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# EXPERIMENTAL COMPARISON OF STAGGERED AND IN-LINE TUBE-BANK THERMAL PERFORMANCE

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## ABSTRACT

*Experimental investigation of fluid flow and heat transfer were performed for cross-flow tube-type heat exchanger in staggered and in-line arrangement. The purpose of this work was to determine the process of forced convection heat transfer of air over a heated cylindrical pure copper rod. Also, the work was to compare the performance of both staggered and in-line tube bundle configurations. Tube bank with staggered arrangement consisted of 25 mm transverse pitch, 18.75 mm longitudinal pitch, 20 cylindrical rods of diameter12.5 mm, in a 125 x 125 mm cross section. An in-line tube arrangement was constructed based on the staggered tube bank dimensions. During the test, the working sections were placed one after the other in a specially design testing rig air channel through which the fluid flowed normal to the tube. A complete set of test were taken with the heated element in each of the four banks of tube for ten different throttle openings in the range of 10 to 100%. The test results show that the heat transfer coefficient associated with the tube is a function of its position in the tube bank. Higher heat transfer coefficients were obtained in the subsequent tube bank downstream of the first row. However, the heat transfer coefficients stabilize, such that little change occurs for tube beyond the fourth row. The overall heat transfer coefficient for staggered tube arrangement was found to be 4.28% higher than of the in-line tube arrangement.*

**Key words:** Forced Convective, Thermal Energy, Staggered Tube Bank And In-Line Tube Bank.

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# 1. INTRODUCTION

Heat transfer between two or more fluids in motion is indispensable in many industrial processes, and the device used to exchange the heat is called heat exchanger. In numerous applications, the tube-tube type heat exchangers are the most widely used and they represent the major share of heat exchanger market [1, 2]. The tubes are arranged in a column of array as either in-line or staggered configuration so as to accommodate many tubes as possible in order to achieve maximum heat transfer surface area [3, 4]. A tube bank is characterized by the dimensionless transverse ($S_T/D$), longitudinal ($S_L/D$), and diagonal pitches measure between tube centers where $S_T$ = Transverse pitch, $S_L$ = Longitudinal Pitch and D = diameter of the tube.

The flow conditions within the bank are dominated by boundary layer phenomena which include turbulence, flow separation and wake interaction and these influences convective heat transfer [5]. This occurrence is determined by the relative pitches and geometry of the bank. A more compact bank gives a better heat transfer performance when compared to a single-tube situation. The tube-tube type heat exchanger performance is also dependent on the number of longitudinal rows because of the inlet-outlet effects [6].

The excessive heat loss in modern thermal engineering applications coupled with the challenges of climate change, energy crisis, and instability in fuel prices have fueled research focus on improving the rate of heat transfer efficiency as well as conserving energy and heat transfer surfaces [7, 8]. As a result, many experts have developed several strategies for increasing thermal efficiency and at the same time cutting down on the material and energy utilization, and waste [9-13]. One of these strategies involves reduction in heat transfer surface material which in turn affects the cost of design and operation of heat transfer units. Bases on this premise, many studies have dealt with the process of forced convection heat transfer in tube banks. These results were used to develop heat transfer correlation for the total number of tubes [14]. Furthermore, studies have shown that the heat transfer coefficient associated with tube in tube-type heat exchanger is a function of its configuration and position in the bank of tube. The heat transfer varies from tube to tube for the first few rows downstream, after which no significant changes takes place [15-16].

However, there are limited works with respect to heat transfer coefficient variation in tube bank per position. The corresponding heat transfer correlations are also limited. This implies that the heat transfer from a tube as a function of its position in tube bank, with the appropriate correlations is needed in order to determine the maximum number of heat transfer surface such as tubes is required for optimum heat transfer and minimum pressure drop.

The purpose of this work is to carry out an experimental investigation to determine the heat transfer coefficient between a heated cylindrical copper element and the air flowing past it in a specially designed test rig composed of a Perspex working sections of staggered and in-line tube arrangements exposed to forced convection of air provided by a centrifugal fan. The experimental scheme also permits determination of the pressure loss coefficient, which represents the pressure drop imposed on the flow by each successive row of tubes expressed as a proportion of the velocity head.

# 2. EXPERIMENTAL SET-UP AND DESCRIPTION

## 2.1. Equipment details

The experimental set-up is depicted in fig.1. The working sections and rod are made of Perspex glass. The heated element is copper tube and its properties are presented in table 1. Air was used as working fluid and the properties are presented in table 2. The electrical heater is employed for heating the copper element in isolation from the working section. The temperature of the heated element was measured by a K-type 0.2mm diameter thermocouple probe. The thermocouple output was connected to a digital multi-meter which in turn was connected to personal computer to record the temperature per unit time.

The set-up also consisted of a centrifugal fan driven by 1hp electric motor at a constant speed of 2,500rpm and its inlet was connected to the working section. Air entered the working section through the bell-mouth to minimize aerodynamic loss. The air exiting the working section passed through a honeycomb flow straightener (not shown) to prevent transmission of swirl from the fan back into the working section.

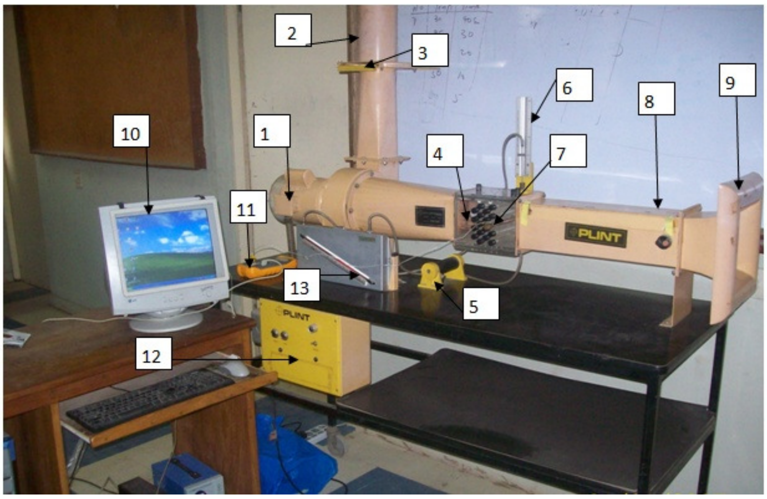

**Figure1.** Experimental setup:

1-Fan, 2-Air discharge tube, 3-Throttle valve, 4-Working section, 5-Electric heater, 6-Total head tube, 7-Test element, 8-Thermometer, 9-Bellmouth, 10- Computer Monitor, 11-Digital multimeter, 12-Control panel, 13-Inclined manometer

**Table 1.** Geometrical and properties of the heated element (copper tube)

| Descriptions | Quantity |
|---|---|
| External diameter ($d_o$) | 12.45 mm |
| Internal diameter ($d_i$) | 11.5 mm |
| Thickness of tube (t) | 0.5 mm |
| Length of tube (l) | 95 mm |
| Effective length ($l_1$) | 0.1304 m |
| Surface area (A) | 0.00371 $m^2$ |
| Effective surface area ($A_1$) | 0.00404 $m^2$ |
| Specific heat (c) | 380 J/kg.K |
| Mass | 0.0274 kg |

**Table 2.** Properties of the fluid (air)

| Descriptions | Quantity |
|---|---|
| Ambient temperature (TA) | $300^oK$ |
| Barometric pressure (PA) | 99,042 $N/m^2$ |
| Density of air at TA | 1.1614 $kg/m^3$ |
| Dynamic viscosity | $1.846x10^{-5}$ kg/ms |
| Thermal conductivity (K) | 0.0263 $J/ms^oC$ |

## 2.2. Experimental procedure

The experimental technique developed in this work, was geared to produce cooling curves for the heated element under various flow conditions. The fan was constantly in operation during the experiment while the flow conditions were regulated by the throttle valve.

As the fan was set in operation, the throttle valve was set to the required flow rates, air at ambient temperature powered by the centrifugal fan blown normal over the tube banks of five rows and four columns inside the working section. The velocity head upstream of the working section was observed equal to the pressure drop between the atmosphere and the upstream static pressure tapping. As soon as this was established, the depression at the static tapping was employed as a measure of velocity head upstream ($H_1$) via an inclined water manometer.

The total head tube was moved to the traverse position downstream of the tube and the second leg of the manometer was connected to the downstream static pressure tapping. The velocity head downstream ($H_2$) was recorded. The difference between the velocity heads yield the velocity head drop ($H_3$) across the tube was also recorded.

The air inlet temperature was recorded by the mercury-in glass thermometer at the inlet. The copper element was withdrawn from the working section and inserted in the cylindrical heater. The element was moved to the working section of the heat exchanger when heated to temperatures ranging from 95-100$^{o}$C with the fan running at desired throttle openings. The rate of cooling of each the heated elements was measured by a thermocouple embedded at the centre of each of the rows. The recording was taken at the rate of 1 data per second by the computer for ten (10) different flow rates. A logarithm plot for the rate of cooling was obtained using the temperature and time recorded at each opening. The heat transfer coefficients between the copper element and the air flow normal to it were determined with the slopes of the logarithm curves and the knowledge of the thermal capacity of copper, mass and surface area.

# 3. DATA REDUCTION

In these tests, the equivalent additional surface area represented by the plastic extensions was accounted for by making an addition to the true length of the element to give an effective length to be used in the calculations. This correction amounts to 8.4mm [17].

Therefore:

$$L_1 = L + 0.0084 \quad (1)$$

Further, the temperature gradients within the element were assumed to be negligible, this was done in order to obtain the effective surface temperature from the thermocouple embedded at the centre.

The heat transfer from the element to the air stream is expressed as:

$$q = hA(T - T_a) \quad (2)$$

The temperature drop d$T$ per time $dt$ is expressed as:

$$-qdt = \dot{m}cdT \quad (3)$$

Substituting $q$ *in* equation (3) gives:

$$\frac{-dT}{(T - T_a)} = \frac{hA_1}{mc}dt \quad (4)$$

Integrating equation (4) between $T$ and $T_o$ and $t = t$ and $t = 0$ respectively gives:

$$\log_e(T - T_a) - \log_e(T_o - T_a) = \frac{-hA_1 t}{mc} \quad (5)$$

A plot of $\log_e(T - T_a)$ against *t* as shown in Equation (5) will give a straight line of slope

$$M = \frac{-hA_1}{mc} \quad (6)$$

The heat transfer coefficient h can be calculated given that the other factors in this expression are known. Making the heat transfer coefficient h the subject of the formula from equation (6) the expression in equation (7) is obtained:

$$h = \frac{-mc}{A_1} M \quad (7)$$

A plot of $log_{10}(T - T_a)$ against time (t) was used noting that $log_e N = 2.3026\ log_{10} N$. The expression in equation (8) shows the relationship between heat transfer coefficient and the slope *M*.

$$h = \frac{-2.3026mc}{A_1} M \quad (8)$$

Obtaining the slope *M* through the semi-logarithmic plot of rate of cooling and knowing the thermal capacity and surface area of the copper element allows for the calculation of the average heat transfer coefficient between the element (copper) and the air flowing past it.

Upstream velocity was calculated from equation (9) and the effective velocity of air across the tube bank was determined using equation (10).

$$V_1 = 237.3\sqrt{\frac{H_1 T_a}{P_a}} \quad (9)$$

The effective velocity through bank of tubes was determined based on the minimum flow area. When all the tubes are put in place, the region the minimum area occurs is at the transverse plane, therefore, the effective velocity is expressed as:

$$V = 2V_1 \quad (10)$$

The mean of these velocities was then used in equation (11) to determine the Reynolds number. Nusselt and Prandtl numbers were determined with the expressions in equations (12) and (13).

$$Re = \frac{\rho V D}{\mu} \quad (11)$$

$$Nu = \frac{hD}{k} \quad (12)$$

$$Pr = \frac{C_p \mu}{k} \quad (13)$$

The pressure loss coefficient is expressed in terms of the velocity head. It accounts for the pressure drop as a result of the barrier posed by the tubes as air flows through each successive row of tubes. The pressure loss coefficient is expressed as shown in equation (14) since the mean velocity of air flowing past the tubes is twice the mean upstream velocity and there are four rows of tubes:

$$C_p = \frac{\left( H_3 / 4 \right)}{4H_1} \tag{14}$$

## 4. RESULTS AND DISCUSSION

The velocity distribution upstream of the tube banks was determined using the measurement of the static wall pressure at a point. From the obtained data, the relationship between the velocity head $H_1$ (upstream) and $H_3$ (across the tube bank) was established. The results of the test for staggered tube bank as shown in fig.2, gives the relationship between $H_1$ and $H_3$ as shown in equation (15):

$$H_1 = 0.22H_3 \tag{15}$$

For the in-line tube bank, the relationship between $H_1$ and $H_3$ is depicted in fig.3, graphically and is expressed in equation (16) as:

$$H_1 = 0.20H_3 \tag{16}$$

From equations (15) and (16), the pressure loss coefficients associated with staggered and in-line tube banks are 0.22 and 0.20 respectively.

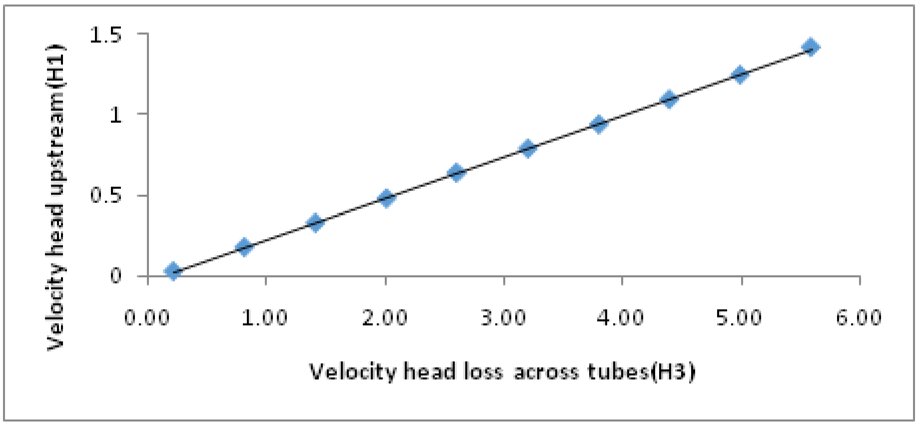


**Figure 2.** Velocity head upstream and velocity head drop across staggered bank

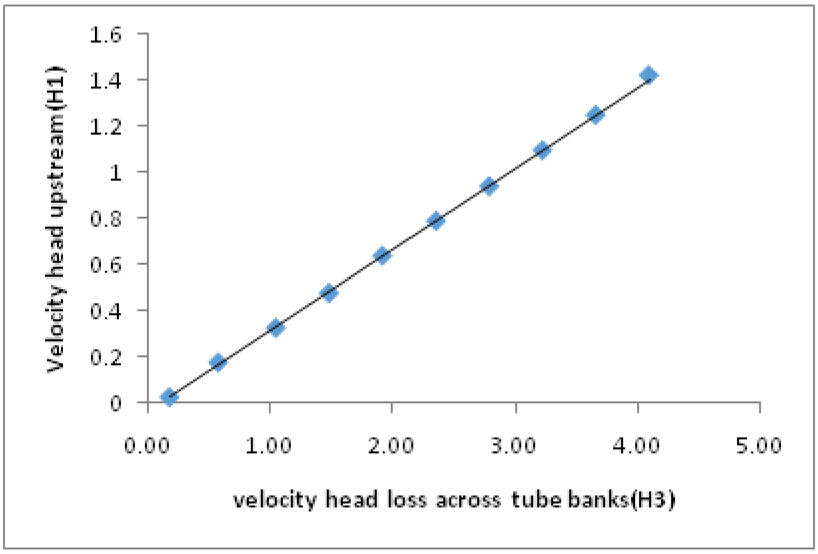


**Figure 3.** Velocity head upstream and velocity head drop across in-line tube bank

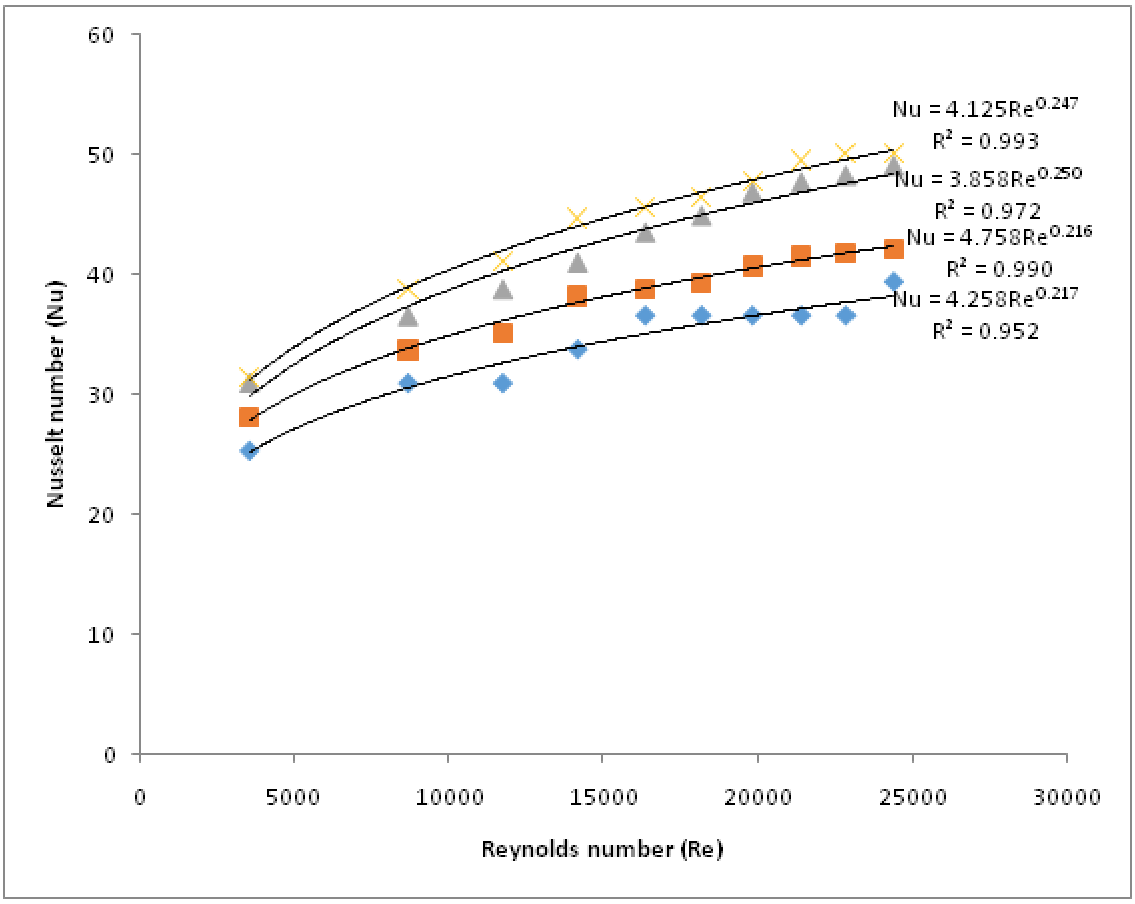


**Figure 4.** Relationship between *Nu* and *Re* for flow past staggered tube bank

The mean heat transfer coefficients were deduced using Equation (8). The Nusselt numbers were evaluated with the equation (12) using the mean heat transfer coefficients obtained. The slopes of the cooling curves were observed to increase in successive rows of tubes. This development increased the heat transfer coefficient downstream of the first row of tube bank.

In the staggered tube bank, the empirical equations derived from when the heated element was positioned at the centre of each of the four rows of the tubes are as presented in fig.4. It was observed that the heat transfer coefficient decreased, at a diminishing rate from row to row. The percentage drop in the heat transfer coefficients from rows one to two, two to three and three to four were 12.48, 10.82 and 4.08% respectively.

Similarly, in the in-line tube bank, the empirical equations derived when the heated element was positioned at the centre of each of the four rows of tubes are shown in fig.5. In this configuration of tube banks, the heat transfer coefficient decreased, at a diminishing rate from row to row. The percentage drop in the heat transfer coefficients from rows one to two, two to three and three to four were 12.49, 11.85 and 5.27% respectively.

From the results obtained, it can be inferred that the heat transfer coefficient of the fluid associated with these configurations will change slightly beyond the fourth row.

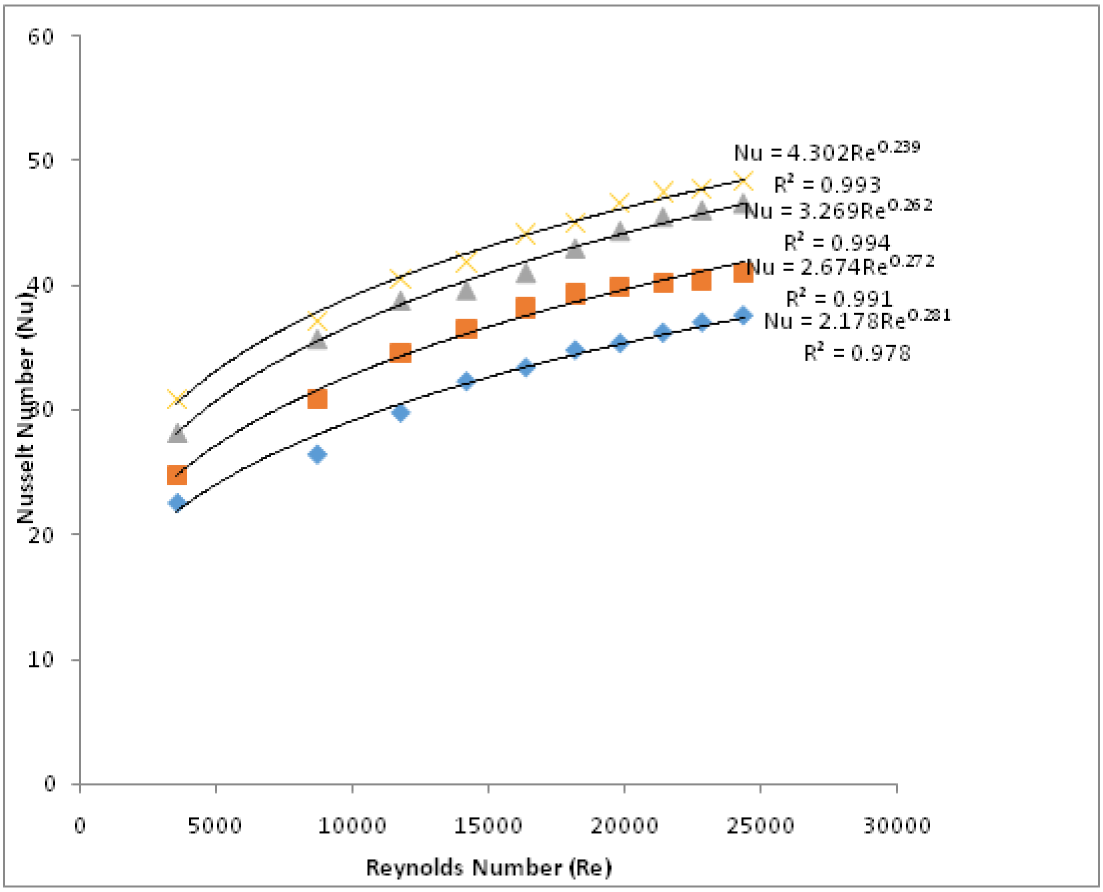


**Figure 5.** Relationship between *Nu* and *Re* for flow past in-line tube bank

A comparative analysis of the performance of the two configurations of tube banks (fig.6) revealed that for the range of Reynolds number, the heat transfer coefficients of staggered tube-bank were 5.32, 3.92, 4.49 and 3.4% higher for first, second, third and fourth rows respectively when compared to that of the in-line tube bank. This was because the heated elements in the staggered configuration were exposed to the main stream unlike the in-line tube-bank where both side of tubes except the first row tubes were in the wake region. However, the pumping power required to move the working fluid through the in-line tube bank was less than that of the staggered tube bank.

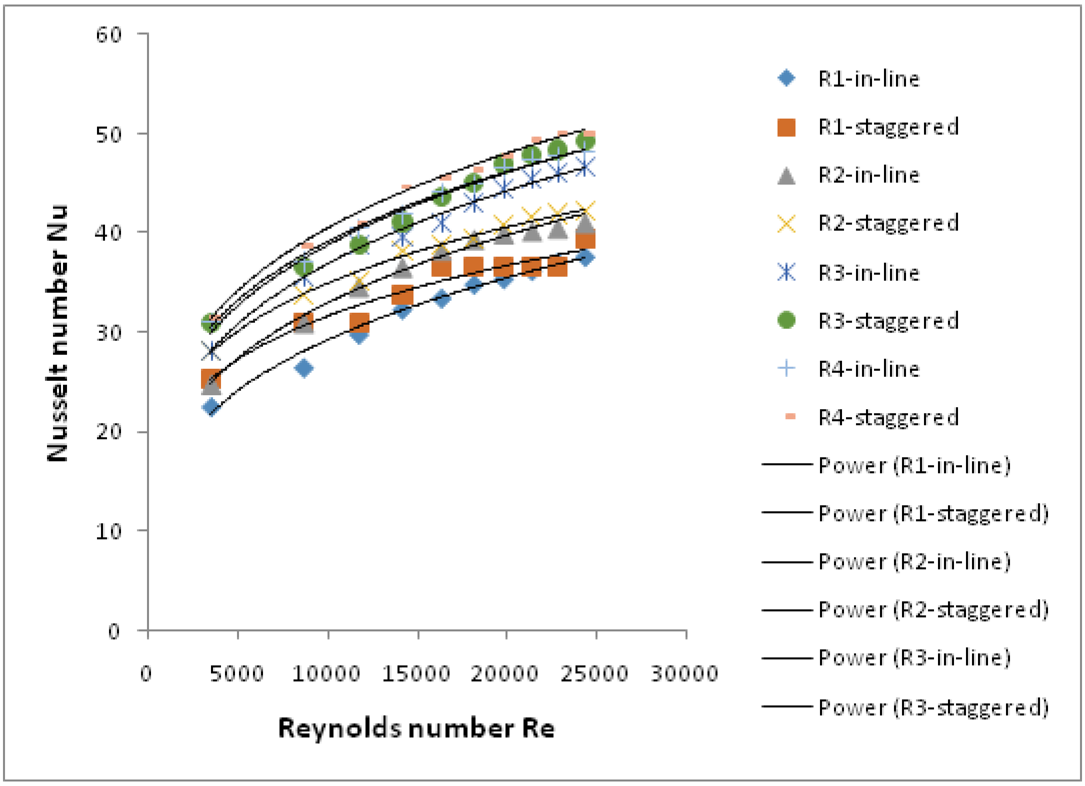


**Figure 6.** Comparison of Nu for staggered and in-line tube banks

## 5. CONCLUSION

Experimental analysis of forced convection heat transfer was performed for staggered and in-line cylindrical tube banks. The purpose of the work was to determine the local dissipation rate and fluid flow characteristics over a heated cylindrical copper tube.

The experimental data obtained showed a linear relationship between the velocity head upstream of the tube banks and the velocity head drop across the banks. The slopes of the graphs were used to calculate the drag coefficients which were 0.257 and 0.351 for the staggered and in-line tube banks configurations respectively. Also, experimental results showed that heat transfer coefficient for tubes are dependent on the position of the tubes in bank. The heat transfer coefficient decreased, at a diminishing rate from row to row. The percentage drop in the heat transfer coefficients from the first row to the fourth were 12.48%, 10.82% and 4.08% from rows one to two, two to three and three to four respectively for staggered tube configuration. Similarly, the heat transfer coefficient for the in-line tube configuration decreased in order of 12.49, 11.85 and 5.27% from rows one to two, two to three and three to four respectively.

A comparative study of two different tube banks showed that the velocity and Nusselt number for staggered tube heat exchanger were higher than that of in-tube heat exchanger in each row of tube banks. From the experimental data, 4.25% increase in Nusselt number was obtained.

The results from both configurations considered show that increasing the rows of the tube beyond five may not make any significant improvement in the performance of the heat exchanger.

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