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Heat Transfer Effectiveness And Coefficient Of Pressure Drop On The Shell Side Of A Staggered Elliptical Tubes Bank.

Budi Utomo Kukuh W. 1,a, Samsul Kama2,b, Suhanan3,c, I Made Suardja4,d

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Keywords: staggered elliptical tube bank, aspect ratio, geometric parameters, effectiveness and coefficient of pressure drop.

Abstract. The effectiveness of heat transfer and the pressure drop coefficient of staggered elliptical tube banks are studied experimentally. The bank consists of 11 elliptical tubes of 0.75" equivalent diameter in an arrangement of 4-3-4. The major and the minor sub-axis of each tube are 24.70 mm and 12.35 mm respectively, and therefore the aspect ratio (AR) of the tube is 2.0. The geometric parameters of the bank are $S_T = 24.70$ mm, $S_L = 37.00$ mm and minimum frontal area $B = 12.35$ mm. Seven mid-tubes are internally heated by electrical heater of 69.6 Watt each. Experiment is conducted in a sub sonic wind tunnel and run with the wind velocities of 1 m/s - 12.6 m/s which correspond with Reynolds number of $= 346-6904$. The results show that the effectiveness ($\varepsilon$) varied from 2144.44 to 15.26. It decreases exponentially at low Reynolds numbers and tended asymptotically at higher Reynolds number. The coefficient of pressure drop ($Cdp$) ranges from 7.21 to 4.41 decreases continuously at low Reynolds number and asymptotic at higher one.

Introduction
The core of a compact heat exchanger is a tube bank where on the so called shell side, gas flows across the bank. And it is therefore the performance of any compact heat exchanger depends on the gas side mostly. The upmost characteristic of a compact heat exchanger is its very large surface area compared to its volume which achieves $A/V \geq 700$ m$^2$/m$^3$. Due to its compact size this type of heat exchanger is used widely in the energy industry and industrial process systems. Yet, the pressure drop characteristic is quite large on the gas side while its coefficient of convection is relatively small. The gas side heat transfer coefficient can be improved by at least two ways. The first one is increasing gas flow rate, and the second one is modifying the geometrical aspect of the tube. The increase of gas flow rate will directly increase the fan power to run it, while the geometrical aspects of the tubes can be modified by changing the cross section of the tubes and tube spacing as well. The change of tube cross section may also affect the heat transfer coefficient on tube side. Lenticular, oval and elliptical cross sections are receiving increased attentions due to their advantages. Among them, Ota et al. [1,2] studied the heat transfer and flow over an elliptical cylinder of 2.0 and 3.0 aspect ratio. The value of the tube pressure coefficient is smaller in the upstream stagnation point ($\alpha = 0^\circ$) up to $\alpha = 60^\circ$ than that of the circular ones. The best mean Nusselt numbers revealed on the angle of attack of $\alpha = 60^\circ$ and in the tube of 2 - 2.50 aspect ratio (AR). Further increase of aspect ratio showed no significant increase in Nusselt number. There are numerous works dealing with elliptical tube heat exchanger in cross-flow. A number of studies on heat transfer across tube banks reported that the staggered arrangement of the banks revealed the mean Nusselt number higher than those of aligned one [3,4,5]. This phenomenon occurs in all variations of the pitch ratio and Reynolds number. Kritikos et al. [6] investigated the performance of a staggered elliptic-tube heat exchanger for aero engine applications. Merker and Hanke [7] compared the effectiveness and pressure drop of some different configurations of oval tube banks.
Experiment Apparatus and Procedure

Figure 1 here under shows the schematic diagram of the experiments carried in this present study. A bank consists of 11 elliptical tubes and is arranged in a Armsfield sub sonic wind tunnel. Seven of the tubes are electrically heated by inserting electrical heater of 750-760 Ω electrical resistance. Variations in the wind tunnel air speed is set via a Regavolot potentiometer that controls the voltage electric fan motor. Air speed and temperature at the upstream position were measured by Lutron hot wire anemometer. Pressure difference of air flow inside the tunnel before and after the bank was measured using a mikromanometer 4u.5 Kofferprufsatz Modell.

![Fig. 1. Schematic diagram of the experimental apparatus](image1)

![Fig. 2. Schematic diagram of the test section](image2)

The tubes were made from circular copper tube of 0.75 inches diameter. The major and the minor sub-axis of each tube are 24.70 mm and 12.35 mm respectively, and therefore the aspect ratio (AR) of the tube is 2.0. The geometric parameters of the tube-bank are transversal and longitudinal pitches $S_T = 24.70$ mm and $S_L = 37.00$ mm respectively, and minimum spacing between tubes $B = 12.35$ mm. The characteristic length to calculate Reynolds number is the minimum spacing between two adjacent tubes (B). There are $2 - 3$ nodes on each heated tube, and hence there are 14 nodes altogether, to measure the surface temperature of the tubes as shown in Figure 3. Nodes 1-5 represent the first tube (upstream) temperature distribution. The temperature distribution of the second tube was represented by nodes: 6-10, and the third tube (downstream) was represented by nodes 11-14. Setting temperature reference is located above node 10, to operate electric heater when the temperature is below target and stop when the temperature exceeds the target. The offset allowance is 0.5 °C under and/or above the setting temperature. The temperatures of the nodes are acquired by Advantech Adam 4018 data logger of 300-500 ms interval simultaneously.
Experiments were performed on three consecutive setting temperature 49 °C, 59 °C and 69 °C for air velocity of 1 – 10 m/s. Assuming \( \cos \theta = 1 \) and all of the heat generated by the electric heater, \( P_{\text{elec}} \), was transferred by convection, \( q_{\text{kone}} \), to the air flow, the following relationship is valid therefore.

\[
P_{\text{elec}} = E \cdot I \cos \theta = q_{\text{kone}} = \dot{m}_{\text{air}} \cdot c_p \cdot \Delta T_{\text{air}}
\]

where \( E \) is the electric voltage, \( I \) is the electric current, \( \dot{m}_{\text{air}} \) is the mass flowrate of air, \( c_p \) is the specific heat at constant pressure and \( \Delta T_{\text{air}} \) is the temperature difference of air before and after the bank. Assume that air flow is incompressible with constant properties and steady state, then the pumping power required to force the air flow across the tube bank can be expressed as

\[
P_{\text{fan}} = \dot{v} \Delta p \text{ Watt}
\]

where \( \dot{v} \) = the air flow rate in \( \text{m}^3/\text{s} \) and \( \Delta p \) = the pressure drop in \( \text{N/m}^2 \).

### Table 1. Experimental data

<table>
<thead>
<tr>
<th>( T ) setting</th>
<th>( E ) (Volts)</th>
<th>( I ) (Amp)</th>
<th>( v ) [m/s]</th>
<th>( T_{eq} ) [°C]</th>
<th>( \Delta T_{eq} ) [°C]</th>
<th>( \Delta p ) [Pa]</th>
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<td>4912</td>
<td>15.42</td>
<td>174</td>
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</table>
Effectiveness of heat exchanger is defined as the ratio between heat transfer rate and power of fan to deliver cooling air, thus the effectiveness of heat exchangers can be summarized as

\[ \varepsilon = \frac{q_{\text{conv}}}{P_{\text{fan}} \Delta T_{\text{air}}} = \frac{h A_s (T_s - T_{\text{air}})}{\nu \Delta p} = \frac{m_{\text{air}} c_p \Delta T_{\text{air}}}{\nu \Delta p} = \frac{P c_p \Delta T_{\text{air}}}{\Delta p} \]  

(3)

Pressure drop coefficient is defined as the ratio between the pressure drop (\( \Delta p \)) and the dynamic pressure (0.5 \( \rho V^2 \)) of the airflow across tube bank

\[ C\Delta p = \frac{\Delta p}{0.5 \rho V^2} \]  

(4)

Both parameters, effectiveness and pressure drop coefficient are then used as the performance indicators of the heat exchanger. In general, both performance indicators are the functions of geometry, Prandtl number and Reynolds number. Heat exchanger with high effectiveness and low pressure drop coefficient is expected.

Results and discussion

The parameters of the tube bank are expressed as the function of Reynolds number which based on free stream velocity (\( V_s \)) and the minimum spacing between the tubes (\( B \)). The experimental data is summarized in the table 1. By substituting the value in table 1 into equation 2, 3 and 4 the result of the experiment is then summarized in table 2 and plotted in figure 3 and 5.

<table>
<thead>
<tr>
<th>Table 2. Result of experiment</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T ) setting = 49 °C</td>
</tr>
<tr>
<td>( Re_s ) 690   1381  2071  2762  3452  4833  5524  6214  6904</td>
</tr>
<tr>
<td>( \varepsilon )  2144.44  538.31  234.23  121.22  64.89  32.45  35.19  24.39  15.26</td>
</tr>
<tr>
<td>( C\Delta p )  7.21  5.41  4.80  4.50  4.47  4.56  4.56  4.45  4.94</td>
</tr>
<tr>
<td>( T ) setting = 59 °C</td>
</tr>
<tr>
<td>( Re_s ) 720   1308  2028  2682  2944  3401  5364  5587</td>
</tr>
<tr>
<td>( \varepsilon )  1032.23  242.18  127.51  61.49  69.24  42.40  26.80  21.06</td>
</tr>
<tr>
<td>( C\Delta p )  6.14  6.51  4.84  4.42  4.41  4.67  5.14  5.00</td>
</tr>
<tr>
<td>( T ) setting = 69 °C</td>
</tr>
<tr>
<td>( Re_s ) 1228  1842  2456  3070  3684  4298  4912</td>
</tr>
<tr>
<td>( \varepsilon )  1521.58  760.79  526.82  322.26  220.79  128.72  91.00</td>
</tr>
<tr>
<td>( C\Delta p )  6.71  5.97  5.03  5.22  5.01  5.34  5.21</td>
</tr>
</tbody>
</table>

![Figure 4. Effectiveness versus Reynolds number](image)

It can be depicted from table 2 and figure 4 that the effectiveness of the heat exchanger is high at very low Reynolds number and on the contrary, very low at high Reynolds number. The lower the setting temperature, the steeper gradient of the effectiveness curve in Reynolds number <2000. The
effectiveness tends to be asymptotic at Reynolds number of 3000 and or higher. The difference in operating temperature did not affect the value of the effectiveness at higher Reynolds number. The above phenomenon is in agreement with the trends observed by G.P. Merker and H. Hanke as shown in Figure 5. The effectiveness according to G.P. Merker and H. Hanke is the ratio between average heat transfer coefficient and power to flow the air. It therefore has unit of \( \text{K}^{-1} \).

![Graph](image1)

**Fig. 5. Effectiveness versus mass flowrate (G.P. Merker and H. Hanke)**

Such fashion is the coefficient of pressure drop as shown in Figure 6. The value of coefficient of pressure drop is high in low Reynolds number and decreases to minimum value at Reynolds number approximately 3000. It then increases parabolically to higher value at Reynolds number approximately 5000. Compare to the work G.P. Merker and H. Hanke shown in Figure 7, the recent experiment showed a good agreement.

![Graph](image2)

**Fig. 6. Coefficient of pressure drop versus Reynolds number**
Fig. 7. Coefficient of pressure drop versus Reynolds number (G.P. Merker and H. Hanke)

Summary
The recent study has examined the influence of $Re_B$ and the temperature setting on the effectiveness and pressure drop coefficients of a staggered elliptical tubes bank. The effectiveness of the bank decreases rapidly from 2144 on $Re_B = 690$ to 121-322 on $Re_B$ about 3000 due to the increase of $Re_B$. The effectiveness does not change significantly at higher Reynolds number. Pressure drop coefficients then increases, and reaches the maximum on $Re_B$ of about 5000.

References