



## TROCADOR DE CALOR DE CORRENTE CRUZADA DE AR

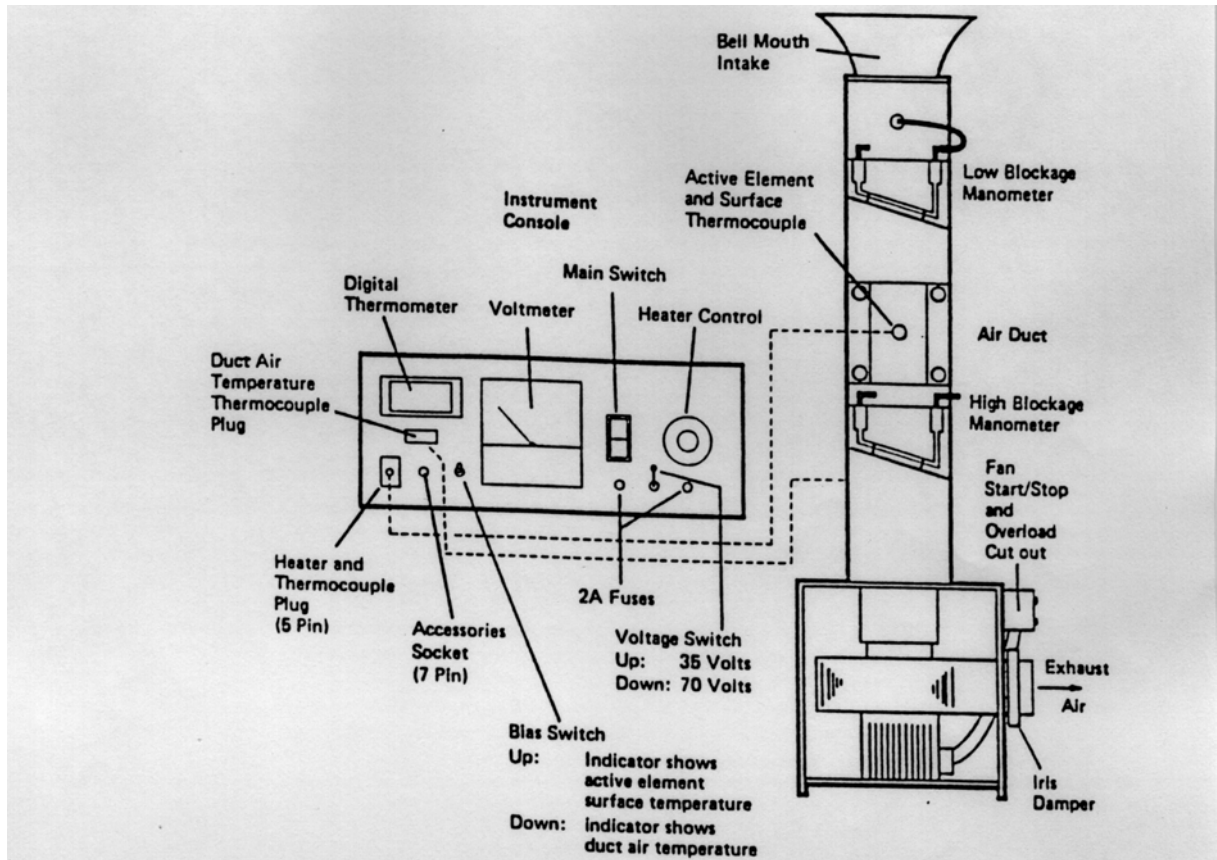
### Procedimento Experimental

1. Verifique que o cilindro aquecedor esteja na posição a montante em relação ao escoamento descendente de ar na seção de testes.
2. Verifique que uma extremidade da mangueira de medida de pressão esteja conectada no manômetro inferior e a outra extremidade na entrada da seção de testes.
3. Nunca ligue o ventilador antes de verificar que a válvula com diafragma, localizada na saída do escoamento de ar, esteja na posição de abertura mínima.
4. Após ligar o ventilador, ajuste a abertura da válvula com diafragma para obter a menor altura manométrica desejada ( $H = 5 \text{ mm H}_2\text{O}$ ).
5. Selecione a posição 70 V no seletor de voltagem.
6. Ajuste a voltagem fornecida ao cilindro aquecedor até que a sua temperatura atinja um valor em torno de  $60 \text{ }^\circ\text{C}$ . Este ajuste deve ser feito em etapas, aguardando alguns minutos até que a temperatura do cilindro atinja um valor de equilíbrio.
7. Quando o regime permanente for atingido, registre os valores de  $T_s$ ,  $T_a$ ,  $H$  e  $V$ .
8. Abra a válvula de diafragma do ventilador até obter o novo valor desejado da altura manométrica ( $H = 6 \text{ mm H}_2\text{O}$ ).
9. Varie agora novamente a voltagem fornecida ao cilindro para restabelecer, na medida do possível, a temperatura original do cilindro aquecido.
10. Quando a operação em regime permanente for obtida, registre novamente os valores de  $T_s$ ,  $T_a$ ,  $H$  e  $V$ .
11. Repita a sequência com incrementos de 1 mm no valor de  $H$  até o valor máximo de  $10 \text{ mm H}_2\text{O}$ .
12. Ajuste a voltagem fornecida ao cilindro para um valor mínimo e aguarde o seu resfriamento por alguns minutos.
13. Ajuste agora o valor da altura manométrica  $H$  para o valor  $8 \text{ mm H}_2\text{O}$ .
14. Ajuste a voltagem fornecida ao cilindro até que ele atinja um valor em torno de  $60 \text{ }^\circ\text{C}$ .
15. Registre novamente os valores de  $T_s$ ,  $T_a$ ,  $H$  e  $V$ .
16. Ajuste a voltagem fornecida ao cilindro para um valor mínimo e aguarde o seu resfriamento por alguns minutos.
17. Coloque agora o cilindro aquecido na fila a jusante do trocador, movendo o cilindro não aquecido desta fila para a posição anteriormente ocupada pelo cilindro aquecido.
18. Mantendo o valor de  $H$  em  $8 \text{ mm H}_2\text{O}$ , repita agora as etapas 14, 15, 16 e 17 até obter as leituras para o cilindro aquecido na sexta fila do trocador de calor.
19. Ajuste novamente a voltagem fornecida ao cilindro para um mínimo e aguarde o seu resfriamento por alguns minutos.
20. Desligue a fonte de potência e em seguida o ventilador, fechando também a sua válvula diafragma.

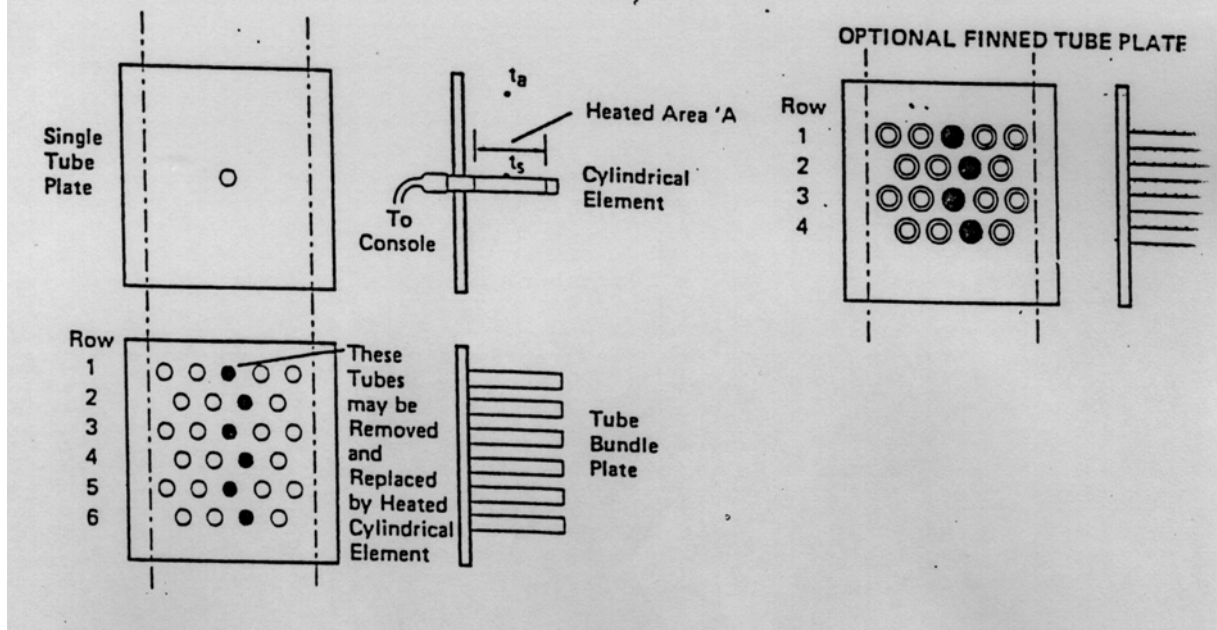


### Apresentação de Resultados

1. Faça um gráfico (log-log) com os valores experimentais do número de Nusselt em função do número de Reynolds para o cilindro aquecido na primeira fila do trocador.
2. Relacione os seus dados do item anterior na forma da equação  $Nu = C.Re^m.Pr^n$ .
3. Compare, num gráfico, a sua correlação com a correlação  $Nu = 0,273.Re^{0,635}.Pr^{0,34}$ .
4. Faça um gráfico da variação dos valores experimentais de Nu em função da posição na fila do trocador, considerando o número de Reynolds médio no qual as medidas foram tomadas neste caso.
5. Calcule, através de uma média aritmética, o valor médio experimental de Nu para o cilindro nas primeiras quatro filas do trocador.
6. Considerando o valor médio de Re em que as medidas das primeiras quatro filas do trocador foram tomadas, compare o resultado do item anterior com a correlação  $Nu = 0,273.Re^{0,635}.Pr^{0,34}.Fn$   
Determine o valor de Fn obtido desta comparação.



### Cross Flow Heat Exchanger H350





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USEFUL DATA

Plain Tubes

Diameter of Active Element = 15.8mm

Length of Active Element Heated Surface = 50mm

Heated Surface Area of Active Element =  $2.482 \times 10^{-3} \text{ m}^2$

Duct Cross Sectional Area =  $9.75 \times 10^{-3} \text{ m}^2$

Minimum Flow Area through the Plain Tube Bundle =  $4.160 \times 10^{-3} \text{ m}^2$

Duct Air Velocity =  $74.294 \sqrt{\frac{H Ta}{Pa}} \text{ m s}^{-1}$

The resistance of the active element heater may be found engraved on the five pin plug attached.

**INTRODUCTION**

In order to transfer heat between two fluids many forms of heat exchanger have been devised. In one of the most common arrangements, heat is transferred between a fluid flowing through a bundle of tubes and another fluid flowing transversely over the outside of the tubes.

This configuration is known as a Cross Flow Heat Exchanger and is shown schematically in Fig.I.

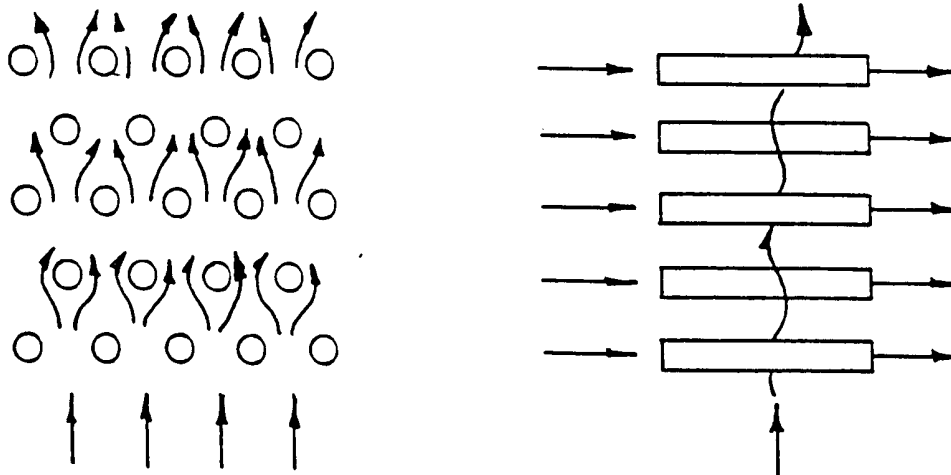


FIG. I

Various tube layouts have been devised in order to improve the efficiency of the cross flow heat exchanger and thereby reduce the physical size for a given heat transfer rate. However, the objective of all of the arrangements is to promote turbulence in the fluid flowing across the tube bundle.

The reason for this lies in the fact that the overall heat transfer coefficient for a cross flow heat exchanger is made up of three components. Firstly the surface heat transfer coefficient for the fluid flowing through the tubes, secondly the thermal conductivity and thickness of the tube material, and thirdly the surface heat transfer coefficient for the fluid flowing over the external surface of the tubes.

Enhancement of the first two components may be achieved by increasing flow velocity in the tubes and reducing the tube wall thickness, or using a material of higher thermal conductivity.

The third component may be increased by raising the stream velocity, thereby increasing the external Reynolds Number of each individual tube. Alternatively, the tube layout may be changed in order to maximise turbulence. This is achieved by ensuring that each row of tubes is positioned such that turbulence induced by the preceding row is incident upon the next row. Hence a cascade effect is produced such that the degree of turbulence increases with the depth of the tube bundle.

The effect of turbulence is to enhance the surface heat transfer coefficient beyond the level achieved by increased Reynolds Number alone.

If the fluid flowing over the outside of the tubes is a gas, then the effective heat transfer coefficient may be further increased by the use of extended surfaces, e.g. fins.

As cross flow heat exchangers occur in many varied forms throughout industry, it is essential that engineers and technologists should be aware of the performance of such units.



**Determination of the Duct Air Stream Velocity**

By application of Bernoulli's equation between the pressure tapping point and the external atmosphere, the following equation may be obtained:

$$U = C_d \sqrt{\frac{2 \ln g HRTa}{Pa}}$$

Where  $R$  is the specific gas constant for air.

For the intake type used on the duct  $C_d$  is taken as 0.98. Substituting in the equation for the known quantities gives:

$$U = 74.294 \sqrt{\frac{HTa}{Pa}} \text{ m s}^{-1}$$

Note that the velocity obtained is the velocity existing in the unobstructed duct.

Where the obstruction is small as in the case of the single tube, the effects of such a blockage may be ignored. However, when the multi-tube plate is in position, the blockage effect must be accounted for and in fact the velocity through the minimum area available to flow is taken.

For the plain multi-tube plate this results in an effective velocity through the tube bundle of,

$$U' = U \times 2.343$$

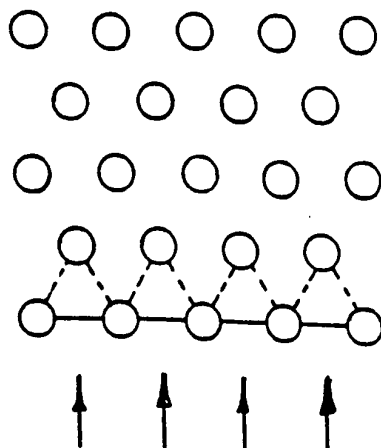
**B. Tube Bundles in Cross Flow**

In the case of an isolated cylinder in cross flow, the velocity used to calculate the Reynolds Number of the flow is that of the stream approaching the cylinder.

However, for the case of a tube bundle obstructing the duct, it can be readily appreciated that the velocity of the flow approaching the bundle will be far lower than the velocity between the rows of tubes, the duct area having been reduced by some function of the transverse plan area of the tubes.

A characteristic reference velocity for a particular tube bundle is therefore taken, and an accepted value is the stream velocity at the minimum free area.

For the schematic arrangement shown below, this would be established by determining the minimum of the two areas denoted by the dotted or continuous lines.





Hence, if the empty duct has a cross sectional area of  $A_D$  and the minimum inter tube area is  $A_T$ , the velocity through the heat exchanger will be,

$$U' = U \times \frac{A_D}{A_T}$$

It is this velocity that is used to calculate the Reynolds Numbers used in the correlations.

As in the case of a single tube in cross flow, determination of a correlation for the mean convective heat transfer coefficient for the tubes forming a cross flow heat exchanger must be carried out experimentally.

The tube position within the bundle adds a further variable to the general turbulent flow equation (II) and this then has the form,

$$(V) \quad Nu = C Re^m Pr^n Fn$$

Where  $Fn$  is a function of the number of tube rows crossed by the transverse stream.

An accepted form of the generalised equation (V) is,

$$Nu = 0.273 Re^{.655} Pr^{.34} Fn$$

The Nusselt Number obtained from this correlation is a mean value for all of the tubes within a bundle. Hence for design purposes a prediction may be obtained for the overall heat transfer rate of a cross flow heat exchanger of a particular size and number of rows.

The above equation is applicable to a staggered arrangement of tubes, as in the previous figure, for Reynolds Numbers between 300 and 200,000.

For a staggered tube bundle,  $Fn$  is found to vary in a manner shown in Fig.II below.

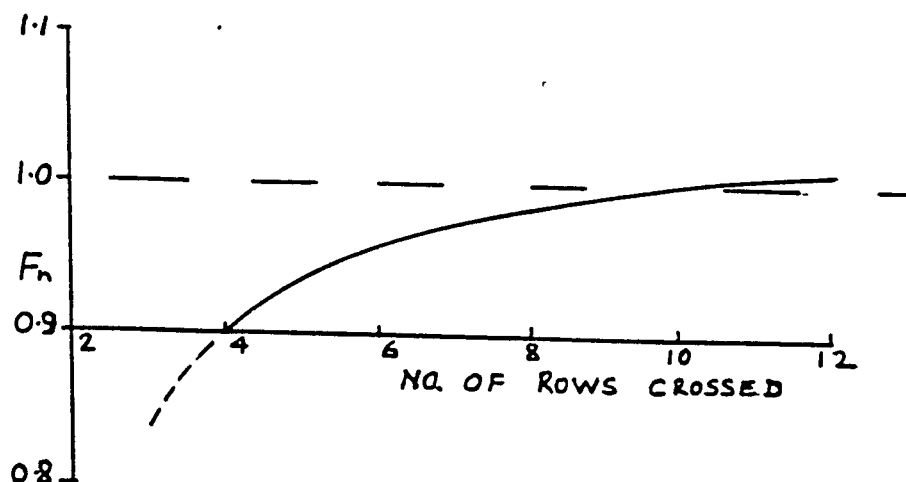


FIG.II

Thus it may be seen that as the number of rows crossed increases, the effect is an increase in the convective heat transfer coefficient. This is effectively due to the turbulence behind each row adding to the turbulence due to the stream velocity. However, as may be seen in Fig.II, a reduction in the rate of increase occurs beyond approximately 10 rows. The stream effectively becomes "saturated" in terms of turbulence.



SPECIMEN CALCULATIONS

Test No. 1 - Tube Row No. 1

Rate of Heat Transfer from Active Element,

$$\begin{aligned} &= \frac{V^2}{R} \\ &= \frac{35.2^2}{70.5} \\ &= \underline{17.6 \text{ Watts}} \end{aligned}$$

Area of Heat Transfer Surface =  $2.491 \times 10^{-3} \text{ m}^2$  (See Page 4)

Heat Flux,

$$\begin{aligned} \phi &= \frac{\dot{Q}}{A} \\ &= \frac{17.6}{2.482 \times 10^{-3}} \text{ Watts m}^{-2} \\ &= \underline{7.091 \text{ kW m}^{-2}} \end{aligned}$$

Active Element Surface to Air Temperature Difference,

$$\begin{aligned} &= t_s - t_a \\ &= 96 - 23 \text{ K} \\ &= \underline{73 \text{ K}} \end{aligned}$$

Mean Surface Heat Transfer Coefficient,

$$\begin{aligned} h &= \frac{\phi}{(t_s - t_a)} \\ &= \frac{7.091}{73} \\ &= \underline{0.0971 \text{ kW m}^{-2} \text{ K}^{-1}} \end{aligned}$$

Duct Air Velocity,

$$\begin{aligned} U &= 74.294 \sqrt{\frac{T_a H}{P_a}} \\ &= 74.294 \sqrt{\frac{(273 + 23) \times 1.5}{1.009 \times 10^5}} \text{ m s}^{-1} \\ &= \underline{4.9 \text{ m s}^{-1}} \end{aligned}$$

Ratio of free duct area to effective inter-tube area = 2.343 (See Page 14).

Effective air velocity,

$$\begin{aligned} U' &= U \times 2.343 \\ &= 4.9 \times 2.343 \\ &= \underline{11.5 \text{ m s}^{-1}} \end{aligned}$$

From the graph on Page 46, the kinematic velocity of air at 14°C,

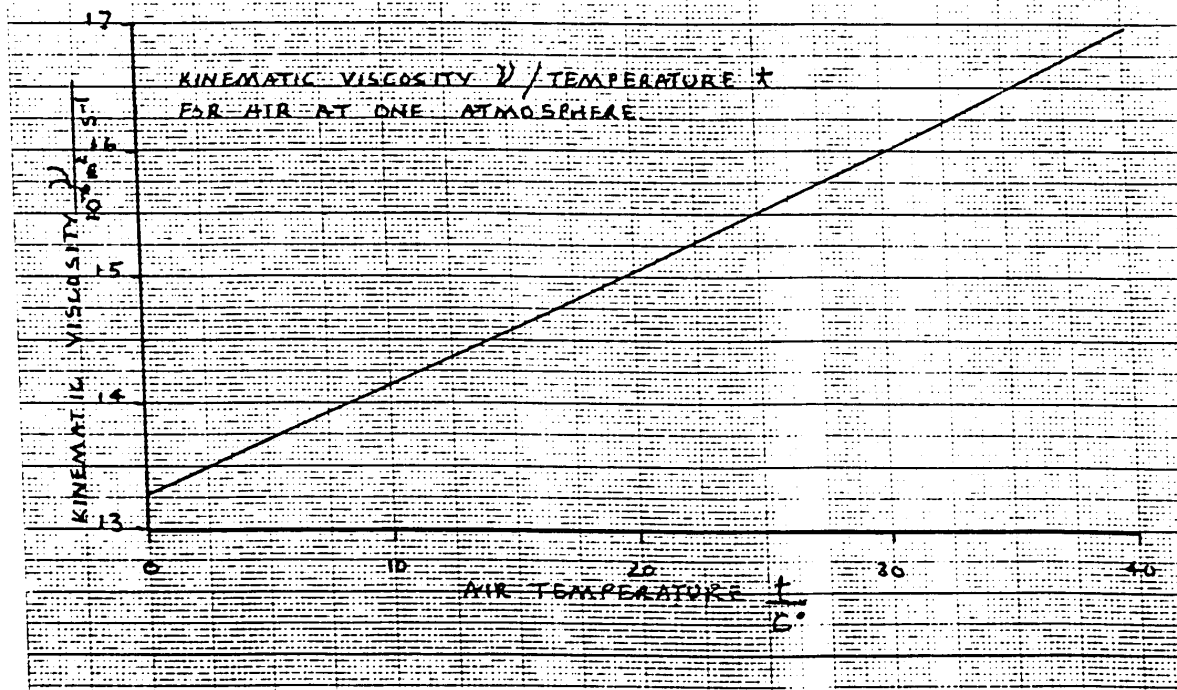
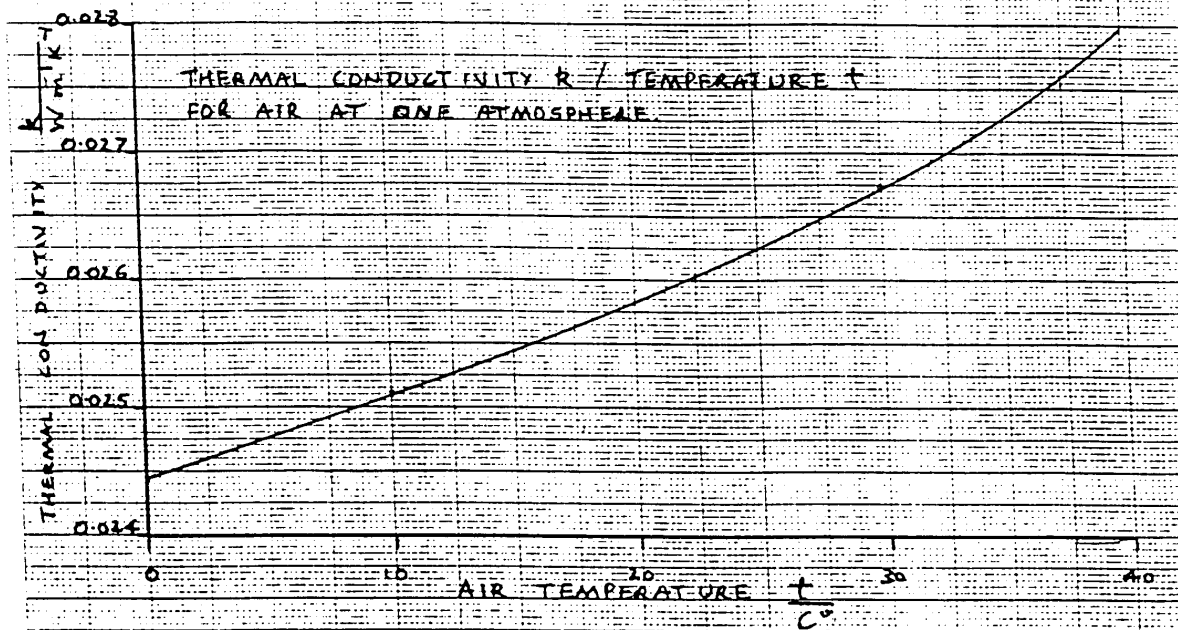
$$v = 15.34 \times 10^{-6} \text{ m}^2 \text{ s}^{-1}$$

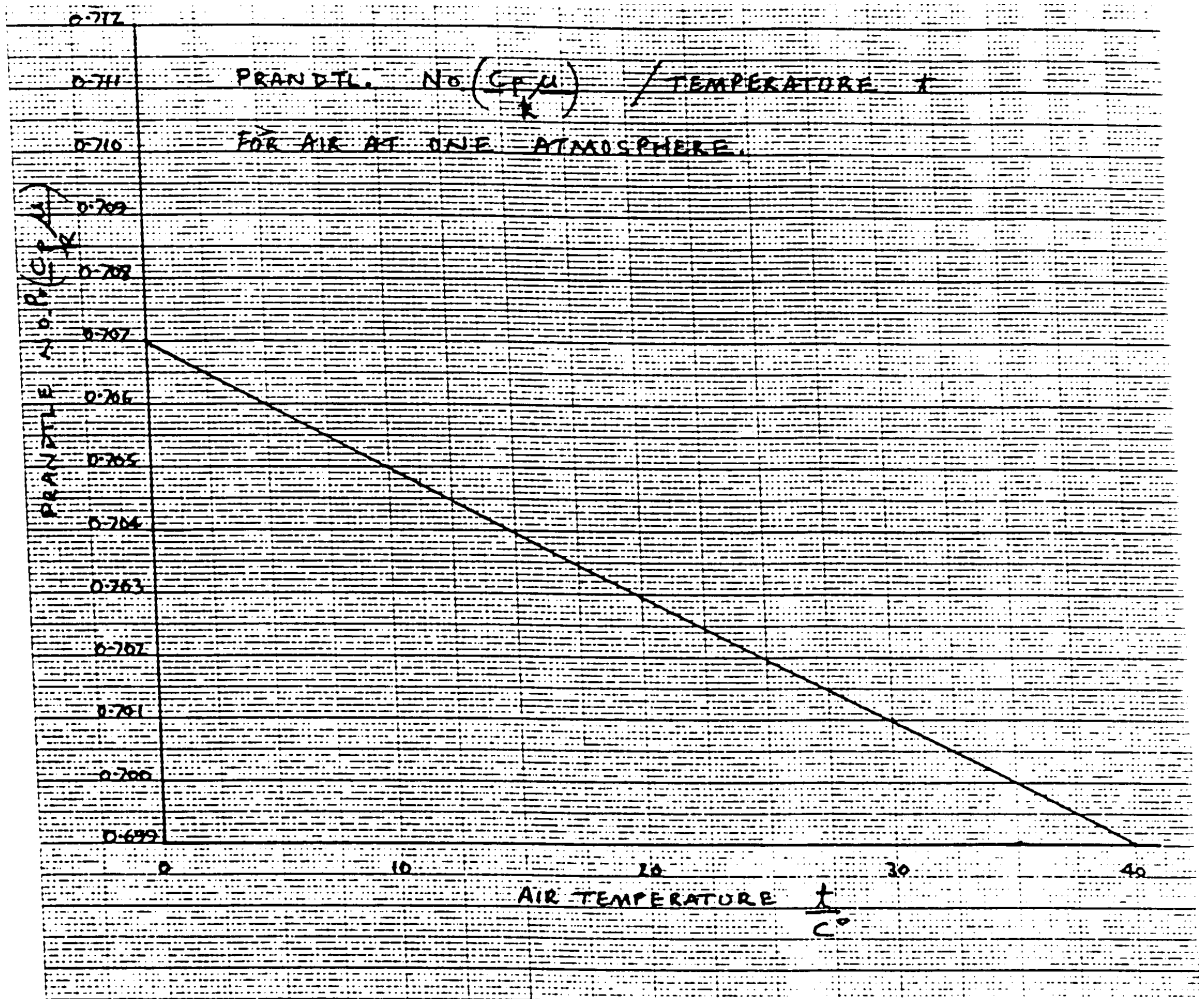




$$\begin{aligned}\text{Reynolds Number} &= \frac{U' d}{\nu} \\ &= \frac{115 \times 158 \times 10^{-3}}{15.34 \times 10^{-6}} \\ &= 11844\end{aligned}$$

Similar calculations apply to the remaining data.







HILTON CROSS FLOW HEAT EXCHANGER

OBSERVATION SHEET

Date:

Atmospheric Pressure:

mbar

Heater Element Resistance R:  
(engraved on connection plug)

Ohms

TEST No	1	2	3	4	5	6	7	8
Active Element Surface Temperature $t_s / ^\circ\text{C}$								
Duct Air Temperature $t_a / ^\circ\text{C}$								
Intake Air Depression $H / \text{mm H}_2\text{O}$								
Active Element Heater Voltage $V / \text{Volts}$								
Tube Row								

**DERIVED RESULTS:**

Heat Transfer Rate $\dot{Q} / \text{Watts}$								
Heat Flux $\phi / \text{Watts m}^{-2}$								
Active Element Surface to Air Temperature Difference $t_s - t_a / \text{K}$								
Mean Surface Heat Transfer Coefficient $h / \text{W m}^{-2} \text{K}^{-1}$								
Duct Air Velocity $U / \text{m s}^{-1}$								
Effective Air Velocity $U' / \text{m s}^{-1}$								
Reynolds Number								
Nusselt Number $h d / k$								