

Experimental Investigation of Forced Convection Heat Transfer for Turbulent Air Flow Inside Horizontal Heated Pipe

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Abstract:

An experimental study presented in this paper about the forced convection in the ranges of turbulent flow. The experiments were made on dry air and moist air of different moisture content. Different turbulent air flow and heat flux considered and analyzed. The distribution of temperature and local Nusselt number along the heated copper pipe, the average Reynolds number is $7.4 \times 10^4 - 9.1 \times 10^4$ against average Nusselt number, in addition to the velocity and temperature distribution across a horizontal pipe were presented in the results and discussed. Finally, it can be concluded that the effect of moisture content in the air on Nusselt number was little.

Nomenclature:

A_i	internal pipe area (m ²)
A_{orifice}	orifice area (m ²)
B	length of heated pipe up to chosen section (mm)
C_d	the orifice discharge coefficient it is equal 0.613
C_p	specific heat of air at inlet temperature (W/m.K)
d	copper pipe diameter (m)
H	heat transfer coefficient (W/m ² .K)
K	thermal conductivity of insulating material (fiber glass) (W/m.K)
k_{air}	thermal conductivity of air (W/m.K)
k_c	thermal conductivity of the copper (W/m.K)
L	length of insulating pipe (test pipe length) (m)
\dot{m}_a	air mass flow rate (kg/sec)
Q_1	heat input tape (W)
Q_2	heat lost through lagging (W)
Q_3	heat input by conduction (W)

\bar{r}	mean radius of copper tube (m)
r_i, r_o	inside and outside radii of the lagging (m)
T	copper pipe wall thickness (m)
T_b	bulk mean air temperature (°C)
T_{in}	air inlet temperature (°C)
ΔT	mean temperature drop across lagging (K)
Δp	pressure drop across orifice (N/m ²)
ρ	air density at orifice, and it is calculated for dry and moist air (kg/m ³)
w	Water content (kg _{wat} /kg _{dryair})

1. INTRODUCTION

The analyses of heat transfer by convection are more difficult than that by conduction. In convection, it must be keep the mass and momentum conservation in addition to energy conservation.

Forced convection is more important than free convection because of its wide industrial applications, such as most of heating equipments, heat exchangers, boilers, etc. The theoretical study of turbulent forced convection is difficult because of the energy results from eddies and turbulences of the flow. Therefore, the dependence was on empirical data and correlations. A lot of studies were made on forced convection and they need great area to review. Many of these studies are presented in the references[1-4]. The present work was made on the forced convection inside tubes. The apparatus consists from a heated pipe by uniform heat flux. The air flow inside the pipe with high velocities at turbulent limits. Some important parameters studied, Reynolds number, heat flux and Nusselt number. The experiments had been done on the dry air and moist air with different moisture content.

2. EXPERIMENTAL APPARATUS and PROCEDURE

The apparatus is shown in fig. 1. It consists of an electrically driven centrifugal fan which draws air through a control valve and discharges into a 76.2 mm diameter, U-shaped pipe. The fan speed remains constant throughout. An orifice plate is fixed in this pipe to measure the air flow rate. This pipe is connected to a copper test pipe which discharges to atmosphere and is electrically heated by a heating tape wrapped around the outside of the pipe. The power input to the tape can be varied by means of a variable transformer fitted to the apparatus. The test pipe is insulated with 25mm thick fiberglass lagging. The test length, situated within the heated length of the test pipe, has pressure tapping at each end which are connected to a manometer on the instrument panel. Other manometers fixed to the instrument panel measure fan discharge pressure and the orifice pressure drop. Water boiler, not shown in fig.1, is added to the apparatus to supply water vapor in the inlet air.

Seven thermocouples are fixed to the wall of the copper test pipe at various points along the heated length. A further six thermocouples are situated at points within the lagging. The positions of all the thermocouples and relevant dimensions, are shown on a mimic display on the instrument panel and shown in fig. 2. A thermometer measures the air temperature at the inlet to the test pipe. The output from any thermocouple may be chosen with a selector switch fitted to the instrument panel and measured with an electronic thermometer.

It must be mentioned here, that the calibration was made on the measurement tools before doing the experiments.

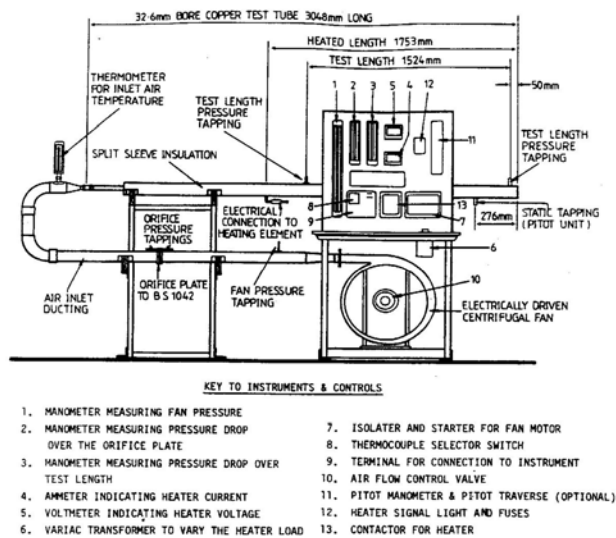


Fig. 1 General arrangement of the apparatus

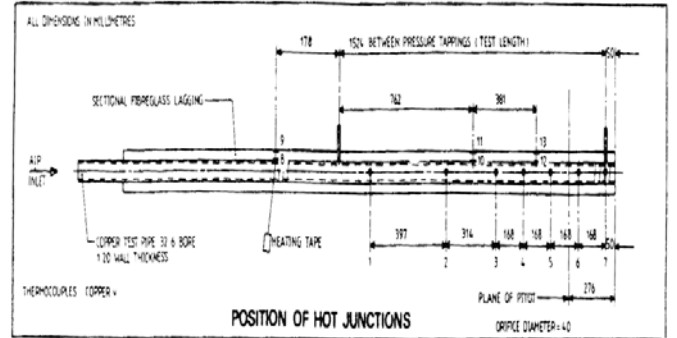


Fig. 2 Dimensions and positions of thermocouples

3. CALCULATIONS and THEORY

The calculation falls into five parts as described in the following sections

3.1. Mass Flow Rate

$$m_a = \rho A_{orifice} \cdot C_d \sqrt{\frac{2\Delta p}{\rho}} \quad (1)$$

3.2. Heat Flux

It can be calculated from heat input tape, and heat lost through lagging, as follows:

$$Q_1 = I * V$$

$$Q_2 = 2 \pi k l \frac{\Delta T}{\ln \frac{r_o}{r_i}} \quad (2)$$

Thus, heat flux (q') through tube wall is:

$$q' = \frac{Q_1 - Q_2}{A_i} \quad (3)$$

3.3. Mean Air Temperature at Chosen Heat transfer Section

The thermocouple positions are shown on the diagram on the instrument panel. From the temperature readings it seen that the section between 2 and 5 is free of exit and entrance effects. Hydrodynamically, the turbulent flow becomes fully developed and paralleled after small distance from entrance, approximately ten times of pipe diameter[5]. Therefore, it is suggested that the heat transfer calculations are made around section 4, also the bulk mean air temperature at this point 4. Total heat input includes heat input by the heating tape plus heat input by conduction in the pipe less the heat lost through the lagging.

$$Q_3 = 2\pi k_c \bar{r} t \times \frac{\text{temp.} - \text{drop}}{\text{meter}} \quad (4)$$

Total heat input (Q_{in}) to chosen section

$$= (Q_1 - Q_2) \times \frac{b}{1753} + Q_3 \quad (5)$$

$$T_b = T_{in} + \frac{Q_{in}}{m_a C_p} \quad (6)$$

3.4. Heat Transfer Coefficient

It can be calculated from the following equation:

$$h = \frac{q'}{T_w - T_b} \quad (7)$$

The wall temperature T_w is given by the thermocouple at the point at which the heat balance is taken.

3.5. Nusselt Number

To calculate Nusselt number, it must be know the thermal conductivity of the air at bulk mean air temperature. Nusselt number can be calculated from the following equation

$$Nu = \frac{hd}{k_{air}}$$

Also here, Nusselt number was calculated from empirical correlations suggested by many investigators. These correlations for turbulent flow inside pipes. They are listed in the following table:

$Nu = 0.023$ $Re^{0.8} Pr^{0.4}$	Most familiar equation [6]
$Nu = 0.27$ $Re^{1/2} Pr^{1/4} \left(\frac{\mu}{\mu_b} \right)^{0.14}$	$Re > 10^4$ and $0.7 < Pr < 16700$, it is used when the temperature has significant effect on fluid thermal properties [1]
$Nu = 0.0214$ $(Re^{0.8} - 100) Pr^{0.4}$	It is applicable at the ranges: $0.5 \leq Pr \leq 1.5$ and $10^4 \leq Re \leq 5 \times 10^6$ [7]

4. RESULTS and DISCUSSION

4.1. Temperature Distribution along Heated Pipe

The figures(3,4,5) show the temperature distribution along heated pipe at different values of heat flux, Reynolds number and moisture contents. It can be observed the parabola shaped in all curves with peak point at the distance around point four (the region of the balance). The approximation was to polynomial of second order with correlation coefficients between 0.91 – 0.93 .

At peak point, the temperatures of air and pipe wall are equal, after that, wall temperature decreases. This can be explained as follow: usually here, heat transfer occurs when there is temperature difference between wall and air. At inlet section, this difference was high and reduced gradually when the flow continuous inside the pipe. At balance section, it must be zero (highest values for air and wall temperatures). When the heat flux raised manually, 2.28, 3.77 and 5.1 kW/m², wall temperature at different sections will be higher

More turbulence in flow leads to reduce wall temperature because there is more heat removed from the pipe wall. As moisture content increases at a particular values of Reynolds number and heat flux, the wall temperature increased. For lowest value of heat flux, it can be seen little variation in the temperature along the pipe wall.

4.2. Local Nusselt number (Nu_x)

The local Nusselt number (Nu_x) is proportional reversely with the thermal conductivity of air (k_{air}), that varied with temperature and it was taken at bulk mean temperature, and proportionally with local heat transfer coefficient.

Figures(6,7,8) represent variation of Nu_x along the test tube at different values of heat flux and Re with varied moisture contents. It can be observed that Nu_x has maximum value at entrance and be less at inner sections till the balance section (minimum value of Nu_x) after that it is increased toward the exit sections. This variation in Nu_x comes from the temperature difference ($T_w - T_b$) and thermal conductivity of air which varied little with temperature. The increasing in heat flux leads to reduce Nu_x at different sections, whereas the increasing in Re leads to rise the values of Nu_x .

For dry air, it can be observed from figure(6) the change in Re causes a clear variation in Nu_x , whereas for wet air, there is a little variation in Nu_x with Re.

4.3. Average Nusselt number

The figures (4-9)-(4-11) , represent the relation between average Nusselt number (Nu) and Reynolds number for dry and wet air. The values of Nu for the present work were less than that calculated from equations in table above. For dry air, figure(4.9), Nu increased with Re and it is between 110 to 127 for high heat flux equal 5.1kW/m², whereas for wet air, the value of Nu changes little with Re×w. The increasing in moisture content leads to reduce heat transfer coefficient which increase when Re increased.

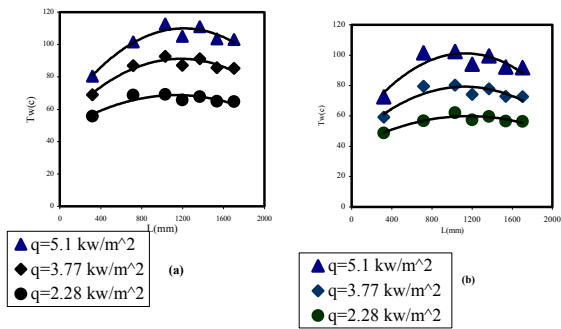


Fig. 3 Temperature Distribution along Heated Pipe, Dry Air ($w=0.0\text{kg}_{\text{wat}}/\text{kg}_{\text{dry air}}$)
 (a) $\text{Re}=7.4 \times 10^4$ (b) $\text{Re}=9.1 \times 10^4$

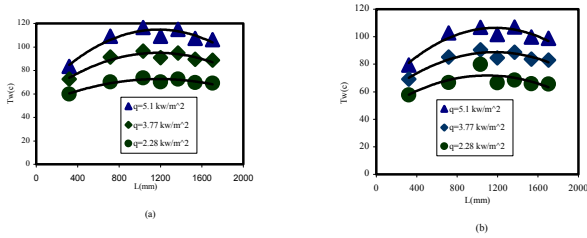


Fig. 4 Temperature Distribution along Heated Pipe, Wet Air ($w=0.013\text{kg}_{\text{wat}}/\text{kg}_{\text{dry air}}$)
 (a) $\text{Re}=7.4 \times 10^4$ (b) $\text{Re}=8.5 \times 10^4$

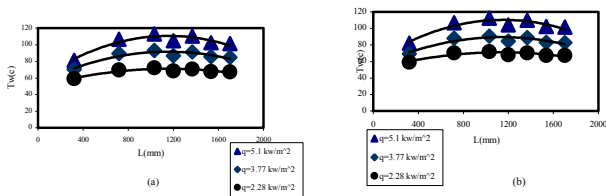


Fig. 5 Temperature Distribution along Heated Pipe, Wet Air ($w=0.032\text{kg}_{\text{wat}}/\text{kg}_{\text{dry air}}$)
 (a) $\text{Re}=7.4 \times 10^4$ (b) $\text{Re}=8.5 \times 10^4$

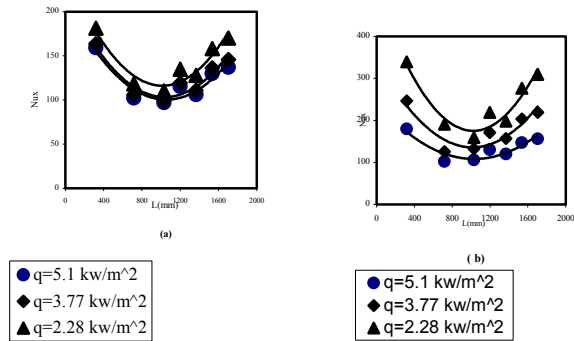


Fig. 6 Local Nusselt number along Heated Pipe, Dry Air ($w=0.0\text{kg}_{\text{wat}}/\text{kg}_{\text{dry air}}$)
 (a) $\text{Re}=7.4 \times 10^4$ (b) $\text{Re}=9.1 \times 10^4$

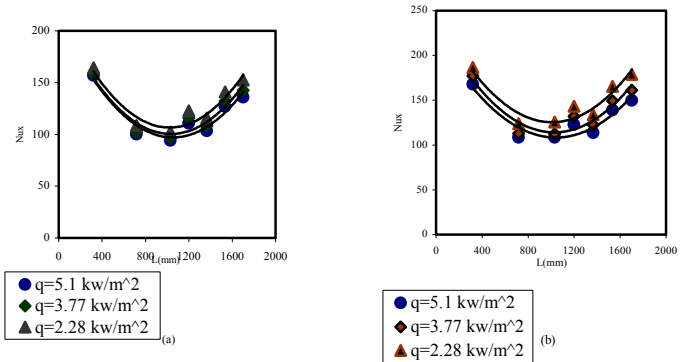


Fig. 7 Local Nusselt number along Heated Pipe, Wet Air ($w=0.013\text{kg}_{\text{wat}}/\text{kg}_{\text{dry air}}$)
 (a) $\text{Re}=7.4 \times 10^4$ (b) $\text{Re}=8.5 \times 10^4$

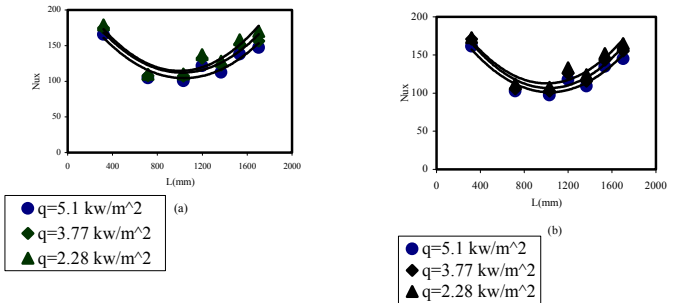


Fig. 8 Local Nusselt number along Heated Pipe, Wet Air ($w=0.032\text{kg}_{\text{wat}}/\text{kg}_{\text{dry air}}$)
 (a) $\text{Re}=7.4 \times 10^4$ (b) $\text{Re}=9.1 \times 10^4$

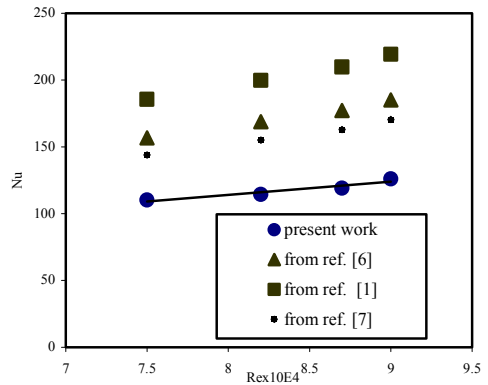


Fig. 9 Relation between Reynolds number and Average Nusselt number
 Dry air, $q=5.1\text{kW}/\text{m}^2$

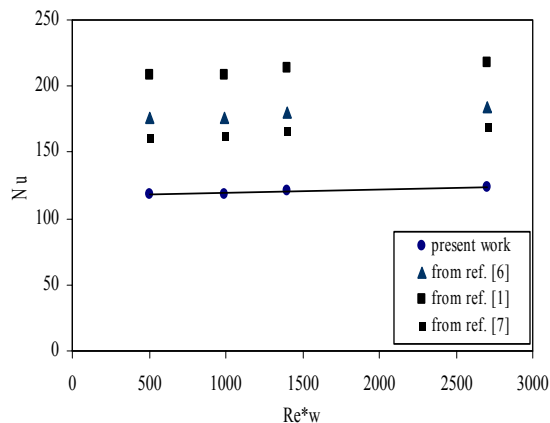


Fig. 10 Relation between Reynolds number average Nusselt number, wet air, $q'=5.1\text{kW/m}^2$

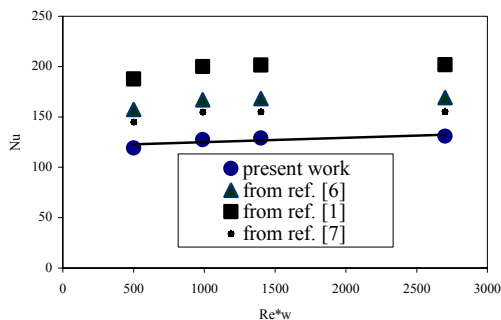


Fig. 11 Relation between Reynolds number multiplying moisture content and Average Nusselt number, wet air, $q=2.28\text{kW/m}^2$

5. CONCLUSIONS

1. Pipe wall temperature is a strong function of Reynolds number and heat flux.
2. More moist air leads to reduce the average Nusselt number. This refers to the decreasing in heat transfer by convection and increasing heat transfer by conduction.
3. Also for more moist air the velocity has greater values for any section through the heated pipe.

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