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Design of Indirect Evaporative Cooler

Flat Plate-Counter Flow Type

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***Abstract*—This paper deals with the Designing of Indirect Evaporative Cooler of flat plate, countercurrent type. The design of Indirect Evaporative Coolers consists of Developing the Governing differential equations which are based on mass and energy conservation law, then solving these differential equations and then going through the iterative process to select the values of the parameters like velocity of air, air flow rate, thickness of plates, channel width etc to get the desired cooling. A MATLAB program has been generated for the calculation and iteration purpose. CFD analysis has also been carried out to verify the theoretical results obtained.**

Keywords—Indirect Evaporative Cooler; Primary air; Secondary air ; Mass flow rate; Counter flow

I. INTRODUCTION

The cooling energy demand worldwide has increased tremendously in recent decades. This has created serious concern for which, in some countries, further utilities and hence additional supply had to be taken into account, thus increasing the average cost of electricity. Of course, this increase of energy consumption has environmental side-effects related to the increased Co2 emissions and to the ozone-depleting Chlorofluorocarbons (CFCs) used in air conditioners. The Kyoto protocol binds the developed countries to reduce the collective emissions of six key greenhouse gases - among which Co2- at least by 5 % by 2008-2012. This protocol encourages the governments, amongst others, to improve energy efficiency and to promote renewable energy (EU, 2003). Therefore, counterbalancing the energy and environmental effects of air conditioning is a strong requirement for the future. Lately, research has been oriented towards low-energy techniques, one of which is evaporative cooling technique [1].

Over the past decades, evaporative cooling, utilizing the principle of water evaporation for heat absorbing, has gained growing popularity for use in air conditioning, owing to its simplicity in structure and good use of natural energy (i.e., latent heat of water) existing in ambient. This led to enhanced system COP in the range 15–20, which is significantly higher than that for conventional vapour compression and adsorption/absorption air conditioning systems. Direct Evaporative Cooling (DEC) keeps the primary (product) air in direct contact with water, causing evaporation of the water and reduction of temperature of the air simultaneously. As a result, the vaporized water, in form of vapour, is added into the air, which often creates wetter air condition and causes discomfort to the residents. To overcome this difficulty, Indirect Evaporative Cooling (IEC) was brought into consideration. In an IEC, water is separated from the primary (product) air using the heat exchanging plate. During operation, evaporation of the water occurs in one side of the plate where the secondary (working) air moves along; while the primary (product) air flows across the other side. Evaporation of the water causes reduction of the temperature of the plate, resulting in heat transfer between the primary air and the plate. Meanwhile, the vaporized water is taken away by the secondary (working) air across the wet-side of plate. By doing so, the primary air is cooled but no moisture is added into it, which is ideal for purpose of building air conditioning. Owing to this advantage, Indirect Evaporative Cooling (IEC) has potential to become a feasible alternative to conventional mechanical vapour compression systems, which would lead to realization of low (zero) carbon air conditioning served for buildings.

Biggest problem facing the Indirect Evaporative Cooling (IEC) technology lies in its high level of dependency to ambient air condition. The driving force of either direct or indirect evaporative cooling is the temperature difference between the dry-bulb and wet-bulb (or dew point) of the process air, which, in humid or mild climate regions, is very small and thus leads to very limited system cooling capacity. Further, instability of the ambient air condition (temperature/humidity) also causes unsteady operation of the Indirect Evaporative Cooling (IEC) system. A general comparison among the above mentioned air conditioning systems is indicated in Table 1 [2].

Table I: A general comparison between the currently available air conditioning systems

System type	Mechanical vapour compression	Absorption/adsorption	Evaporative cooling
Features	Dominate the air conditioning market Mature technology with stable performance and low cost Driven by electricity with COP of 2 to 3 Energy intensive	Driven by heat (waste, renewable or gas) with limited application Complex system configuration and operational status Low COP: 0.4–1.2 Energy intensive	Using water as the cooling medium Simple system configuration and operational status High dependency to the ambient condition High COP: 15 to 20 Energy economic

II. BASIC CONCEPT AND OPERATING PRINCIPLE

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.

If the product air of the Indirect Evaporative Cooler (IEC) system travels in a counter flow manner to the working air at an appropriate air-flow-ratio and across an infinite surface area, the temperature of the product air in the dry side of the plate will reach the wet-bulb temperature of the incoming working air. The temperature of the working air in the wet side of the plate will be lowered from its incoming dry- bulb temperature to the incoming wet bulb temperature. However, the actual effect is that only 40-80% of the incoming air wet-bulb temperature can be achieved. The reasons for the reduced cooling effectiveness are investigated, giving identification of several attributing facts: (1) there is limited heat exchanging surface area; (2) none pure counter flow pattern could be achievable; and (3) uniform and even water distribution over the wet sides of the plate is hard to obtain.

Fig. 1 presents the working principle and psychometric illustration of the air treatment process relating to an indirect evaporative cooling operation. During operation, the primary (product) air enters in to the dry channel while the secondary (working) air enters in to the adjacent wet channel. The primary air is cooled by the sensible heat transfer between the primary air and the plate, which is induced by the latent heat transfer relating to water evaporation from the plate's wet surface to secondary air. As a result, the primary air (state1) is cooled at the constant moisture content and moves towards the wet-bulb temperature of the inlet secondary air; whereas the secondary air of sate 1 is gradually saturated and changed in to state 20 at its earlier flow path, then heated when moving along the flow path and finally discharged to atmosphere in the saturated state 3. It should be noted that to enable heat transfer between the dry side air to wet side air, the state 3 should have a lower temperature than the state 2 and theoretically speaking, the enthalpy decrease of the air within the dry side channel is equal to the enthalpy increase of the air within the wet side channel i.e., h1-h2=h3-h1 [2].

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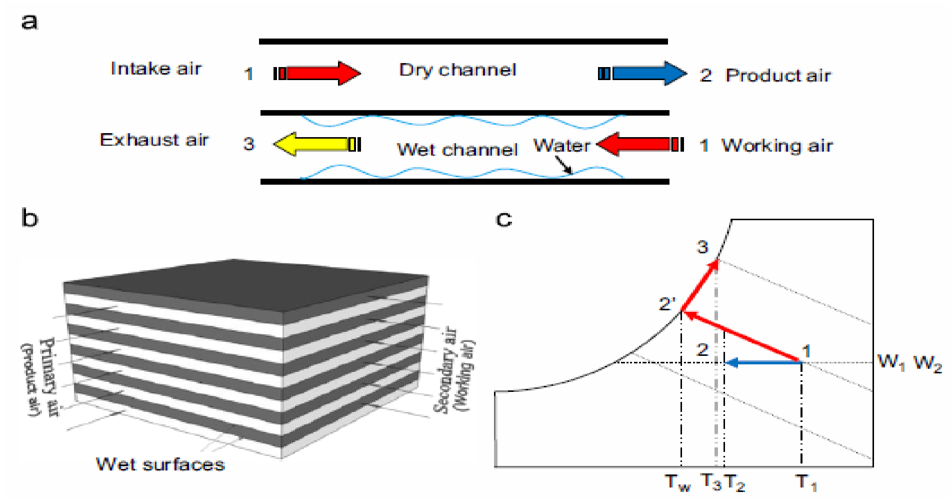

Fig. 1. Conventional IEC heat and mass exchanger - (a) Working principle of the indirect evaporative cooling (b) Configuration of the type IEC heat and mass exchanger (c) Psychrometric illustration of the air treatment process in the IEC heat and mass exchange

III. IEC MODELLING

A schematic diagram of the countercurrent type IEC is shown in the fig. 2.

In principle and based on the energy and mass conservation law, a set of differential equations are to be considered along with the length of IEC as follows, according to the schematic diagram of heat and mass transfer in an IEC (Fig. 2).

- The heat transfer from the water film into the secondary air flow:

$$dQ_S = h_s\,(T_W - T_S)dA \qquad (1)$$

- The mass flow of water that is evaporated into the secondary air:

$$dW = h_m(\omega_{(Tw)} - \omega)dA \qquad (2)$$

- The heat transfer from the primary air into the water film:

$$dQ_P = U_Z\,(T_P - T_W)dA \qquad (3.a)$$

$$dQ_p = -m_s dE_p \qquad (3.b)$$

- The overall heat transfer coefficient is:

$$U_Z = \frac{1}{\frac{1}{h_P}+\frac{\delta_{wall}}{k_{wall}}+\frac{1}{h_W}} \qquad (4)$$

- The water mass balance (refer to Fig.2) yields:

$$dm_w = dW \qquad (5)$$

- The water and air mass balance (refer to Fig.2), yields:

$$m_s d\omega = dm_w \qquad (6)$$

- The overall energy balance on the process for the A and B control surface can be expressed as:

$$m_s dH_S = -H_{pw}dW - dQ_s \qquad (7.a)$$

$$m_s dH_s + dQ_p = m_W dH_W + H_w dm_W \qquad (7.b)$$

The enthalpy of humid air equals the sum of the enthalpies of the dry air and water vapor. The specific enthalpy of humid air is also defined per unit mass of dry air. For lower pressure, the specific enthalpy of water vapor is almost a linear function of temperature. Therefore, the enthalpy of humid air can be expressed as:

$$\mathrm{H(T)} = C_p T + \omega(2.501 + 1.805 \times 10^{-3} T) \qquad (8)$$

By using equations (1) -(3) and rearrangement of equations (5) -(8), a set of ordinary differential equations are described below:

$$\frac{d\omega}{dx} = -\frac{h_m f_m a(\omega_{(T_W)} - \omega)}{m_s} \qquad (9)$$

$$\frac{dT_S}{dx} = -\frac{a}{m_s C_p}\left(h_m f_m C_{pw}\left[\omega_{(T_W)} - \omega\right] + h_s\right) \qquad (10)$$

$$\frac{dT_W}{dx} = \frac{a}{m_s C_W}\Big(h_m f_m\left[T_w\left(C_W - C_{pw}\right) - h_{fg}\right]\cdot\left[\omega_{T_w} - \omega\right] - h_s(T_w - T_s) + U_z(T_p - T_w)\Big) \qquad (11)$$

$$\frac{dT_P}{dx} = -\frac{U_Z a(T_P - T_W)}{m_P C_P} \quad (12)$$

Equations 9-12 are the full description of the system. In the above equations, f_m represent wettability of the plate.

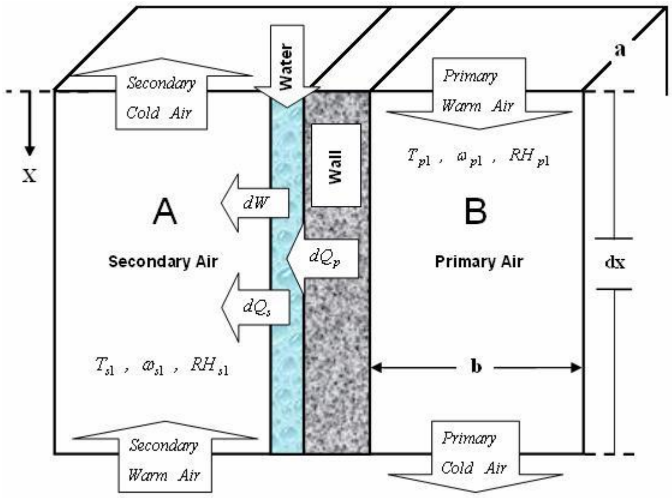

Fig. 2. Schematic Diagram of heat and mass transfer in IEC

IV. ANALYTICAL SOLUTION

On solving the above differential equation, we obtain the following relation:

$$T_{po} = (T_{pi} - T_W)e^{Ax} + T_w$$

(13)

Where,

$$A = \frac{-U_Z \times a \times n}{m_p \times C_p}$$

(14)

A large no. of materials are present, like metals and its alloys, fibres, polycarbonate [4], ceramics, zeolites etc, which can be used for Indirect Evaporative Cooling systems. In this paper Aluminium has been selected because of it's following advantages [3]:

- High thermal conductivity (229 W/m k)
- Low cost
- Low density which helps in reducing the overall weight of the IEC system.
- Easy availability

Equation (4), because of higher thermal conductivity of Aluminium, low thickness of plates and higher convective heat transfer coefficient on the wet side, is reduced as follows:

$$U_Z = h_P$$

(15)

The MATLAB program used for various iterations to find out the optimum values of the design parameters is shown below:

```
clc
clear all
Tpi = input (' Temperature of Primary air at inlet (°C) = ');
V = input (' Velocity of primary air at inlet (m/s) = ');
Tw = input (' Temperature of Water (°C) = ');
n = input ('No. of plates = ');
t = input (Thickness of plates (mm) = ');
a = input (Width of plates (m) = ');
L = input (Length of plates (m) = ');
δ = input (Channel width (m) = ');
Cp = 1005;
Pr = 0.711;     k = 0.027;
ν = 16.97×10^-6;
ρ = 1.127;
Re = (V×L)/ ν;
hp =  (0.664*(Re^0.5)) × (Pr^(1/3) ×k)/L;
CA = ( δ /2) × a× n;
mp = ( ρ ×CA×V) ×3600;
Vp = mp/ ρ;
At = n×a×L;
Uz = hp;
A = (-Uz × a × n × 3600) / ( mp × Cp) ;
x=0 ;
for i = 1:60;
   x = x + 0.01;
   T = ( Tpi – Tw ) × exp( A × x ) + Tw ;
if i==1
fprintf(' T(0.000000)=%f\n ', Tpi);
end
fprintf(' T(%f)=%f\n ',x,T);
end
fprintf(' T(0.000000) = %f  °C\n ', Tp i);
fprintf(' T(%f) = %f  °C\n ', x , T );
fprintf(' Volume flow rate = %f  m3/hr\n ', Vp ) ;
fprintf(' Mass flow rate = %f  kg/hr\n ',  mp ) ;
fprintf(' Total heat exchange Area = %f\n ',  At ) ;
x =  [ 0.0:0.01:0.6 ] ;
y =  ( Tpi-  Tw )  × exp(A × x) + Tw ;
plot (x,y) ;
```

The Theoretical results are summarized below:

Thickness of plates, t = 0.4 mm
Length of plates, L = 0.6 m
Width of plates, a = 0.4 m
Channel width, δ = 4 mm
No of plates, n = 70
Total heat exchange area, At = 16.8 m^2
Velocity of primary air, V = 1 m/s
Velocity of secondary air, Vs = 0.5 m/s

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Size of plates = L × a = 0.6 ×0.4 m^2
Volume flow rate of primary air = 175 m^3/hr
Mass flow rate of primary air = 200 kg/hr
Mass flow rate of secondary primary air/ primary air = 0.5

CASE	INLET TEMPERATURE OF PRIMARY AIR (°C)	SUPPLY AIR TEMPERATURE (°C)
I	40	22.0
II	35	20.4
III	30	19.2
IV	25	17.7

The Temperature drop in the primary air in the dry channel, for various cases, is shown below:

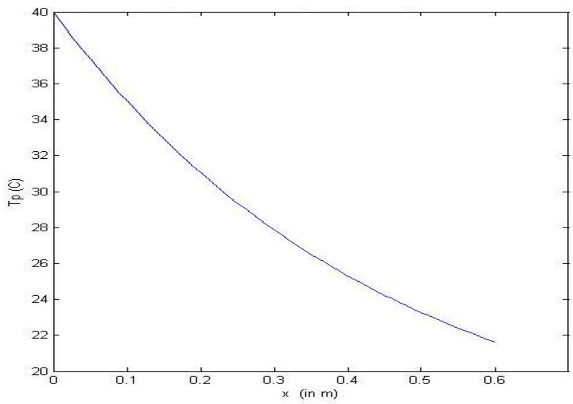


Fig. 3- Variation in temperature of primary air in the dry channel when Tpi = 40 °C

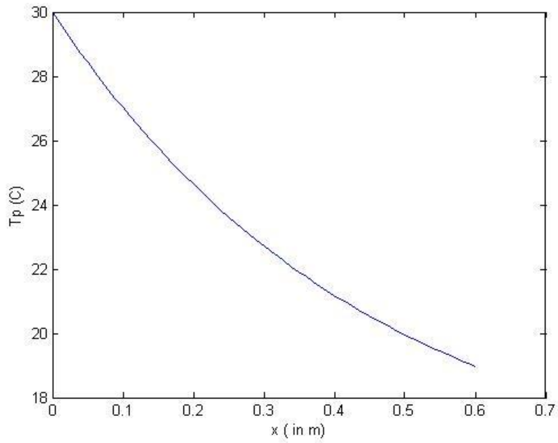


Fig. 4- Variation in temperature of primary air in the dry channel when Tpi = 35 °C

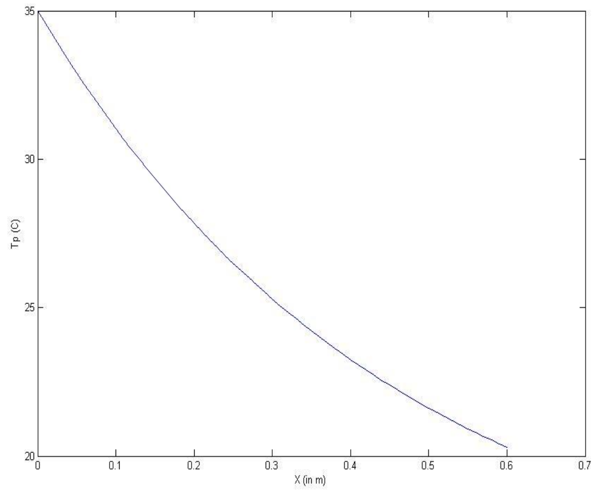


Fig. 5- Variation in temperature of primary air in the dry channel when Tpi = 30 °C

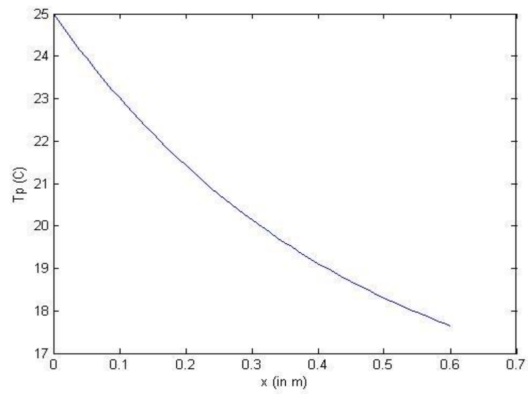
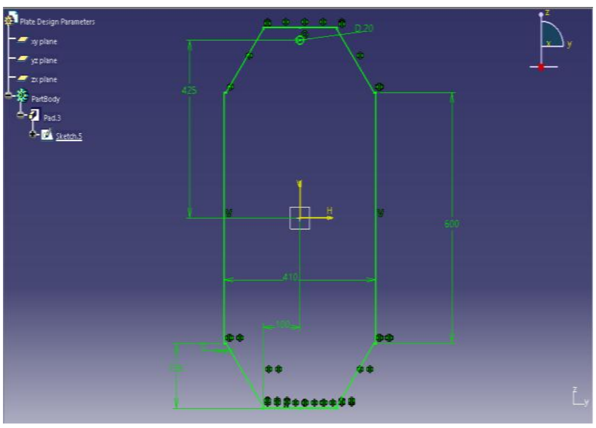


Fig.6. Variation in temperature of primary air in the dry channel when Tpi = 25°C

V. CAD MODEL

The CAD model of the Indirect Evaporative Cooler is shown below:



(i)
Fig. 7– Plate

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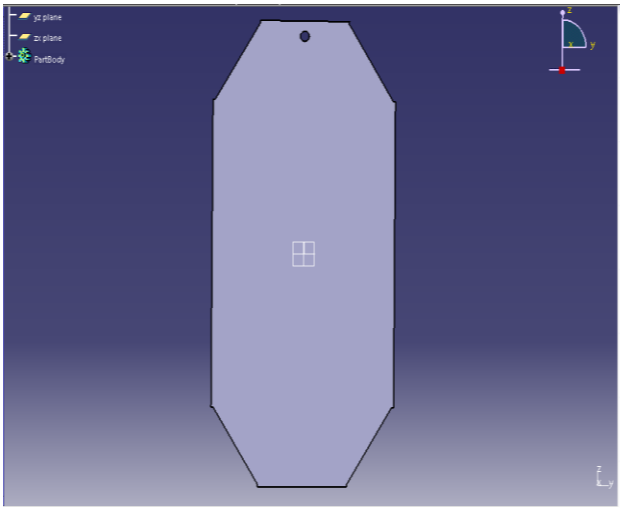


(ii)

Fig. 7– Plate

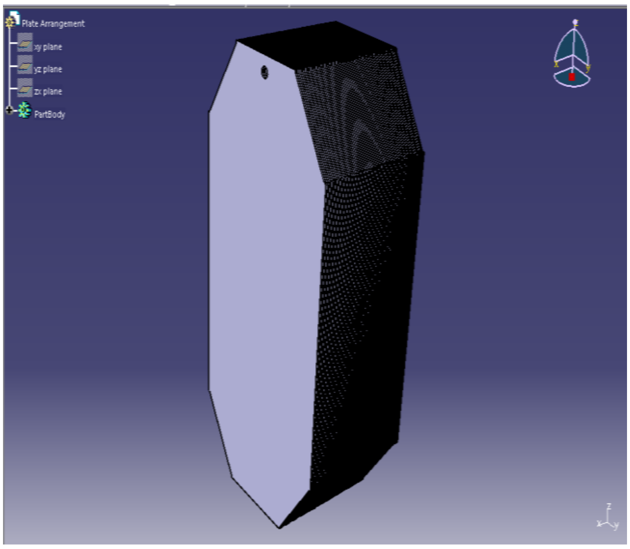


Fig. 8 – Plate Arrangement

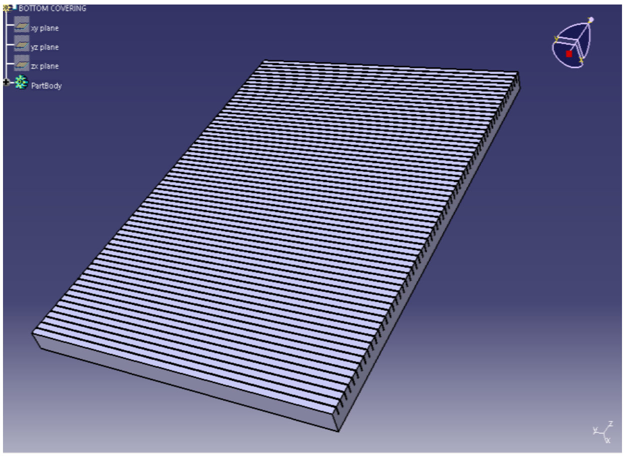


Fig. 9 – Bottom and Top Covering

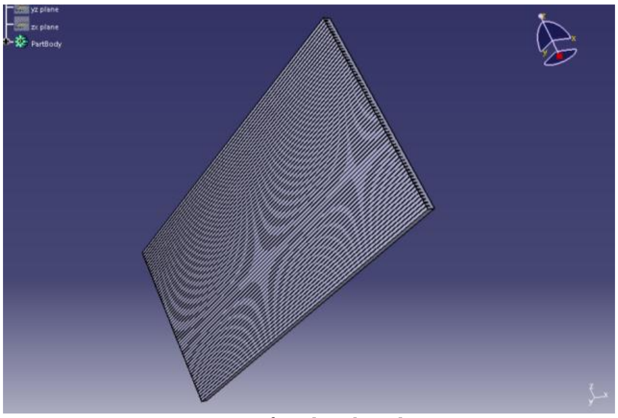


Fig. 10 – Left and Right Side Covering

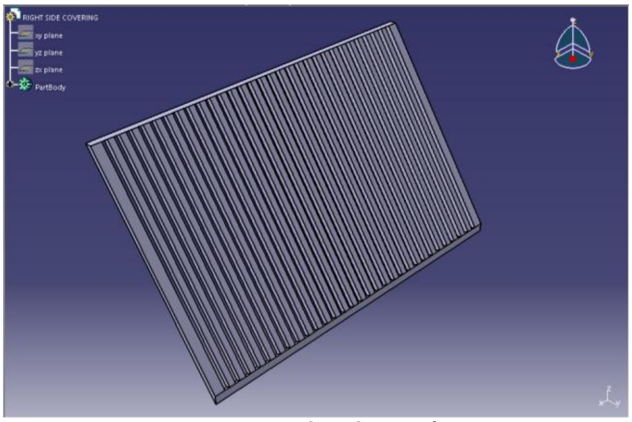


Fig. 11 – Bottom right and Top Left Covering

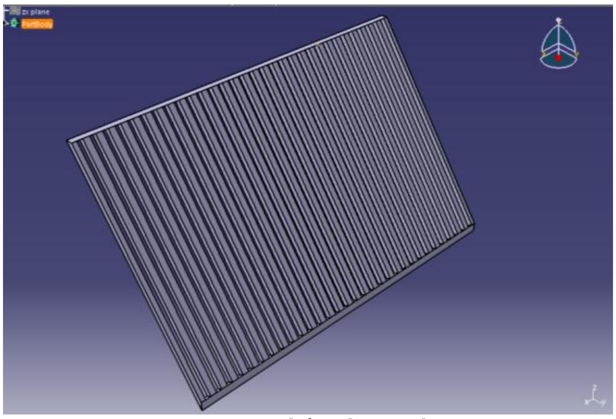


Fig. 12 – Bottom left and Top right Covering

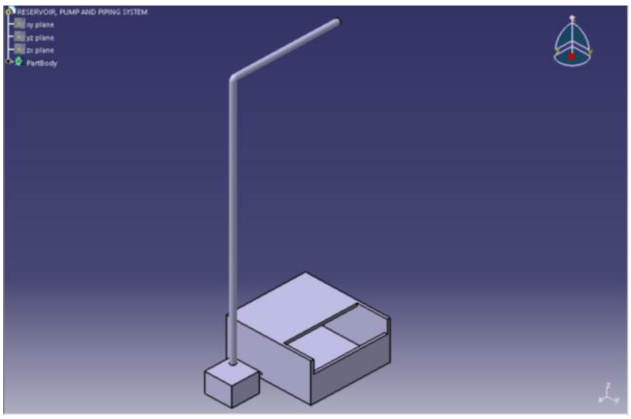


Fig. 13 – Reservoir, pump and piping system

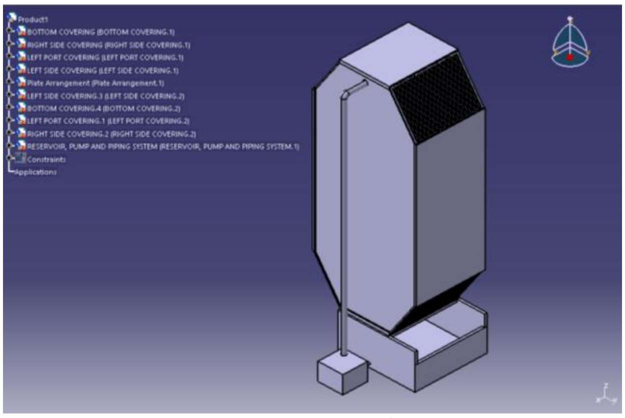
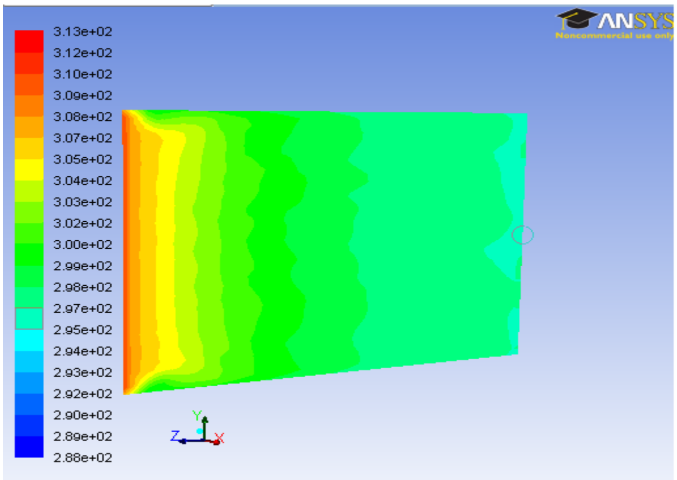


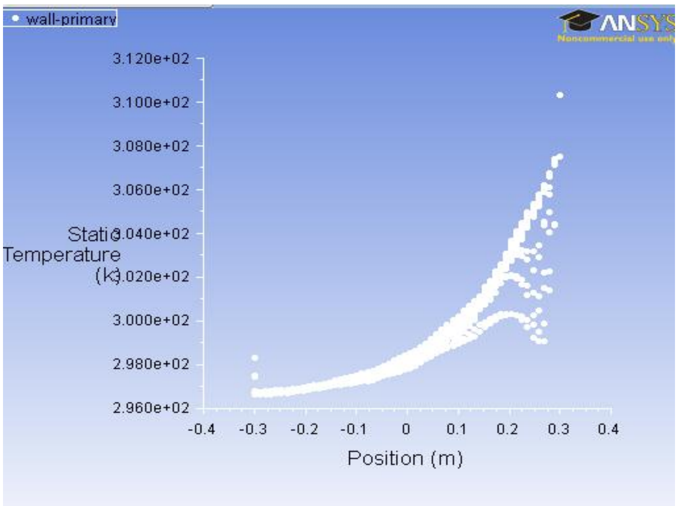
Fig. 14 – Assembly of all parts

VI. CFD ANALYSIS

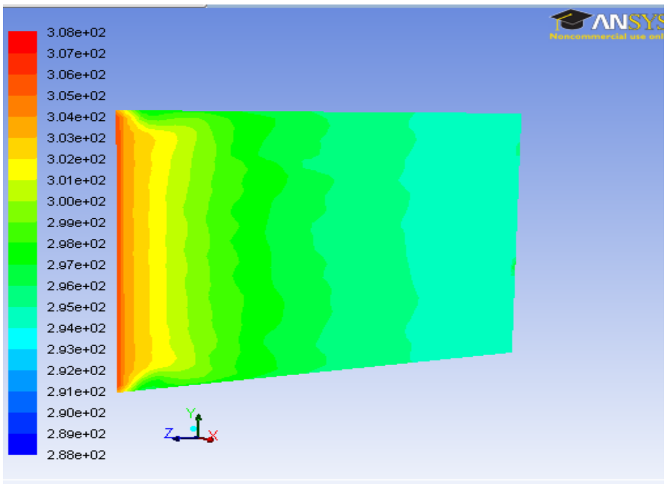
The CFD analysis of the Indirect Evaporative Cooler is carried
out and the results obtained are shown below:



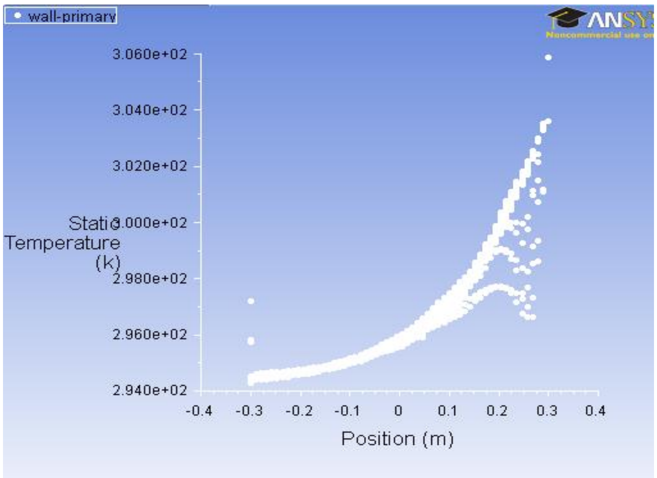
(a) Contours of Temperature along the length of plate



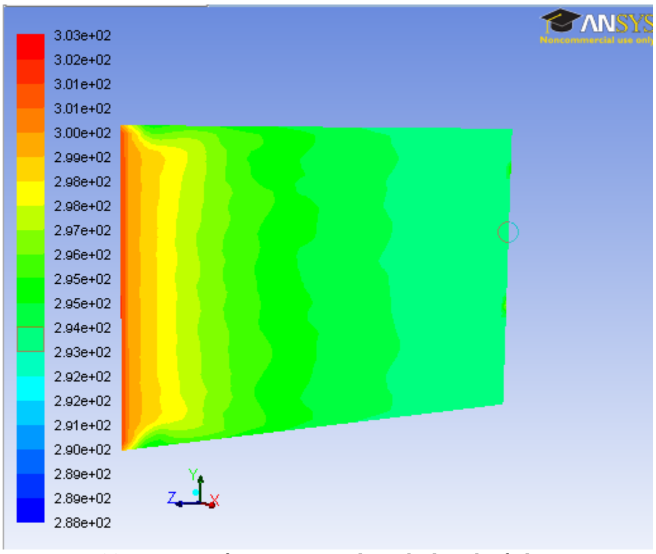
(b) Graphical plot
Fig. 15- Variation in temperature of primary air in the dry channel when Tpi = 40 °C



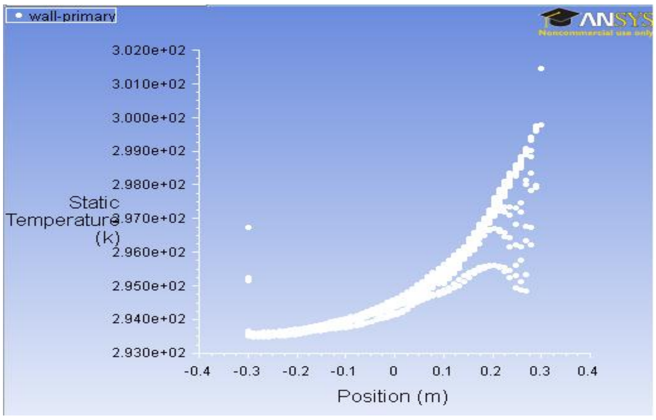
(a) Contours of Temperature along the length of plate



(b) Graphical plot
Fig. 16- Variation in temperature of primary air in the dry channel when Tpi = 35 °C

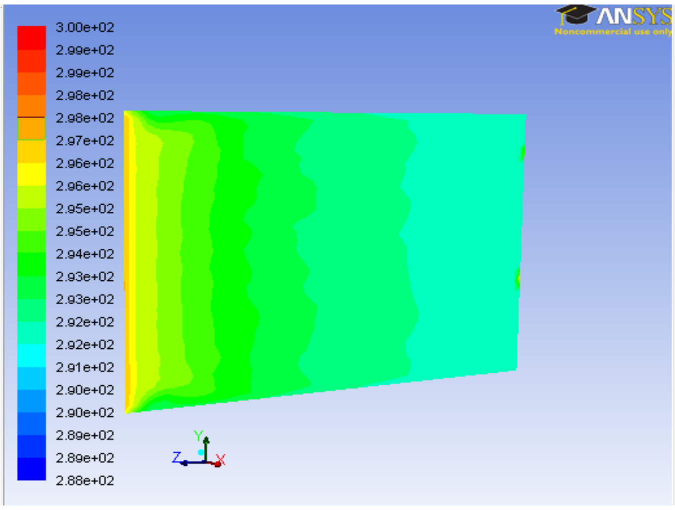


(a) Contours of Temperature along the length of plate

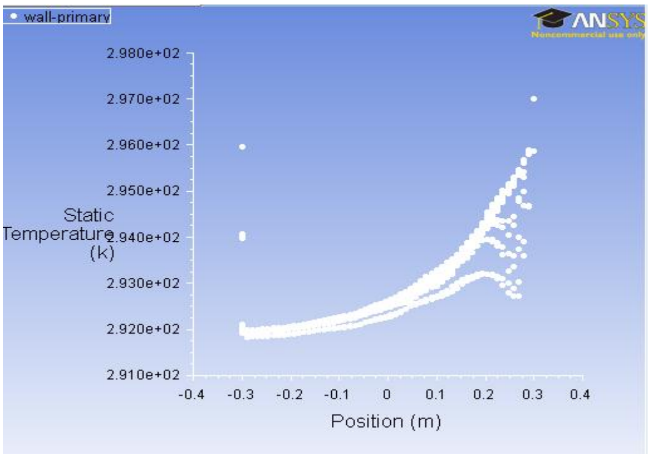


(b) Graphical plot

Fig. 17- Variation in temperature of primary air in the dry channel when Tpi = 30 °C



(a) Contours of Temperature along the length of plate



(b) Graphical plot

Fig. 18- Variation in temperature of primary air in the dry channel when Tpi = 25 ° C

CASE	INLET TEMPERATURE OF PRIMARY AIR (°C)	SUPPLY AIR TEMPERATURE (°C)
I	40	23.0
II	35	21.1
III	30	20.4
IV	25	18.8

VII. CONCLUSION

The proposed model of Indirect Evaporative Cooler (IEC) is Flat Plate – Concurrent type. A total of 70 flat plates are used. Size of the plate used is 0.6m*0.4m with thickness of 0.4 mm. The material used for the plates is Aluminum due to its higher thermal conductivity, easy availability, low density and low cost. The plates used are wick attained on one side to increase the water retaining capacity of the Aluminum plates. These plates are arranged parallel with spacing of 4mm. The primary air and secondary air is flowing alternatively through the flow passages. The proposed design is giving temperature reduction of the supply air to upto 18-22 °C with volume and mass flow rate of 175 m3/hr and 200 kg/hr. The theoretical results obtained by solving the differential equations where validated by the CFD simulation.

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