Demonstrative Unit of Dew Point Evaporative Cooling System

Authors

Asad Ali

10-NUST-SMME-BE-ME-25

Karamullah

10-NUST-SMME-BE-ME-41

Danyal Adil Khan

10-NUST-SMME-BE-ME-31

Supervisor: Lec. Waqas Khalid

Co-Advisor: Dr. Adeel Waqas

DEPARTMENT OF MECHANICAL ENGINEERING

SCHOOL OF MECHANICAL & MANUFACTURING ENGINEERING

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Authors

Asad Ali

10-NUST-SMME-BE-ME-25

Karamullah Rasheed

10-NUST-SMME-BE-ME-41

Danyal Adil Khan

10-NUST-SMME-BE-ME-31

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Thesis Supervisor:

Lec. Waqas Khalid

Thesis Supervisor's Signature:___________________________________

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Declaration

We certify that this research work titled "***Demonstrative unit of Dew point evaporative cooling system***" is our own work. The work has not been presented elsewhere for assessment. The material that has been used from other sources is acknowledged and has been properly referred in the appendices of this report.

Signature of Students

Asad Ali
10-NUST-SMME-BE-ME-25

Karamullah
10-NUST-SMME-BE-ME-41

Danyal Adil Khan
10-NUST-SMME-BE-ME-31

Language Correctness Certificate

This thesis is based on the format which has been provided by the university. It has been assessed by an English Language expert and it is free of typing, syntax, semantic, grammatical and spelling mistakes.

Signature of Students

Asad Ali
10-NUST-SMME-BE-ME-25

Karamullah Rasheed
10-NUST-SMME-BE-ME-48

Danyal Adil Khan
10-NUST-SMME-BE-ME-31

Signature of Supervisor

Lec. Waqas Khalid

Copyright Statement

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Abstract

Dew point evaporative cooling system is an alternative to vapor compression air conditioning system for sensible cooling of air. Project is based on theoretical and experimental study of dew point evaporative cooling system. There are two systems for evaporative cooling; direct and indirect systems. This demonstrative unit will be based on indirect evaporative cooling system because it provides only sensible cooling to the process air keeping humidity constant. Indirect evaporative cooling system uses two streams of ambient air in the process that is dry channel for intake ambient air and wet channel of secondary air separated by thin-film cotton wall to prevent moisture penetration. Along the flow path, the intake air loses sensible heat to the wet side for water evaporation and the secondary air is cooled in direct contact with water. The outlet temperature can be decreased theoretically close to ambient dew point temperature. In this way the wet bulb and dew point effectiveness and coefficient of performance of the system increases.

The design considerations are summarized as follows;

- For maximum heat transfer and mass flow for all dry and wet channels through exchanger using counter flow arrangements.
- Corrosion avoidance by using water in the cooling process.
- Evenly saturation of all wet surfaces in the wet channel by using a reservoir.
- Extraction of certain fraction of outlet air as working air in the cooling process.

Key Words: *Dew point evaporative cooling, wet bulb, dew point effectiveness, humidity, vapor compression, air conditioning, ventilation, moisture, cotton wall, primary air, secondary air, dew point temperature*

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CHAPTER 1: INTRODUCTION

1.1 Background

Due to rapid increase in population, natural resources are depleting exponentially. To meet the energy requirements for the world's population, technology is being shifted towards renewable energy. Major energy consumption for a building is for heating and cooling. In developed countries, the energy consumption in heating ventilation and air-conditioning (HVAC) systems has accounted for 50% of the energy consumption in buildings and 20% of total energy consumption. In some developing countries, such as China, due to the facts of poor insulation, inefficient HVAC systems and inefficient transformation of energy to heat, the energy consumption of HVAC accounted for 50-70% of energy consumption in buildings. In the HVAC systems, air conditioning proportion has grown rapidly due to global warming, improved insulation of buildings, and increased indoor facilities.[1]

In evaporative cooling, air is cooled by evaporation of water. Currently two types of evaporative cooling systems are available in the market i.e. direct evaporative cooling systems and indirect evaporative cooling systems. Both of them have some limitations. Direct evaporative cooling adds moisture to air and indirect evaporative cooling has low wet-bulb effectiveness and theoretically it can only approach to wet-bulb temperature of the incoming air. To overcome these two limitations, dew point evaporative cooling systems can be used. A dew point evaporative cooling system can sensibly cool and supply the air below the wet bulb temperature of ambient without humidity increase. In the psychometric path of indirect evaporative cooling system, the outlet air leaving the dry channel of the system has essentially lower dry bulb and wet bulb temperature than the intake air (ambient). Therefore, some fraction of this outlet air would be used as the working air for the wet channel. In an ideal process, the outlet air temperature can be reduced theoretically toward the dew point of intake air (ambient). This leads to the possibility of the wet bulb effectiveness obtained which can be greater than 100%.

1.2 Project Objective

Main objective of our project was to develop an efficient cooling system for ventilation air. Conventional cooling systems have problem of CFCs emissions and high energy consumptions. To meeting energy requirements, cooling system should have high COP and to keep the environment clean it should have low CO_2 emissions. This project is based on the purpose of designing and fabricating a cooling system which causes no CO_2 emissions and has high COP as compared to conventional units.

Evaporative cooling is a type of air cooling which causes no harm to environment i.e. no CO_2 and CFCs emissions. It sensibly cools air by using latent heat of water evaporation. Dew point evaporative cooling system is indirect evaporative cooling which can cool air below its wet bulb temperature. We aimed development of demonstrative unit of dew point evaporative cooling to find the responses of unit at different input conditions.

1.3 Project Methodology

Project was started with literature review of evaporative cooling, types of evaporative cooling, dew point evaporative cooling, heat and mass transfer, numerical and analytical methods for equations. Then design and simulation for heat and mass exchanger was build. Matlab solver, BVP4C was used for solving heat and mass equations. After solving the equations, relationships between different input parameters were plotted and optimum design was selected. After which we developed prototype using aluminum foil, cotton sheet and sealant. In the end experimental and theoretical results were compared.

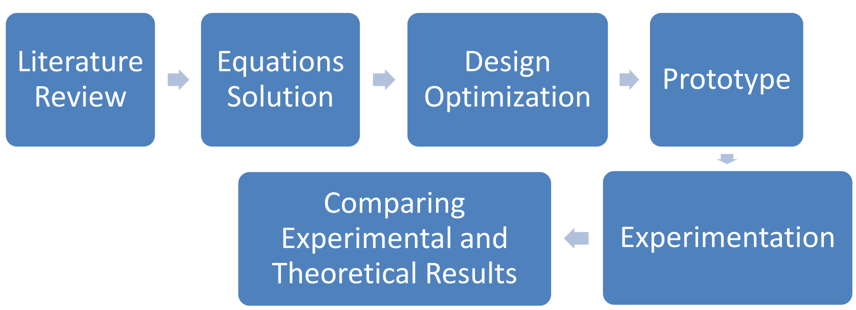


Figure 1.1 Methodology

CHAPTER 2: LITERATURE REVIEW

2.1 Background:

Compared with the popular conventional vapor-compression and absorption refrigeration, the evaporative cooling has a few significant advantages cover:

- large capacity of energy saving and carbon dioxide emission reduction because no energy intensive compressors exist in evaporative cooling systems
- More environmental friendly cooling technology because only water participate in the cooling process rather than pollutant refrigerants
- More simple in terms of structure, construction, and control strategies

As two major types of evaporative cooling, direct evaporative cooling (DEC) adds moisture to room air, which causes discomfort and health problems. Indirect evaporative cooling (IEC) lowers air temperature and avoids adding moisture to the air, but it limits the temperature of supply air to some degrees (2 to 5^{o}C) above the wet bulb of the outdoor air (usually 40-60% wet-bulb effectiveness), which is too high to perform air conditioning of buildings.[1] To solve this problem dew point evaporative cooling is used.

2.2 Vapor Compression Air-Conditioning:

The vapor-compression uses a circulating liquid refrigerant as the medium which absorbs and removes heat from the space to be cooled and subsequently rejects that heat elsewhere. Figure 2.1[3] depicts a typical, single-stage vapor-compression system. All such systems have four components: a compressor, a condenser, a thermal expansion valve (also called a throttle valve or metering device), and an evaporator. Circulating refrigerant enters the compressor in the thermodynamic state known as a saturated vapor and is compressed to a higher pressure, resulting in a higher temperature as well. The hot, compressed vapor is then in the thermodynamic state known as a superheated vapor and it is at a temperature and pressure at which it can be condensed with either cooling water or cooling air. That hot vapor is routed through a condenser where it is cooled and condensed into a liquid by flowing through a coil or tubes with cool water or cool air flowing across the coil or tubes. This is where the circulating refrigerant rejects heat from the system and the rejected heat is carried away by either the water or the air (whichever may be the case).

The condensed liquid refrigerant, in the thermodynamic state known as a saturated liquid, is next routed through an expansion valve where it undergoes an abrupt reduction in pressure. That pressure reduction results in the adiabatic flash evaporation of a part of the liquid refrigerant. The auto-refrigeration effect of the adiabatic flash evaporation lowers the temperature of the liquid and vapor refrigerant mixture to where it is colder than the temperature of the enclosed space to be refrigerated.

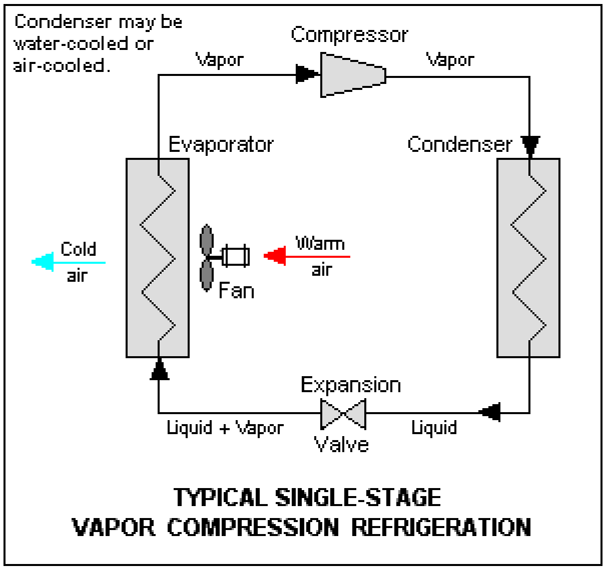


Figure 2.1 Vapor Compression Cycle

The cold mixture is then routed through the coil or tubes in the evaporator. A fan circulates the warm air in the enclosed space across the coil or tubes carrying the cold refrigerant liquid and vapor mixture. That warm air evaporates the liquid part of the cold refrigerant mixture. At the same time, the circulating air is cooled and thus lowers the temperature of the enclosed space to the desired temperature. The evaporator is where the circulating refrigerant absorbs and removes heat which is subsequently rejected in the condenser and transferred elsewhere by the water or air used in the condenser. To complete the refrigeration cycle, the refrigerant vapor from the evaporator is again a saturated vapor and is routed back into the compressor. The major energy consumption for vapor compression cycle is for driving compressor.

2.3 Working Principles of Evaporative Cooling Systems:

Evaporative cooling is responsible for the chill you feel when a breeze strikes your skin—-the air evaporates the water on your skin, with your body heat providing the energy. The ancient Egyptians hung wet mats in their doors and windows, and wind blowing through the mats cooled the air—-the first attempt at air conditioning. This basic idea was refined through the centuries: mechanical fans to provide air movement in the 16th century, cooling towers with fans that blew water-cooled air inside factories in the early 19th century, swamp coolers in the

20th century.[2] Evaporative cooling is economical, effective, environmentally friendly, and healthy. ASHRAE standard 55 for comfort conditions in summer recommends 25 °C and 50–60% relative humidity (around 10–12 g/kg humidity ratio) [5]. Since, air temperature and humidity are the two major parameters affecting thermal comfort significantly, and only sensible load can be handled by an evaporative cooling system, conventional evaporative cooling system is suitable for dry and temperate climate where the humidity is low.[4]

2.3.1 Direct Evaporative cooling system

In direct evaporative cooling system, water and product air are in direct contact with each other i.e. moisture is added to the product air. The working diagram of direct evaporative cooling and the working process representing on the psychometric chart are shown in Figure. 2.2[4]

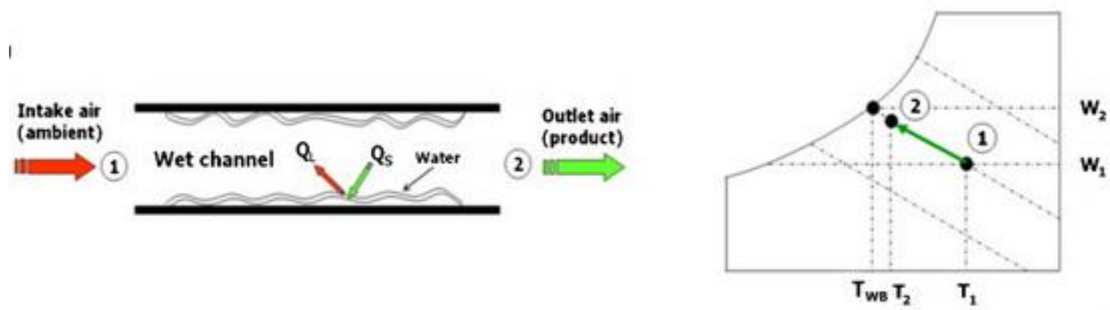


Figure 2.2 Direct Evaporative Cooling

Inlet air (state 1) enters into the wet channel and exchanges the heat and moisture with water. As water is continuously evaporated into the airstream along the wet channel, consequently, the dry bulb temperature of entering air has been reduced with rising moisture content (state 2). In the process, the sensible heat of air decreased with latent heat increases due to the evaporation of water. Theoretically, the thermodynamic process of direct evaporative cooling is limited by the wet- bulb temperature of entering air. The wet-bulb effectiveness of current direct evaporative cooling systems is ranged from 70% to 95% depending on the configurations and the air velocity of passing the evaporative medium. This type of systems has been widely used for the reasons of simple structure, cheap initial and operating costs.[1]

2.3.2 Indirect Evaporative cooling system

Indirect evaporative cooling (IEC) systems can lower air temperature without adding moisture into the air, making them the more attractive option over the direct ones. In an indirect evaporative air cooling system, the primary (product) air passes over the dry side of a plate, and the secondary (working) air passes over the opposite wet side. The wet side air absorbs heat from the dry side air with aid of water evaporation on the wet surface of the plate and thus cools the dry side air; while the latent heat of the vaporized water is transmitted into the working air in the wet side[6]. The working diagram of indirect evaporative cooling and the working process representing on the psychometric chart are shown in Figure. 2.2[4]

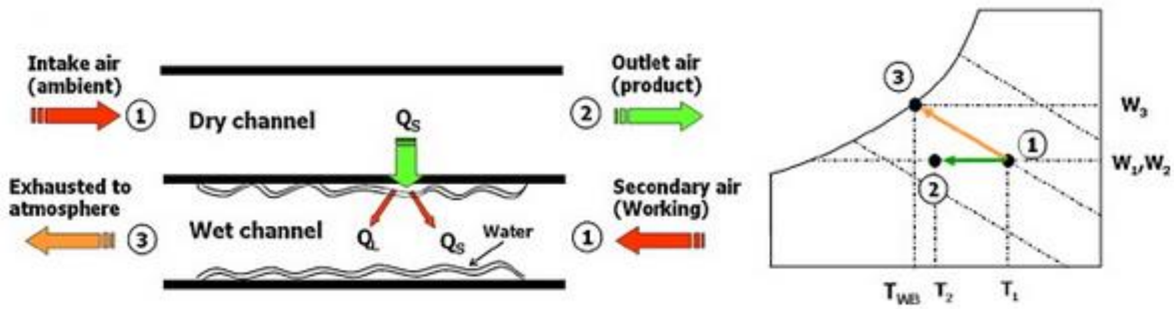


Figure 2.3 Indirect Evaporative Cooling

The wet-bulb effectiveness of current typical indirect evaporative cooling systems may range between 55% and 75% or higher, which is lower than that of direct evaporative cooling (ASHRAE 1996). However, due to no moisture addition to the supply air, the IEC system doesn't have to supply the same cooling with the DEC system to provide the same comfort feeling[1].

2.4 Dew Point Evaporative Cooling System

A dew point evaporative cooling system can be used to supply the outlet air temperature below wet bulb temperature of ambient air. In dew point evaporative cooling, there are two types of channels for air, one is dry channel and the other is wet channel. In wet channel, some porous material is applied to both sides on which water is sprayed. Wet surface should be evenly saturated all the time. Walls of the channels should be of small thickness such that their thermal resistance can be neglected. Intake air or primary air passes through the dry channel and loses its sensible heat to wet side. The air at the end of dry channel must have lower wet and dry bulb temperature, as shown in figure 2.4[4], a part of it is extracted and directed towards the wet channel in which it gains the latent and sensible heat of water and exhausted to the air. With time the temperature of primary air leaving dry channel approaches its dew point temperature.

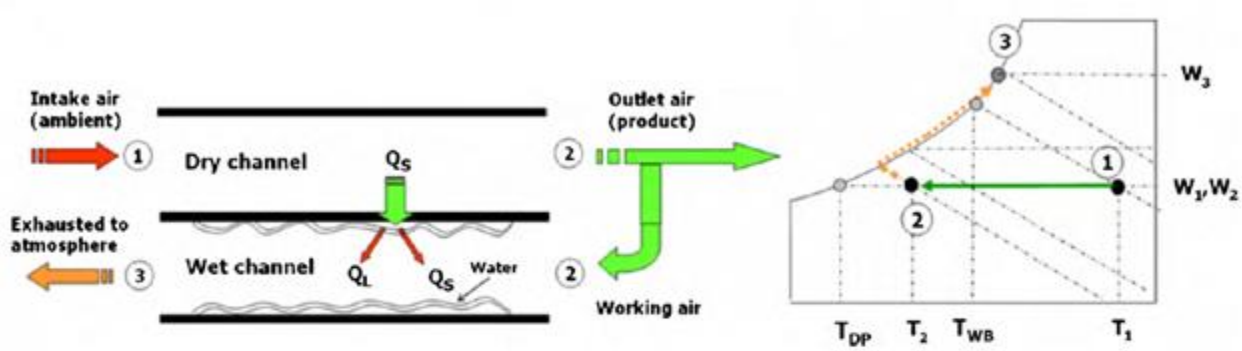


Figure 2.4 Dew Point Evaporative Cooling

A dew point evaporative cooling system can sensibly cool and supply the air below the wet bulb temperature of ambient without humidity increase. Therefore, the system effectiveness can be improved compared to conventional systems

2.5 Wet bulb/Dew Point Effectiveness

The wet bulb effectiveness is defined as the ratio of the difference between intake and outlet air temperature to the difference between intake air temperature and its wet bulb temperature, which is expressed in the following equation:

$$\varepsilon_{\mathrm{wb}} = \frac{(t_{\mathrm{a,in}} - t_{\mathrm{a,out}})}{(t_{\mathrm{a,in}} - t_{\mathrm{wb,in}})}$$

Since the dew point evaporative cooling system can sensibly cool the outlet air temperature below the ambient wet bulb temperature, the dew point effectiveness is used to indicate (a) its cooling performance compared to its theoretical limitation, and (b) how close the outlet air temperature is to the dew point temperature of the intake air (ambient). The dew point effectiveness can be expressed as:

$$\varepsilon_{\text{dew}} = \frac{(t_{\text{a,in}} - t_{\text{a,out}})}{(t_{\text{a,in}} - t_{\text{dew,in}})}$$

2.6 Secondary to Primary Air Ratio

The ratio of secondary airflow rate to primary airflow rate has an effect on the performances of IEC systems. Assuming the total airflow rate is constant, the cooling effectiveness of an IEC system is enhanced when the secondary to primary ratio increases due to more secondary air working in the direct section to absorb sensible and latent heat from the primary air. However, the supply airflow rate of the IEC system is reduced.

2.7 Cooling Capacity

The cooling capacity of coolers can be evaluated using the sensible cooling of intake air. The calculation equation is given as below:

$$Q_{\text{cooling,IA}} = \frac{c_{p,f}\rho_f V_2\left(t_{db,1} - t_{db,2}\right)}{3.6}$$

where,

$Q_{\text{cooling,IA}}$- Sensible cooling of intake air, W;

$c_{p,f}$- Specific heat of air at constant pressure, kJ/(kg k);

ρ_f- Density of air, kg/m^3;

V_2- Airflow rate of supply air, m^3/h;

$t_{db,1}$- Dry-bulb temperature of intake air, °C.

$t_{db,2}$- Dry-bulb temperature of supply air, °C.

2.8 Energy Efficiency

COP of the system is the ratio of cooling capacity (Q) to total power consumption (W).

$$\text{Energy efficiency} = \frac{Q_{\text{cooling,IA}}}{W} = \frac{c_{p,f}\rho_f V_2\left(t_{db,1} - t_{db,2}\right)}{3.6W}$$

2.9 Water Evaporation Rate

The water consumption depends on the airflow, the contaminant load, the effectiveness of the evaporator medium, and the dry-bulb and wet-bulb difference of the intake air. It can be calculated by the following formula:

$$V_w = \frac{1000V_3\rho_{w,f}}{\rho_w}\,(w_3 - w_1)$$

where,

V_w - Water evaporation rate, litre/h;

V_3 - Secondary air flow rate, m^3/h;

$\rho_{w,f}$ - Secondary air density, kg/m^3;

ρ_w - Water film density, kg/m^3;

w_1 - Inlet moisture content of secondary air, kg/kg (dry air);

w_3 - Outlet moisture content of secondary air, kg/kg (dry air).

2.10 Evaporative Medium Type

A wide range of material types can be used as a medium to evaporate water, i.e. metal, fiber, ceramics, zeolite and carbon. The following properties of the wicking material (evaporative medium) should be considered in selecting the preferable wicking materials for the heat and mass exchanger:

- A wicking material with high wicking ability enable a fast, thin and uniform wetting on the wet surface of the plate

- A wicking material with high thermal conductivity allows a large amount of heat to be conducted from the dry side of the plate to the wet side
- The tensile strength of the wicking material should be well enough to process or shape into various geometries
- The wicking material should be inexpensive and be ease of cleaning and replacement.

CHAPTER 3: THEORETICAL ANALYSIS

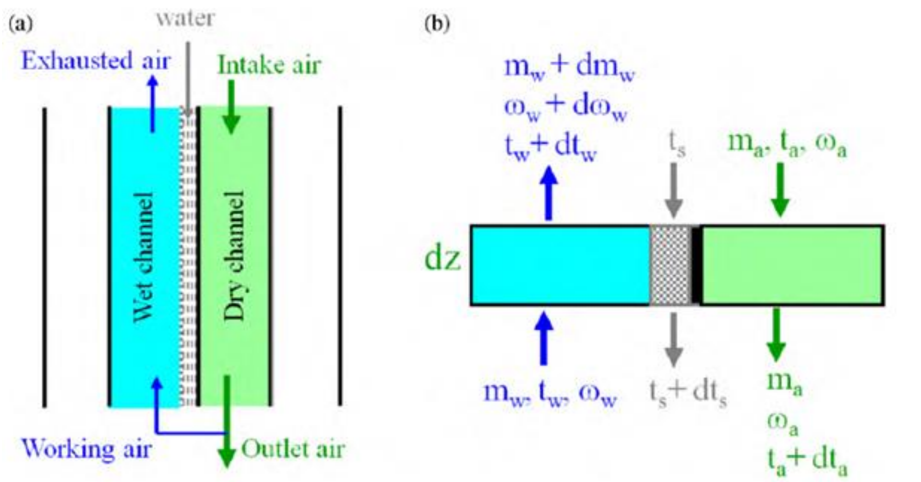
3.1 Model of Heat/Mass Exchanger

Model consists of layer of dry and wet channel with channel walls in between. The primary air (the sum of product and working air) of dry channel is cooled through exchanging sensible and latent heat (Q_s and Q_l) with the working air in the adjacent wet channel. The air velocity and temperature distribution over the cross section of flow channels were assumed uniform. To simplify the modeling process and mathematical analysis, the following assumptions were made:

- The heat and mass transfer process is considered as steady state. The exchanger enclosure is considered as the system boundary.
- The heat and mass transfer process is adiabatic. No heat transfer to the surroundings.
- No heat and mass transfer in the airflow direction. Within the air streams, the convective heat transfer is considered the dominant mechanism for heat transfer. The channel walls are impervious to mass transfer.
- The wet surface of the heat transfer sheet is completely saturated. The water film is distributed uniformly across the wet channel.
- Air is treated as an incompressible gas. The velocity and properties of all air streams are considered to be uniform in a differential control volume.
- Due to relatively low thickness (0.5 mm), the thermal conductivity of the wall and temperature difference of wall surfaces between dry and wet side can be neglected.
- Diffusion is considered negligible.
- Lewis factor is unity.

3.2 Mass and Energy Conservation

A set of differential equations were established for a control volume (Figure. 3.1[4]) using the energy and mass conservation laws.



Schematic of dew point evaporative cooling (a) unit configuration and (b) a differential control volume.

Figure 3.1 Control Volume

Along the dry channel, there is only sensible heat transfer taking place. An energy balance of a differential control volume on the dry side, which sensible heat losses from the process air to the wet side passing the parallel wall yields the following equation:

$$\dot{m}_a C_{pm} dt_a = h_a a(t_a - t_s)dz \tag{1}$$

Where, C_{pm} is the specific heat of moist air and is defined as:

$$C_{pm} = C_p + \omega \cdot C_v \tag{2}$$

Rearranging this equation provides the local gradients of process air along the dry channel to be calculated in successive steps of overall length.

$$\frac{dt_a}{dz} = \frac{h_a a(t_a - t_s)}{\dot{m}_a C_{pm}} \tag{3}$$

For a differential control volume on the wet side, the rate of heat transfer across the air film from the wet surface interface to the bulk air can be written as:

$$\dot{m}_w C_{pm}\, dt_w = h_w a(t_s - t_w)dz \tag{4}$$

Where, $\dot{m}_w$ is the mass flow of working air inside the wet channel defined as:

$$\dot{m}_{\mathrm{w}} = r \cdot \dot{m}_{\mathrm{a}} \tag{5}$$

Mass transfer takes place only at wetted surface under the driving force of vapor partial pressure difference. The mass exchange is balanced and written as:

$$\dot{m}_{\mathrm{w}}\, \mathrm{d}\omega_{\mathrm{w}} = h_{\mathrm{m}} a(\rho_{\mathrm{s}} - \rho_{\mathrm{w}})\,\mathrm{d}z \tag{6}$$

Rearranging Equations (4) and (6) gives the following relationships:

$$\frac{\mathrm{d}t_{\mathrm{w}}}{\mathrm{d}z} = \frac{h_{\mathrm{w}} a(t_{\mathrm{s}} - t_{\mathrm{w}})}{\dot{m}_w C_{pm}} \tag{7}$$

$$\frac{\mathrm{d}\omega_{\mathrm{w}}}{\mathrm{d}z} = \frac{h_{\mathrm{m}} a(\rho_{\mathrm{s}} - \rho_{\mathrm{w}})}{\dot{m}_{\mathrm{w}}} \tag{8}$$

These governing equations demonstrate the profiles of the air temperature and humidity along the wet channel length which depend upon the driving forces of temperature and mass concentration difference, respectively.

For air–water vapor mixtures in the wet passage, the relation of heat and mass transfer coefficients with Lewis number can be expressed as:

$$\frac{h_{\mathrm{c}}}{h_{\mathrm{m}}} = \rho C_{\mathrm{pm}} L_{\mathrm{e}}{}^{2/3} \tag{9}$$

Nussle number for laminar flow and convective heat transfer coefficient can be expressed as:

$$Nu = 0.66 Re^{0.5} Pr^{0.33} \tag{10}$$

$$h = \frac{k}{L} Nu \tag{11}$$

The overall heat transfer coefficient between the water film and intake air is given by:

$$k = \frac{1}{\frac{1}{h_a} + \frac{\delta_{wall}}{k_{wall}} + \frac{1}{h_w}} \tag{12}$$

Where δ_{wall} is wall thickness and k_{wall} is thermal conductivity of channel walls, $h_w$is the heat transfer coefficient between the water film and channel walls.

From Eq (12); if wall thickness is less than 0.5mm, the thermal conductance resistance will be very small and can be ignored. So the temperature at wet side of wall and the dry side of the wall will be the same.

Eqs. (3), (7) and (8) are the differential governing equations describing the heat and mass transfer process of the dew point evaporative cooling system. Numerical results of the air temperature and humidity for a differential control volume can be obtained by solving the set of discretized equations. BVP4C (boundary value problems for ordinary differential equations) solver was used to solve for temperature and humidity distributions of the air with the desired boundary conditions.

CHAPTER 4: SIMULATION AND DESIGN

Simulation of the heat and mass equations was performed for different input parameters. Simulation was conducted to find the influence of inlet air temperature (by varying inlet air temperature between 28 and 42˚C, and inlet air humidity 7g/kg and 10g/kg) on the outlet air temperature. The outlet air conditions affect significantly the comfort condition of a cooling space. The performance of dew point evaporative cooling system, i.e. wet bulb and dew point effectiveness, is essentially dependent upon the outlet air temperature (product air) and the inlet air condition. The outlet air humidity is not changed owing to the thin film cotton wall sheets preventing the moisture exchange between dry and wet channels. The increasing inlet air temperature rise the outlet air temperature.

4.1 Temperature Distribution along the length

Temperature distribution along the length of the channel is shown in figure 4.1 and figure 4.2 at 7g/kg and 10g/kg of humidity ratio respectively.

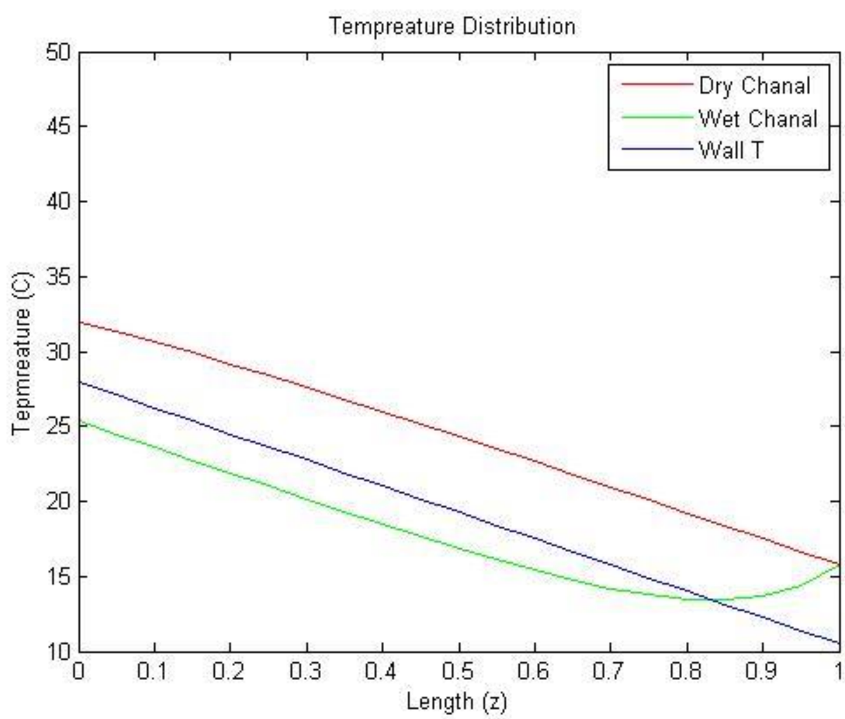


Figure 4.1 Temperature Distribution (humidity-7g/kg)

Table 4.1 Parameters for Figure 4.1

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^0C)	Inlet temperature (^0C)
2.0	0.006	0.5	1	28	32

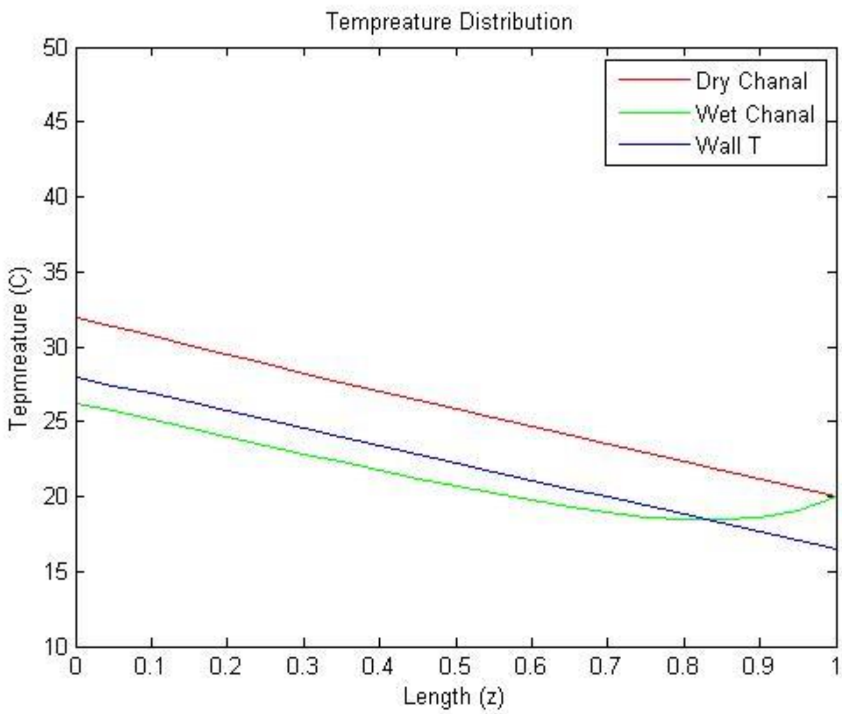


Figure 5.2 Temperature Distribution (humidity-10g/kg)

Table 4.2 Parameters for Figure 4.2

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^0C)	Inlet temperature (^0C)
2.0	0.006	0.5	1	28	32

4.2 Velocity Vs Outlet Temperature

Outlet temperature of air changes by varying velocity of air through the channels. Figure 4.3 shows the relationship between velocity and outlet temperature.

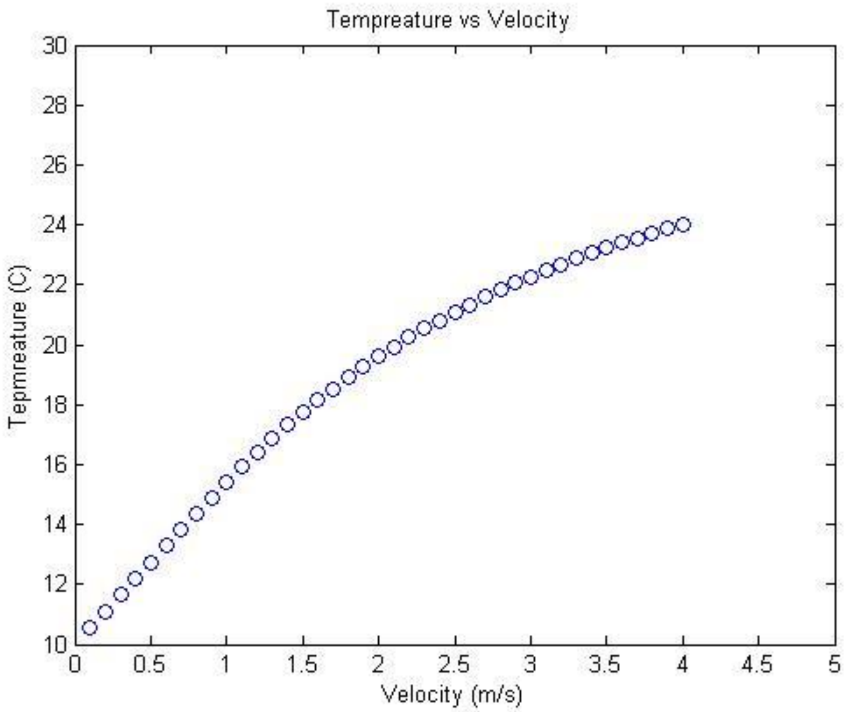


Figure 4.3 Outlet Temperature Vs Velocity

Table 4.3 Parameters for Figure 4.3

Humidity Ratio (g/kg)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)	Inlet temperature (^{0}C)
10	0.006	0.5	1	28	32

4.3 Channel Gap Vs Outlet Temperature

Outlet temperature of air changes by varying channel gap of unit. Figure 4.4 shows the relationship between channel gap and outlet temperature.

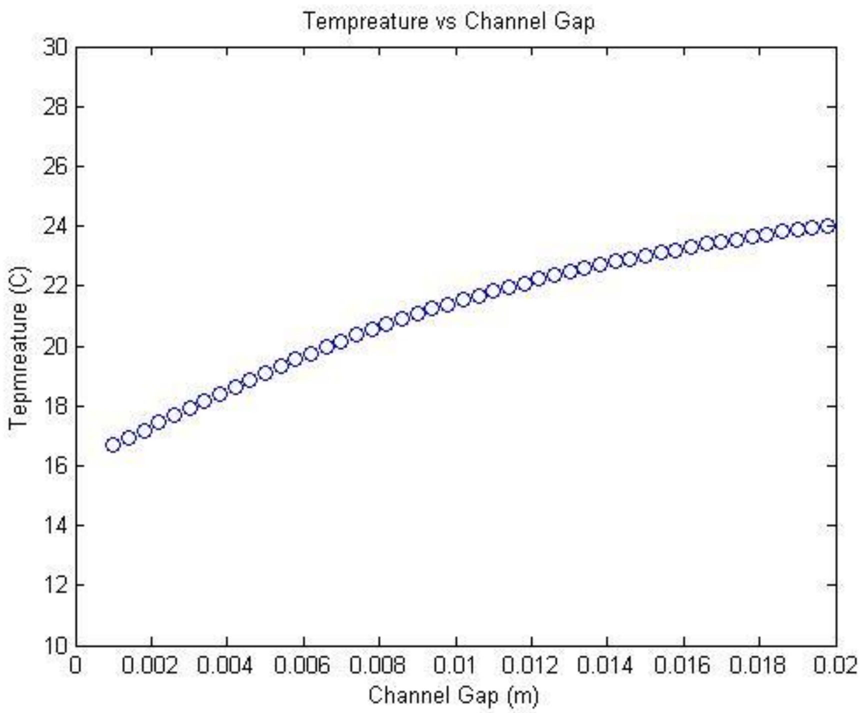


Figure 4.4 Outlet Temperature Vs Channel Gap

Table 4.4 Parameters for Figure 4.4

Humidity Ratio (g/kg)	Air Velocity (m/s)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)	Inlet temperature (^{0}C)
10	2	0.5	1	28	32

4.4 Inlet Temperature Vs Outlet Temperature

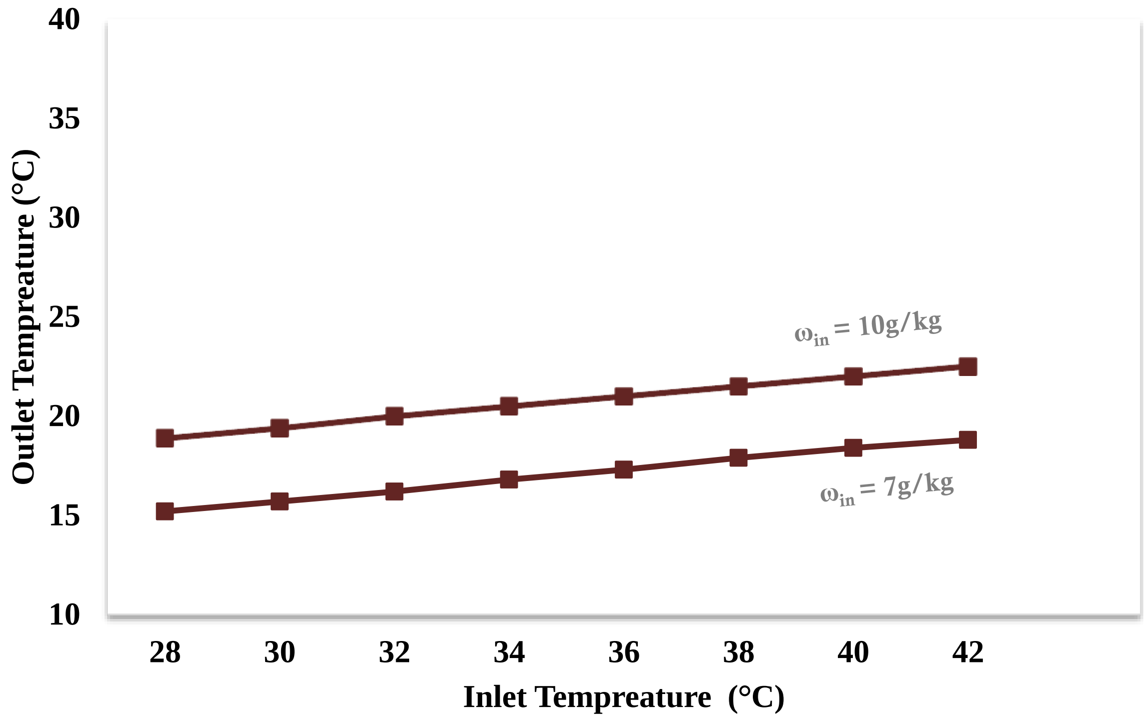


Figure 4.5 Outlet Temperature Vs Inlet Temperature

Table 4.5 Parameters for Figure 4.5

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)
2.0	0.006	0.5	1	28

4.5 Wet Bulb Effectiveness Vs Inlet Temperature

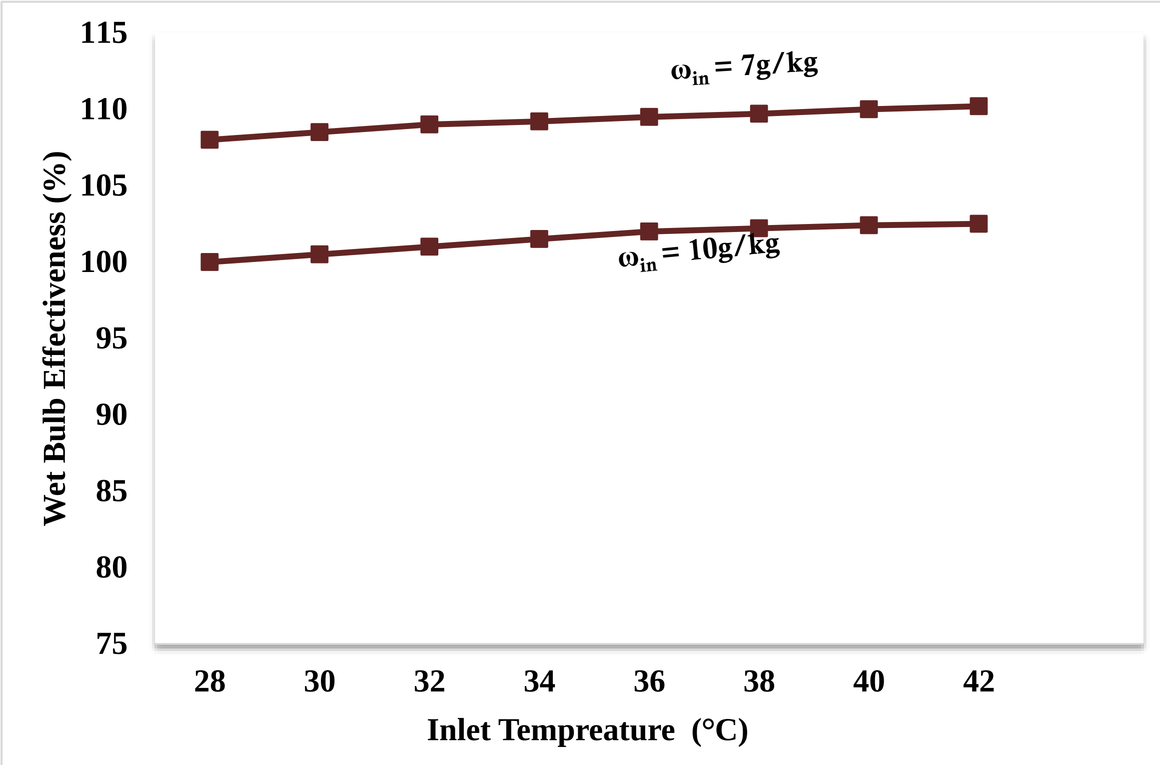


Figure 4.6 Wet Bulb Effectiveness Vs Inlet Temperature

Table 4.6 Parameters for Figure 4.6

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)
2.0	0.006	0.5	1	28

4.6 Dew Point Effectiveness Vs Inlet Temperature

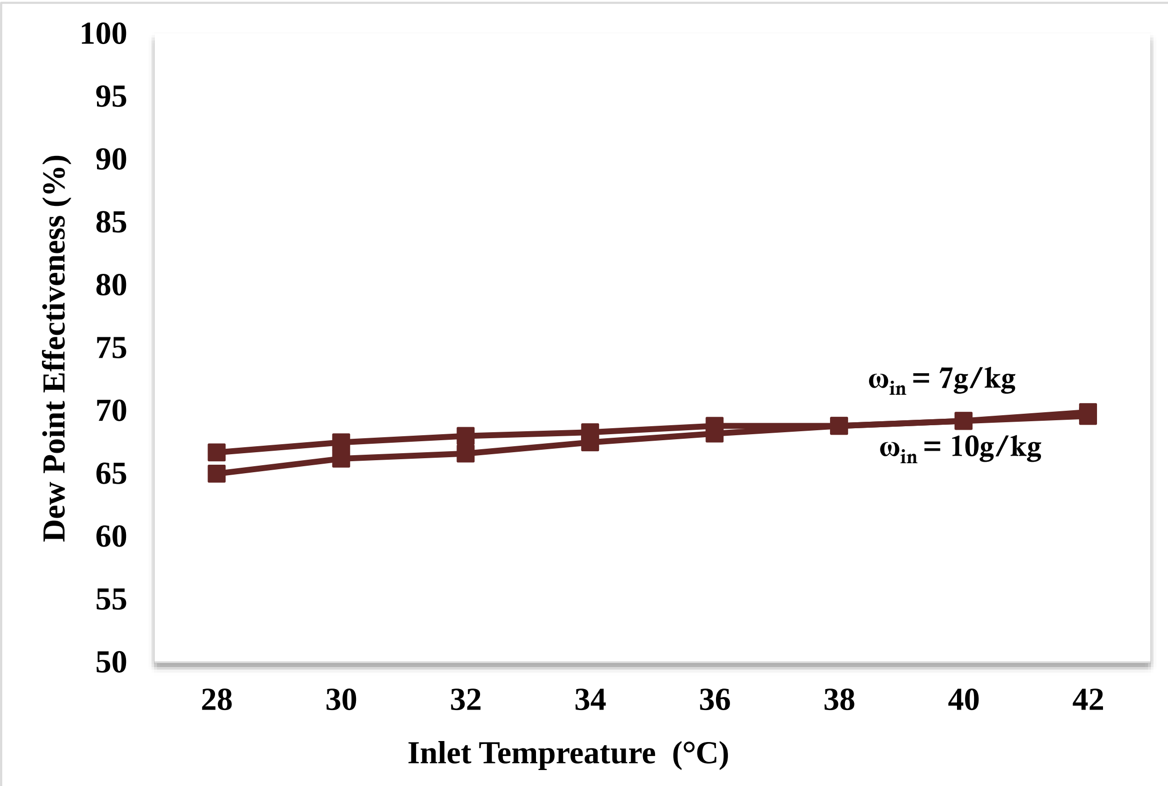


Figure 4.7 Dew Point Effectiveness Vs Inlet Temperature

Table 4.7 Parameters for Figure 4.7

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)
2.0	0.006	0.5	1	28

4.7 COP Vs Inlet Temperature

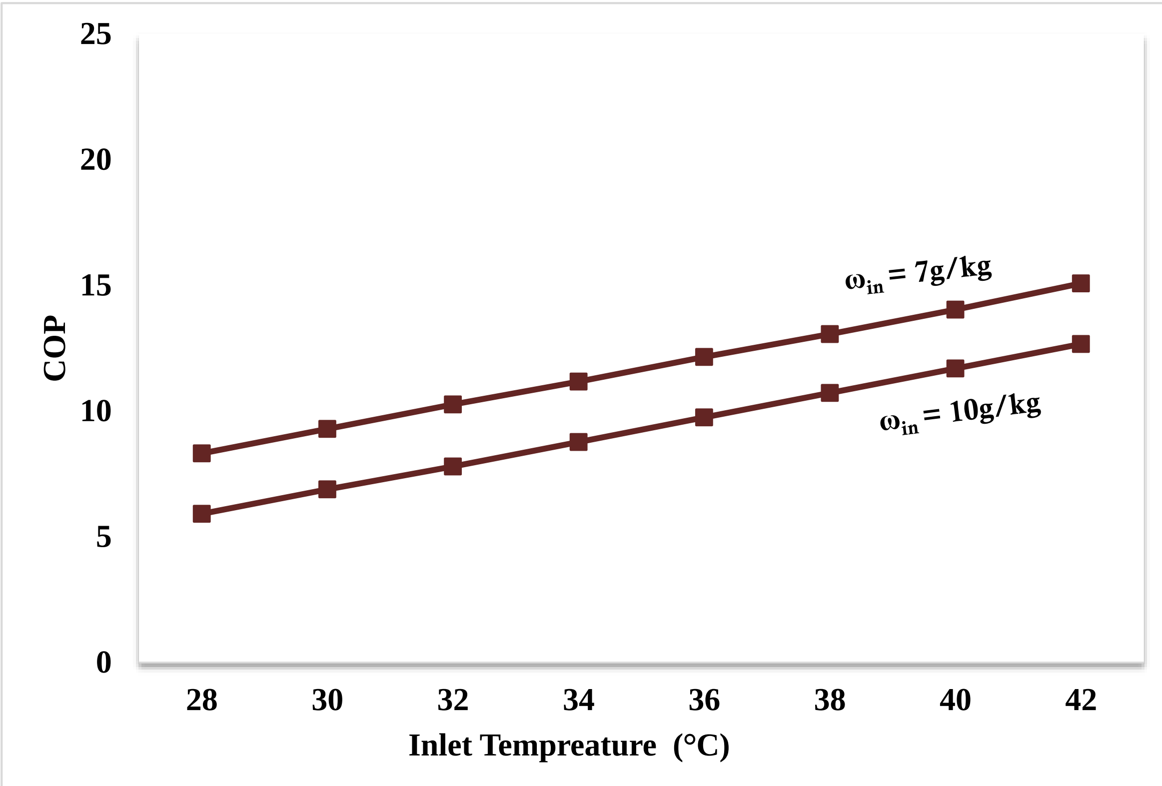


Figure 4.8 COP Vs Inlet Temperature

Table 4.8 Parameters for Figure 4.8

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)
2.0	0.006	0.5	1	28

CHAPTER 5: CONSTRUCTION AND EXPERIMENTATION OF DEW POINT AIR COOLER PROTOTYPE

Prototype is made for testing and experimentation of dew point evaporative cooling process. We will evaluate and compare the theoretical and experimental data by this method. Prototype has 4 dry and 2 wet channels. Channels are made using wood strips stacked together with help of silicone. Aluminum foil is place in between channels and cotton sheets have been place in wet channels. Water is sprayed on the cotton sheets to saturate evenly all wet surface in wet channel. Silicone is used for preventing water leakage. The whole body is then insulated with glass wool insulation so then heat must not enter the system from outside.

5.1 Components of prototype

The Prototype consists of the fallowing components

- Heat and mass exchanger
- Supply air fan
- Exhaust air fan
- Water container
- Water pump
- Stand
- Sensors

Figure 5.1 Prototype

5.1.1 Heat and mass exchanger

Heat and mass exchange portion is made by combining two dry and one wet channel. Wet channel is placed in between two dry channels and it’s both walls are wetted by cotton sheets. Air from one dry channel is by diverted in to wet channel, this results in evaporative cooling and lowering of temperature of wet channel from which other dry channel exchanges heat and it’s temperature is reduced effectively. Similarly two sets of two dry and one wet are combined for construction of heat and mass exchanger

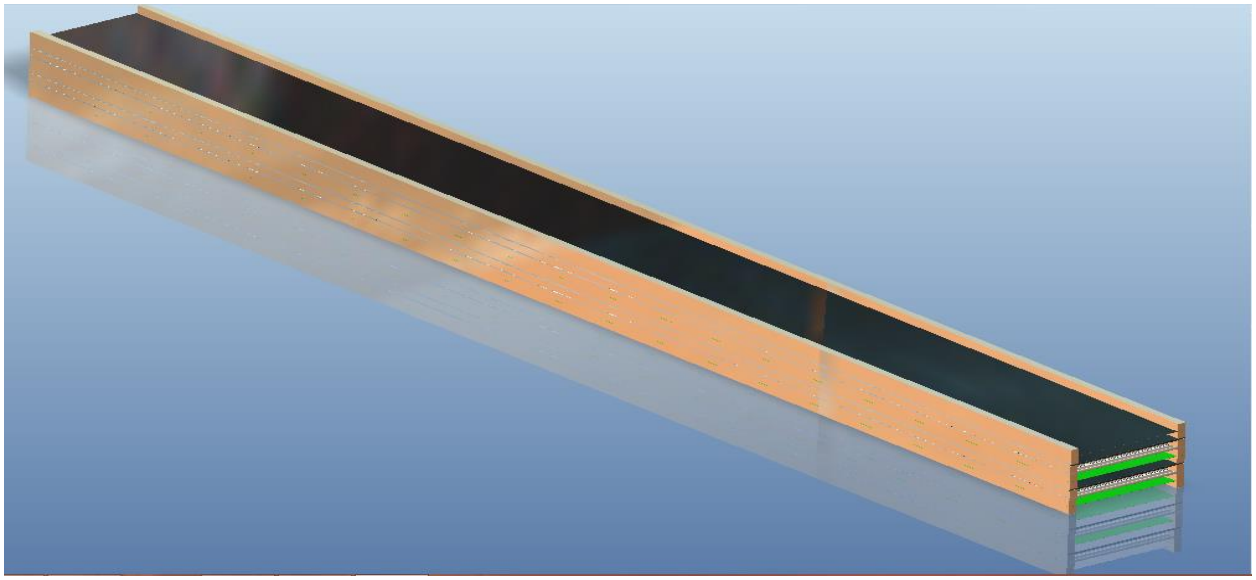


Figure 5.2 CAD Model of Heat Exchanger

5.1.2 Supply air fan

Inlet fan is installed to supply air to the heat and mass exchanger. It supplies air to four dry channels. It has power rating of 2 Watt. As this fan supplies steady flow of air to the dry channels and keeps flow laminar.

5.1.3 Exhaust air fan

This fan is installed at exit of wet channels as it creates suction of air through wet channel which contain humid air in it and the power rating of this fan is 2 Watt

5.1.4 Water container

As water container works as reservoir of the system it contains water with is sprayed through the wet channels

5.1.5 Water pump

20 watt water pump is placed inside the water container as its sucks the water from container and supply it to the wet channels constantly.

5.1.6 Casing

Casing is used which is made of hard board around the water container and then sprayed so the heat transfer from external environment to water container should be minimum as feed water has to be maintained at constant temperature

5.1.7 Stand

The whole prototype is the fixed into a stand which carries it and stand is manufactured by welding pipes together.

5.1.8 Sensors

K-type thermocouples are used to take measurement of temperature at inlet and out let of the prototype. Water level sensor is used to check the level of water in the water container as water is needed continuously.

5.2 Experimental Vs Theoretical Results

5.2.1 Inlet Vs Outlet Temperature

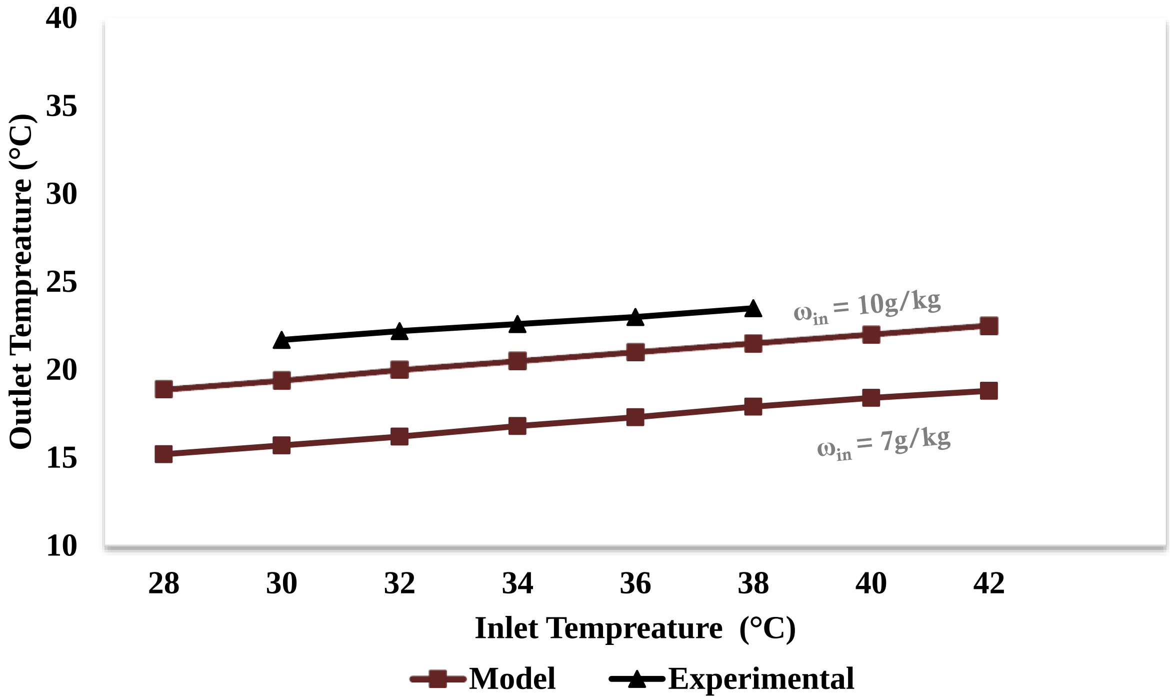


Figure 5.3 Inlet Vs Outlet Temperature (Experimental)

Table 5.1 Parameters for Figure 5.3

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)
2.0	0.006	0.5	1	28

5.2.2 Wet Bulb Effectiveness (Experimental)

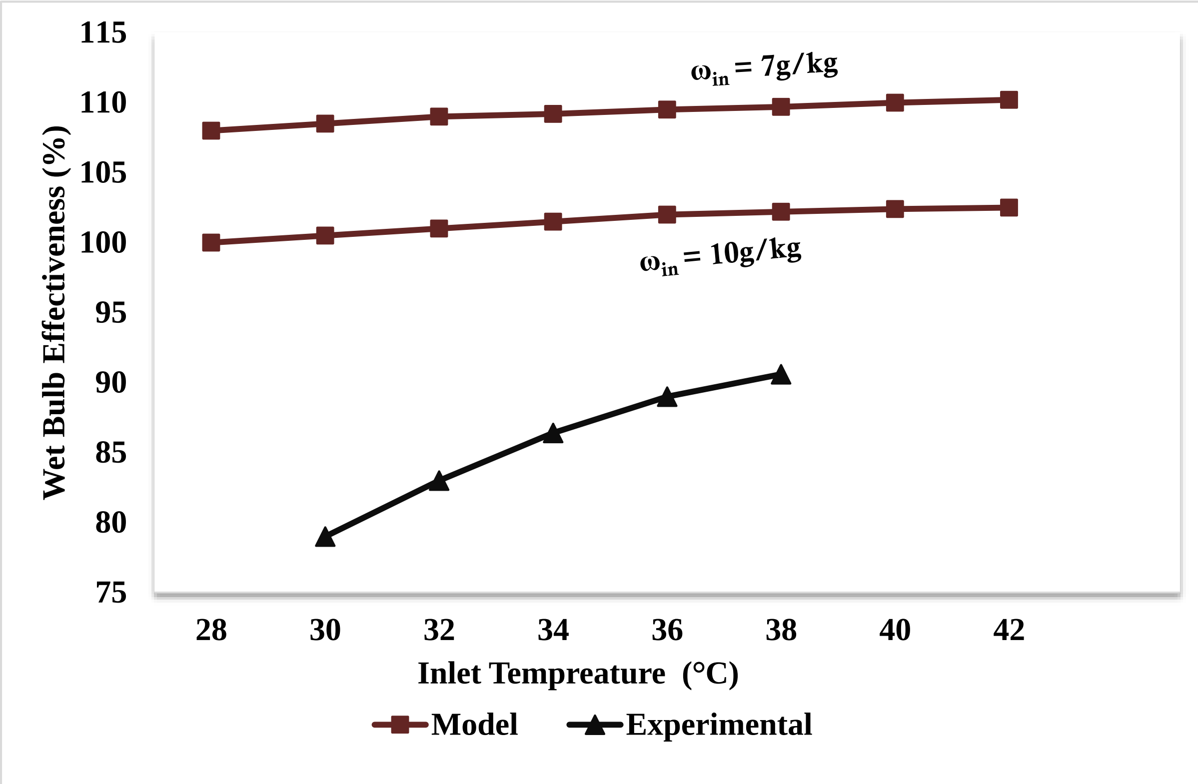


Figure 5.4 Wet bulb effectiveness Vs Inlet Temperature (Experimental)

Table 5.2 Parameters for Figure 5.4

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)
2.0	0.006	0.5	1	28

5.2.3 Dew Point Effectiveness (Experimental)

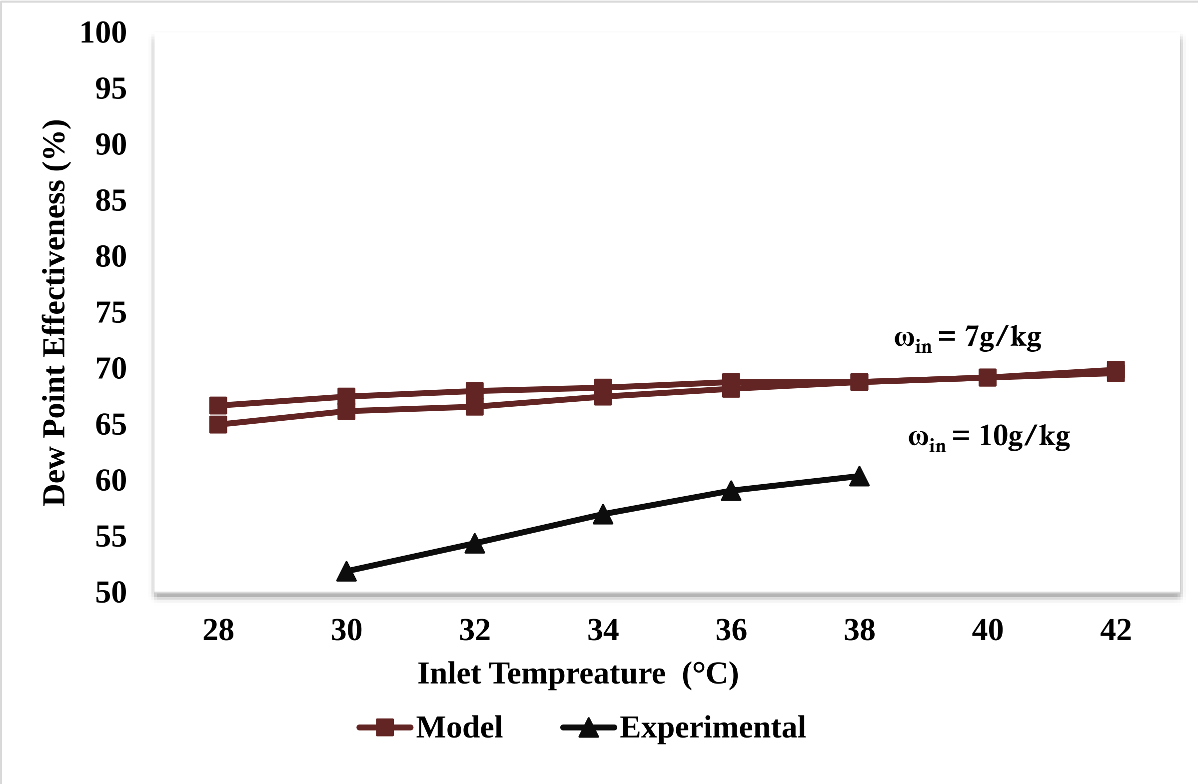


Figure 5.5 Dew Point effectiveness Vs Inlet Temperature (Experimental)

Table 5.3 Parameters for Figure 5.5

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)
2.0	0.006	0.5	1	28

5.2.4 COP (Experimental)

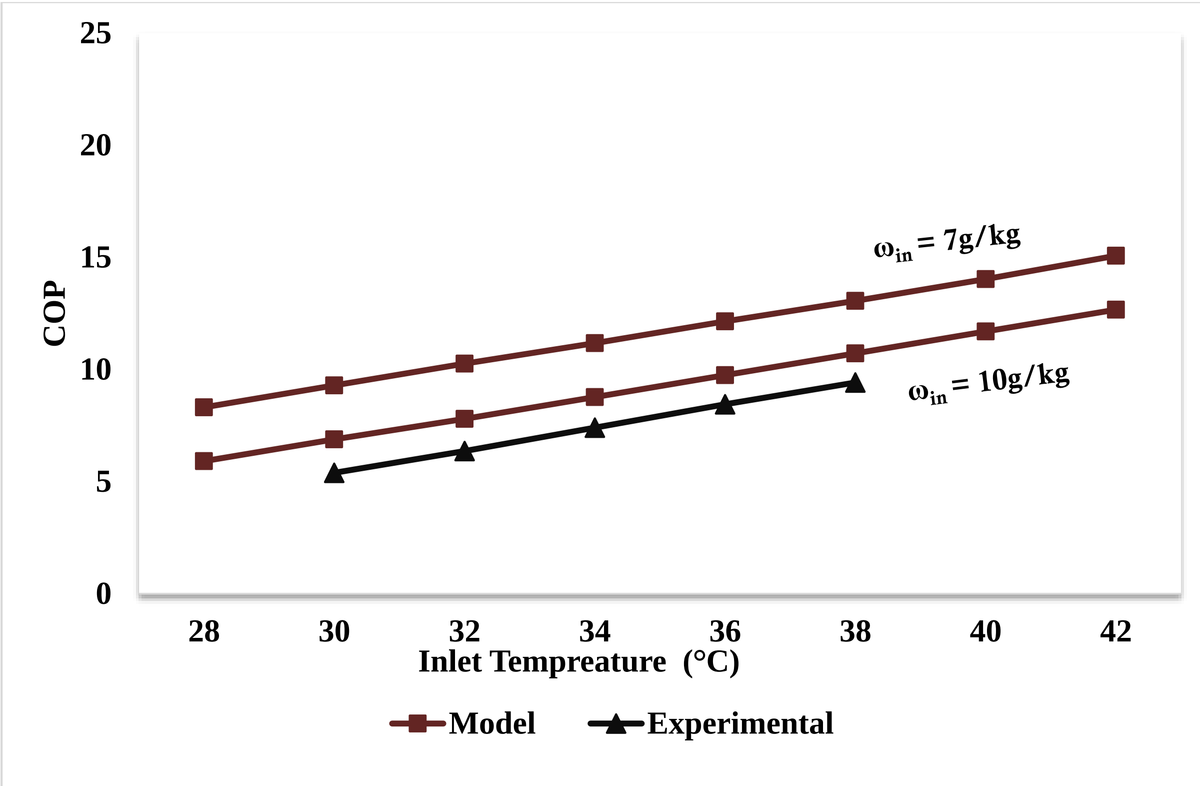


Figure 5.6 COP Vs Inlet Temperature (Experimental)

Table 5.4 Parameters for Figure 5.6

Inlet Air Velocity (m/s)	Channel Gap (m)	Working air to intake air ratio (kg/kg)	Channel length (m)	Feed water temperature (^{0}C)
2.0	0.006	0.5	1	28

RECOMMENDATIONS

- The material used for wetting the surface of wet channel can be replaced by some kind of material which has better quality to absorb water by this we can achieve higher dew point effectiveness as the wet bulb temperature decreases in wet channel.
- We ignored thickness of the insulation but by graphs we can see that there is a relationship between channel gap and outlet temperature, as insulation increases channel gap which results in increase in outlet temperature
- To have better heat transfer, fins can be used with aluminum foil as they increase heat transfer.

NOMENCLATURE

A	width (m)
C_p	specific heat of dry air (kJ/kg ˚C)
C_{pm}	specific heat of moist air (kJ/kg ˚C)
C_v	specific heat of water vapor (kJ/kg ˚C)
D	equivalent diameter of air passage (m)
DB	dry bulb temperature (˚C)
h_a	heat transfer coefficient of the air in dry side (W/m^2˚C)
h_m	mass transfer coefficient (kg/m^2 s)
h_w	heat transfer coefficient of the air in wet side (W/m^2˚C)
L	channel length (m)
Le	Lewis number (dimensionless)
$\dot{m}_a$	mass flow rate of intake air in dry side (kg/s)
$\dot{m}_w$	mass flow rate of working air in wet side (kg/s)
Nu	Nusselt number (dimensionless)
Pr	Prandlt number (dimensionless)
R	working air to intake air ratio (kg/kg)
Re	Reynolds number (dimensionless)
t_a	air temperature in dry side (˚C)
t_s	wall surface temperature (˚C)
t_w	air temperature in wet side (˚C)
WB	wet bulb temperature (˚C)
Z	height of control volume (m)
P	density (kg/m^3)
E	effectiveness or efficiency
Ω	humidity ratio (kg water/kg dry air)

Appendix A: Design Optimization

Channel Gap:

From simulated results, it's clear that outlet temperature is directly proportional to channel gap as shown in figure 4.4. Graph shows that by reducing channel gap, outlet temperature also reduces but smaller channel gap creates complexities in construction of heat exchanger so we have selected optimized channel gap of 6mm for our prototype.

Velocity of air through channels:

Flow velocity is also directly proportional to outlet temperature of the air (shown if figure 4.3) because at high velocity there is less time available for streams to exchange heat but for flow there is also a limitation, there is problem of pressure drop throughout the length of channel. By trade off analysis we have selected 2m/s air velocity for our prototype.

Length of Heat Exchanger:

Length is inversely proportional to outlet temperature of air, by increasing length of heat exchanger, outlet air temperature decreases. But for easy handling we have chosen 1m length for our prototype.

Appendix B: MATLAB Code

Numerical analysis of the equations governing heat and mass transfer, thermodynamics and refrigeration is done on MATLAB; all the theoretical results graph are obtained by using that method.
By varying certain variables like Humidity ratio, Air velocity, channel, inlet temperature and channel gap graphs are plotted to know the effect of these variables on the outlet condition of the numerical model and afterwards results are compared with experimental data collected and the code for solving these equations is given as fallows

```
%

function dew

clc;

clear all


syms ta ts a mdota Cpm z tw mdotw Cpm hm ps pw Ww hm

 S=dsolve('Dta=ha*a*(ta-ts)/(mdota*Cpm)','Dtw=hw*a*(ts-tw)/(mdotw*Cpm)',...

 'Dts=(mdotw*Cpm*Dtw/mdotw+mdota*Cpm*Dta/mdotw)','ta(L)=32','tw(0)=ta(0)',...

  'ts(L)=28','z');

Ww=dsolve('DWw=-hm*a*(Ws-Ww)/mdotw','Ww(1)=.0085','z');

Cp=1005; %specific heat of dry air (J/kg °C)

Cpm=1020.7; %specific heat of moist air (J/kg °C)

DB=0; %dry bulb temperature (°C)

% ha heat transfer coefficient of the air in dry side (W/m2 °C)

% hm mass transfer coefficient (kg/m2 s)

% hw heat transfer coefficient of the air in wet side (W/m2 °C)

L=1; %channel length (m)

Le=1; %Lewis number (dimensionless)

Lt=1; %length of thermal entry region(m)

mdotv=.001;

 %Na Nusselt number (dimensionless)

Pr=.7; %Prandlt number (dimensionless)

r=.5; %working air to intake air ratio (kg/kg)

Re=0; %Reynolds number (dimensionless)
```

```
ta=0; %air temperature in dry side (°C)
ts=0; %wall surface temperature (°C)
tw=0; %air temperature in wet side (°C)
to=28; %Feed water Tempreature
k1=.0262;     %300k
k2=.0262;     %300k
mew=.00001983;
p=1.2;
patm=101.325;
w=.05;
format short
   Ws=1;
   ta=S.ta;
   tw=S.tw;
   ts=S.ts;
   i=1;

for a=.001:.0004:.02;
    v1=1;
    v2=v1;
    Dh=4*a*w/(2*w+2*a);
    Cp=1005;
    Nu1=8.24;
    Nu2=8.24;

ha=10*k1*Nu1/Dh;
hw=10*k2*Nu2/Dh;
hm=hw/(p*Cpm*Le^(.666));

Ws=.024;
mdota=p*a*w*v1; %mass flow rate of intake air in dry side (kg/s)
```

```
mdotw=mdota*r; %mass flow rate of working air in wet side (kg/s)
mdots=(.024-.0085)*mdotw;

    L=1;
    con=(to-16)/L;
    z=[0:.05:L];
    ta=subs(ta);
    tw=subs(tw);
    ts=subs(ts);
    Ww=subs(Ww);
    out(i)=a;
    outtt(i)=v1;
    outt(i)=ta(1);
    i=i+1;
    ta=S.ta;
    tw=S.tw;
    ts=S.ts;
end

 out
 outt
%  xlswrite('Tvsmf.xls',outt','B1');
%  xlswrite('Tvsmf.xls',out','C1');
%  xlswrite('Tvsmf.xls',outtt','D1');
 plot(out,outt,'o')
 title('Tempreature vs Channel Gap ')
 axis([0 .02 10 30]);
 xlabel('Channel Gap (m)')
 ylabel('Tepmreature (C)')

% plot(1-z,ta,'r-')
```

```
% title('Tempreature Distribution ')
% axis([0 L 10 50]);
% xlabel('Length (z)')
% ylabel('Tepmreature (C)')
% hold on
% plot(1-z,tw,'g-')
%  plot(1-z,ts,'b-')
% legend('Dry Chanal','Wet Chanal','Wall T')
% ta
hm=.00001;
ps=200;
pw=1;

 plot(z,Ww)
 eta_wb=(ta(length(z))-ta(1))/(ta(length(z))-25)
 eta_db=(ta(length(z))-ta(1))/(ta(length(z))-20)

%
```

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