Humidification System using Aerodynamic Principles
Chapter 5
Humidification System using Aerodynamic Principles

5.1 Introduction

Humidification plays a vital role in different industrial sectors like textile, tobacco industry, automotive industry, spray paint booths, industrial humidifiers, hospital surgery and nursery rooms, photographic film manufacturing plants, and aircraft industry clean rooms. This kind of industrial sectors requires much more humidity. In order to achieve the high levels of humidity, these kinds of industrial sectors spend considerable amount of energy on installing and running artificial humidification devices like air washers and coolers. In these kind of conventional artificial devices, the combined action of pump and sprayer is used for humidification. In the case of large size companies, these kinds of humidification devices consume enormous energy. This energy consumption necessitates the development of an alternative humidification arrangement for saving the energy. However, the alternate devices should be simple in construction and easy to maintain in order to avoid the increase in their manufacturing and maintenance costs. In the light of these ideas, the principles of “aerodynamic humidifier” were adopted during this module of doctoral work. Accordingly, a setup using aerodynamic humidifier concept has been adopted from Bhat (2003) and subjected to validation studies. The details of this work are presented in this chapter.

5.2 Methodology

The methodology depicted in Figure 5.1 was adopted to accomplish this module of the doctoral work.
The steps of this methodology are enumerated below:

1. The humidification using aerodynamic principles was studied by referring to Bhat (2003).

2. The conceptual features of the humidifying setup incorporated with aerodynamic principle were adopted from Bhat (2003).

3. The FOILSIM software was used to select a suitable standard airfoil based on the pressure distribution over the airfoil surface. The airfoil with minimum pressure (below atmospheric pressure) over the upper surface was chosen.
4. The maximum pressure difference between the upper and lower surfaces of the airfoil was found out by fixing the angle of attack and varying the camber and thickness of the airfoil.

5. The coordinates of the selected airfoil profile were generated using Winfoil software.

6. Venturi section was designed using the continuity equation and Bernoulli’s Equation.

7. Suction tube was designed.

8. RH values were calculated against three different pressures at the throat of the venturi section.

9. The selected airfoil was fabricated with the help of Altech industries, Coimbatore, India.

10. Venturi section was fabricated.

11. Suction tube was chosen.

12. Different components of the setup were assembled.

13. Experiments were conducted using wind tunnel and the results were compared with the theoretical work.

14. The setup was simulated using CFD software ‘FLUENT’.

15. The simulated results were compared with the theoretically calculated values.

5.3 Importance of RH

The importance of RH is appraised here in order to enable the reader to appreciate the importance of this module of doctoral work. During the manufacturing process of textile yams, the atmospheric conditions with respect to temperature and humidity play a key role. The properties like dimensions, weight, tensile strength, elastic resistance, rigidity etc, of all textile fibers whether
natural or synthetic, are influenced by RH. Clean air, which is dirt and debris free, is highly important to the textile industry. The textile industry is the chief user of humidification devices. Some examples of physical properties of textile materials, which are influenced by the RH values, are given in Table 5.1 (Batliboi, 1992). The strength of cotton increases when RH value increases. Elongation percentage goes up with increased RH value for most textile fibers. The tendency for generation of static electricity due to friction decreases, as RH value increases.

In most unhumidified areas, static electricity poses a major problem. It is a condition caused by the static charges of electricity and friction. When the fibers are dry, the problem of static electricity increases proportionately with the speed of production machinery. Without sufficient humidification, high-speed machinery will not serve its purpose. The static electricity results in reduced efficiency. Humidification reduces fly and micro-dust, giving a healthier and more comfortable working environment.

Table 5.1. Recommended RH Levels (Batliboi, 1992)

<table>
<thead>
<tr>
<th>Department</th>
<th>Cotton %</th>
<th>Wool %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carding</td>
<td>50±5</td>
<td>65±5</td>
</tr>
<tr>
<td>Spinning</td>
<td>60±5</td>
<td>55±5</td>
</tr>
<tr>
<td>Weaving</td>
<td>80±5</td>
<td>60±5</td>
</tr>
<tr>
<td>Combing</td>
<td>60±5</td>
<td>70±5</td>
</tr>
<tr>
<td>Drawing</td>
<td>55±5</td>
<td>65±5</td>
</tr>
</tbody>
</table>
5.4 Study and Adoption of the Humidifier Using Aerodynamic Principles

As mentioned earlier, the motivation of this module of doctoral work was the contribution of Bhat (2003). The theoretical principles comprising experimental setup presented in this paper are described here. The experimental setup presented in Bhat (2003) is shown in Figure 5.2.

As shown, it comprises an airfoil, a venturi section, an air blower and a water reservoir. The airfoil structure is fixed at the exit point of connecting tube. A suction tube connects the reservoir placed below the venturi section. A blower is used to provide the sufficient suction pressure at venturi and airfoil.

Blower is used to create vacuum pressure in the venturi throat section. Due to the venturi effect, the pressure difference is created between free surface of water in the reservoir and at the end of the suction tube, kept at the throat of the venturi section. This results in flow of water from the reservoir to the suction tube. Now the water is sprayed in the air stream at the throat of the venturi section. When the same air is allowed to pass over the airfoil, a low-pressure partial vacuum is created over the upper surface. Due to this low pressure, the
mixture of air and water vapour is sucked into the low-pressure region through the connecting tube, which is inserted in the airfoil by means of a through hole in the airfoil. The air stream gets mixed thoroughly with the mainstream and flows out of the device, (i.e.) the airflows into the room.

The amount of water sprayed into the air stream is proportional to the low pressure created at the throat of venturi section. The mixture of air stream and water, flowing out of the venturi setup is forced towards a very low pressure region created over the upper surface of the airfoil and gets mixed with the main stream. Thus the main stream of air is effectively humidified, which is then made to flow and spread evenly in the room/space to be humidified, with the help of a duct.

5.5 Design of the Setup

Using the principles mentioned in the previous section, the design of the setup was carried out. This process was carried out in the following stages:

1. Selection of airfoil
2. Design of venturi section
3. Design of suction tube.

5.5.1 Selection of Airfoil

Airfoil section is used for creating the required pressure difference to suck the humidifying air formed at the venturi throat section. Selection of airfoil is an important step that determines the pressure difference being created. However, this task was challenging with thousands of such airfoils being available in both research and practice environments.

The following factors were considered while choosing one among them, by referring to Anderson, (2001). Lift is the basic factor by which an airfoil was
selected for an application. Lift is a function of angle of attack, wing geometry, that is, section shape and the viscosity and compressibility of the air. However, during this module of the doctoral work, the effects of compressibility and viscosity were neglected since the maximum velocity of air would be only 30 m/s. Lift is also a measure of the pressure difference produced by the airfoil. A cambered airfoil has greater coefficient of lift compared to a symmetric airfoil at any angle of attack (Anderson, 2001).

![Figure 5.3. Comparisons of lift coefficients of symmetric and cambered airfoils.](image)

1. For any airfoil section, the coefficient of lift is directly proportional to the angle of attack. The greater the angle of attack, higher is the lift produced. When the angle of attack is greater than 14 degrees, the lift coefficient starts decreasing. This process is called stalling and this reduces the lift coefficient and increases the drag coefficient. This phenomenon is due to the boundary layer separation along the upper surface of the airfoil and is highly pronounced in the cambered airfoils. So the angle of attack was fixed as 12 degrees in the setup and a cambered airfoil was chosen because, it produces a higher lift coefficient. Further boundary layer separation is associated with
these foils only at higher angles of attack. This was also another reason for choosing cambered airfoil with 12-degree angle of attack.

2. A high-speed, low-lift airfoil has very little camber. A low-speed, high-lift airfoil has a pronounced camber. Because the speed of the blower available at the researcher's laboratory was low, an airfoil with pronounced camber was chosen.

On inputting all the above factors and values in FOILSIM software, the airfoil NACA 2304 was selected. (NACA stands for National Advisory Committee for Aeronautics). Using FOILSIM software, it was found that the maximum coefficient of lift is produced by NACA 2304, for the boundary conditions of the required application. The boundary conditions were derived based on the specifications of the wind tunnel. The amount of lift produced and the pressure difference produced by the airfoil were found out using FOILSIM software. As shown in Figure 5.4, parameters like angle of attack, camber and maximum thickness, were given as input variables in FOILSIM software to get the pressure distribution on the upper and lower surface of the airfoil. The pressures on upper surface of airfoil obtained by inputting different values of the parameters of four different airfoils are indicated in Table 5.2.

Table 5.2. Pressure on the upper surface of different airfoils at 12° angle of attack

<table>
<thead>
<tr>
<th>Trial No</th>
<th>Name of Airfoil</th>
<th>Angle of Attack (Degree)</th>
<th>Camber (% C)</th>
<th>Thickness (% C)</th>
<th>Pressure on upper Surface (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>NACA 43012</td>
<td>12</td>
<td>4</td>
<td>12</td>
<td>98.294</td>
</tr>
<tr>
<td>2</td>
<td>NACA 4308</td>
<td>12</td>
<td>4</td>
<td>8</td>
<td>95.849</td>
</tr>
<tr>
<td>3</td>
<td>NACA 2304</td>
<td>12</td>
<td>2</td>
<td>4</td>
<td>84.695</td>
</tr>
<tr>
<td>4</td>
<td>NACA 3304</td>
<td>12</td>
<td>3</td>
<td>4</td>
<td>86.992</td>
</tr>
</tbody>
</table>
Joukowski Airfoil in Standard Earth Atmosphere
Pressure = 101.213kPa, Temperature = 14°C, Density = 1.223kg/m³
Camber = 2.0 % chord, Thickness = 4.0 % chord, Angle of attack = 12° degrees.
Upper Surface
Lower Surface
X Y P V X Y P V
-0.489 0.105 84.695 601 -0.489 0.105 84.695 601
-0.478 0.11 96.618 331 -0.487 0.098 96.576 217
-0.454 0.113 98.959 245 -0.47 0.096 98.917 142
-0.417 0.112 99.749 208 -0.439 0.08 101.791 15
-0.368 0.109 100.371 174 -0.337 0.078 101.725 40
-0.31 0.102 100.841 142 -0.269 0.069 101.597 65
-0.091 0.061 100.755 148 -0.108 0.017 101.528 76
0.071 0.021 100.917 136 0.069 0.012 101.486 81
0.151 0.0 100.985 131 0.156 -0.028 101.469 84
0.227 -0.022 101.046 125 0.239 -0.044 101.453 86
0.297 -0.044 101.104 121 0.313 -0.06 101.436 89
0.359 -0.063 101.156 116 0.377 -0.079 101.418 90
0.411 -0.073 101.203 112 0.427 -0.097 101.388 92
0.451 -0.092 101.246 108 0.463 -0.106 101.375 95
0.477 -0.1 101.294 104 0.484 -0.102 101.349 96
0.488 -0.103 101.494 80 0.488 -0.103 101.494 80

As indicated in Table 5.2, it is clear that for the required boundary conditions, the selected airfoil NACA 2304 produces the maximum coefficient of lift for 12° angle of attack.

The specifications of NACA 2304 are as follows:

- **Maximum camber**: 2% of chord length.
- **Maximum thickness position**: 30% of chord length from the leading edge
- **Maximum thickness**: 04% of chord length.
- **Chord**: 100 mm
- **Span**: 250 mm
The influence of chord length on the coefficient of lift for an airfoil is negligible. Owing to space constraints, the chord length was selected as 100 mm and the span of the airfoil was selected as 250 mm so that the airfoil could be accommodated in the wind tunnel section (300 mm X 300 mm), available in the laboratory of the researcher.

5.6 Theoretical Attainment of Higher RH

First, the venturi section and suction tube were designed. Next, the RH value was computed. In order to attain the higher RH, three trials of design were carried out. Each trial of design began by assuming an appropriate pressure at the throat of the venturi section. The computation details under each trial are presented in the following section. The exploded view of the venturi section and section tube arrangement is shown in Figure 5.5.

![Venturi section and suction tube arrangement](image)

Figure 5.5 Venturi section and suction tube arrangement

The design details are presented below:

5.6.1 Design of Venturi Section and Suction Tube: Trial 1

Design of Venturi Section

Pressure assumed at the throat, $P_2 = 99$ kPa.

Pressure at the inlet of venturi section, $P_3 = 101.3$ kPa

From blower specifications,
Velocity of air at the inlet of venturi section, $V_3 = 30 \text{ m/s}$

Density of air $\rho_{\text{air}} = 1.12 \text{ kg/m}^3$

As sections 2 and 3 lie at the same datum, $Z_2 = Z_3$

Substituting in Bernoulli’s equation,

$$P_3/\rho g + V_3^2/2g + Z_3 = P_2/\rho g + V_2^2/2g + Z_2$$

Velocity of air at throat section, $V_2 = 70.76 \text{ m/s}$

Assuming diameter of venturi inlet, $D_3 = 25.4 \text{ mm (1")}$

Cross sectional area of venturi inlet, $A_3 = 5.067 \times 10^{-4} \text{ m}^2$

Now discharge of air, $Q_{\text{air}} = A_3 \times V_3 = 0.0152 \text{ m}^3/\text{s}$

$$= 0.0152 \times 10^6 \text{ cc/s}$$

The throat diameter $D_2$, is found as

$$\frac{\pi}{4} \times D_2^2 \times 70.76 = 0.0152$$

$$D_2 = 16.54 \text{ mm}$$

**Design of Suction Tube**

Pressure at the surface of the reservoir, $P_1 = 101.3 \text{ kPa}$

Pressure at the throat, $P_2 = 99 \text{ kPa}$

Head available, $H = (P_1 - P_2) / (\rho g)$

$$= ((101.3 - 99) \times 1000) / (1000 \times 9.81)$$

$$= 0.2345 \text{ m of water}$$

Considering 20% Friction Losses,

Effective head available after friction losses $H_{\text{eff}}$

$$(\text{or}) \text{ Length of the suction tube, } l = 0.8 \times 0.2345 = 0.1876 \text{ m of water}$$

The frictional losses associated with a fluid flow in a pipe of diameter ‘d’ and length ‘l’ is given by

$$h_{\text{fl}} = 4f\nu^2/2gd$$

Where, ‘f’ is the coefficient of friction.
So, greater the diameter, lesser is the frictional loss. These losses are minimal when the diameter is greater than or equal to 9 mm (Fox and Mc Donald, 2001). Hence, the length and diameter of the suction pipe are assumed as 187.6 x 9 mm.

**Determination of Discharge of Water**

Applying Bernoulli's equation between section 1 and 2,

\[ \frac{P_1}{\rho g} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + Z_2 \]

Pressure at the surface of the reservoir, \( P_1 \) = 101.3 kPa
Velocity of the still water, \( V_1 \) = 0 m/s
Pressure at the throat, \( P_2 \) = 99 kPa
Effective Head, \( H_{\text{eff}} = Z_2 - Z_1 \) = 0.1876 m
Density of water, \( \rho_{\text{water}} \) = 1000 kg/m³

Substituting in Bernoulli’s equation,

Velocity of water at the throat, \( V_2 \) = 0.958 m/s

To find the discharge of water,

Suction tube diameter, \( d_2 \) = 9 mm.
Area = \( \frac{\pi}{4} \times d_2^2 \) = 6.362 x 10⁻⁵ m²
Discharge of water, \( Q_{\text{water}} \) = 6.095 x 10⁻⁵ m³/s
So amount of water added = \( \rho Q_{\text{water}} \) = 10⁶ x 6.095 x 10⁻⁵
= 60.95 g/s

**Determination of RH**

\[ \text{RH} = \frac{\text{Amount of moisture present in unit volume of dry air}}{\text{Amount of moisture needed to saturate it at the same temperature}} \]

Discharge of air, \( Q_{\text{air}} \) = 0.0152 x 10⁶ cc/s
Amount of water added to dry air = 60.95 g/s
For a saturated vapour,

For a room condition of 60% RH and temperature 30 °C,

\[ \text{Amount of moisture in 1 cc of dry air at 30 °C} = 0.118 \text{ g} \]

(Ananthanarayanan, 2001)

Assuming 60% RH of air at room condition,

Amount of moisture actually present in
1 cc of dry air before section 2 \( = 0.6 \times 0.118 \) \( = 0.0708 \text{ g/cc.} \)

Amount of water added in 1 cc of dry air at the section 2 = (Amount of water added per second) / (Volume of dry air absorbed by the venturi section per second)

\[ = \frac{60.95}{(0.0152 \times 10^6)} \]
\[ = 4.009 \times 10^{-3} \text{ g/cc} \]

Amount of moisture present in 1 cc
of dry air after section 2 \( = 0.0708 + 4.009 \times 10^{-3} = 0.07481 \text{ g/cc.} \)

\[ \text{RH available after section 2} \]
\[ = \frac{0.07481}{0.118} \]
\[ = 0.6339 \]
\[ = 63.39\% \]

So, an increase of 3.39% in RH is achieved at a pressure of 99 kPa at throat section. To increase the RH further, the pressure at throat section has to be reduced which was done in subsequent trials.

5.6.2 Design of Venturi Section and Suction Tube: Trial 2

Design of Venturi Section

Pressure assumed at the throat, \( P_2 = 98 \text{ kPa.} \)
Pressure at the inlet of venturi, \( P_3 = 101.3 \text{ kPa} \)
From blower specifications,

Velocity of air at the inlet of venturi section, \( V_3 = 30 \text{ m/s} \)

Density of air \( \rho_{\text{air}} = 1.12 \text{ kg/m}^3 \)

As sections 2 and 3 lie at the same datum, \( Z_2 = Z_3 \)

Substituting in Bernoulli's equation,

\[
P_3/\rho g + V_3^2/2g + Z_3 = P_2/\rho g + V_2^2/2g + Z_2
\]

Velocity of air at throat section, \( V_2 = 82.41 \text{ m/s} \)

Assuming diameter of venturi inlet, \( D_3 = 25.4 \text{ mm (1")} \)

Cross sectional area of venturi inlet, \( A_3 = 5.067 \times 10^{-4} \text{ m}^2 \)

Now discharge of air, \( Q_{\text{air}} = A_3xV_3 = 0.0152 \text{ m}^3/\text{s} \)

\[= 0.0152 \times 10^6 \text{ cc/s} \]

The throat diameter \( D_2 \), is found as

\[
\pi / 4 \times D_2^2 \times 82.41 = 0.0152
\]

\[D_2 = 15.33 \text{ mm} \]

**Design of Suction Tube**

Pressure at the surface of the reservoir, \( P_1 = 101.3 \text{ kPa} \)

Pressure at the throat, \( P_2 = 98 \text{ kPa} \)

Head available, \( H = (P_1-P_2) / (\rho g) \)

\[= ((101.3 - 98) \times 1000) / (1000 \times 9.81) \]

\[= 0.34 \text{ m of water} \]

Considering 20% Friction Losses,

Effective head available after friction losses \( H_{\text{eff}} \)

(or) Length of the suction tube, \( l \)

\[= 0.8 \times 0.34 = 0.27 \text{ m of water} \]

Hence, the length and diameter of the suction pipe are assumed as 270 x 9 mm.
Determination of Discharge of Water

Applying Bernoulli's equation between section 1 and 2,

$$\frac{P_i}{\rho g} + \frac{V_i^2}{2g} + Z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + Z_2$$

Pressure at the surface of the reservoir, $P_1 = 101.3$ kPa

Velocity of the still water, $V_1 = 0$ m/s

Pressure at the throat, $P_2 = 98$ kPa

Effective Head, $H_{eff} = Z_2 - Z_1 = 0.27$ m

Density of water, $\rho_{water} = 1000$ kg/m$^3$

Substituting in Bernoulli's equation,

Velocity of water at the throat, $V_2 = 1.3$ m/s

To find the discharge of water,

Suction tube diameter, $d_2 = 9$ mm.

Area $= \left(\frac{\pi}{4}\right) \times d_2^2 = 6.362 \times 10^{-5}$ m$^2$

Discharge of water, $Q_{water} = 8.27 \times 10^{-5}$ m$^3$/s

So, amount of water added $= \rho Q_{water} = 10^6 \times 8.27 \times 10^{-5}$

$= 82.702$ g/s

Determination of RH

$$RH = \frac{\text{Amount of moisture present in unit volume of dry air}}{\text{Amount of moisture needed to saturate it at the same temperature}}$$

Discharge of air, $Q_{air} = 0.0152 \times 10^6$ cc/s

Amount of water added to dry air $= 82.7$ g/s

For a saturated vapour,

For a room condition of 60% RH and temperature 30°C.
Amount of moisture in 1 cc of dry air at 30°C = 0.118 g

Assuming 60% relative humidity of air at room condition,

Amount of moisture actually present in
1 cc of dry air before section 2 = 0.6 × 0.118
= 0.0708 g/cc.

Amount of water added in 1 cc of dry air at the section 2 = \(\frac{\text{Amount of water added per second}}{\text{Volume of dry air absorbed by the venturi section per second}}\)
= 82.7 / (0.0152 × 10^6)
= 5.441 × 10^3 g/cc

Amount of moisture present in
1 cc of dry air after section 2 = 0.0708 + 5.441 × 10^3
= 0.07624 g/cc.

RH after section 2 = \(\frac{0.07624}{0.118}\)
= 0.6461
= 64.61%

So, an increase of 4.61% in RH is achieved at a pressure of 98 kPa at throat section. To increase the RH further, the pressure at the throat section has to be reduced which was done in the subsequent trial.

5.6.3 Design of Venturi Section and Suction Tube: Trial 3

Design of Venturi Section

Pressure assumed at the throat, \(P_2\) = 97 kPa.
Pressure at the inlet of venturi, \(P_3\) = 101.3 kPa

From blower specifications,

Velocity of air at the inlet of venturi section, \(V_3\) = 30 m/s
Density of air, \(\rho_{\text{air}}\) = 1.12 kg/m^3
As sections 2 and 3 lie at the same datum, \( Z_2 = Z_3 \)

Substituting in Bernoulli’s equation,

\[
P_2/\rho g + V_2^2/2g + Z_2 = P_2/\rho g + V_2^2/2g + Z_3
\]

Velocity of air at throat section, \( V_2 = 92.62 \text{ m/s} \)

Assuming diameter of venturi inlet \( D_3 = 25.4 \text{ mm (1")} \)

Cross sectional area of venturi inlet, \( A_3 = 5.067 \times 10^{-4} \text{ m}^2 \)

Now discharge of air, \( Q_{\text{air}} = A_3xV_3 = 0.0152 \text{ m}^3/\text{s} \)
\[= 0.0152 \times 10^6 \text{cc/s} \]

The throat diameter \( D_2 \), is found as

\[
\frac{\pi}{4} \times D_2^2 \times 92.62 = 0.0152
\]

\[D_2 = 14.45 \text{ mm} \]

**Design of Suction Tube**

Pressure at the surface of the reservoir, \( P_1 = 101.3 \text{ kPa} \)

Pressure at the throat, \( P_2 = 97 \text{ kPa} \)

Head available, \( H = (P_1 - P_2) / (\rho g) \)
\[= (101.3 - 97) \times 1000 / (1000 \times 9.81) \]
\[= 0.438 \text{ m of water} \]

Considering 20% Friction Losses,

Effective head available after friction losses, \( H_{\text{eff}} \)

(or) Length of the suction tube, \( l \)
\[= 0.8 \times 0.438 = 0.35 \text{ m of water} \]

The longer the tube, the lesser will be the pressure in which water comes out of the tube. To ensure that water comes out of the tube at sufficiently high pressure, length obtained in previous trial (where length of the suction tube is 270mm for the pressure, \( P_2 \) of 98 kPa) itself is maintained. Hence, the length and diameter of the suction pipe are assumed as 270 x 9 mm.
Determination of Discharge of Water

Applying Bernoulli's equation between section 1 and 2,

\[ P_1/\rho g + V_1^2/2g + Z_1 = P_2/\rho g + V_2^2/2g + Z_2 \]

Pressure at the surface of the reservoir, \( P_1 = 101.3 \) kPa
Velocity of the still water, \( V_1 = 0 \) m/s
Pressure at the throat, \( P_2 = 97 \) kPa
Effective Head, \( H_{\text{eff}} = Z_2 - Z_1 \) = 0.27 m
Density of water, \( \rho_{\text{water}} = 1000 \) kg/m\(^3\)

Substituting in Bernoulli's equation,

Velocity of water at the throat, \( V_2 = 1.82 \) m/s

To find the discharge of water,

Suction tube diameter, \( d_2 = 9 \) mm.

Area = \((\pi / 4) \times d_2^2\) = \(6.362 \times 10^{-5}\) m\(^2\)

Discharge of water, \( Q_{\text{water}} = 1.157 \times 10^{-4}\) m\(^3\)/s

So, amount of water added = \( \rho Q_{\text{water}}\)
= \(10^6 \times 1.157 \times 10^{-4}\)
= 115.76 g/s

Determination of RH

\[ \text{RH} = \frac{\text{Amount of moisture present in unit volume of dry air}}{\text{Amount of moisture needed to saturate it at the same temperature}} \]

Discharge of air, \( Q_{\text{air}} = 0.0152 \times 10^6 \) cc/s

Amount of water added to dry air = 115.7 g/s

For a saturated vapour,

For a room condition of 60% RH and temperature 30°C.

Amount of moisture in 1cc of dry air at 30°C = 0.118 g
Assuming 60% relative humidity of air at room condition,

Amount of moisture actually present in 1 cc of dry air before section 2 = 0.6 \times 0.11 = 0.0708 \text{ g/cc}.

Amount of water added in 1 cc of dry air at the section 2 = \frac{\text{Amount of water added in per second}}{\text{Volume of dry air absorbed by the venturi section per second}}

= \frac{115.7}{(0.0152 \times 10^6)} = 7.578 \times 10^{-3} \text{ g/cc}

Amount of moisture present in 1 cc of dry air after section 2

= 0.0708 + 7.578 \times 10^{-3}

= 0.07837 \text{ g/cc}.

RH after section 2

= \frac{0.07837}{0.118}

= 0.6642

= 66.42\%

Hence, an increase of 6.42% in RH is achieved at a pressure of 97 kPa at the throat section. The RH values obtained in the above three trials for the three different throat pressures are presented in Table 5.3.

<table>
<thead>
<tr>
<th>Trials</th>
<th>Throat Pressure $P_2$ in kPa</th>
<th>RH in %</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>99</td>
<td>63.39</td>
</tr>
<tr>
<td>2</td>
<td>98</td>
<td>64.61</td>
</tr>
<tr>
<td>3</td>
<td>97</td>
<td>66.42</td>
</tr>
</tbody>
</table>
As shown, the higher RH of 66.42% was obtained for the throat pressure of 97 kPa.

5.7 Experimentation

In order to test the theoretical design, a setup was fabricated for experimenting with the humidifier using aerodynamic principles. However, a perfect fabrication of the required setup was not possible due to the lack of funding support. Hence the setup had to be diluted in many aspects. The airfoil is fabricated using many materials like titanium alloy and duralumin. Keeping this cost factor in mind, the airfoil was fabricated using teak wood during this experimentation. A good surface finish was achieved to enhance the accuracy of the results. The venturi section was fabricated using C40 Steel. The stock was machined to obtain the venturi section dimensions. A commercial copper tubing of 3 mm was used. The entire setup was then assembled in a test section of a wind tunnel. Many of the design parameters could not be adhered to because of the reasons mentioned above. For example, the diameter of the suction tube chosen was 3 mm against the actual requirement of 9 mm. The experimentation was carried out under these diluted conditions.

5.7.1 Wind Tunnel Experimentation

The entire fabricated setup was assembled in the test section of wind tunnel. Using a copper tube, the reservoir containing water was connected to the throat of venturi section. The airfoil was kept inside the test section. Some of the photographs of the wind tunnel and the arrangement of the setup experimental are shown in Figures 5.6 and 5.7.
The specifications of the experimental setup are presented below:

<table>
<thead>
<tr>
<th>Type</th>
<th>Low speed open circuit suction type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test section</td>
<td>300mm X 300mm</td>
</tr>
<tr>
<td>Wind velocity</td>
<td>30m/s</td>
</tr>
<tr>
<td>Drive</td>
<td>Axial flow driven DC Motor (3 HP)</td>
</tr>
</tbody>
</table>

Once the setup was made ready, the experiment was conducted. The blower was switched on and it was allowed to become stabilized. The blower speed was maintained at 30 m/s. The inlet air conditions including DBT and WBT were recorded just before air enters the test section using a Psychrometer.
However, the results obtained did not coincide with that projected through theoretical calculation. Particularly, the increase in RH value was not observable. This might be due to the dilution of the fabricated setup from the actual requirement. At one point of time, the author of the paper Bhat (2003), Mr. S.K. Bhat was approached through email. He has responded that, he had not attempted the experimentation. At this juncture, it was decided to conduct the experimentation by adopting the simulation technique.

5.8 Simulation

To begin with, the coordinates for the selected airfoil using FOILSIM software were generated using the WINFOIL V 2.2 software. These details are shown in Figure 5.8. As shown, for generating the coordinates, certain parameters like camber, camber position, the thickness of the airfoil and the number of coordinates required were used in the modeling of the setup in Gambit software. Any airfoil designed using winfoil is stored as coordinates in the database. These coordinates are saved as a dat file, which is used in the modeling of the airfoil using GAMBIT software. The model created using Gambit software was subjected to simulation using CFD software. The steps followed during this process are described in the following subsections.

5.8.1 Creation of Far Field Boundary

As the flow over the airfoil was of external type, a farfield boundary was defined and the region between the airfoil geometry and the farfield boundary had to be meshed. It was preferred to place the farfield boundary well away from the airfoil since the ambient conditions were used to define the boundary conditions at the farfield.
5.8.2 Meshing the Far Field Boundary and the Setup Model

The setup modeled using Gambit software was meshed fully before analysis. This involved two stages. First, every edge in the setup was meshed using Gambit. Then the faces were created which was followed by face meshing. Triangular mesh elements were used for the analysis because any model involving complex curves like airfoil could be fully meshed only by employing these elements. The meshed model was then imported to Fluent.

5.8.3 Loading the Boundary Condition

The boundary conditions like pressure and velocity of air from the blower were given as inputs to the FLUENT solver. The analysis of the airfoil was carried out for the following boundary conditions.

- Velocity of air \(= 30 \text{ m/s} = 108 \text{ km/h}\)
- Pressure of air \(= 101.13 \text{ kPa}\)
- Angle of attack \(= 12 \text{ deg}\)
- Temperature of air \(= 30 \text{ °C}\)
The data regarding velocity of air depended on the specifications of the blower used in the wind tunnel. Since it was difficult to model the airfoil tilted at 12 degrees, it was modeled horizontally and the angle at which the air strikes the airfoil is set to 12 degrees, which produced the same effect.

5.8.4 Solving in Fluent Solver

Once the boundary conditions were applied, the problem could be solved easily using the fluent solver. The convergence of the solution is important in any analysis. A numerical method is said to be convergent if the solution obtained tends to the exact solution as the grid spacing tends to zero. Therefore, convergence is usually checked by repeating the calculation on a series of successively refined grids. In this simulation study, the refinement was done till 0.01. The number of elements in the mesh reached about 50,000 on an average. The analysis of this finer meshed model using a P4 processor with 2 GB RAM capacities, consumed 2 hours.

5.8.5 Confirmation of Results

The results could be obtained in different forms like the pressure distribution along the airfoil and velocity contours in the venturi section. These results could be compared and verified using the results obtained theoretically. The results obtained were found to be closer to the theoretical results obtained earlier. The simulated results of the trial 3 are depicted in Figures 5.9 and 5.10.
Figure 5.9. Contours of static pressure at throat section

From the Figure 5.9, it is clear that the pressure at the throat of the venturi section lies between $-5.5 \times 10^3$ Pa and $-3.3 \times 10^3$ Pa which is approximately in line with the results obtained theoretically. The corresponding absolute pressures for this range are 95.8 kPa and 98 kPa respectively, which is closer to the theoretical result of 97 kPa obtained earlier.

Figure 5.10. Contours of velocity at throat section
As shown in Figure 5.10, velocity of air at the throat section lies between 90 and 95 m/s, which also confirm the result of 92.62 m/s obtained theoretically.

5.9 Conclusion

The module of the doctoral work presented in this chapter has opened up, an avenue for increasing RH in industrial atmosphere using the aerodynamic principles. The proposed humidifier device using aerodynamic principle enjoys several merits when compared with those of the conventional devices. As this device does not contain any advanced technology and mechanism, it is easier to fabricate and maintain it. The other highlights of this module of doctoral work are enumerated below:

- Practically feasible and efficient device has been developed to replace the conventional cooler and humidifier set up.
- The RH of the model can be increased to a maximum of 6.42% from the RH at the room condition.
- The degree of humidifying can be easily controlled to get the required degree of comfort.
- This adopted device reduces the number of moving parts and hence results in lesser wear and tear.

The experiences of conducting this module of the doctoral work hinted that the help of practitioners with commercial background is necessary to successfully fabricate the proposed device and depicts out for real time applications. Usage of this device in industrial sectors requiring high humidity levels will reduce the expenditure on energy because the operation of this device results in energy savings.