

Experimental Investigation of the Effect of Extended Surfaces on the Performance of Tube Banks in Cross Flow

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Abstract. Experimental results of the heat transfer and pressure drop of an air flow across different configurations of extended surfaces of in-line and staggered tube bank arrangements are reported.

Empirical equations relating the heat transfer, pressure drop and flow velocity are obtained. Results indicate that extended surfaces may enhance the heat transfer, but at the same time, they will increase the pressure drop.

It is concluded that for Reynolds number (Re) $\leq 6 \times 10^3$, the simple basic configuration of a circular cylinder in the in-line arrangement is recommended since it has the highest ratio of heat transfer to pressure drop ($Nu/\Delta P^*$). For $Re > 6 \times 10^3$, the configuration with the extended surface in the downstream end (straight fin) of a staggered arrangement is recommended.

Nomenclature

A	surface area of the tube
D_p	tube diameter
h	heat transfer coefficient
k	thermal conductivity
L	length of the fin
Q	steady state rate of heat transfer
T_a	free air temperature
T_f	film temperature
\bar{T}_s	average tube temperature
t	fin thickness
V_o	velocity of air entering the tube bank
V_{max}	maximum velocity of air
ΔP	pressure drop through the tube bank

ΔP^*	non-dimensional pressure drop
Nu	Nusselt number
Re_{max}	Reynolds number, $V_{max} D_p \rho / \mu$
μ	viscosity
ρ	density

1. Introduction

Enhancement of heat transfer is of vital importance in many industrial applications [1,2,3]. In the fields of new energy development and energy conservation, it is realized that various schemes (especially on commercial basis) depend on the development of high performance heat exchangers [4,5]. That is increasing the heat transfer and reducing the pumping power, are some of the measures taken in many engineering fields and applications [6].

One of the means of enhancing heat transfer is the use of extended surfaces or fins. A drawback of this technique is that it increases the pressure drop. Therefore the two should be weighed against each other; the gain from the increased heat transfer and the penalty of increased pressure drop. The use of extended surfaces has been under extensive investigation by many workers [7 - 26].

Sparrow and Kang [7] performed heat transfer and pressure drop experiments for cross-flow tube banks in which the individual tubes were equipped with longitudinal fins. In these experiments, they investigated the effect of the fin tip geometry (blunt and contoured), the fin thickness ($t/L = 0.16$ and 0.25 where t is the thickness and L is the length of the fin), and the fin configuration or placement (at the rear, the front, and the rear and front of the tube). Experiments were also performed for increased tube diameters to yield the same surface area. Their experiment was performed for Reynolds number range from 1000 to 8600. The results indicated that a high degree of heat transfer enhancement can be obtained by finning. Comparing the results of the finned tubes to those of increased diameter unfinned tubes showed that finning yields significantly greater heat transfer enhancements for the same pressure drop. They also concluded that for the same mass flow, greater enhancements are attained with increased diameter unfinned tubes but at a high penalty in pressure drop and pumping power. However for the same pumping power, the two methods of adding surface area yield comparable enhancements with that for finning being slightly greater.

Eckels and Rabas [8] conducted an experimental study for the heat transfer and pressure drop performance of two types of finned tubes used in the air conditioning industry for 14 different bundle configurations. Twelve of the configurations contained an unconventional type of finned tubes called K-Y and the other two contained the conventional pin fin. They compared the performance of these air conditioning type finned tubes with that of air cooled heat exchanger tubes used in the process and power industries. They concluded that both the heat transferred and the

pressure drop are substantially higher with the air conditioning configurations; with the heat transfer coefficient being superior even to plain tubes (the pressure drop is about five times greater than plain tubes).

In their experimental investigation on the heat transfer and pressure drop of typical air cooler finned tubes, Eckels and Rabas [9] used helically wrapped finned tubes with an equilateral triangular pitch arrangement (one through five rows). The experiments were conducted on two finned tube types, one with a "T" foot and the other with an overlapped "L" foot. The dimensions of both finned tubes were similar. Eckels and Rabas could not detect any difference in the thermal performance of the two finned tube types. They also concluded that both the heat transfer coefficient and pressure drop increase with the number of tube rows, and that the heat transfer performance of both was greater than other air cooler tubes with similar tube geometry.

Sparrow and Vemuri [10] performed experiments to determine the combined-mode natural convection/radiation heat transfer characteristics of highly populated arrays of pin fins. In their experiments, the fins were oriented with their axes horizontal and were attached to a vertical heated baseplate. The investigation concentrated on the following parameters; number of fins in the array, the fin length and diameter, the baseplate-to-ambient temperature difference, and the presence or absence of adjacent shrouding surfaces. They concluded that finning was highly enhancing compared to an unfinned convecting-radiating baseplate (upto a sixfold increase in heat transfer) and that even the longest fins were highly efficient. They found, however, that increasing the number of fins for fixed values of the other parameters, increases the heat transfer at first, until it attains a maximum, and then decreases. Furthermore, Sparrow and Vemuri concluded that arrays having different diameter fins yielded about the same performance when the surface area of the fin-baseplate assembly was held fixed. The contribution of radiation was especially important for more populous arrays, for longer fins, and at small temperature differences.

Kroger [11] proposed a method for studying the heat transfer and pressure drop characteristics of industrial finned tubes. In his investigation, he determined the performance characteristics of a four-row bundle of closely packed, extruded, helically finned, bi-metallic tubes experimentally. Kroger presented his results in terms of the conventional colburn and friction factors and in the form of dimensional heat transfer and pressure drop parameters.

Yau *et al.* [12] investigated (experimentally) the dependence of heat transfer performance on fin spacing for condensation of steam on horizontal integral fin tubes. For comparison purposes they performed tests using a plain tube having the same inside diameter and an outside diameter equal to that at the root of the fins for the finned tubes. They concluded that heat transfer enhancement for the finned tubes significantly exceeded that from increasing the area only.

It is here emphasized that in actual applications of heat exchangers, various constraints are to be taken into account. These may be specified by some parameters such as the pumping power, the pressure drop, the flow rate, the fluid temperature and the heat exchanger size. The constraints may be put as fixed parameters, then the performance of extended surfaces may be evaluated in terms of the reduction of heat exchanger size, the reduction of pumping power, or the increase of heat exchange rate. In other words, one attempts to find a fin configuration which produces the maximum or minimum of the objective parameter under the specified condition.

This paper investigates the performance characteristics of a tube bank when some simple extended surfaces of different geometries other than those used by the previous investigations are attached to its circular tubes.

The experimental study would, therefore, determine the specific configuration which offers much less resistance to flow while being more effective in transferring heat from the standpoint of power required.

2. Experimental Setup and Procedure

Four tube banks have been tested in this investigation, each consisted of tubes with different geometries. Figure 1 shows the cross-section of sample of the tubes used and the extended surfaces attached to them.

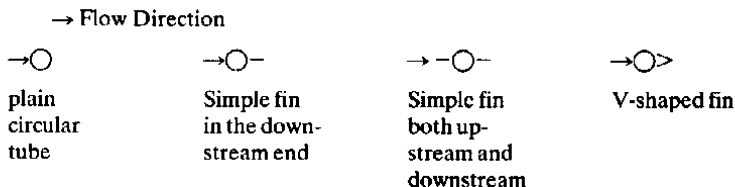


Fig. 1. Tube bank cross sections

The investigation has been carried out in two phases; in the first, the tube banks were arranged in the in-line fashion, while in the second, the staggered arrangement was used. The extended surfaces were of the same material (copper) as that of the tubes and were welded to the tubes to ensure smooth attachments.

A summary of the pertinent data for the tube banks is given in Table 1.

Electric heaters, 600 watt each, were fixed axially inside the tubes of the bank to raise their temperature to the required level by regulating the input power to the heaters. One of the central tubes was provided with three copper constantan thermocouples which were fixed at the top (at the forward stagnation point), middle (90° from stagnation point — i.e. 90° to flow direction) and bottom (at the leeward position) of the tube for temperature measurements at these locations (to account for both longitudinal and circumferential temperature variations along the tube).

Table 1. Summary of the physical data of the tube banks tested

Physical data	In-Line arrangement	Staggered arrangement
- Total number of tubes	30	27
- Inside diameter of the tube, mm	16	16
- Outside diameter of the tube, mm	18.4	18.4
- Tube length, cm	30.5	30.5
- Tube thickness, mm	1.2	1.2
- Number of rows	6	6
- Number of tubes in each row	5	5.4 alternately
- Longitudinal distance S_L , cm	4.51	4.51
- Transverse distance S_T , cm	3.0	3.0
<i>Fin Dimensions:</i>		
- Length, mm		295
- Width (simple fin), mm		20
- Width (v-shaped fin)	- two legs 25 mm each at an angle of 37.3 degrees	
- Thickness, mm		1.5
Minimum temperature, °C		30
Maximum temperature, °C		450

A test section (15 × 29 cm) containing the tube bank arrangement was connected to a wind tunnel provided with flow control, static pressure and velocity measurement facilities. The basic apparatus consists of a rectangular duct (200 × 35 cm) which is constructed in four sections clipped tightly together with snap-action fasteners and supported at four points along its length. The identical entry and exit sections (50 × 35 cm) are separated by a plain centre section which can easily be replaced with the optional heat exchangers. Inspection windows (12 × 16 cm), made from double glazed glass are provided upstream and downstream on both sides of the interchangeable centre section; for clear observation of the heat exchanger elements during test.

Air flow measurements are computed using the inlet pressure drop results taken on the standard precision, multi range, inclined manometer. The manometer inclination angle during the experiments was 18 degrees. This was accounted for in the pressure drop calculations. Four pressure taps, one in the centre of each side of the duct are combined to provide a single connection to the manometer and hence an average (static) pressure reading. The average inlet velocity to the duct is calculated from the readings of these tappings along with those from a pitot tube located upstream of the test section as shown in Fig. 2. Two pressure tappings (just before and just after the test section) are used to measure and calculate the pressure drop (ΔP). The standard

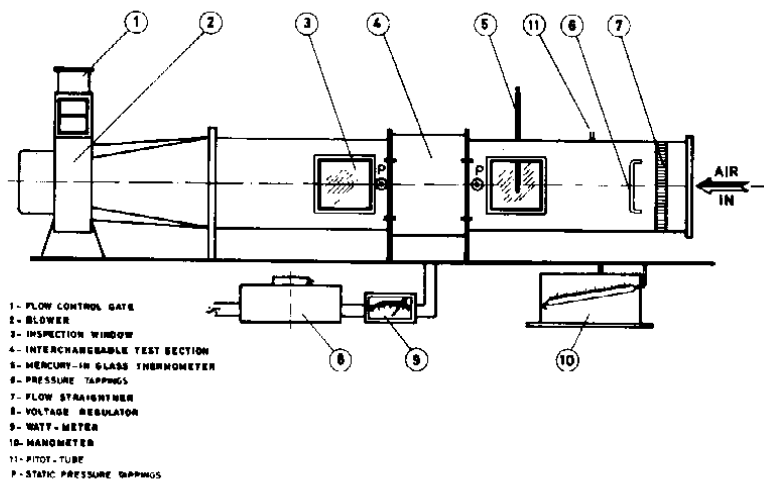


Fig. 2. Experimental setup

apparatus is provided with a centrifugal fan complete with its own electrical starter and reducing section connected to the duct exit. The fan discharges directly to the atmosphere through an adjustable throttle plate which can be used to vary the volume flow rate. (The maximum air velocity in the duct is 7 m/s). Various resistance screens, for insertion into the conical inlet are provided as an alternative means of varying the flow. Figure 2 shows the schematic diagram of the experimental setup.

3. Results and Discussion

The experimental data were used to determine Reynolds number $(Re)_{\max}$ for flow over the tubes based on the minimum free area available for fluid flow in each arrangement and the tube diameter

$$Re_{\max} = \frac{V_{\max} D_p \rho}{\mu} \quad (1)$$

The properties of air were determined based on the film temperature T_f

$$T_f = \frac{\bar{T}_s + T_a}{2} \quad (2)$$

where \bar{T}_s = the average temperature of the tube

T_a = the free air temperature at the inlet to the duct.

The average convection heat transfer coefficient (h_{av}) was calculated using Newton's law of cooling

$$Q = h A (\bar{T}_s - T_a) \quad (3)$$

where A = the surface area of the tube (including the extended surface)

Q = the steady state rate of heat transfer

It was necessary to estimate the energy loss from the electrical heaters through the walls in order to determine the actual heat transfer through the test tubes.

Heat transfer takes place in such a situation by means of two mechanisms; radiation and natural convection. In each test run the wall temperature as well as the laboratory air temperature were measured. The heat transfer from the walls to the outside air was calculated using the basic equations of natural convection and radiation heat transfer. The results showed that - in average - about 15% of the energy input to the heaters was lost through the wall of the apparatus leaving only 85% of the energy to be transferred through the test tubes.

Therefore the value of Q in equation (3) was taken as 85% of the electric power input to the heater. Nusselt number (Nu) was then calculated for each run where

$$Nu_{av} = \frac{V_{av} \cdot D_p}{k} \quad (4)$$

where k is the thermal conductivity of air based on T_f .

The pressure drop (ΔP) has been nondimensionalized by means of the following equation

$$\Delta P^* = \Delta P / \frac{1}{2} \rho V_0^2 \quad (5)$$

where V_0 = velocity of air entering the tube bank.

ρ = air density.

Figures 3 - 6 show the experimental results for the heat transfer coefficient and the pressure drop (both in dimensionless form, Nu and ΔP^* respectively) as a function of Re_{max} . These figures show a common intuitive result for both heat transfer and pressure drop, with respect to the type of tube arrangements. That is, for staggered arrangements, both heat transfer and pressure drop are higher than those for in-line arrangements. This indeed is expected, due to the contracted flow and the higher turbulence levels encountered in the staggered tube arrangements. The results as shown, so far, show the general trends of heat transfer and pressure drop and the individual values for each tube bank. Comparative analysis of all results, however, is needed to determine the advantages of each bank configuration over the

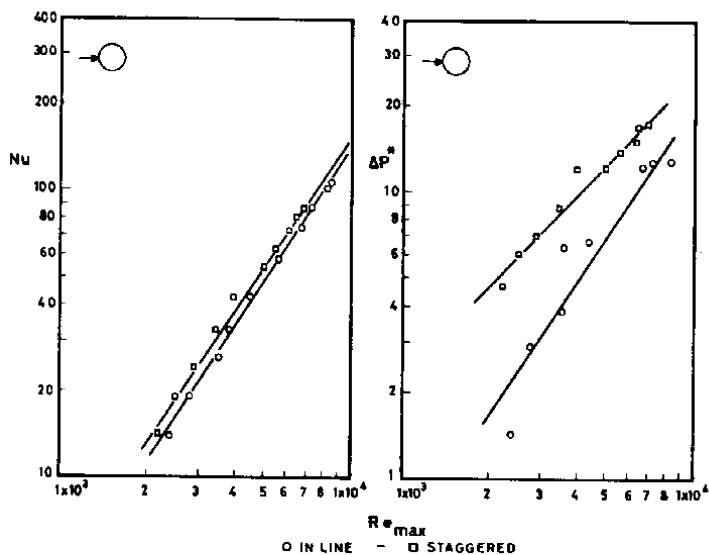


Fig. 3. Experimental heat transfer and pressure drop for circular tube

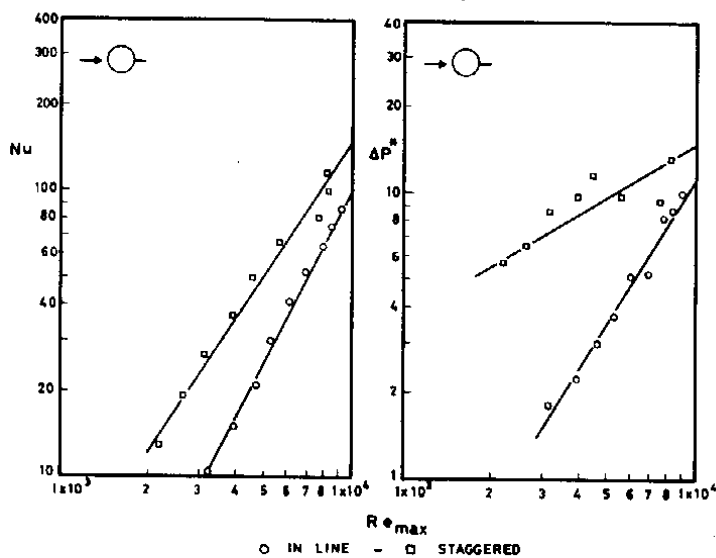


Fig. 4. Experimental heat transfer and pressure drop - one-sided fin tube

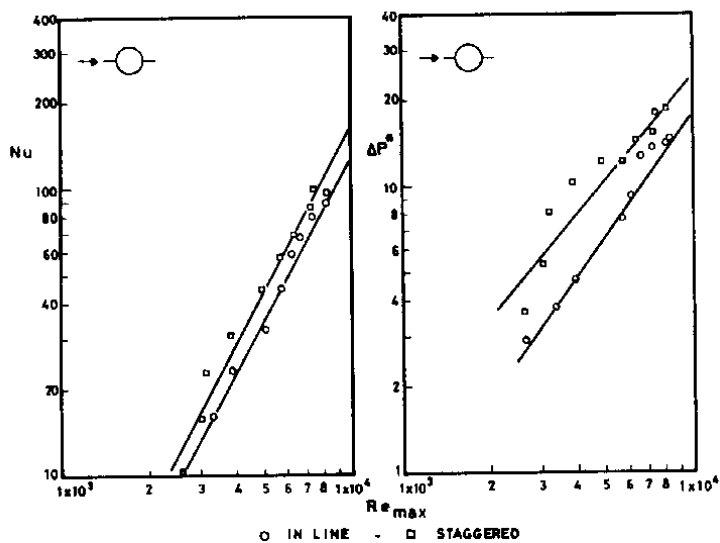


Fig. 5. Experimental heat transfer and pressure drop - two-sided fin tube

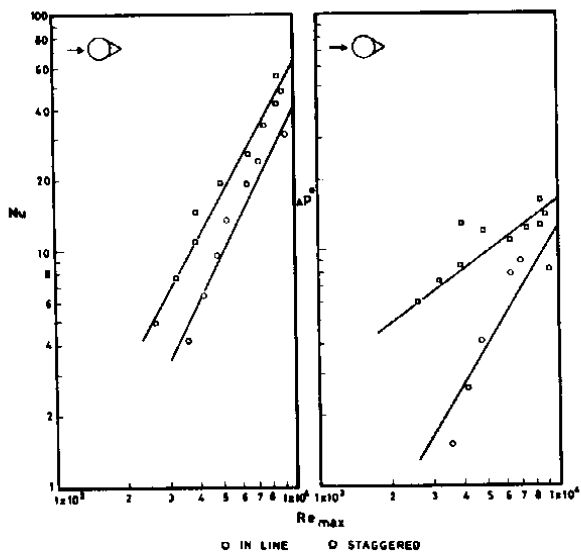


Fig. 6. Experimental heat transfer and pressure drop - V-shaped fin tube

other at different flow conditions (Re). For such a purpose the relationship between the ratio $Nu/\Delta P^*$ (rather than separate values) and Re_{max} has to be compared for all arrangements. Higher or lower value of the ratio $Nu/\Delta P^*$ indicates higher or lower heat transfer with respect to pressure loss encountered, such a parameter is considered to be a suitable one for the present investigation.

For the purpose of comparative analysis of the results, a stepwise regression technique was used to develop predictive equations for both heat transfer and pressure drop as a function of Re_{max} . The fitted $Nu/\Delta P^*$ values can then be used for comparison.

Tables 2 and 3 summarize the results of analysis for both arrangements, in-line and staggered respectively. The ratios $Nu/\Delta P^*$ were determined from the equations in Tables 2 and 3 and are plotted versus Re_{max} for the tube arrangements under investigation, the results are shown in Fig. 7.

Among the different results shown in Fig. 7, lower and higher $Nu/\Delta P^*$ ratio may be discussed. The lowest values of $Nu/\Delta P^*$ have been obtained for the V-shaped finned tubes in their in-line and staggered arrangements. The reason is that the V-shaped surface in the rear side of the tube suppresses the turbulence and damps the wakes, and heat transfer is known to be enhanced by turbulence, wakes and stirred fluids. Therefore the heat transfer is lower for the V-shaped finned tube

Fig. 7. Variation of heat transfer per unit pressure drop with Re number for different tube configurations.

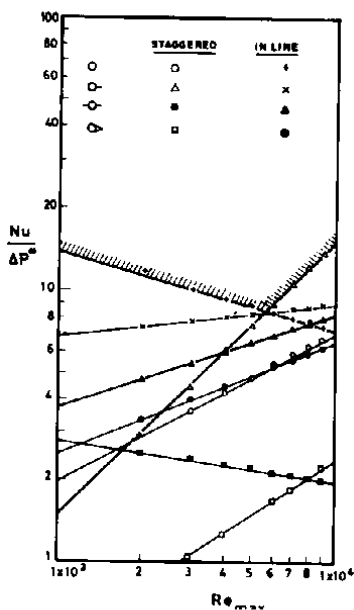


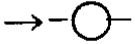



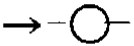



Table 2. Predictive equations for the in-line arrangement

Configuration	Nu Vs Re_{max}	Re_{max} Vs ΔP^*
	$Nu = 2.401 \times 10^{-6} Re_{max}^{1.995} Pr^{0.33}$	$Re_{max} = 2306 \Delta P^{0.441}$
	$Nu = 1.09 \times 10^{-6} Re_{max}^{2.05} Pr^{0.33}$	$Re_{max} = 2270 \Delta P^{0.517}$
	$Nu = 1.84 \times 10^{-7} Re_{max}^{2.28} Pr^{0.33}$	$Re_{max} = 1858 \Delta P^{0.562}$
	$Nu = 2.08 \times 10^{-5} Re_{max}^{1.58} Pr^{0.33}$	$Re_{max} = 1823 \Delta P^{0.58}$

→ Flow Direction

Table 3. Predictive equations for the staggered arrangement

Configuration	Nu Vs Re_{max}	Re_{max} Vs ΔP^*
	$Nu = 5.8 \times 10^{-5} Re_{max}^{1.64} Pr^{0.33}$	$Re_{max} = 507 \Delta P^{0.92}$
	$Nu = 8.4 \times 10^{-7} Re_{max}^{2.16} Pr^{0.33}$	$Re_{max} = 698 \Delta P^{0.86}$
	$Nu = 9.2 \times 10^{-7} Re_{max}^{2.1} Pr^{0.33}$	$Re_{max} = 1306 \Delta P^{0.6}$
	$Nu = 3.8 \times 10^{-6} Re_{max}^{1.8} Pr^{0.33}$	$Re_{max} = 609 \Delta P^{0.89}$

→ Flow Direction

and consequently the ratio $Nu/\Delta P^*$ is always lower than that for the simple plain circular tube.

On the other hand in-line arrangement of circular tubes has resulted in the highest values of $Nu/\Delta P^*$ at low Re_{max} . However, this ratio decreases as Re_{max} increases. Opposite results have been obtained for the one-sided straight finned tube staggered arrangement. Figure 7 shows that the curves representing the above results intersect at Re_{max} value of about 5.6×10^3 .

The results indicate that higher values of $Nu/\Delta P^*$ can be achieved in circular tube in the in-line tube bank arrangements with Re_{max} values up to 6.5×10^3 , above this value, single-sided finned tube staggered bank can achieve higher values of $Nu/\Delta P^*$. Similar results were obtained by Sparrow and Kang [7] where they found that the pressure drop in the back fin configuration is about one third greater than the unfinned tube values, and that the pressure drop for the forward fin configuration is substantially high. They also found that at the low end of the Reynolds number range the finned tube Nusselt numbers lie below those for the unfinned tube, while the opposite is true at the high end of the range. Among the finned configurations, the highest Nusselt numbers are obtained by the front finned tube bank.

4. Conclusions

Extended surfaces may have appreciable effects on the performance of tube banks when attached to their circular tubes. The coefficient of heat transfer and the pressure drop across the bank are the two main factors to be considered in this kind of investigations.

Straight and V-shaped fins have been fixed to the rear stagnation point of circular tubes and their effects on heat transfer: pressure drop ratio were tested.

The in-line plain circular tube bank has offered the highest heat transferred per unit pressure drop ($Nu/\Delta P^*$) at lower Re_{max} , however, continuous decrease in $Nu/\Delta P^*$ has been observed as Re_{max} increases. For the bank of staggered single-sided finned tubes, on the other hand the heat transfer per unit pressure drop is relatively low at low values of Re_{max} and increases rapidly as Re_{max} increases. At a Re_{max} value of 5.6×10^3 the heat transfer per unit pressure drop for both banks of in-line circular and staggered single-sided finned tubes are equal. With further increase in Re , the bank of single-sided finned tubes showed superiority over the bank of circular tubes.

The banks of in-line and staggered double-sided finned tubes are seen to have close values of heat transfer per unit pressure drop at all values of Re . (with those for the in-line arrangement being always higher by about 30%). The trend showed continuous increase of $Nu/\Delta P^*$ with Re .

The lowest values of heat transfer per unit pressure drop which have been observed among the different arrangements are for the in-line and staggered bank of V-shaped finned tubes. However both have shown opposite trends, that is, for in-line arrangement $Nu/\Delta P^*$ decreases with Re , while for the staggered arrangement the heat transfer per unit pressure drop increases with Re .

It may be concluded that for tube banks designed to work at higher Re_{max} , single sided straight fins (fixed at the back of the tubes - leeward) will be advantageous. Fortunately from manufacturing point of view, such a fin can be fixed at relatively low costs.

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دراسة عملية (تجريبية) لتأثير الزعانف على أداء مصفوفة أنابيب في تيار متعامد

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ملخص البحث: في هذا البحث تمت دراسة عملية لتأثير أشكال وترتيبات مختلفة لمصفوفات أنابيب ذات زعانف مختلفة موجودة في تيار متعامد من الهواء على كل من معدل انتقال الحرارة ومعدل تغير (تناقص) الضغط.

وتم الحصول على معادلات تجريبية تربط معامل انتقال الحرارة ومعامل تناقص الضغط مع سرعة السريان - وتؤكد النتائج التي تم الحصول عليها أن الزعانف تزيد معدل انتقال الحرارة ولكنها في الوقت نفسه تزيد معدل انخفاض الضغط.

وتم استنتاج ما يلي: وهو أنه لأرقام رينولد تساوي أو تقل عن 3×10^3 فإن التركيب المبسط للأنابيب النحاسية (بدون زعانف) والمرتبطة في صفوف متتالية هو أحسن التركيبات من حيث انتقال الحرارة والضغط لأنه يعطي أكبر قيمة للنسبة بين انتقال الحرارة والضغط. أما لأرقام رينولد التي تزيد على 3×10^3 فإن الشكل ذو الزعانف المستقيمة والمركبة في الجهة الخلفية لاتجاه التيار في التركيب المتخالف هو المفضل.