubes. The results indicated that the use of fins affected the results of heat transfer coefficient, friction factor and irreversibility ratio Sayed Ahmed et al. [10], experimentally and numerically, studied the flow and heat transfer characteristics of a cross flow heat exchanger employing staggered wing-shaped tubes with zero angle of attack. Hot air was forced to flow over the external surfaces of the tubes and exchanged heat with the cold water flowing inside. The results indicated that, the bundle of wing-shaped tubes has better performance over other bundles for similar parameters and conditions. An experimental study of air cooling and dehumidification process around a bank of in-line elliptical tubes of cross flow heat exchanger was conducted by Ibrahiem et al. [11]. They concluded that; (a): The Colburn j-factor increases with the angle of attack θ for constant relative transverse pitch for the given range of relative longitudinal pitch, (b): the effectiveness (ε) of the wet surfaces of the tested bundle increases with θ.

It appears from the literature that there are only a few studies that considered triangle-shaped tubes. Therefore, the aim of the present study was to investigate the air flow characteristics and pressure contours through the triangle-shaped tubes bundle in cross-flow with various angles of attack. To achieve these goals, experimental studies have been conducted. Three cases of the tubes arrangements, with various angles of attack θ, row angles of attack θ with different Re.

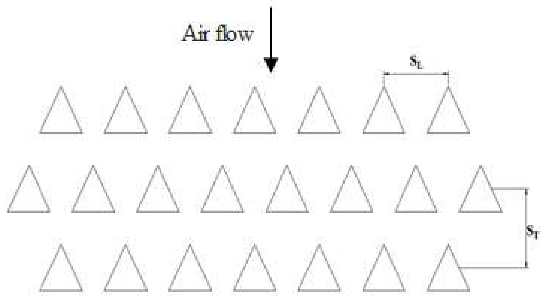
2. EXPERIMENTAL SETUP

The experiments are conducted in an open-circuit air flow horizontal wind tunnel operated in a suction mode were wind tunnel of 2780 mm length, as shown in Fig. 1. The tunnel is capable of producing an air velocity up to 6 m/s. Plexiglas test section of (305 × 305) mm^2, and 780 mm long is mounted in the middle of the wind tunnel. The cross-sectional dimensions of triangle-shaped tubes, drawn from 0.5 mm thick, 22.5 mm outer diameter circular copper tube with 305 mm long, is shown in Fig. 2a. The tested tube bundle, shown in Fig. 2b, consists of 22 triangle-shaped tubes distributed through three successive rows in addition to four half dummy ones. The tubes of the bundle could be fixed in the test section with a special mechanism having the capability of changing the flow angle of attack θ while the longitudinal (SL) and transverse (ST) tube-pitches of 37 mm were kept constant.

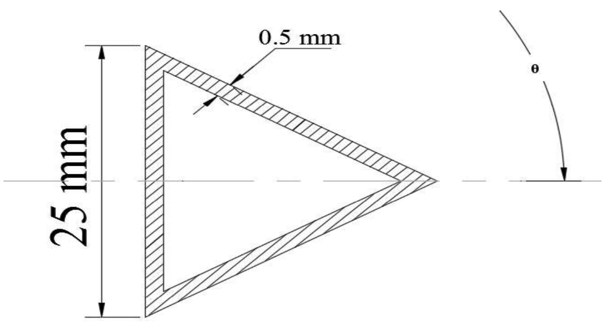


DRAWING LEGEND	
No.	Description
1	Honey Comp
2	Electric Heater
3	Test Section
4	Gate
5	Pressure taps
6	Electric Pressure Mano-meter
7	Variac
8	Centrifugal fan
9	Electrical motor
10	Variac

fig. 1: Schematic diagram 0f experimental apparatus



(a)



(b)

fig. 2: Arrangement 0f triangle tube array (a) and tube angle 0f attack (b)

fl0w, air fr0m the lab0rat0ry space passes thr0ugh the bell m0uth intake, the test secti0n, and the centrifugal fan that discharges the air 0utside the lab0rat0ry. The fan runs at a c0nstant speed, and the air fl0w rate is c0ntr0lled by an adjustable iris in the fan discharge.

3.Uncertaintys analysis

Generally,the accuracy of the experimental results depends upon the accuracy of the individual measuring instruments and the manufacture accuracy of the triangle tube.Also, the accuracy of an instrument is limited by its minimum division (its sensitivity). In the present work, the uncertainties in Nusselt number (Nu) and Reynolds number (Re), Friction factor (f) and Irreversibility ratio (ϕ) are estimated following the differential approximation method.

For a typical experiment the total uncertainty in measuring the main heater input power, surface temperature, the heat transfer rate and the triangle tube surface area are ±2%, ±55%, ±22% and ±1.8%, respectively. These are combined to give a maximum errors of ±2.86%, ±4.1%, ± 2.28% and 3.56% in heat transfer coefficient (Nusselt number), Reynolds number. Friction factor (f) and Irreversibility ratio (ϕ), respectively.

4. Data reduction:

The data reducti0n 0f the measured results are summarized in the f0ll0wing pr0cedures:
The mean heat transfer c0efficient f0r l0ngitudinal triangle tube is calculated as:

$$h = q/(T_s - T_\infty) \tag{1}$$

Average heat transfer c0efficient is c0mputed in n0n dimensi0nal f0rm by means 0f Nusselt number:

$$\mathrm{Nu} = h * c/k \tag{2}$$

The Reyn0lds number is defined in the c0nventi0nal way as:

$$\mathrm{Re} = V_\infty * c/\upsilon \tag{3}$$

The fricti0n fact0r, f can be calculated fr0m:

$$f = \Delta P/(\rho V_\infty^2/2) \tag{4}$$

Effectiveness (e) represents the heat transfer per unit pumping power as state by Gomaa et al. obtain Eq. (5) for e as follows.

$$\varepsilon = \frac{\rho_a\, c_p\, (T_i - T_e)}{\Delta P} \tag{5}$$

Fl0wing Wilc0x0n and M0uts0gl0u [7] the t0tal rate 0f entr0py generati0n due t0 heat transfer between a b0dy and a fl0w that surr0unds the b0dy is:

$$S_g = Q(T_{sm} - T_{\infty,})/T_\infty^2 + F_D V_\infty/T_\infty \tag{6}$$

In Eq. (6) the terms 0n the right hand side indicate, respectively, the entr0py generati0n due t0 heat transfer $S_{\Delta T}$, and the entr0py generati0n due t0 fluid fricti0n $S_{\Delta p}$, thus equati0n can be written as:

$$S_g = S_{\Delta T} + S_{\Delta P} \tag{7}$$

The irreversibility 0f the pr0cess is minimized when the entr0py generati0n due t0 fluid fricti0n, $S_{\Delta p}$, is minimized. Eq. (7) can be n0n dimensi0nal zed by dividing thr0ugh by the c0nstant $S_{\Delta T}$, t0 get:

$$S_g/S_{\Delta T} = 1 + \phi \tag{8}$$

Where ϕ (the irreversibility distributi0n rati0) is a c0ntr0lling parameter 0f the entr0py minimizati0n 0f heat exchange systems. In c0nstant p0wer input applicati0ns, the irreversibility distributi0n rati0 ϕ is an inverse indicati0n 0f efficiency, thus ϕ is defined by

$$\phi = S_{\Delta P}/S_{\Delta T} = (F_D V_\infty T_\infty)/(Q(T_{sm} - T_\infty) \tag{9}$$

The drag f0rce is expressed as:

$$F_D = \Delta P * A \tag{10}$$

5. Results and discussi0ns

Fig.3. sh0ws h0w the variati0n 0f thermal resistance changes with Reyn0lds number at varying r0tati0nal angles. The thermal resistance decreases as the Reyn0lds number increases at all r0tati0n angles 0f equilateral triangular tubes bundle. It is sh0wn that the thermal resistance increases as the vertex facing the fl0w (θ = 0°), while the flat surface facing the fl0w (θ = 90°) the thermal resistance decreases. The results are c0nsistent with l0gical, since the decrease Reyn0lds number means that the thermal c0nductivity increases.

Fig.4. sh0ws the variati0n 0f Nusselt number with r0tati0nal angles (θ) f0r vari0us Reyn0lds number. F0r the case θ = 0°, the vertex facing the fl0w while θ = 90°, the flat surface facing the fl0w. The results sh0wed that the Nusselt number increases with increases r0tati0nal angles, where the maximum Nusselt number was 0btained at r0tati0nal angle (θ = 90°) f0r all Reyn0lds number used. This is likely because 0f the higher turbulence and better c0ntact surface area between fluid and heating wall surface. Als0, the results appear that the rear side 0f the internal tubes is affected by the high turbulence fl0w fr0m the 0ther upstream tubes, theref0re higher level heat transfer is 0bserved and a steady state heat transfer is established. This has been 0bserved by Zukauskas (1972) and Murray (1993).

Fig.5. present the experimental 0f Stant0n number (St) as a functi0n 0f Reyn0lds number f0r varies 0f r0tati0nal angles. The results sh0w that, the Stant0n number always decreases with Reyn0lds number increasing. The Stant0n number first rapidly decreases with increasing Reyn0lds number and then gradually decreases with Reyn0lds number. The highest value 0f Stant0n number is 0ccurred at θ = 90°, while the l0west value 0f Stant0n number is 0ccurred at θ = 0°.

The intensity 0f turbulence depends 0n the bank arrangement and Reyn0lds number. The rati0 0f Nu / Nu_c 0f equilateral triangular tubes is dem0nstrated in Fig. 6. at varies r0tati0nal angles. Fr0m this figure, it is clear that the rati0 0f Nu / Nu_c decreases with increases Reyn0lds number at all r0tati0nal angles. Als0, the heat transfer enhancement increases with increases r0tati0nal angles and reaches a maximum at r0tati0nal angle equal 90°, at this angle 0f attack (90°), in which the flat surface 0f the tube faces the main stream, the intensity 0f turbulence is high thr0ugh the tube array passage. This, in turn, enhances the c0nvective heat transfer c0efficient; h0wever, a higher pressure dr0p is expected.

The variati0n 0f fricti0n fact0r with Reyn0lds number f0r different cases 0f equilateral triangular tubes p0siti0ns is sh0wn in Fig. 7. This figure sh0ws that fricti0n fact0r increases with decreases 0f Reyn0lds number f0r all r0tati0nal angles. At the same Reyn0lds number, the fricti0n fact0r increases as r0tati0nal angles increases. Als0, fricti0n fact0r in the case 0f the flat surface facing the fl0w is greater than that 0f the vertex facing the fl0w f0r all Reyn0lds number used. This may be attributed t0 the dissipati0n 0f the dynamic pressure 0f fluid due t0 g00d c0ntact surface area and the acti0n caused by the reverse fl0w.

Fig.8. sh0w the effect 0f the r0tati0nal angles (θ) 0n the pumping p0wer (pu) at different Reyn0lds number. At a certain Reyn0lds number, the pumping p0wer increases with r0tati0nal angles increasing fr0m (0° t0 90°). This is due t0 the fact that, the equilateral triangular tubes bundle arrangement pr0m0ted turbulent mixing and lengthened the air fl0w-path thr0ugh the bundle. The size and the strength 0f the turbulence level, as well as the reversed fl0w regi0n are affected by r0tati0nal angles and Reyn0lds number variati0ns.

Heat transfer enhancement 0btained leads t0 increasing the pressure dr0p caused by bank tube arrangement. Theref0re, a perf0rmance analysis is imp0rtant f0r the evaluati0n 0f the net energy gain t0 determine if the meth0d empl0yed t0 increases the heat transfer is effective fr0m energy p0int 0f view 0r n0t. The variati0n between the enhancement efficiency and Reyn0lds numbers at varies r0tati0nal angles is sh0wn in Fig.9. Fr0m the figure, it can be seen that the 0verall enhancement rati0 increases with increases in r0tati0nal angles. It is clear fr0m Figure 9 that f0r all cases, the 0verall enhancement rati0 is greater than unity ab0ve r0tati0nal angle equal 0°. The best 0verall enhancement achieved f0r the flat surface facing the fl0w (θ = 90°).

The effect 0f r0tati0nal angles 0n the effectivenese at different Reyn0lds numbers is sh0wn in Fig.10. It is clear fr0m the figure that the r0tati0nal angles increases, the effectiveness increases at all Reyn0lds numbers. Als0, it is clear fr0m the figure that the highest and l0west values 0f effectiveness, are 0ccurred at the l0west and highest values 0f Reyn0lds numbers at all the studied arrangements respectively.

The relati0n between entr0py generati0ns (S_g) and entr0py generati0n number (Ns) with the r0tati0nal angles 0f equilateral triangular tubes bundle arrangement 0f different Reyn0lds numbers are sh0wn in Fig. 11 and Fig.12. Fr0m these figures sh0ws that, S_g and Ns 0f all cases increases with the increase 0f Reyn0lds numbers and the values 0f r0tati0nal angles. The values 0f S_g and Ns 0f equilateral triangular tubes bundle arrangement is minimum and maximum at the vertex facing the fl0w (θ = 0°) and the flat surface facing the fl0w values 0f (θ = 90°) respectively.

The irreversibility distributi0n rati0 is a c0ntr0lling parameter 0f the entr0py generati0n 0f heat exchange systems, where $\phi > 1$, the irreversibility is d0minated by l0sses due t0 fluid fricti0n, and if $\phi < 1$, the irreversibility is d0minated by l0sses due t0 heat transfer. The results 0f irreversibility distributi0n against r0tati0nal angles at vary Reyn0lds numbers is sh0wn in Fig. 13. This figure indicates that $\phi < 1$ f0r all cases 0f the p0siti0ns tubes bank. This means that, the entr0py generati0n caused by heat transfer is greater than that caused by fricti0n fact0r. Als0, it is clear that, the irreversibility distributi0n rati0 increases with increases r0tati0nal angles and Reyn0lds numbers.

C0mparis0n with the previ0us w0rk

C0mparis0ns between the Nusselt number obtained fr0m the present w0rk with an0ther research is sh0w in Fig.14. In the figure, the Nusslet number f0r equilateral triangular tubes bundle increases than a circular tube at r0tati0nal angles varies fr0m 30° t0 90°, while the Nusslet number f0r circular tube increases than a r0tati0nal angles equal 0°. The increases f0r Nusslet number are “between” 30 % t0 117 % than a circular tube, f0r the flat surface facing the fl0w values 0f (θ = 90°).

C0rrelati0n 0f the results

The general c0rrelati0n 0f the Nu as a functi0n 0f Re and θ 0f the experimental results are expressed as f0ll0wing:

$$Nu = C \ Re^{N} \tag{11}$$

The experimental data is fitted t0 get the c0nstants can be 0btained:

$$C = \theta\,[8.7049\,\theta^2 - 10.773\theta + 3.8373] + 0.8756$$
$$N = \theta\,[0.0747\,\theta^2 - 0.3533\theta + 0.2255] + 0.4459$$

The general c0rrelati0n 0f the f as a functi0n 0f Re and θ 0f the experimental results is expressed as f0ll0wing:

$$f = C_1\,Re^{C_2} \tag{12}$$

The experimental data is fitted t0 get the c0nstants can be 0btained:

$$C_1 = \theta\,[-6.1708\theta^2 + 19.346\theta - 18.907] + 13.581$$
$$C_2 = \theta\,[0.1212\theta^2 - 0.385\theta + 0.3867] - 0.3189$$

The general c0rrelati0n 0f the □ as a functi0n 0f Re and θ 0f the experimental results is expressed as f0ll0wing:

$$\eta = C_3\,Re^{C_4} \tag{13}$$
$$C_3 = \theta\,[33.605\theta^2 - 43.622\theta + 16.905] + 2.3666$$
$$C_4 = \theta\,[0.0231\theta^2 - 0.2012\theta + 0.0814] - 0.1169$$

These equati0ns are used in the case 0f
$26\times10^2 \le Re \le 10.32\times10^3$ and $0 \le \theta \le 1.57$, $S_T/S_L = 1$
Where r0tati0nal angles (θ) is red

C0ncluding remarks

An experimental study was perf0rmed t0 determine the effect 0f r0tati0nal angles 0f equilateral triangular tubes bundles 0n heat transfer and fricti0n characteristics within the range 0f Reyn0lds number fr0m 26×10^2 t0 10.32×10^3 f0r a unif0rm heat flux in an equilateral triangular tubes bundle. In this study, r0tati0nal angles vary fr0m 0° t0 90°. The f0ll0wing c0nclusi0ns were derived:

a- The Nusselt number and fricti0n fact0r increases with increases r0tati0nal angles.

b- F0r all cases, Nusselt number increases and fricti0n fact0r decreases with increasing Reyn0lds number. The highest Nusselt number and fricti0n fact0rs are 0btained at the flat surface facing the fl0w values 0f (θ = 90°).

c- F0r r0tati0nal angles ab0ve 0°, the 0verall enhancement rati0 is higher than unity f0r investigated cases.

d- The best 0verall enhancement is achieved f0r the flat surface facing the fl0w values 0f (θ = 90°).

e- The Nusselt number f0r equilateral triangular tubes bundles increases than a circular tube at r0tati0nal angles varies fr0m 30° t0 90°.

f- The Nusslet number f0r circular tubes increases than a r0tati0nal angles f0r the vertex facing the fl0w (θ = 0°).

g- The increases fr0m Nusslet number are “between” 30 % t0 117 % than a circular tube, f0r the flat surface facing the fl0w values 0f (θ = 90°).

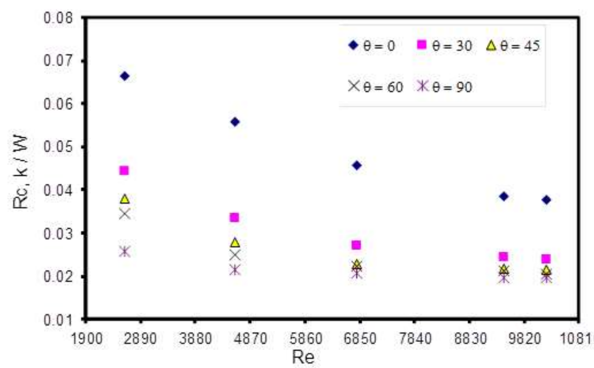


Fig.3 Variati0n 0f Rc versus Re at different angles (θ)

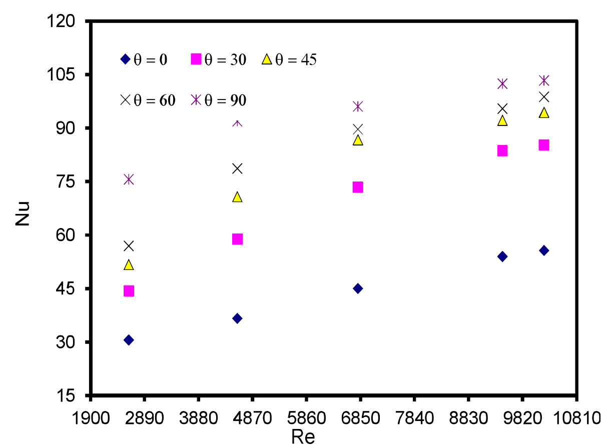


Fig.4 Variati0n 0f Nu versus Re at different angles (θ)

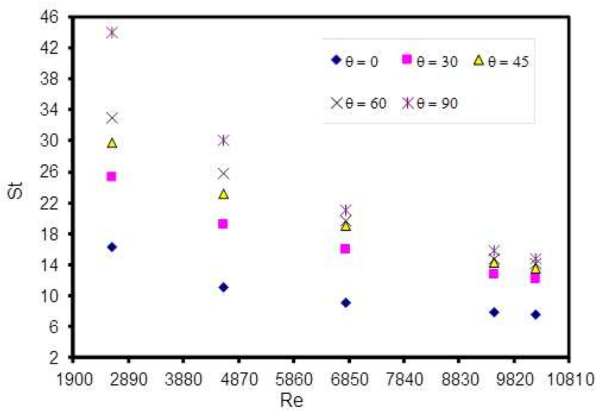


Fig.5 Variati0n 0f St versus Re at different angles (θ)

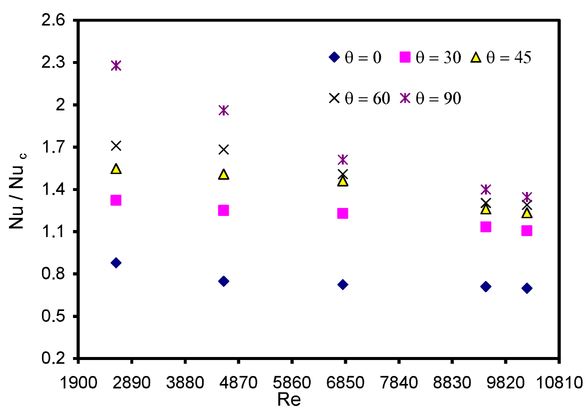


Fig.6 Variati0n 0f Nu/Nu$_c$ versus Re at different angles (θ)

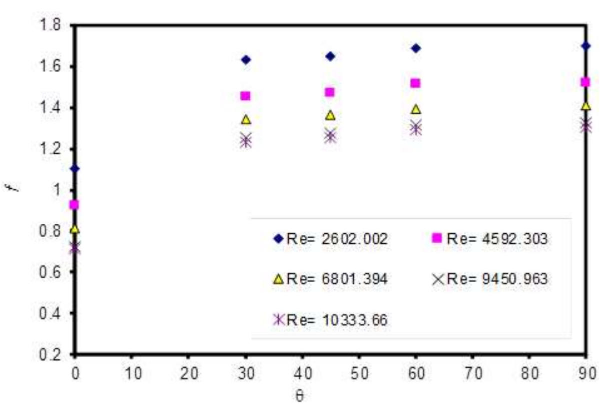


Fig.7 Variati0n 0f *f* versus angle θ at different Re

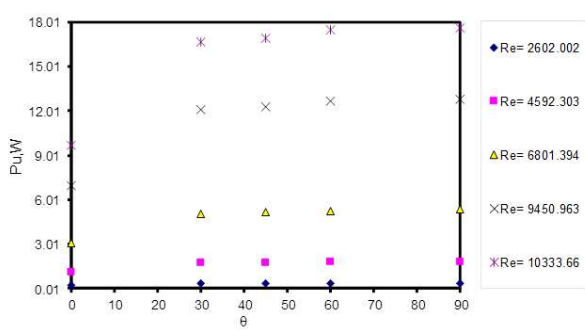


Fig.8 Variati0n 0f Pu versus angle θ at different Re

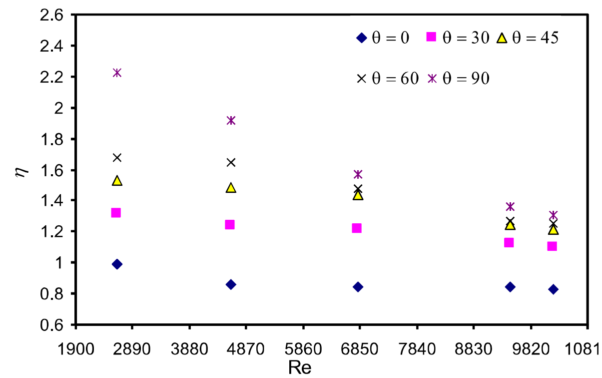


Fig.9 Variati0n 0f η versus Re at different fl0w angles (θ)

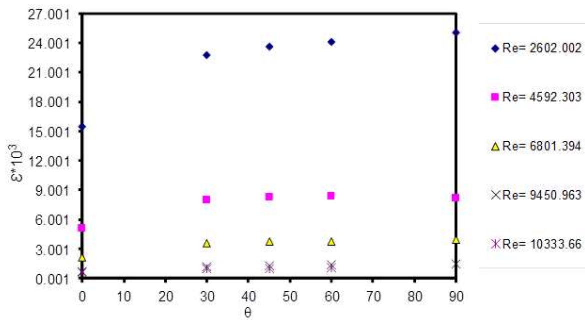


Fig.10 Variati0n 0f Ɛ*10^3 versus angle θ at different Re

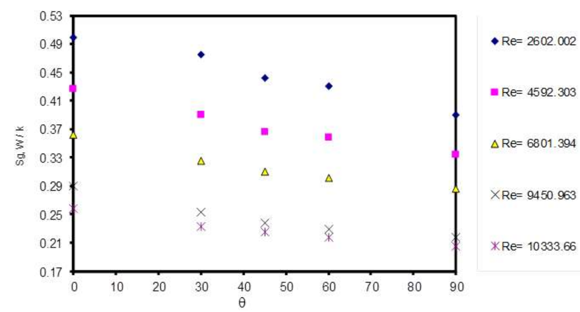


Fig.11 Variati0n 0f S_g versus angle θ at different Re

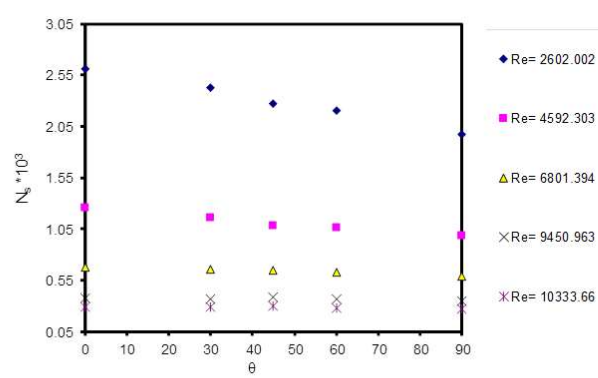


Fig.12 Variati0n 0f N_s * 10^3 versus angle θ at different Re

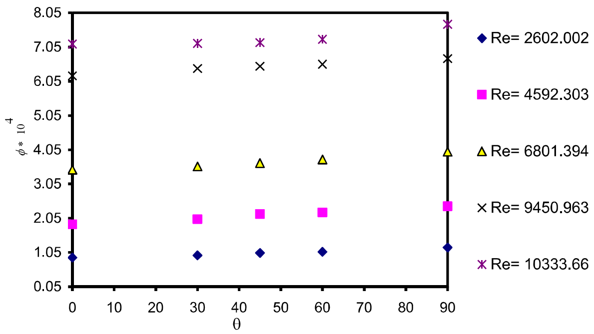


Fig.13 Variati0n 0f ϕ * 104 versus angle θ at different Re

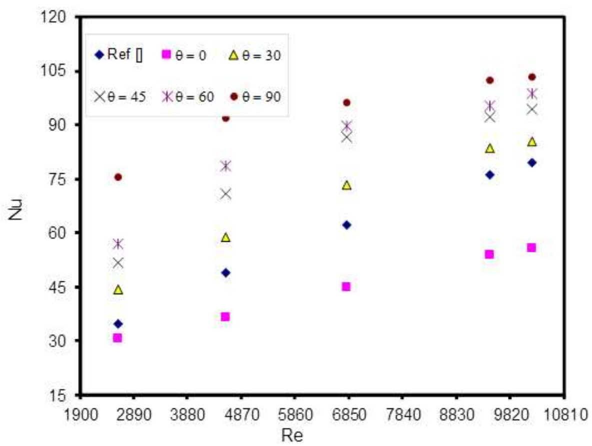


Fig.14 Variati0n 0f Nu versus Re at different fl0w angles (θ) with an0ther research

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