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AN EXPERIMENTAL STUDY AIR FLOW AND HEAT TRANSFER OF AIR OVER IN-LINE FLAT TUBEBANK

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ABSTRACT

An experimental study has been made to investigate heat transfer and air flow around flat tube of in-line flat tube banks with laminar forced convection. Measurements were conducted for sixteen tubes in the flow direction;four tubes in the rows, the three air velocity (0.6, 0.8 and 1.0 ms^{-1}) and Reynolds number are Re_{Dh} = 527, 703 and 880, where D_h isthe hydraulic diameter of the tube. The total heat flux supplies in all tubes are 967.92, 2258.48 and 3629.70 Wm^{-2}, respectively. The study results indicate that the average Nusselt number of all flat tubes has increased 23.7%-36.7% with Reynolds numbers varying from 527 to 880 at the fixed heat flux; also averageNusselt number has increased11.78% - 23.75% at the heat flux are varying 967.92, 2258.48 and 3629.70 Wm^{-2}, respectively at the Reynolds number Re = 703. In addition, the pressure drop decreaseswith the increase of Reynolds number. The Nusselt number-Reynolds number correlation was found to be $\overline{Nu} = C_1 \times Re^{C_2}$, the correlation yielded good predictions of the measured data with the mean error R^2 = 0.992.

Keyword: Experimental study; Forced convection; In-line flat tube; Measurement.

INTRODUCTION AND PREVIOUS WORKS

The heat transfer and fluid flow in tube bundles the represent an idealization of many industrially important processes. Tube banks are openly employed in cross-flow heat exchangers, the designstill relies on empirical correlations of pressure drop and heat transfer. Heat exchangers with tube bundles in cross-flow are of a major operation interest in manychemicals and thermal engineering processes (Buyruk et al., 1998; Incropera et al., 2007; Mandhani et al., 2002; Liang and Papadakis, 2007; Kaptan et al., 2008; Wang et al., 2000; Žukauskas, 1972). Flat tubes, despite, have not been achieved to the same extent, through the fact that they play a significant role in manytechnical applications, such as automotive radiators and modernheat exchangers. It is designs have recently been provided use air conditioning for automotive evaporators and condensers.The recent developments in automotive aluminium manufacturing technology have made the cost of the flat tube heat exchanger building more propitious (Webb and Kim, 1993). In addition,the flat tube heat exchangers are expected to be the best air-side heat transfer coefficients and minimum air-side pressure drop compared with circular tube heat exchangers, the pressure drop in flat tube is expected to be less of circular tubes, due to a smaller wakearea. For the same reason, noise and vibration are expected they are less in flat tube heat exchangers compared to circular tube heat exchangers. Ay et al.(2002) presented an experimental studyinplate finned-tube heat exchangers. The test result on the strategy zone of both a staggered and in-line configuration, the results show that the averaged heat transfer coefficient of staggered arrangement is 14 - 32% greater than that of the in-lined arrangement. The experimental and numerical study of the laminar heat transfer and fluid flow over the cylinder in cross-flow presented by Buyruk et al. (1998), the variation of local pressure drop, Nusselt number, streamline contour and isotherm line contour are expected with two Reynolds number are 120 and 390. In addition, the results show increasing Reynolds number leads to a state of separation to move upstream and overall heat transfer to increase. Heat transfer and fluid flow over a fourth-row of elliptic finned-tube heat exchanger is studied numerically and experimentally from Jang and Yang (1998). The change values of inlet velocity ranging 2 - 7 m s^{-1}. The two arrangements in-line and staggered of elliptic finned-tube and one circular finned tube with staggered configuration. The experimental results show that the average heat transfer coefficient incensed 35 - 50% at an elliptical finned tube with identical circular finned tube. The pressure drop in elliptic finned-tube bundle is only 25 - 30% of the circular finned-tube bundlearray. An experimentalstudy was carried out to investigate heat transfer and flow characteristics from one tube within a staggered tube bundle and a row of similar tubes. A variation of a local Nusselt number and local pressure coefficients were shown with different blockages and Reynolds numbers (Matos et al., 2004).This is an experimental, numerical and analytical study of the optimal spacing between cylinders in cross-flow forced convection. The first part, experimental Re_D range of 50 - 4000 and the second part, similar results are developed based on numerical simulations for $Pr = 0.72$ and $40 \leq Re_D \leq 200$. The experimental and numerical results for optimal spacing and maximum thermal conductance are explained and correlatedanalytically by intersecting the small-spacing and large-spacing asymptotes of the thermal conductance function (Stanescuet al., 1996).

Tahseen et al. (2012a, 2013b) have numerical studies incompressible, steady state flow and using the body fitted coordinate (BFC). The first study heat transfer over a series of the flat tube between two parallel plates and second study the heat transfer over in-line bank of a circular tube.The two studies show the effect of the Reynolds number on the Nusselt number also, the Nusselt number increase with an increase of Reynolds number always. The flow over elliptic cylinders banks presented from Yianneskis (2001) both numerically and experimentally. The average velocities for array fined experimentally by flow visualization and using a laser Doppler anemometer (LDA), the results show the arrangement. The result shows that the arrangement generates much lower than turbulence levels than an equivalent array with circular cylinders. Turbulence levels to continue nearly constant along the flow sections across successive rows, referring to the absence of confusion between adjacent columns.

In the present study, the cooling process was experimentally examined. An array of in-line flat tubes of diameter ratio 1.85. The external flow of air Reynolds number, based on the mean freestream air velocity and outer hydraulic of the tube, varied from 527 to 880, also the heat flux supply to all tubes between the range 967.92W m^{-2}-3629.70W m^{-2} to detect the effects of Reynolds number on the air pressure drop across the flat tube banks and heat transfer rate.

EXPERIMENTAL DETAILS

The flattube arrangement was made from Aluminum with the small diameter 10mm and bigger diameter 18.5mm, with the thickness of tube 1mm and the hydraulic diameter D_h = 13.5mm all tubes had the length of 200mm. The double electric heaters were inserted the inside of the tubes to simulate the heat flux originated from a hot fluid. All the arrangements four rows in the direction of external flow over all tubes, as shown in Figure 1. Sixteen flat tubes were then assembled according to the design presented in Figure 1, in a drawer from Teflon type Polytetrafluoroethylene (PTFE), which is the test module. Losses minimized by holding the end of the flat tube between two Teflon walls separated by the distance L = 200mm.

Thirty two heating elements, the consisted of cylindrical electric heaters each rated at50W up to 850W with 220V AC, the electric heaters outside diameter 8 mm and a length of 200mm. A diameter small enough to be inserted in the aluminum tubes, the heaters were connected in parallel and two AC voltage source variable that produced voltages range 0 to 125 V, the maximum power supply 220V and a maximum current of 2.5A, the model LOADSTAR 850. The current and voltage measurements were performed with a current clam meter model U1191A and voltmeasurement meter model TENMA9272, respectively.Fifteen thermistors of type EPCOS B57164K0102J NTC (resistance 1000Ω at 25^{o}C) were placed in the test module. All the thermistors were placed in the middle between the side walls of the wind tunnel and on the midline of the elemental channels. Four Thermistors were placed at the arrangementinlet ($T_{in,1}$ - $T_{in,4}$),four at the surfaces of tubes ($T_{s,1}$ – $T_{s,4}$), five at the outlet ($T_{out,1}$ - $T_{out,5}$) in one elementalchannel. An additional thermistor (T_{bef}) was placed on the extended region at 400mm before the test module to measure the temperature of free stream.

Thermistors in surfaces of tube showed that the difference between the tubes in one elementchannel is negligible, and are within ±0.57°C margin with respect to average four thermistors. Finally, the thermistor put in the extended region for purpose measured the free stream temperatures within ±0.22°C margin with respect to the average temperature measured inlet arrangement, in all the tests carried out in this work.The velocity measurements were taken with a vane type hot wire anemometer; model YK-2004AH, which was placed in the extended flow region, as shown in Figure 1, for the range of 0.2m s^{-1} - 20ms^{-1}. The resolution 0.1 m $s^{-1}$of readingand uncertainty in freesteam velocity U_∞ was ±(1%-5% +0.1 m s^{-1}). The velocity of free streamU_∞ was varied 0.6ms^{-1}-1.0 ms^{-1} in this study. The pressure drop measurements were taken with a differential pressure meter; model TESTO 510 (TESTO, Inc.), the nominal range of the (0-100)kPa, the differential pressure resolution 1Pa of reading and the accuracy ±0.3 Pa. The differential pressure measurements had the finality of measuring the pressure drop across each of change the free stream velocity in all experiments as shown in Figure 1.The experimental work includes the attainment of temperature data using Testo highly accurate thermometer, model TESTO 110 (TESTO, Inc.), the nominal range of (-50°C to +150°C), the temperature thermometer resolution 0.1°C of reading and the accuracy, ±0.2°C. Also, the thermistor was calibrated in a laboratory to find the deviation limits. The thermistor was immersed in the four types of material distilled water, n-Hexane, Metghylated Spirit and Toluene. After that, each material is heated separately to reach the boiling point and then record the values as shown in Figure 2 (Collett and Hope, 1983).

Started each run by selecting the voltage and current for the cartridge heaters and air velocity of free stream, then we waited for (2.5 - 3.0) hour while monitor the changes in voltage, current, T_{bef}, $T_{in,1}$ - $T_{in,4}$, $T_{s,1}$ - $T_{s,4}$ and $T_{out,1}$ - $T_{out,5}$.We had taken final readings when the relative changes in the voltage, current and temperature were less than 0.5%-0.8%, 2%-2.2% and 0.044%-0.075%, respectively.

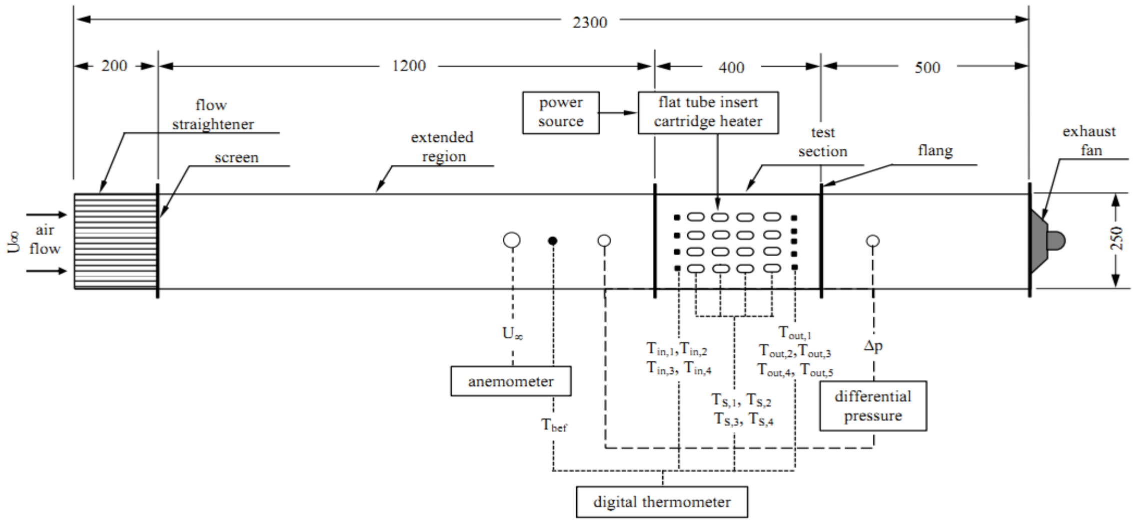

Figure 1. The schematic display of the experimental approach.

These changes were estimated relative by repeating the same value Re_{Dh} value, between (7.5 - 9) hours. It should be noted that these relative changes are small when compared with the uncertainties in the relevant measurements.

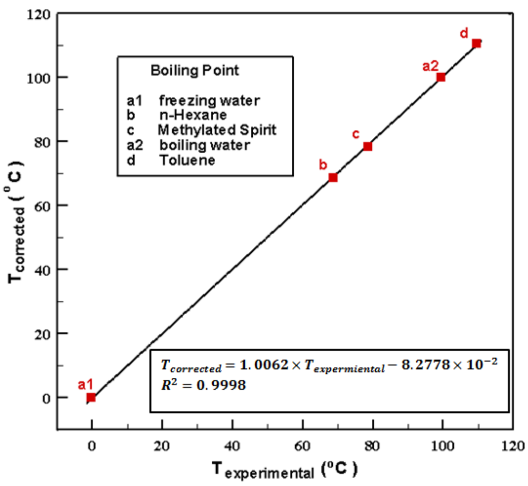

Figure 2. Calibration of the thermistor.

Data Collection

In this experiment assumed steady state flow. To investigate and used the following relations of the relevant properties of air at the calculations follow, they are based on data (Rogers and Mayhew, 1995) and valid for the range of temperature. $250K \le (\bar{T}_{in} + \bar{T}_{out})/2 \le 400K$:

$$\rho = 1.05269 + 4.895\times10^{-4}\left(\frac{\overline{T}_{in}+\overline{T}_{out}}{2}\right) \quad (1a)$$

$$k = \left[3.71452 + 7.495\times10^{-2}\left(\frac{\overline{T}_{in}+\overline{T}_{out}}{2}\right)\right]\times10^{-3} \quad (1b)$$

$$\mu = \left[4.99343 + 4.483\times10^{-2}\left(\frac{\overline{T}_{in}+\overline{T}_{out}}{2}\right)\right]\times10^{-6} \quad (1c)$$

$$Cp = \left[9.8185 + 7.7\times10^{-4}\left(\frac{\overline{T}_{in}+\overline{T}_{out}}{2}\right)\right]\times10^{2} \quad (1d)$$

Where $\overline{T}_{in} = (T_{in,1}+T_{in,2}+T_{in,3}+T_{in,4})/4$ and $\overline{T}_{out} = (T_{out,1}+T_{out,2}+T_{out,3}+T_{out,4}+T_{out,5})/5$
The electrical heat gain rate was calculated by:

$$Q_{elect} = V \times I \quad (2)$$

The steady state heat balance of the electrically heat test surface can be written as:

$$Q_{elect} = Q_{conduction} + Q_{radiation} + Q_{convection} \quad (3)$$

Heat transfer from the system may be (i) conduction between lab and wall of the tubes was neglected because of the extremely low thermal conductivity of air (0.23 $Wm^{-1o}C^{-1}$) of Teflon andnegligible temperature difference between the lab outer walls of the Teflon (ii) radiation heat transfer between surfaces of tubes and surrounding were also neglected. Based on the measurement of $(\overline{T}_{in}+\overline{T}_{out})/2$ and mean $\overline{T}_s$ was estimated the radiation transfer coefficients as (Khan et al., 2004):

$$\overline{h}_{rad} = \varepsilon\times\sigma\times\left[\left(\left(\frac{\overline{T}_{in}+\overline{T}_{out}}{2}\right)^2 + \overline{T}_s^{\,2}\right)\times\left(\left(\frac{\overline{T}_{in}+\overline{T}_{out}}{2}\right)+\overline{T}_s\right)\right] \quad (4)$$

Where $\overline{T}_s = (T_{s,1}+T_{s,2}+T_{s,3}+T_{s,4})/4$

For a commercial aluminum tube with emissivity, $\varepsilon \approx 0.028$ (Collett and Hope, 1983), $\overline{h}_{rad}$ was found to a range between (0.174-0205)$Wm^{-2o}C^{-1}$, in approximately (0.0102-0.15)% of the air side convection heat transfer coefficient, $\overline{h}$. Thus, the heat transfer between the air and the surface of the tubes was actually due to the convection and the mechanism of the equation (3) was rewritten (Naik et al., 1987):

$$Q_{total} \cong \overline{h}\times A_s\times\left[\overline{T}_s - \left(\frac{\overline{T}_{out}+\overline{T}_{in}}{2}\right)\right] \quad (5)$$

Where $$A_s = nt\times\left[\pi\times d + 2\times(D-d)\right]\times L$$

Where *nt* number of tubes and the inlet air temperature $\overline{T}_{in}$ was the variable 22.86°C - 26.01°C and the average temperature of the surfaces of tubes $\overline{T}_s$.For the steady state condition, the overall heat transfer rate, Q_{elect}, was equal to the electrical heat supply, Q_{elect}. From equation (3), the average heat transfer coefficient was determined as:

$$\bar{h} = \frac{Q_{elect}}{A_s \times \left[\bar{T}_s - \left(\frac{\bar{T}_{out} + \bar{T}_{in}}{2} \right) \right]} \tag{6}$$

The dimensionless average heat transfer coefficient of air, namely, the Nusselt number, was calculated from:

$$\overline{Nu} = \frac{\bar{h} \times D_h}{k} \tag{7}$$

Where the hydraulic diameter:

$$D_h = \frac{4 \times (cross - \text{sec}tional\ area)}{wetted\ \ perimeter} = \frac{4 \times \left[\frac{\pi}{4} d^2 + \left(D - d \right) \times d \right]}{\pi \times d + 2 \times \left(D - d \right)} \tag{8}$$

The Reynolds number is defined as follow:

$$Re_{Dh} = \frac{\rho \times U_\infty \times D_h}{\mu} \tag{9}$$

The dimensionless pressure drop:

$$C_p = \frac{\Delta P}{0.5 \times nt \times \rho \times U_\infty^2} \tag{10}$$

The experimentaluncertainty issues were dealt by (Editorial, 1993; Dieck, 1997; Holman, 2012), etc.There is more than one way to estimate the uncertainty in the experimental results and has presented by Kline and McClintock. The few sample calculations here.Independent parameters (such as dimensions, temperature, velocity, etc.) Found different prejudices (P) and accuracy errors (A) using the root sum square (RSS) method:

$$P = \sqrt{P_1^2 + P_2^2 + P_3^2 + \cdots + P_n^2} \tag{11}$$

and

$$A = \sqrt{A_1^2 + A_2^2 + A_3^2 + \cdots + A_n^2} \tag{12}$$

The sum of component errors to get their 95% certitude uncertainty (U), the following the equation was used:

$$U = \sqrt{P^2 + A^2} \tag{13}$$

The dependent parameters (like D_h, Re, Q_{elect}, Nu, etc.), that are independent functions of other measured parameters, the uncertainty of the independent variable spreads in them according to their functional relationship. The example, in case of the electric power (Q_{elect}),

$$Q_{elect} = V \times I \tag{14}$$

The uncertainties of current and voltage propagate into Q_{elect}, and can be estimated in terms of relative or absolutevalues (%) as follows:
relative

$$\left(\frac{U_{Q_{elect}}}{Q_{elect}}\right)=\sqrt{\left(\frac{U_v}{V}\right)^2+\left(\frac{U_I}{I}\right)^2} \tag{15a}$$

or absolute

$$U_{Q_{elect}}=\sqrt{\left(\frac{\partial Q_{elect}}{\partial V}U_v\right)^2+\left(\frac{\partial Q_{elect}}{\partial I}U_I\right)^2} \tag{15b}$$

The uncertainties in finding the Q_{elect},D_h,Re, Nu and Cpwere estimated and found to remain approximately within1.6%,1.14%, 8.42%, 8.41%, and 3.5%, respectively.

RESULTS AND DISCUSSION

This section presents the experimental result of laminar forced convection heat transfer across tube bundles with in-line arrays. The main objective of this study is to conclude Reynolds number-Nusseltnumberrelationship. Figure 3 shows the variations of meanNusselt number $\overline{Nu}$ with Reynolds number Re for different heat flux supply.It is clear that $\overline{Nu}$ number, increases almost linearly with increase Re number.

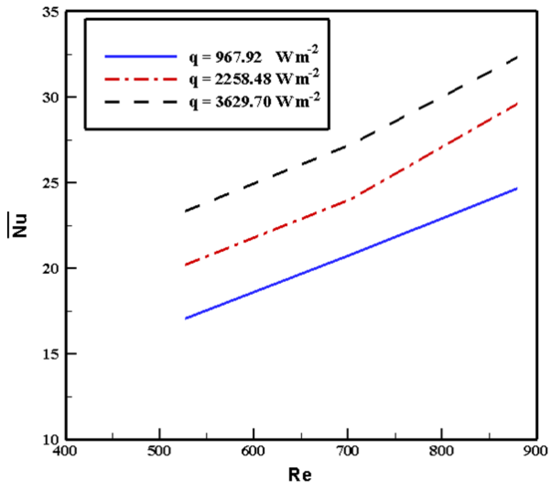

Figure 3. Variations of $\overline{Nu}$ with $Re$for different heat flux supply.

The variation of pressure drop ΔP and the dimensionless pressure drop C_p with the Reynolds number as shown in Figure 4, it is clear from the figure the ΔP increase linearly with increase Re, also, the C_p decreases with increase Re. The Figure 5 shows the relationship of average Nusseltnumber with Reynolds number.In addition, from thefigure can see the comparison the experimentalresult with two previous work Hausen, 1983; Wilson and Bassiouny, 2000. The meanNusselt number increased with the increase of Reynolds number in the power law formalize,

$$\overline{Nu} = C_1 \times Re^{C2} \tag{16}$$

CONCLUSION

In this paper, an experimental study of heat transfer and air flow over in-line flat tube bundle. The study results can be summarized as:

1- As the average Nusselt number of air flow increases with the increase of Reynolds number.

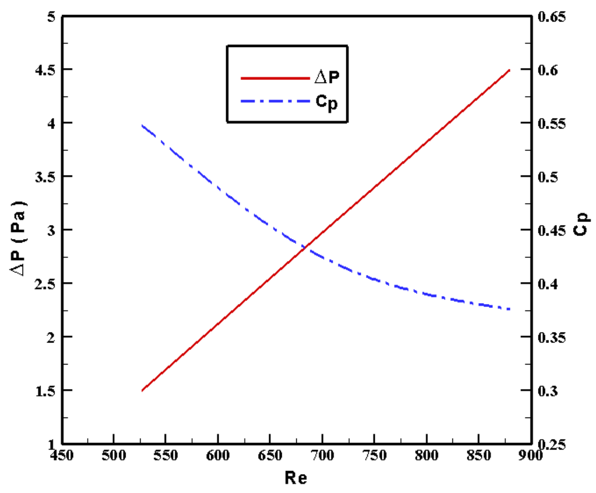

Figure 4. The variations of ΔP and Cp across test section corresponding to Re.

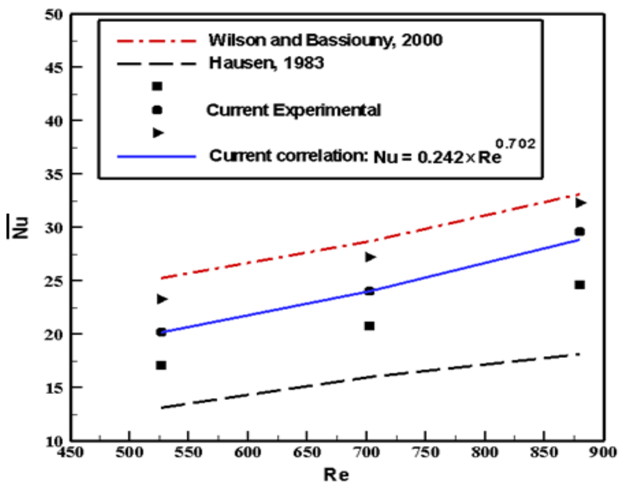

Figure 5. Variations of $\overline{Nu}$ with Re and comparison with different previous studies and correlation.

2- The average Nusselt number of all flat tubes is increased23.7%-36.7% with Reynolds numbers varying from 527 to 880 at the fixed heat flux.

3- The pressure drop increase with the increase Reynolds number, also dimensionless pressure drop decreases with the increase Reynolds number.

4- Finally, the Nusselt number-Reynolds number correlation was found to be $\overline{Nu} = 0.242 \times Re^{0.702}$ with the mean error $R^2 = 0.992$.

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NOMENCLATURE

A_s	tube surface area, m^2	T	temperature,^{o}C
C_1-C_2	correlation/curvefitcoefficients	U	velocity, m s^{-1}
C_p	pressurecoefficient	V	voltage, Volt
c_p	specific heat capacity at constant pressure, kJ kg^{-1o}C^{-1}	***Greeks***	
D_h	hydraulic diameter of flat tube, m	ΔP	pressure drop across the tube array, Pa
$\overline{h}$	average heat transfer coefficient, W m^{-2o}C^{-1}	ε	emissivity
k	thermal conductivity, W m^{-1o}C^{-1}	μ	dynamic viscosity,kg m^{-1}s^{-1}
I	Current, A	σ	Stefan-Boltzmann constant, 5.670373×10^{-8}W m^{-2} K^{-4}
L	total length of the tubes, m	ρ	density, kg m^{-3}
nt	number of tubes	***Subscripts***	
$\overline{Nu}$	averageNusseltnumber	*air*	air
P	perimeter of the flat tube, m	*in*	inlet
Pr	Prandtl number	*out*	out
Q	heat transfer rate,W	s	tubesurface
Re	Reynolds number	∞	free stream