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EXPERIMENTAL INVESTIGATION OF THERMAL AND HYDRAULIC PERFORMANCE OF OVAL-TUBE BANKS IN CROSS-FLOW

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بحث تجريبي للاداء الحراري والهيدروليكي
لمجموعة انابيب بيضاوية في السريان العمودي

يتناول البحث دراسه معمليه للاداء الحراري والهيدروليكي لمجموعة انابيب بيضاوية المقطع بنسبة محاور ١ :٢,٥ ، ويبلغ عدد الانابيب ٤٥ انبوبه مرتبه في ستة صفوف تتعرض لسريان عمودي وبحيث تكون الوحده الاساسيه على شكل مثلث متساوي الاضلاع . وللمقارنه تم دراسة مجموعه اخرى من الانابيب مستديرة المقطع مرتبه بنفس الشكل الهندسي. ولقد تمت التجارب في نفق هواء يسمح بتيار هواء سرعته ٢٦ م/ث وكان عدد رينولد يتراوح من ١٢٠٠٠ الى ٥٣٧٧٧.
وتدل النتائج ان هناك تحسنا ملحوظا في اداء مجموعة الانابيب البيضاويه وذلك من خلال زيادة انتقال الحراره و تخفيض قدرة الضخ . كما تم الحصول على علاقات جديده لعدد نوسلت كداله في عدد رينولد وكذلك علاقات جديده اخرى لعدد نسلت كداله في عدد رينولد لكل من الانابيب المستديرة المقطع والبيضاوية المقطع.
كما تم استحداث مقياس جديد للاداء ياخذ الطاقه الحراريه المتبادله والطاقة المبذوله في الضخ في الاعتبار، ولقد تم التعبير عن هذا المقياس في صورة منحنى وعلاقه رياضيه كداله في عدد رينولد.

ABSTRACT

In this work, experimental investigation is conducted to clarify heat transfer and pressure drop for staggered oval tube banks in cross flow. For comparison, experimental data for staggered plain tube banks are obtained as reference. Oval shaped tubes are formed from plain tubes and have an axis ratio of 1:2.5. The test fluid is air and Reynolds number based on the major axis ranged from 12000 to 53777. The results show that there is a high degree of heat transfer enhancement for oval tube bank when the comparison is made at fixed pressure drop, while the plain tube bank is found to have higher value of heat transfer at fixed mass flow rate. It is found that the performance of tube banks can be evaluated by a dimensionless parameter (effectiveness) which is the ratio between the total heat energy and pressure drop energy. Correlations to calculate this parameter are presented.

INTRODUCTION

The present work on heat transfer from staggered tube banks in crossflow is based on the preceding research on flow characteristics of an oval tube bank [1]. The heat transfer results, complete the flow investigations and contribute to the understanding of what occurs in the performance evaluation for a tube bank. The exploration of compact and high performance heat exchangers for conserving energy is of vital importance. Among the different types of tube banks, those made of circular tubes are used in many industries. Flow around the tubes, however, is not always normal to the tube axis. In such a situation, the cross section of the tube in the flow direction becomes an ellipse. An elliptic and oval tube is a basic and general shape, which can become, a flat plate in a special case and also a circular tube depending on its axis ratio. In recent years. oval-shaped tubes instead of circular ones have been taken into consideration for crossflow heat exchangers utilizing heat recovery from the exhaust gas of gas turbines.

There have been many works on the flow around elliptic and oval-shaped tubes reported in [2] and the flow characteristics are found to change considerably with axis ratio and angle of attack. Furthermore, it is well known that its fluid dynamic drag for oval tubes is lower than that of circular tube. This may be an advantageous feature when using elliptic tubes as a heat transfer element. Banks with elliptic or oval-shaped tubes require considerably less pumping power on the shell-side compared to those with circular tubes. Hence, keeping the pumping power constant, it is expected, that the front area on the shell-side of such heat exchangers will be much smaller and the overall design more compact. Recently, few experimental studies have been made for elliptic and oval-shaped tubes to reduce the pressure drop [1,3,4].

As far as forced convection heat transfer characteristics of the elliptic and oval tubes are concerned, few research activities are available. For staggered tube banks in crossflow the only data available are those from Merker and Hanke [3]. In order to employ elliptic tubes as a heat transfer surface element of heat exchangers, it is basically important to examine the heat transfer characteristics of elliptic or oval-shaped tube banks to explore the performance of such heat exchangers. For this reason, the present experimental study was carried out to obtain mean heat transfer coefficient and pressure loss along the shell-side of plain and oval-shaped tubes arranged in staggered equilateral triangular centers.

EXPERIMENTAL APPARATUS AND TECHNIQUE

The experimental test circuit consists mainly of an air circuit, test heat exchangers, and water circuit which are described below.

(1) Air circuit:

The available open wind tunnel allows flowing large amount of fresh air at atmospheric conditions in order to cool the tested heat exchangers-Fig.(1). Air enters the tunnel through a convergent part by way of a protective screen. The working section is constructed from perespex and has a length of 650 mm. Down stream of the working section is a divergent part leading to a fan, and a double butterfly control valve. The tunnel is constructed from aluminum metal and rests on tubular steel supports. A parallel flow with sufficient uniformity is obtained, at the inlet of the test section. At the exit of the test section, a short diverging section is provided and the air flow is led to the atmosphere. The purpose of the diverging section is to convert the velocity of the exit stream to pressure head and to extend the capacity range of the air fan. The mean value of the temperatures t_{ai} and t_{ao} at the inlet and exit of the test section is used to determine the physical properties of the flowing air.

The main important technical data of the used wind tunnel are : working section 30.4 cm x 30.4 cm - maximum air speed 36 m/s, - 2900 rpm 7.5 HP fan - 3-phase 6 KW motor - overall dimensions 442 cm x 107 cm x 160 cm.

(2) Tested Heat Exchangers :

The plain tube bank which serves as the reference for all experiments, is illustrated schematically in Fig.2(a). The plain tubes which are made from copper were positioned on equilateral triangular centers. The tube diameter is D, the transverse and longitudinal pitches are denoted by p_t and p_l respectively, and the length of the tubes is S. In terms of these quantities, the investigated tube bank may be specified by the following dimension ratios:

$p_t/D = 2$, $p_l/D = \sqrt{3}$, $S/D = 15.24$, $D = 20$ mm, $D_i = 18$ mm.

The oval shaped tubes are formed from the same plain tubes with the minor to major axis ratio of 1 : 2.5. The oval tubes are also arranged in equilateral triangular center-Fig.2(b)- which is the same geometry of the plain tube bank. Each of the tube banks has two collecting boxes and the tubes are in the vertical direction. In all tests, the surface areas for both plain and oval tube-banks are the same, the longitudinal and transverse pitches are also the same. There are 45 tubes in each bank arranged in six rows and the overall dimensions are kept the same.

(3) Water Circuit :

A water tank is designed to get an enough amount of hot water with temperature lower than 90 ^{o}C (less than the saturation temperature at the atmospheric pressure). The tank is provided with three electric heaters placed at different levels and positions inside it. The electrical capacity of each heater is 2.5 KW. Also, each heater is equipped with a thermostat to help getting the hot water at desired temperature. The tank is insulated by fiber glass to minimize the heat losses to the surrounding. The circuit is equipped with a water pump to deliver the hot water from a heater tank as a heating fluid through the heat exchanger. Water is returned again to the heater tank. The water discharge is controlled by a gate valve. As the discharge of the water can be controlled, the flow meter is used to measure the amount of the discharge in lit/hr. The type of the flow meter is Precision Bare Flow meter Type No. FP1-35-G-10183. The maximum reading of the flow meter is 2300 lit/hr.

The pressure loss is measured with the help of alcohol differential inclined manometer. Error in measuring the pressure has been analyzed and the maximum error was calculated to be 3%. Copper-constantan thermocouples were used to measure the temperature of air side and water side with the help of the temperature recorder Model 4156, a member of the μR100 series. The maximum error in reading the temperature is found to be 0.3%.

EXPERIMENTAL RESULTS AND DISCUSSIONS

1- Pressure drop

As is known, the total pressure drop across a tube bank is a function of flow velocity, tubes arrangement, and physical properties of the fluid. The flow resistance of a bank handling viscous fluid of constant density can be expressed by the following functional relation [5]:

$$\Delta P = f\ (u,\ \Delta P,\ D,\ 2a,\ 2b,\ Z,\ \mu,\ \rho) \qquad (1)$$

The dimensionless form of this relation will be :

$$Eu = \Delta P/(0.5\ \rho\ u^2\ Z)$$

Normally, the pressure loss is represented as Euler number against Reynolds number, which is based on the reference velocity in whose definition there is considerable flexibility. If m represents the air flow rate per lane of the tube bank, and u is the velocity at the minimum free flow area for tube bank, then u can be defined as:

$$u = m/[\rho\ (p_T - D)\ S] \qquad \text{in case of the plain tube bank}$$

$$u = m/[\rho\ (p_T - 2b)S] \qquad \text{in case of the oval tube bank}$$

For a given tube bank, the relation between Euler and Reynolds number is expressed in a power law as follows:

$$Eu = c\ Re^{d} \qquad (2)$$

where the constants c and d are determined experimentally.

Figure (3) illustrates the relation between Euler and Reynolds numbers according to the experimental data reported by Ibrahim and Mahgoub [1] for four types of tube banks. The results of plain tube bank are in good agreement with that obtained by Zukauskas [5]. For tube bank with downstream splitter of L/D = 0.5 there is a reduction in pressure drop of 9 to 13% compared of that of plain tube bank. This trend agrees fairly with the data obtained by Ibrahim et.al [6] for 12 rows tube bank with different reduction values. This difference in the reduction value in pressure drop may be due to the difference in number of rows and number of tubes in each row. The results of the plain tube bank with upstream fin show a marginally higher value for pressure drop. On other hand, the results of the oval-shaped tube bank show a greet reduction in pressure drop of 60 to 63% compared with the data obtained for plain tube bank. According to the obtained data, the correlation for 6 rows heat exchanger can be written as the following:

$$\text{For plain tube bank,} \quad Eu = 10.398\ Re^{-0.358} \qquad (3)$$

$$\text{For oval tube bank,} \quad Eu = 4.10\ Re^{-0.308} \qquad (4)$$

2- Heat Transfer

To study the heat transfer behavior, the plain and oval tube banks without splitters are selected. The tested tube bank is heated by flowing hot water inside the tube bank. For each run, the mean heat transfer coefficient is calculated for each bank. The system was allowed to approach a steady state condition by running for one to two hours.

Nusselt and Reynolds numbers for oval tube bank based on the major axis are calculated using the following relations :

$$Nu = \alpha\ .\ (2a)/K\ , \qquad Re = u\ (2a)/\nu \qquad (5)$$

Generally, the mean Nusselt number for the hydrodynamically developed regime is given as a function of Reynolds number as follows:

$$Nu = A\ Re^{n} \qquad (6)$$

The constants A and n in the above equation depend on tube shape, mean stream velocity for each bank. In Fig.4 the Nusselt number is plotted as a function of Reynolds number over the range $20000 < Re < 60000$. It may be noted that a comparison between the Nusselt number for oval and plain tubes at a fixed Reynolds number actually compares the heat transferred per unit surface area at fixed flow rate. The obtained heat transfer results for plain tube bank are in good agreement with that obtained by Zukauskas [5]. It is important to notice that curve for the oval tubes lies blow that of for the plain tubes. According to the obtained experimental data, the power fit correlations for the 6 rows plain and oval banks can be given by:

For plain tube bank: $$Nu = 0.185\ Re^{0.652} \tag{7}$$

For oval tube bank: $$Nu = 0.143\ Re^{0.640} \tag{8}$$

For comparison, the correlation given by Zukauskas [5] for plain tubes is given by :

$$Nu = 0.317\ Re^{0.6} \tag{9}$$

It is obvious that the oval and plain tubes exhibit different pressure drop value and different Nusselt number for the same mass flow rate m. Figure 5 shows the relation between Nusselt number and pressure drop ratio- $\Delta P_{oval} / \Delta P_{plain}$ for the same mass flow rate- . Again, the curve for the oval tube bank lies under that of the plain tube bank. The behavior of Nusselt number shown in Figs. 4 and 5 gives an indication that the plain tubes exhibit higher thermal performance. Actually, this behavior can be changed if the Nusselt number is plotted against the Reynolds number for the same pressure drop as will be described below.

Now, the Nusselt number corresponding to a fixed pressure drop are plotted in Fig.6. Results are given for two different levels of pressure drop. Examination of Fig.6 shows that for fixed pressure drop, oval shaped tube bank yields higher heat transfer coefficient value by 21 to 23% than does the plain tube bank. This result approves that the oval tube bank exhibits higher thermal performance.

3- Performance Evaluation of the Tested Tube Banks

According to the analysis of the presented results of heat transfer as well as pressure drop on the tested tube banks , the thermal and hydraulic performance of heat exchangers can be evaluated by introducing a dimensionless parameter ψ which is the ratio between the total heat transfer energy and pressure drop energy and is given by:

$$\psi = \frac{Q}{\Delta P.\ V}$$

where Q is the heat transfer rate, ΔP is the pressure drop, and V is the volumetric flow rate. This ratio is termed here as effectiveness. Figure 7 illustrates a comparison between the effectiveness behavior for the oval tube bank and for the plain tube bank. It is clear that the oval tube banks exhibit higher performance than plain tube banks for the same configuration and overall dimensions. The curves presented in figure 7 can be expressed by the following best fit relations:

For plain tube bank, $$\psi = 1.822*10^{12} Re^{-2.545} \tag{10}$$

For oval tube bank, $$\psi = 8.878*10^{11} Re^{-2.410} \tag{11}$$

CONCLUSION

From the above discussion it can be concluded that the oval tube banks produce a reduction in pressure drop of 60 to 63% compared with plain tube banks. At fixed flow rate Nusselt number for plain tube bank is the higher than that for oval tube bank, while at fixed pressure drop the the oval shaped tube bank is the higher. Values of dimensionless parameter ψ as a ratio between total heat energy and pressure drop energy, are greater for oval tube bank by 44 49% over that for plain tube bank.

NOMENCLATURE

2a	length of major axis (m)
2b	length of minor axis (m)
D	diameter of the circular cylinder (m)
k	thermal conductivity (W/m.K)
Q	heat transfer rate (J/s)
p_T	transverse pitch (m)
p_l	longitudinal pitch (m)
ΔP	pressure drop (N/m^2)
S	tube length (m)
u	velocity at minimum free flow area (m/s)
V	volumetric flow rate (m^3/s)
Z	number of rows
α	heat transfer coefficient ($W/m^2.K$)
ρ	density (Kg/m^3)
ν	kinematic viscosity (m^2/s)
ψ	effectiveness or dimensionless ratio
Eu	Euler number
Nu	Nusselt number
Re	Reynolds Number

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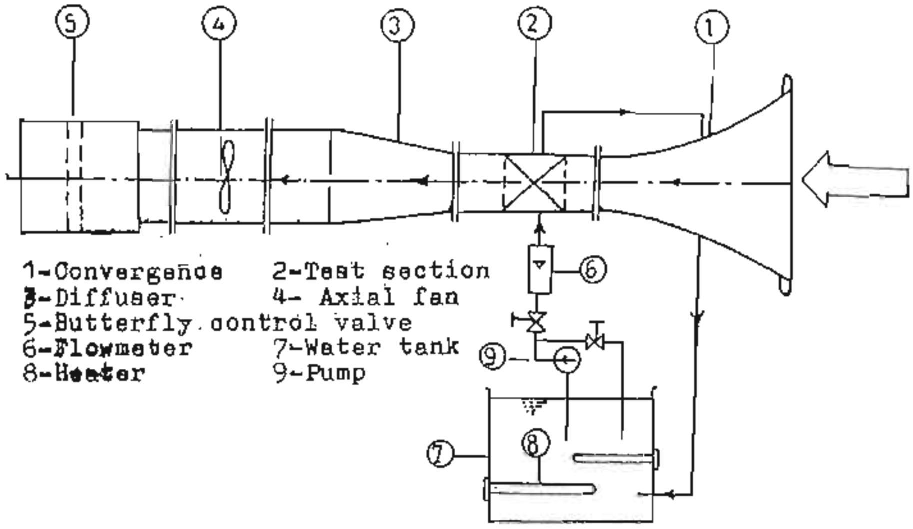

Fig.1 Schematic Diagram of the Experimental Set-up.

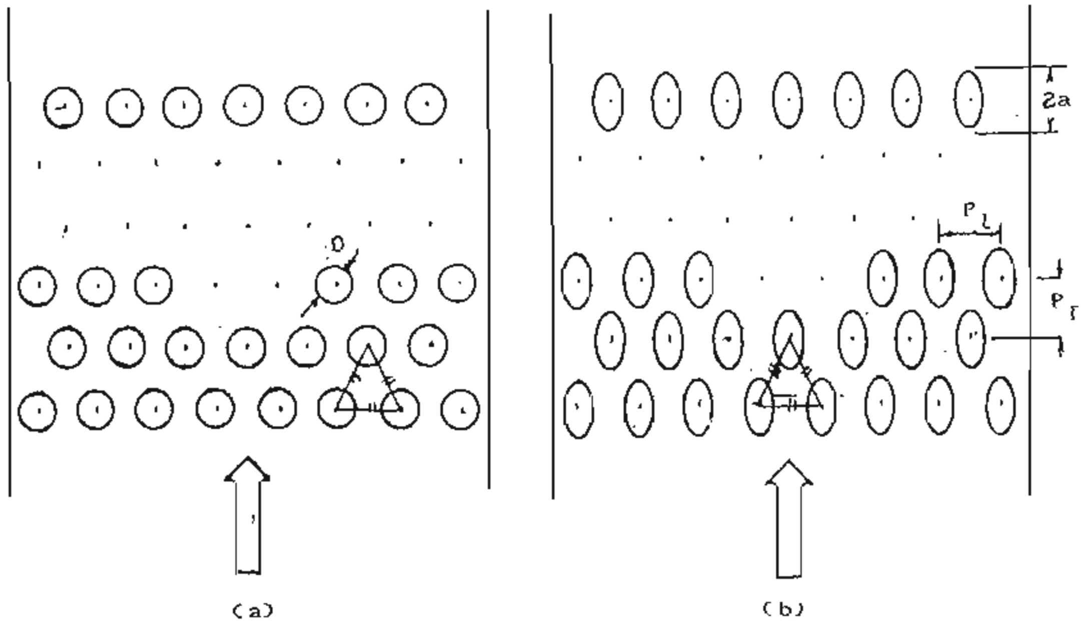

Fig. 2 Schematic diagram of tube banks

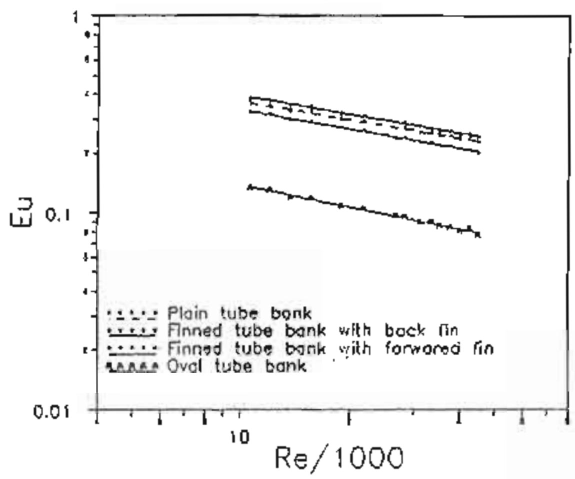

Fig.3 Euler number as a function of Reynolds number

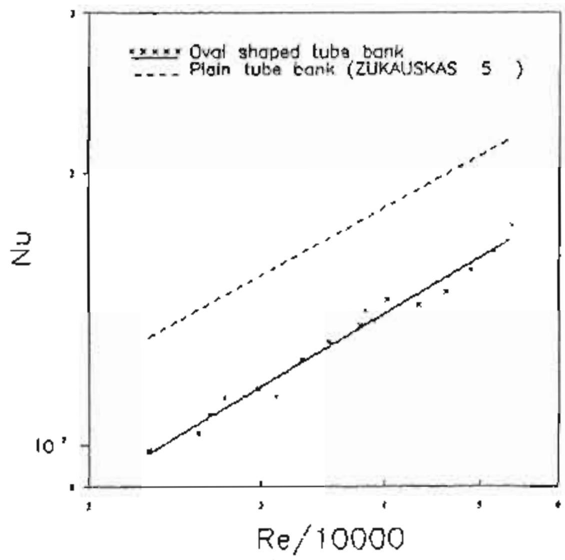

Fig 4 Nusselt number aginst Reynolds Number

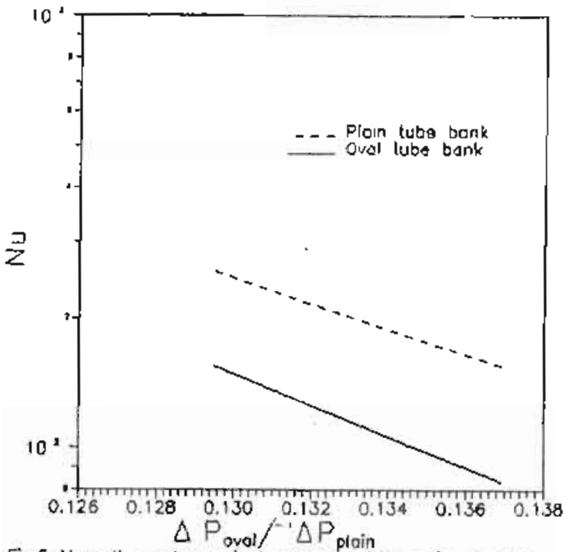

Fig.5 Nusselt number against pressure drop ratio of oval tube to plain tube banks at fixed mass flow rate

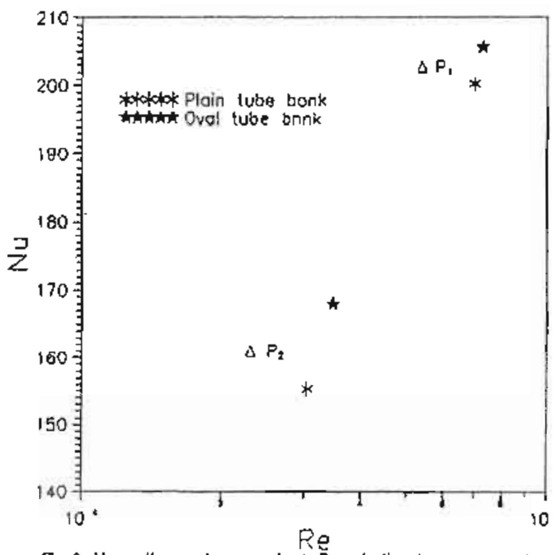

Fig.6 Nussell number against Re at fixed pressure drop (for plain & oval tube banks)

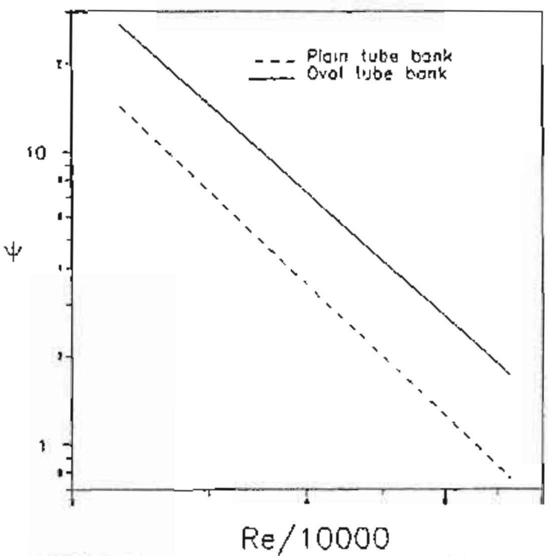

Fig.7 Effectiveness against Reynolds number for plain and oval tube banks