Natural convection in tubes – a solar water heating application

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Abstract

The present work deals with experimental studies on heat transfer and flow characteristic for buoyancy induced flow through inclined tubes. The parameters that were varied during the experimentation are tube inclination and heat supply. It was found that the mass flow rate and the heat transfer coefficient increase with an increase in heat flux. The flow rate decreases for an increase in the tube inclination.

Keywords: thermo siphon, natural convection, buoyancy induced flow, solar water heating system, uniform wall heat flux, renewable energy.

1 Introduction

Heat transfer by natural convection inside the tube (circular or square) has a large number of applications in industries. These include solar water heating systems, cooling of gas turbine blades, heat exchangers etc. The heat transfer and flow characteristics with respect to various parameters play a vital role for the design of such systems. The present work is carried out with a specific application in mind, i.e. domestic solar water heating systems.

Presently most of the dimensions of solar water heaters are standardized from some other commercial consideration and not necessarily giving the best thermo-siphon results. Little information is available in the literature on the heat transfer characteristics for buoyancy-induced flow through inclined tubes [1–4]. The present work aims to study the effect of tube inclination and heat flux on heat transfer and flow characteristics for buoyancy induced flow through inclined tubes, and to develop an experimental model. Hence this work is undertaken.
2 Nomenclature

\( a \) = Side of the square tube (m)
\( h \) = Heat transfer coefficient (W/m\(^2\)°C)
\( q \) = Heat flux (W/m\(^2\))
\( m' \) = Mass flow rate (ml/min)
\( g \) = Acceleration due to gravity (m/sec\(^2\))
\( t \) = Temperature, (°C)
\( k \) = Thermal conductivity (W/m°C)
\( C_p \) = Specific heat at constant pressure, (kJ / kg K)

**DIMENSIONLESS NUMBER**

\( Gr = \text{Grashof Number} \left( \beta g \Delta T a^3 / \nu^2 \right) \)
\( Nu = \text{Nusselt Number} \left( h a / k \right) \)
\( Pr = \text{Prandtl Number} \left( \mu C_p / k \right) \)
\( Ra = \text{Rayleigh Number} \left( Gr Pr \right) \)
\( Re = \text{Reynolds Number} \left( 4 m' / (\pi a \mu) \right) \)

**GREEK SYMBOLS**

\( \theta \) = Angle of Inclination (degree)
\( \beta \) = Coefficient of Thermal Expansion, K\(^{-1}\)
\( \nu \) = Kinematic Viscosity, (m\(^2\) / sec)
\( \mu \) = Dynamic Viscosity, (N·s /m\(^2\))

3 Experimental set-up

A schematic of the test set up is shown in Fig.1. It consists of a long aluminum tube. The tube is wound uniformly with a 90-gauge Nichrome wire to provide a uniform heat flux condition on the tube surface. The power supplied is measured directly with the help of a pre-calibrated digital wattmeter. The tube was insulated to reduce the external heat loss. The tube is connected to a constant headwater tank and the flow is measured with the help of a calibrated measuring jar. Thermocouples were used to measure the wall and water inlet and outlet temperatures. The arrangement and methodology suggested by Prayagi and Thombre [4] was adopted for this purpose. Thermocouples were used to measure the wall temperature.

![Figure 1: Schematic of experimental set-up](image-url)
Thermocouples were used to measure the inlet and outlet temperature of water flowing through the tube. A linear temperature variation was assumed from the inlet to the exit of the flowing water. It is well justified since the boundary condition created at the tube surface is a uniform heat flux boundary condition. The dimensional details of the experimental set-up and the parameters varied during the experimentation are given in table 1.

<table>
<thead>
<tr>
<th>Sr.No.</th>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Tube length</td>
<td>1.5m</td>
</tr>
<tr>
<td>2</td>
<td>Square tube side</td>
<td>0.012 m</td>
</tr>
<tr>
<td>3</td>
<td>Tube inclination ($\theta$)</td>
<td>$20^0, 30^0, 45^0$ and $60^0$</td>
</tr>
<tr>
<td>4</td>
<td>Heat flux supplied (q)</td>
<td>200 W/m$^2$ to 7000 W/m$^2$</td>
</tr>
<tr>
<td>5</td>
<td>Insulation used</td>
<td>Asbestos</td>
</tr>
<tr>
<td>6</td>
<td>Fluid used</td>
<td>Water</td>
</tr>
</tbody>
</table>

4 Results

The steady state data generated in the experimental set-up for different values of tube inclination and heat flux where correlated for heat transfer and flow characteristics as described below.

4.1 Heat transfer characteristics

The increase in the heat transfer coefficient for different values of tube inclination with respect to heat flux is shown in fig.2. It represents the effect of heat supplied on the fully developed heat transfer coefficient for the test section of 20 degree inclination, 12mm side square tube and 1.5 m length.

![Figure 2: Heat transfer characteristics.](image)

It can be seen from the fig.2 that with an increase in the value of heat supplied, the heat transfer coefficient tends to increase. This is due to the fact that with an increase in heat supplied the buoyancy force tends to increase. The net effect of this is to increase the mass flow rate of liquid through the tube. The same trend was observed for the other configurations.
Conventionally the heat supplied (the main driving force) is represented by the Rayleigh number, whereas the heat transfer coefficient is represented by the Nusselt number. Hence in the present study the heat transfer characteristics are obtained by correlating the Nusselt number with the Rayleigh number.

Fig. 3 shows the plot between the Nusselt number and the Rayleigh number for all configurations investigated.

![Nusselt number versus Rayleigh number](image)

Figure 3: Nusselt number versus Rayleigh number.

It can be seen from this figure that the Nusselt number increases with an increase in the Rayleigh number. This is an expected result, since an increase in the Rayleigh number increases the buoyancy force and hence the flow rate and the Nusselt number. Typically for an increase in the Rayleigh number of about 50%, the increase in the Nusselt number is around 30%. This is attributed to the fact that if heat supplied is increased, the corresponding system equilibrium temperature will also increase, thereby leading to more heat loss from the system. Furthermore, the increase in heat loss depends upon the heat capacity of the liquid. The higher the heat capacity, the lower will be the heat loss and vice versa. The correlation (best fit curve) obtained from the fig.3 is as follows.

$$\text{Nu} = 1 \times 10^9 \text{Ra}^{1.442} \quad (1)$$

This equation is valid for Rayleigh numbers ranging from $7 \times 10^5$ to $5 \times 10^6$ and Reynolds numbers ranging from $0.3658 \times 10^2$.

### 4.2 Flow characteristics

Figure 4 represents the effect of heat flux on the flow rate for the test section with 20 degree inclination, 12mm side tube and 1.5m length. The following observations were made:

a. The induced flow rate increases with the heat supplied and the variation is nearly parabolic.
b. The same trend was also observed for other configurations investigated.

Conventionally the heat supplied for the buoyancy force is represented by the Rayleigh number, whereas the induced flow rate established is represented by Reynolds number. Hence to obtain mass flow characteristics, the Reynolds number is correlated with the Rayleigh number. The relevant data is plotted in the fig.5. It can be seen from the figure that the Reynolds number increases with an increase in the value of the Rayleigh number.
The Reynolds number increases with an increase in the value of the Rayleigh number. Typically it was observed that for an increase in the Rayleigh number from $3 \times 10^5$ to $4 \times 10^5$, the percentage increase in the value of the Reynolds number was around 30-35%. This is an expected variation as an increase in the Rayleigh number increases the buoyancy force, which results in a higher mass flow rate. Thus from the above analysis it can be concluded that the Reynolds number is dependent upon the Rayleigh number.

The correlation (best fit curve) obtained from the fig.6 is as follows.

$$\text{Nu} = 0.029 \times \text{Re}^{0.7864}$$

(2)
This correlation is valid for Reynolds numbers ranging from $0.3658 \times 10^2$.

5 Validation

The present experimentation was validated against the laminar forced convection, which is available in the literature i.e.

$$Nu = \frac{ha}{k} = 4.36$$

6 Conclusions

Experiments were performed to study heat transfer and flow characteristics for buoyancy induced flow through inclined square tubes with different inclinations and variations in heat flux. The following observations were made.

6.1 Heat transfer characteristics

- The heat transfer coefficient is strongly influenced by heat flux.
- When the heat flux increases, the heat transfer coefficient also increases.
- The heat transfer coefficient is found to be the weak function of the tube inclination.

The correlation obtained is as follows:

$$Nu = 1 \times 10^{-9} Ra^{1.442}$$

This equation is valid for Rayleigh numbers ranging from $7 \times 10^5$ to $5 \times 10^6$ and Reynolds numbers ranging from $0.3658 \times 10^2$.

6.2 Mass flow rate characteristics

- The mass flow rate increases with the heat supplied and the variation is parabolic.
- The mass flow rate is strongly influenced by heat flux.
- The mass flow rate is found to be practically independent of the tube inclination and tube length.

The correlation obtained is as follows:

$$Nu = 0.029 \times Re^{0.7864}$$

This equation is valid for Reynolds numbers ranging from $0.3658 \times 10^2$.

7 Scope for future work

The present study needs to be extended further as suggested below.

- The experimental data should be generated for air as in the case of air heaters operating under natural convection.
It is also proposed to change the fluid inlet temperature by placing the heater in between the inlet fluid tank and the test section, in order to study the effect of a change in the inlet temperature of fluid on the heat and mass transfer characteristics.

Insertion of twisted strip or fins on the test section in order to enhance the heat transfer.

It is also proposed that using a different aspect ratio (i.e. length to diameter ratio), an experiment should be performed to study the effect of diameter and length on heat and mass transfer characteristics.

Mathematical models should be developed for heat transfer as well as for mass-flow rate and must be compared with the experimental data generated. This mathematical model will be helpful for an optimum design of the solar water collector system.

Extensive experimental data should be generated to study the effect of twisted strip insertion inside the tube on the heat and mass flow characteristics. A mathematical correlation should be developed, in order to strike a balance between the parameters mentioned above for an optimum design.

References


