Proceedings of the ASME 2010 3rd Joint US-European Fluids Engineering Summer Meeting and
8th International Conference on Nanochannels, Microchannels, and Minichannels
FEDSM-ICNMM2010
August 1-5, 2010, Montreal, Canada

FEDSM-ICNMM2010-31236

TUBE BANKS HEAT TRANSFER ENHANCEMENT USING CONICAL FINS

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ABSTRACT

In this work the heat transfer and pressure drop experimental results obtained in a two step finned tube bank with conical fins are presented. The tube bank had an equilateral triangle array composed of nine finned tubes with conical fins inclined 45 degrees in respect with the tube axis. The heat exchange external area of a single tube is approximately 0.07 m^2. All necessary thermal parameters, inlet/outlet temperatures, mass flows, for the heat balance in the tube bank were determined for different air velocities, $Re = 3400 – 18400$, and one constant thermal charge provided by a hot water flow with a temperature of 80 °C. As a result, the correlations for the heat transfer and pressure drop calculation were obtained. The experimental results were compared against the analytical results for a tube bank with annular fins with the same heat exchange area. It was found that the proposed tube bank using finned tubes with conical fins shows an increment of heat transfer up to 58%.

Keywords: fins, heat transfer, pressure drop, finned pipes, conical fins.

INTRODUCTION

Given the constant shortage of water and a growing sense of protecting the environment, currently the industrial cooling systems use air instead of water as coolant. As a result, there is an increased demand for high efficiency air-cooled heat exchangers designed to handle and transfer heat to large amounts of air. Air-cooled heat exchangers are used in industries such as refineries, chemical and food industries and in combined cycle power stations [1]. This kind of heat exchangers consists of one or more horizontal or inclined rows of tubes forming a section through which air is forced by a fan.

Because of its low heat capacity, when air is using as a coolant this leads to a low external convective heat transfer coefficient [2]. Thus, to dissipate the same quantity of heat it is needed a larger area of the surface, so the size of the heat exchanger increases. Using banks of finned tubes is the best way to provide a larger area without increasing the overall size of the heat exchanger. The finned tubes bank heat transfer and pressure drop calculation method is well-known [3].

Increasing the heat dissipation capacity of an air-cooled heat exchanger can be done by augmenting the convective coefficient values. Thus, it is necessary to destroy or break the boundary layer on the fin, disrupting the flow across the surface and creating a turbulent flow with macrovortices generation. These flow conditions can be achieved using new profiles of fins, as several authors have investigated [4, 5, 6]. Among other works, in the paper [7] a new fin profile is proposed, an annular fin with its edges partially bended creating a convergent channel, where the flow accelerates to the rear part of it. Authors found that heat transfer enhancement in a tube bank with a triangular array can be up to 47%.

Another proposal, which also leads to heat transfer enhancement, is to tilt annular fins at an angle so as to form cones [8, 9]. In these works the local thermal and aerodynamic characteristics of heat exchange surfaces with conical fins were studied. However, these local characteristics cannot be used to calculate the heat transfer in a tube bank.

Thus, the motivation of this work is to find the correlations to calculate heat transfer and pressure drop of banks of tubes with conical fins. An extended surface composed of two steps of finned tubes with conical fins, inclined 45 degrees in respect with tube axis, and in equilateral triangle tube array, was experimentally studied.

NOMENCLATURE

A	Heat exchange external area, m
d	Tube diameter, m
e	Fin thickness, m
Eu	Euler number
H	Fin height, m
h	Convective coefficient, W/m^2 K
k	Thermal conductivity, W/m K
L	Tube length, m
Nu	Nusselt number
Re	Reynolds number
S_1	Transversal tube pitch, m
S_2	Parallel tube pitch, m
u	Air flow velocity, m/s
ν	Air kinematical viscosity, m^2/s
ρ	Air density, kg/m^3

Subscripts

ext	external
int	internal

EXPERIMENTAL INVESTIGATION

Experimental installation

The experimental investigation was carried out in a wind tunnel that has a DC electric motor of 15 HP, which drives a centrifugal fan to provide air to the test zone. The test zone was located in a section that was added at the end of a stainless steel chamber, 1.78 m in diameter and 2.70 m in length, which inside has a perforated plate for air flow conditioning porpoises. The shape and dimensions of the added section were calculated to provide all the conditions to investigate heat transfer and pressure drop on the tube bank (Fig. 1).

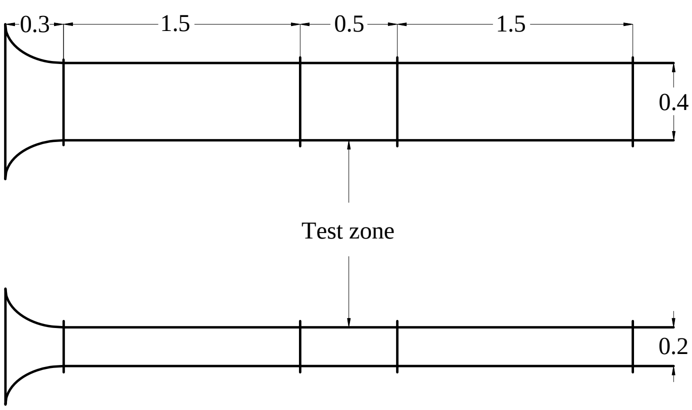

Fig. 1. Test zone.

The study was conducted on a bank of tubes that consists of two steps with a total of 9 finned tubes with conical fins. Five tubes were installed in the first row and 4 tubes in the second one in an equilateral triangle array, S_1 = 0.08 m and S_2 = 0.069 m, that is typical in extended surfaces. The array of the tubes and the tube-pitch dimensions are shown in Figure 2.

As it can be seen in Fig. 2, in the second row, which corresponds to the second step of the tube bank, a half tube was installed on the side walls of the test zone. This half tube was made of acrylic and has the function to simulate a fifth tube, so that the air flowing near the wall interacts with the tubes in the second row and not flow freely. The idea is to avoid disrupting the flow dynamics within the limits of the tubes array.

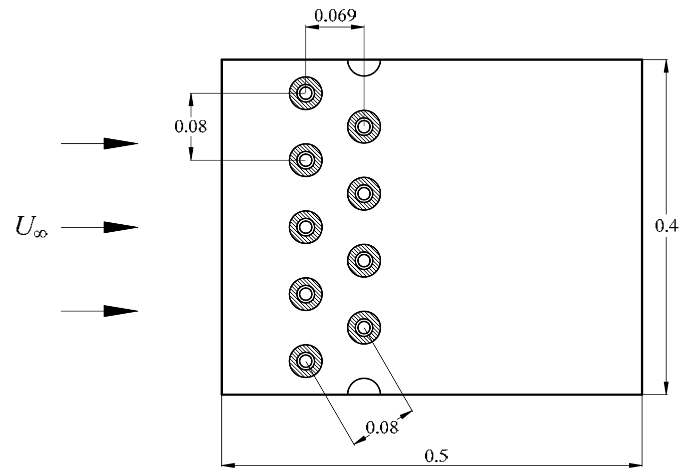

Fig. 2. Top view of the test zone.

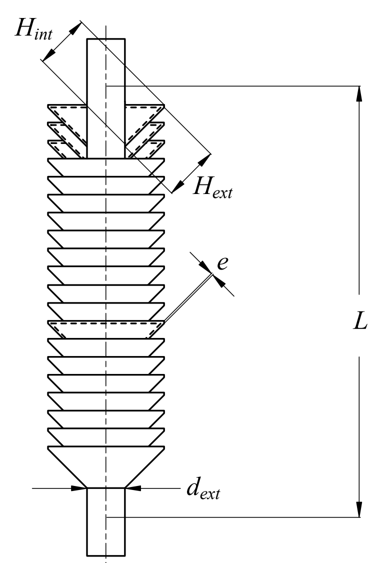

Fig. 3. Finned tube with conical fins.

The finned tubes were turned from aluminum bar, so there is not thermal contact resistance between fins and tube. Each tube has 30 conical fins inclined at an angle of 45° in respect with the axis of the tube. The geometrical characteristics of fins are: internal fin height $H_{int} = 0.008$ m, external fin height $H_{ext} = 0.015$ m, and fin thickness $e = 0.0015$ m. The tube has an external diameter $d_{ext} = 0.022$ m, an internal diameter $d_{int} = 0.015$ m and a length $L = 0.2$ m. The heat exchange external area, A, of a single tube is approximately 0.07 m^2. Figure 3 shows the main geometrical parameters of a finned tube with conical fins.

In order to provide a uniform distribution of water flow within the fin tubes, it was necessary to install a header with an internal area greater than the sum of the internal areas of the fin tubes as recommended in [10]. After performing the required calculations it was chosen a tube 3 inches in diameter to make the entry, exit and return headers. The headers were attached to the tubes with conical fins through hoses resistant to high temperatures. To reduce the heat loses, the headers were insulated. Figure 4 shows the assembled test zone with the tested tube bank, headers and connections.

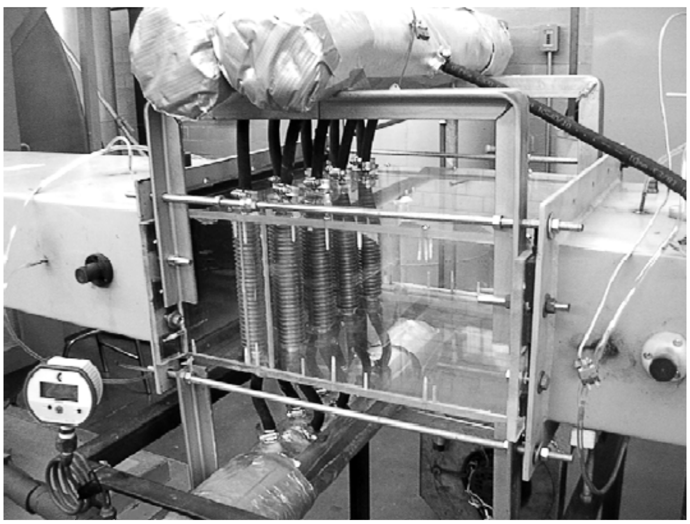

Fig. 4. Test zone assembled with all its elements.

The hydraulic circuit comprises a water circulation system consisting of a centrifugal pump ½ HP, connected to a system of tubes and valves to regulate flow of water entering the tested tube bank. To heat water it was used a boiler which has a set of electrical resistors of 13 kW of power. The boiler raises the fluid temperature up to 90 °C.

The hydraulic system has a vertical flowmeter for measuring the water flow rate in a range of 2 to 20 liters per minute (lpm). The experimental tests were conducted for a flow rate of 14 lpm ($23.34 \cdot 10^{-5}$ m^3/s) of hot water at 80 °C.

Four K type thermocouples (±0.1 °C precision) were installed to measure the temperature of air and water at the entrance and exit of each fluid of the heat exchange surface. As a result, it was possible to measure the temperature difference ΔT of each flow.

The thermocouples were connected to a data acquisition system with support from the Link Scan 2.0 package that displays temperature values in an interval of 4 seconds for each thermocouple.

To calculate the flow velocity two Airflow Pitot tubes, 300 mm long and 4 mm in diameter, were used in connection with an inclined Airflow manometer, which has a measuring range of 0 to 5 kPa depending of its inclination.

To determine the pressure drop caused by the tested tube bank, static pressure was measured using an Airflow inclined manometer connected to perforated static pressure taps on the four walls of the wind tunnel before and after the tube bank.

Experimental data reduction

The experimental results are presented as relations between the dimensionless numbers of Nusselt, Nu, Euler, Eu and Reynolds, Re, whose expressions used in this work are:

$$Nu = \frac{h \times d}{k} \qquad (1)$$

$$Eu = \frac{\Delta p}{\rho \times u^2} \qquad (2)$$

$$Re = \frac{u \times d}{\nu} \qquad (3)$$

In these expressions the outer diameter d was taken as the characteristic length of the finned tube. The hot water and air properties were taken in relation to their average temperature within the limits of the tube bank. The calculation of Nusselt number, Nu, was conducted under condition of constant heat flux. The Euler number, Eu, was calculated considering the air density variations. The range of dimensionless flow velocities, Reynolds numbers, studied is $Re = 3.4 \cdot 10^3$ to $Re = 18.4 \cdot 10^3$, which are representative operational flow velocities of a variety of air-cooled heat exchangers.

Four tests were performed for each flow velocity value to check the variation of obtained results; the experimental data strip deviation was ±8%. The results for Nu and Eu numbers depending of the Re for all four test performed are shown in Annex A. The error in measuring the different parameters was estimated using the recommendations of [11] and not in all cases exceeded 12%.

DISCUSSION

Heat Transfer

Figure 5 shows the relationship between heat transfer and the dimensionless flow velocity, obtained from experimental data for the heat exchange surface of conical finned tubes in an equilateral triangle array.

For comparison are also shown in Figure 5 the theoretical results of the calculated average heat transfer surface area for equivalent conventional annular finned tubes using the methodology reported in [3].

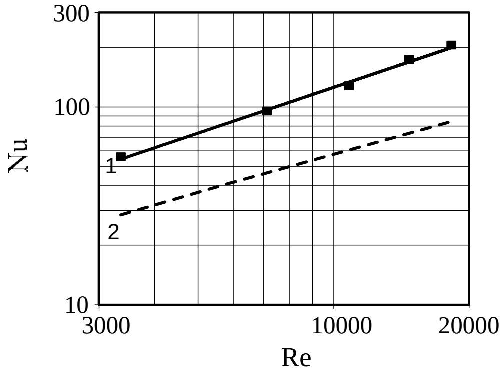

Fig. 5. Performance of dimensionless heat transfer versus dimensionless flow velocity. 1. Experimental data; 2. Theoretical calculated using the methodology of [3].

The theoretical calculations were performed considering the geometrical parameters of annular finned tubes and the heat transfer and flow conditions under which they were exposed as the same as for the conical finned tubes and the experiment. The only difference is that in one case the fins are annular and in the other are inclined at an angle of 45°.

The plot in Figure 5 shows that the behavior of heat transfer corresponds to the expected results, i.e., with increasing speed also increases the convective coefficient. It is also noted that the heat transfer of the tube bank with conical fins at $Re = 3.4 \cdot 10^3$ has a 48% increase, while at $Re = 18.4 \cdot 10^3$ the increment in heat transfer is 58% compared with the calculated results for annular fins.

In the studied range of Reynolds number, $Re = 3.4 \cdot 10^3$ to $Re = 18.4 \cdot 10^3$ the line of approach of the experimental results also is shown in Fig. 5. From this approaching line was obtained the empirical correlation for calculating heat transfer of a bank of tubes with conical fins, in an equilateral triangle array:

$$Nu = 0.0745\ Re^{0.8} \qquad (4)$$

Pressure drop

Figure 6 shows the behavior of the dimensionless pressure drop versus the dimensionless flow velocity. Because the measured pressure at the first flow velocity value was very small and the sensitivity of the instrument was not properly allowed to obtain this reading, a bigger value of the Euler number was obtained in the first point. Therefore, it was made the decision to disregard it.

The correlation for calculating pressure drop of a bank of tubes with conical fins in an equilateral triangle array in the range of $Re = 3.4 \cdot 10^3$ to $Re = 18.4 \cdot 10^3$, which is:

$$Eu = 2.505\ Re^{-0.152} \qquad (5)$$

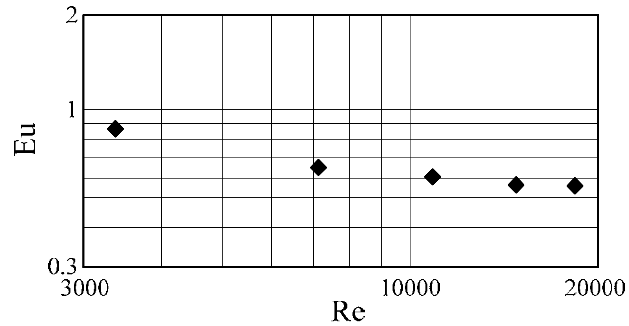

Fig. 6. Behavior of the dimensionless pressure drop versus dimensionless flow velocity.

Tube bank efficiency

Finally, figure 7 shows the plot of the efficiency of the tube bank with conical fins, which can be determined from joint analysis of heat transfer, external convection coefficient *h*, against pressure drop Δp caused by the tube bank.

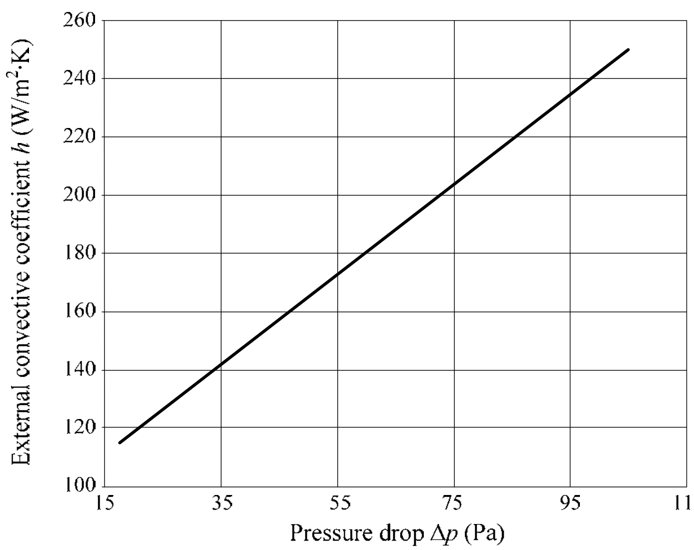

Fig. 7. Efficiency of the bank of tubes with conical fins.

From figure 7 it is possible to determine what will be the pressure drop caused by the tube bank for a given convective heat transfer external coefficient. This plot is valid for the conditions under which was performed the experimental investigation and the geometrical characteristics of the elements studied in this work.

CONCLUSIONS

In the range of $Re = 3.4 \cdot 10^3$ to $Re = 18.4 \cdot 10^3$ a bank of tubes with conical fins in an equilateral triangle array was characterized experimentally.

The experimental results of the thermal behavior of a bank of tubes with conical fins showed an increase from 48% to 58% in heat transfer, compared with the theoretical results of a tube bank with annular finned tubes. Based on the experimental results, the empirical correlation for calculating heat transfer of a bank of tubes with conical fins with an equilateral triangle array was obtained.

After comparison of experimental and theoretical results it is concluded that the surface of conical finned tubes can be applied in air-cooled exchangers because it has better thermal performance than the annular finned surface. However, the increment in pressure drop in the bank of tubes with conical fins has to be taken in account because could be necessary in some cases to change the fans of the air-cooled heat exchanger.

ACKNOWLEDGMENTS

The authors wish to express their thanks to Dr. Pysmennyy, National Technical University of Ukraine, for his useful advice for paper final structure. Thanks are also given to the National Polytechnic Institute of Mexico for their support of this work.

REFERENCES

[1] Kraus A. D., Aziz A. y J. Welty, 2001 “Extended Surface Heat Transfer”, John Wiley & Sons, Inc..

[2] Rohsenow W. M., Hartnett J. P., Cho Y. I., 1998 “Handbook of Heat Transfer”, MacGraw-Hill Companies, Inc..

[3] Pysmennyy Ye., Polupan G., Carvajal Mariscal I., Sánchez Silva F., 2007 “Heat Exchangers and Finned Tube Banks Calculation Handbook” (in Spanish), Ed. Reverté.

[4] Gentry M., Jacobi A., 2002 “Heat Transfer Enhancement by Delta-Wing-Generated Tip Vortices in Flat-Plate and Developing Channel Flows”, Journal of Fluids Engineering, **124**(6), pp 1158-1168.

[5] Torri K., Kwak K., Nishino K., 2002 “Heat Transfer Enhancement Accompanying Pressure-Loss Reduction With Winglet-Type Vortex Generators for Fin-Tube Heat Exchangers”, International Journal of Heat and Mass Transfer, **45**(18), pp. 3795-3801.

[6] O’Brien J., Sohal M., 2005 “Heat Transfer Enhancement for Finned-Tube Heat Exchangers with Winglets”, Journal of Fluids Engineering, **127**(2), pp. 171-178.

[7] Pysmennyy Ye., Terekh A., Polupan G., Sánchez Silva F., 2002 “Thermoaerodynamic Efficiency of Heat Exchangers Created as Tube Banks with Converging Bent Fins”, Proc. International Symposium on Compact Heat Exchangers, G. P. Celata et al., eds., ETS Pisa, Grenoble, France, pp. 323-326.

[8] Carvajal Mariscal I., Sánchez Silva F., P. Quinto Diez, 2001 “Flow Dynamics between the Inclined Fins of a Finned Tube”, International Journal of Heat and Fluid Flow, **22**(5), pp. 519-524.

[9] Carvajal Mariscal I., Sánchez Silva F., Toledo Velazquez M., Pronin V.A., 2001 “Experimental Study on the Local Convective Coefficient Distribution on a Pipe Surface with Inclined Fins”, Experimental Thermal and Fluid Science, **25**(5), pp 293-299.

[10] Lokshin V.A., Peterson D.F., Schwarz A.L., 1988 “Standard Methods of Hydraulic Design for Power Boilers”, Hemisphere Publishing Corporation.

[11] Taylor J.R., 1997 “An Introduction to Error Analysis”, University Science Books.

ANNEX A

EXPERIMENTAL RESULTS

Re	TEST 1		TEST 2	
	Eu	**Nu**	**Eu**	**Nu**
3371	0.9426	56.38	0.9363	54.65
7132	0.6412	91.71	0.6769	101.51
10866	0.5953	126.67	0.6261	133.08
14784	0.58	174.15	0.5928	170.27
18373	0.5733	204.13	0.5739	202.72

Re	TEST 3		TEST 4	
	Eu	**Nu**	**Eu**	**Nu**
3371	0.9373	59.37	0.7811	52.78
7132	0.6769	98.94	0.6412	88.01
10866	0.6215	130.00	0.6012	119.48
14784	0.5844	189.23	0.5625	157.41
18373	0.5690	226.54	0.57	185.59