Investigation the Performance of Cross Flow Heat Exchanger

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Abstract: In this study the performance of cross flow heat exchanger staggered tube bank arrangement are experimentally investigated. Four sets of tube bank consisting of 24 tubes, six rows each four tubes, are used for all experiments. Creating a number of (14, 29, 59) V-grove shape of one mm depth on outer surface of three sets while the fourth remain as it is smooth. Reynolds number based on maximum velocity in tube banks varied from (1380) to (51380) while heat flux supplied from (10) to (120) W/m² by steps of 10's varied. The heat transfer results of a smooth tube bank tests matches with other researchers are in good agreement. Meanwhile, other set shows that the heat transfer increase with increases of Reynolds number, while non dimensional pressure drop decrease with increases of Reynolds number. A new Nusselt number-Reynolds number correlation was found.

Keywords: Cross flow heat exchanger, staggered tube bank, groove,heat transfer,pressure drop.

Nomenclature

| Symbol | Description | Unit |
|--------|-------------|------|
| A      | Surface area of element | m²   |
| d      | Diameter of element | mm   |
| h      | Heat transfer coefficient | W/m².K |
| k      | Thermal Conductivity of air | W/m.K |
| L      | Effective length of element | m   |
| Nu     | Nusselt Number ( H/L / k) |    |
| Reₘ₃₈ | Reynolds number (Vₘ₃₈, L / ρ / μ ) |    |
| S₈     | Transvers pitch of tube bank | mm |
| S₉     | Transvers pitch of tube bank | mm |
| T₈     | Temperature of air | K   |
| T₉     | Temperature of air film | K   |
| V      | Inlet velocity into duct | m/s |
| Vₘ₉₈  | Maximum velocity passes tube bank | m/s |
| ρ      | Density of air | kg / m³ |
| μ      | Viscosity of air | kg . s / m² |
| ΔP     | Pressure difference | Pa   |
| ΔPₙ    | Non dimensional Pressure difference |    |

1. Introduction

Cross flow heat exchanger are found in different industrial sectors,such as steam generation in a boiler or air cooling in the coil of an air conditioner, where heat has transferred between different media. For optimal design of heat exchanger and flow pattern, the determination of its operational parameters and performance. The pressure drop and heat transfer between the fluid flow and the structure have to be known. one of methods to determine the performance is experimental work. Fahmy and Mohamed [1] investigated the characteristics of atube bank when some simple extented surface of different geometries (simple fin in down stream end, simple fin in both up strem and down stream end, and v shaped fin down strem) the investigation has been carried out in two plases,in-line and staggered arrangement .lowest value of Nu/ΔP* have been obtained for v shaped finned tube in there in line and staggered arrangement of other hand in line arrangement of plaine circular tube has resul of high Nu/ΔP* at low Re max however this ratio decresed as Re max incresed .They result showed that single sided straight fin (fixed at the back of tube) will be advantagesose. Sperrow ( as reported in [1] )performed heat transfer and pressure drop experiment for cross flow tube bank in which the individual tube were equipped with longitudinal fins in this exeriments ,they investigated the effect of fin tip geometry (blunt and contoured) and fin configuration or plangement (at the rear,at the front, and atboth rear and front) ,the experiment was performed for reynolds number range from 1000-8600 ,the results indicated that the high degree of heat transfer enhancement can be obtained by finning , comparing the results of the finned tubes to those of increased diamter unfinned tube showed that finning yields significantly greater heat transfer enhancement for the same pressure drop. Edwaed[2] performed experiment consisted of measurement of pressure losses and heat transfer rate for two unconventional cross flow heat exchanger, the first type consisted of an arrangement of circular tube in a sawtooth pattern,performance of this type did not appear to be improvement over more conventional heat exchanger,while the second type used exchanger tube of a speical lenticular cross section,so spaced as to keep the velocity of the fluid flowing out side the tube nearly constant in magnitude reducing seperation and drage. The lenticular tubes performance was superior to conventional type, especially at high reynolds number. Sundus H. Abd[3] experimental study was made on forced and forced heat transfer from three cylinders of different cross-sections circular, triangular and square in cross flow of air. The three cylinders were manufactured from copper for its high conductivity. The three cylinders were made to have equal surface area to compare the effect of their shape on heat transfer coefficient. It was found that in the case of forced convection, the heat transfer of the triangular cylinder is better than that of the square and circular cylinder and several empirical relationships were obtained for the case of forced convection. Igbinosa and Christian [4] They found in numerical and experimental investigations that changes in heat transfer from copper tube was dependent on the position and flow pattern (Reynolds number) in the heat exchanger while the pressure drop was a function of the Reynolds numbers. The heat transfer coefficient from the heated element was found to be a function of its position and
Reynolds number in the tube bank. Nusselt number increase by 12.49% from first to second column, 11.85% from second to third column and 5.27% from third to fourth column. This was due to increasing level of turbulence created by successive column of tubes as the air passes through the bank. However, the increments were at diminishing rate in successive column of tubes. Downstream from the third column, the heat transfer stabilizes, such that little change occurs in the convection coefficient from tube beyond the fourth. Tahseen A.Tahseen,et.,al.[5] In this study, an experimental study of heat transfer and air flow over in-line flat tube bundle. They found the average Nusselt number of air flow increases with the increase of Reynolds number . In present study effect of v-grove in 1 mm dpth which made on the surface tube bank with different number grove ( 14,29, and 59) ,tube bank are arrangined in staggered ,in this study heat transfer and pressure drop are investigated.idea of small groved on surface of tube bank is simulation to finite ribes on the skin of fish and dolphen which reduce friction with water.

2. Equipment and Test Samples

The test rig is coposed of adraw-through wind tunel having a square cross section of (15 *15) cm being set horizontally. The main part of the rig is the working section with spaces provided for the tube as shown in Figure 1 . One of the spaces is occupied by element consisting of tube of pure copper approximately of 15 cm long carried between two extension rods of a fabric-based plastic compound. The element is heated by electric heater built in. The heating element temperature is indicated by thermocouple type K which embieded in surface. The geometric layout of tube bank is staggered with four rows of six tubes. The tube diameter transverse pitch and horizontal pitch are 13 mm, 23 mm and 20mm respectively.

The apparatus includes a centrifugal fan driven by1200W electric motor and having its inlet connected to the working section. Air enters the test section by way of bell-mouth, after working section, a transition piece lead to the fan inlet and carries a honeycomb flous straightenier intended to prevent the transmission of swirl from the fan back into the working section. the fan discharge regulated by flap which regulated air velocity through appartus. Associated static tapping are provided so that the pressure drop across tube bundle are measured,for this purpose the inclined tube manometer is connected to the bottom measuringduct.the total pressure is measured with aid of pitot tube,to determine the flow velocity, the dynamic pressure is determine by inclined manometer,which one side connected to pitot tube “total pressure” and other side connected to tapping “static pressure”,the difference is dynamic pressure ,which used to calculated velocity of air flow. Present study invtigated the prformance of a tube bank when the surface of tubes are groved with different number v-grove with depth of 1 mm as shown in Figure 2.

![Figure 1: Apparature used in test](image_url)
The experimental data were used to determine Reynolds number \( (Re)_{max} \) for flow over tube based on the minimum area available for fluid flow in staggered arrangement and characteristic length of flow, Incropera [6], Buyruk [7]

\[
(Re)_{max} = \frac{V_{max} L}{\mu} \quad \text{(1)}
\]

And

\[
V_{max} = \left( \frac{S_l - S_t}{d} \right) V \quad \text{(2)}
\]

Where:

- \( V_{max} \) = maximum velocity of flow for staggered arrangement according the following Incropera.
- \( S_l \) = longitudinal pitch of tube bank (23 mm).
- \( S_t \) = transvers pitch of tube bank (20 mm).
- \( V \) = inlet velocity (m/sec).
- \( d \) = diameter of tube (13 mm), and
- \( L \) = is path of flow over tube = \( \pi d / 2 \).

The properties of fluid is calculated based on the film air temperature.

\[ T_f = \frac{T_h + T_a}{2} \]

The convection heat transfer coefficient \( h \) was calculated using newtons law of cooling.

\[ Q = h A (T_h-T_a) \quad \text{(3)} \]

Where:

- \( A \) = surface area of heating element.
- \( T_h \) = surface temperature of heating element.
- \( T_a \) = temperature of air at duct inlet.
- \( Q \) = steady state power input to appartuse.

Power input to heating element, this power dissipateded in two mechanisms convection and radiation, for the range of temperature difference during tests, heat transfer by radiation for maximum not more than 3%, according to this ratio heat transfer by radiation neglected, assume heat transfer only by convection.

Nusselt number was calculated for each run as:

\[
Nu No. = \frac{h L}{k} \quad \text{(4)}
\]

Where:

- \( k \) : Is the thermal conductivity of air at film temperature \( T_f \)

Pressure drop was measured between two taps across tube bundle. Pressure drop has nondimensionalizes by manes of following equation, Incropera [6], Buyruk [7].

\[
\Delta P^* = \frac{\Delta P}{0.5 \rho \nu^2} \quad \text{(5)}
\]

3. Results and Discussion

Firstly the heat transfer results, as shown in Figure 3, for smooth surface tube of the geometry arrangement \( (S_l/d=1.77) \) and \( (S_t/d=1.54) \) is found as (index) 0.566 compared (index) 0.568 in ref.[6] which means a good agreement, also, Figure 3 shown, the constant value of 0.564 found to be as higher as 0.429 taken from ref. [6], this difference due to the geometry arrangement \( (S_t/d=2) \) and \( (S_l/d=1.5) \) which means that the arrangement is more compact and suitable for higher heat transfer.

Secondly as shown in Figures 4, 5 and 6 the results of heat transfer for different grooves tube sets of 14, 29 and 59, that gives different \( Re_{max} \) indexes as 0.54, 0.426 and 0.55 respectively.

![Figure 3: Present Results compared with ref. [6].](image-url)

![Figure 4: Nu.No. vs Re_{max} for number of 14 groove.](image-url)
4. Conclusion

Heat transfer and pressure drop in cross flow heat exchanger experimentally investigated, different number of grooves (14, 29 and 59) created on the outer diameter of tube bank. It can summarised as follows:

1) Heat transfer increased as Reynolds number increases for all investigated cases.
2) Pressure drop decreased as Reynolds number increase for all investigated cases.
3) For the sets of case groves 14 and 29 heat transfer higher than smooth at low Reynolds number, but at higher Reynolds number had same heat transfer approximately.
4) For groove number 59 the heat transfer found to be higher than the smooth tube about 8% for all range of Reynolds number investigated.
5) For all cases non dimensional pressure drop increases compared with smooth tube, and that mean more power consumption.

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