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Thermal Behavior of Different Cooling Towers Using Induced Draught Cross Flow

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Abstract

Cooling tower is vastly utilized in industries as well as plants such electrical energy generation field and petrochemical industry to subtract the useless heat . In this paper, a thermal behavior of mechanical draft cross flow wet cooling tower is investigated .The theoretical study based on mathematical model of heat and mass transfer whose analyzed by ANSYS fluent software code employed workbench. Three new configurations of cooling tower are studied at the first time that didn’t studied together in literatures. They are cubic, cylindrical, and pyramid shape. The investigation used several inlet water temperature and different pressures of water at inlet and outlet. The numerical solution is based on the finite volume method. The wet bulb temperature of air has constant value for all cases whose studied . The results of study indicated that the cylindrical configuration is the best geometry. When inlet water pressure get up at constant other variables, the performance is decreased as well as it will improved if it reduces. The water cooling range has no influence on the tower characteristics.

Keywords: cooling tower, cross flow, Ansys, cylindrical, cubic, geometry.

1. Introduction

Cooling tower is a device used to reject the heat from the hot water to environment. It widely employed in industrial plant such as petrochemical or energy generation plant. It divided to three mainly types: natural, induced, forced draft .The induced draft is the counter and cross flow. Bourouni et.al. [1] studied the concentrate on thermic attitude of the cross flow mechanical cooling tower through an improved mathematical model counting the variation of mass flow rate of water inlet cooling tower. The analyses of liquid and gas temperature apportionment confirmed the tower owned passive convection on the tower altitude. The investigation illustrated the variance in water temperature inlet and exit of the device was higher than the air because of the domination vaporization potential matched to convective. So, it was considered the difference of the air moisture on the equipment as well as vaporized liquid quality. The liquid waste quantity was 5.1 percentage of the gross liquid that enter to the device. Ebrahim et.al. [2] presented a mathematical equations that employed to know the thermic attitude of the tower from the cross flow type beneath mutable wet bulb temperature. The result which got from this study are matched with empirical datum in the different running statuses. It concluded when the wet bulb temperature get up the range, approach and loss in evaporation augment considerably . Gudmundsson [3] estimated the execution of wet –tower model fill promoted by Reuter model which derived it in Cartesian coordinates was derived for a rectangular tower as well as matched to counter and cross flow Markel , e-NTU as well as Poppe models. The results appeared the Reuter model was the same as poppe way for current –cross flow fill form and Markel has done used Two models Two and three dimensions by Ansys fluent software . Naik and Muthumar [4] presented paper to study the execution of standard cross flow cooling tower utilized in the water chilled condenser of manufactory in expression of water lack ,range ,approach as well as tower efficiency have surveyed .Empirical investigation have been done in three towers used in 900 Ton refrigeration the capacity of the refrigeration industry over a four months in north-eastern zones of India. It monitored at the rush hour (2 Pm) water lack was about 0.0034 m^3/ h -TR matching 34 ºC dry bulb temperature as well as 31.5 ºC wet bulb enrolled on 22/8/2013.At the trial period the mean value of tower range was 2.6,approach was 4.6 ºC as well as the efficiency was 35 %utilizing a simple arrangement cooling coil owning about 8-12 ºC temperature of the dew point. Juangjandee and Sucharitakul [5] surveyed the cooling tower performance of 300 MW from thier investigation found that the thermal efficiency of cooling tower is 51.63 %,lower than 70.97 % designed efficiency because of the high turbidity whose happened on hot water sink of the tower, it was commended to scour , scrub and erase thallophytic industry in the hot water sink, the second cause is the plugged nozzles at hot water sink required to reform promptly, the final reason mentioned in work was water flow rate ought to be readjusted maximize the water flow to design value. It is recommended to be readjusted all distribution valves because of poor condition of the hot water flow distribution. Sundarand MohanKumar [6] presented study of execution test of Refrigeration(TR) baste enhancement plastic tower. Different parameters were assayed to emend the essential deficiencies. The design imperfections were modified and repaired as well as again analyzed employed both the apparatuses and the suggested system were verified to be efficient .The efficiency was improved from 30 % to 50 %. Immaual and Rajakumar [7] employed cross flow tower in their investigation. The study was based on the first law of thermodynamic to anatomize. Heat and mass transfer rule was utilized to solve the theoretical model by step wise process. The egression case of the water and air were found by theoretical model to use it in energy calculation before this study of tower execution in the plant is impossible. The model is approved with empirical outcomes, the error is 3.6%. From this work the operator can readily analyze the execution of cross flow tower in plant utilizing the datum registered and by mathematical model. The investigation concluded by the empirical at some of data used the best effectiveness at high wet bulb temperature as well as low relative humidity. It lightly relied on the inlet water temperature. To minimize the water evaporation rise the wet bulb temperature at fixed wet bulb reduce the dry bulb temperature, raise the relative humidity. The egression temperature decline at dipped flow ratio as well as dipped entrance water temperature.

Mantelli [8]suggested a modern technology for fractional recovery of water waste to the atmospheric in towers. It was comprises of metally porous mode was fabricated from metal as stripe sponges, moist hot air filtered by it, before departing towers. The porous mode was industrialized of high conductivity material. The cooling tower and air currents were simulated numerically to validated the viability of the suggested technology. The small scale tower empirical install was tested for a porous mode with the refrigeration pipes that is recovered 10 % of the wastewater environment. M.E. Hassen and Gubban,[9], studied Al-Nassiriah thermo-power plant which contain counter flow cooling tower . A new cross flow cooling tower modeled instead of existing counter flow cooling tower. Cross flow tower analyzed by finite difference method. The study used different design parameters for the Two type of cooling tower and used variety operation condition such different weather and changing Liquid to gas ratio. The result of the Two cooling tower types demonstrated that the cross flow tower has high thermal performance . A computer improved to make the design computations divergence with various condition.

2- Governed equations

The partial deferential equation was depicting the heat and mass transfer in the raining region for cross flow wet cooling tower contains the temperature domain of pneumatic hodgepodge as well as the domain of the number density of the water steam .

The temperature domain $T_a(r,z)$ of the wet air is expressed[9] :-

$$v_a \frac{\partial T_a}{\partial r} = \frac{\tau_a}{\rho_{ac_{pa}}} \frac{\partial^2 T_a}{\partial z^2} + Q_t \qquad \text{....... (1)}$$

$$\tau_a = 0.004 \tau_{am} Re_h^{0.8} \qquad [11] \qquad \text{......(2)}$$

$$Re_h = \frac{0.8 \rho_s h_s v_s}{\mu_s} \qquad \text{......(3)}$$

$$R \frac{\partial^2 m}{\partial z^2} + Q_m = v_a \frac{\partial m}{\partial r} \qquad \text{........(4)}$$

$$Le = k/(\rho\, cpD) \qquad [11] \qquad \text{........(5)}$$

$$R = 0.04\, R_m Re_h^{0.8} \qquad \text{............(6)}$$

Rmis The molecular diffusion coefficient water steam in air .Reynolds number[Re] is

$$Re = \left(\frac{2 \rho_a R_d v1}{\mu_a}\right) \qquad \text{...........(7)}$$

$$v_1 = ((v_r - u_a)2 + v_z)0.5 \qquad \text{...........(8)}$$

$$\frac{dR_d}{dl} = -\frac{\gamma(Re)(\rho_{s-\rho})}{\rho_{wv1}} \qquad [12] \qquad \text{...........(9)}$$

$$\gamma(Re) = \frac{D_m (2 + 0.5\, Re^{0.5})}{2R_d} \qquad \text{...........(10)}$$

$$\frac{\partial T_w}{\partial l} = \frac{3[\propto(Re)(T_a - T_w) + \gamma(Re)(U - c_{wT_w})(\rho_{s-\rho})]}{c_w \rho_w R_{dv1}} \qquad \text{.......(11)}$$

$$\propto(Re) = \frac{\tau_{am}(2 + 0.5\, Re^{0.5})}{2R_d} \qquad \text{........(12)}$$

The variation of droplet speed over the path:

$$\frac{dv1}{dl} = \frac{g}{v1} - C(Re)\frac{\rho_a(v_z^2 + (v_r - u_a))}{2v1} \frac{\pi R_d^2}{n} \qquad \text{..............(13)}$$

Estimation of mean value of the difference of enthalpy [13]

$$\vartheta = nwc_{pw}(TWI-TWO) \qquad \text{---------(14)}$$

$$\vartheta = n_a\,(Hg1-Hg2) \qquad \text{------------(15)}$$

From equations (14 and 15)

$$T_2 = T_2 - \left[\left(\frac{n_a}{n_w}\right) * \left(\frac{H_{g1} - H_{g2}}{c_{pw}}\right)\right] \qquad \text{---------(16)}$$

$k_g A$:- the volumetric heat transfer coefficient(kg/m3s) ,

ΔHm :- mean driving force (kJ/kg)

$$k_g A = \frac{n_w c_{pw}(T1 - T2)}{la \Delta H_m} \qquad \text{---------- (18)}$$

$$k_g A = \frac{n_w \times c_{pw} \times (T1 - T2)}{l \times a \times \Delta H_m} \qquad \text{-------- (19)}$$

$$\Delta H_m = f \times \gamma_m = f(H_{mw} - H_{gm}) \qquad \text{-------(20)}$$

Effectiveness [14]

$$\in = \frac{t_{wi} - t_{wo}}{t_{wi} - t_{airwet\ bulb}} \qquad \text{-------(21)}$$

Cyclic of concentration [14]

C O C = Tdsbleed/ Tdsmake up ------- (22)

$$\frac{k_a V}{L}[11] = \int_{T_1}^{T_2} \frac{dT}{h_1 - h} \qquad \text{--------(23)}$$

Evaporation loss[14]=

$$0.00085 \times 1.8 \times (T_{wi} - T_{wo}) \times circulated\ water\ flow\,(m^3/_{hr}) \qquad \text{-----(24)}$$

3-Boundary condition

It is used the pressure and temperature for the inlet water and inlet air respectively. It inserts pressure for the exit water and air respectively. Then it is obtained the temperature values of exit water and exit air. The value of input boundary condition and findings values as following: Firstly:- Cubic design: A:-in the first case datums of the inlet water, inlet air ,exit water and exit air; the pressure and temperature were P= 7500 pascal ,T= 313K; P= 1250 pascal, T=288 K , P=-3000 pascal ,T= 301 K; P =-1500 pascal and T= 290 K respectively.

B:- the second datums the same first case inputs except case inlet water pressure reduced to 1700 pascal and outlet water pressure varied to -800 pascal

C:- The third state equal the first case (A) above excluding the inlet and outlet water pressure changed to 1250 ,400 pascal respectively .

D:-The forth case is as the third case but changed the inlet water pressure to 1100 Pascal.

E:-The fifth case in cubic figures of the pressure and temperature of inlet water, inlet air ,exit water and exit air were P= 1500 pascal ,T= 313K,, P= 1250 pascal , T=288 k, P=-400 pascal ,T= 300 k ,P =-1500 pascal and T= 290 k respectively.

Secondly:- Cylindrical design:

A:- In the first case datums of the inlet water, inlet air ,exit water and exit air were P= 3614 pascal ,T= 312.82 K, , , P= 1254 pascal , T=282,85 k, P=-3607.6 pascal ,T= 300.92 k ,P=-1500 pascal and T= 292 k .

B:-The second datums as the first except the water and air temperatures at inlet and outlet T_{wi} =386.15K ,Two = 382.15 ,T_{ai} = 288.15 K , T_{ao} = 299.15 K respectively .

C:-In the third condition used the same final state in the cubic tower geometry except the inlet water are pressure p_{wi} =7000 pascal.

Thirdly:- Pyramid design:

employed inputs of the inlet water, inlet air ,exit water and exit air were P= 700 pascal ,T= 313K, P= 1250 pascal , T=288 k, P=-400 pascal ,T= 301 k ,P =-1500 pascal and T= 290 k consecutively.

All the previous cases are analyzed under the same condition of wet bulb temperature.

4- Mesh Independent

For the selection of suitable mesh size many sized of mesh were selected to know one of the important parameters in this paper that is the exit temperature. Many iterations were done for them to choose the correct size. The correct cell dimension is about 0.06 which made the exit temperature value is constant approximately as shown in Fig. 1 below.

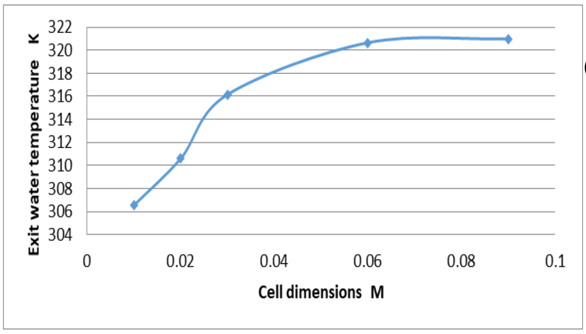


Fig. 1 cell dimensions selection with exit temperature

5- Solution procedures

The solution method, problem procedure flow chart and algorithm are clarified in the following chart of Fig.2.

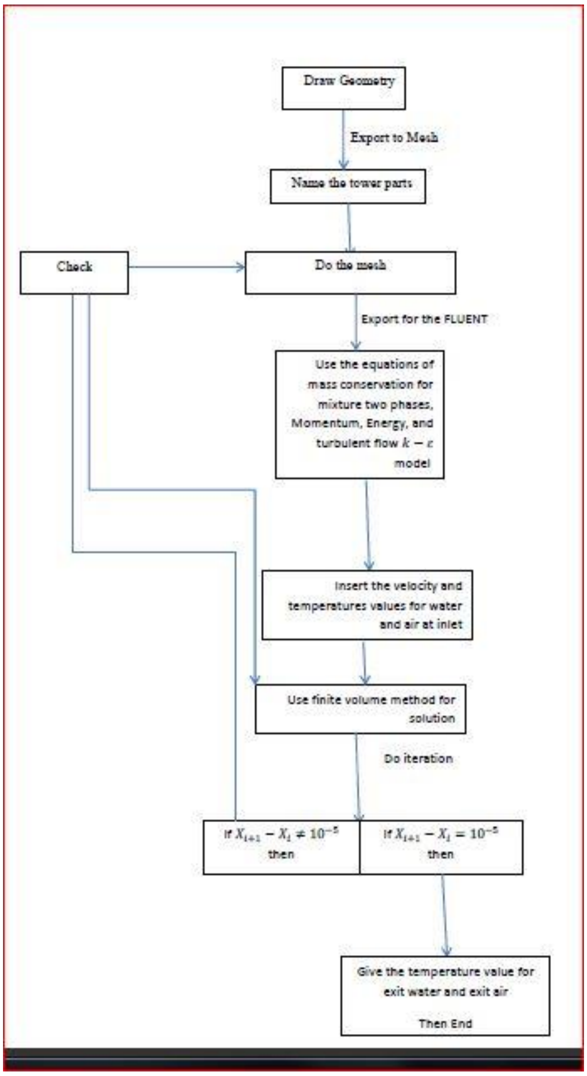


Fig. 2 Algorithm of solution procedures

6- Results and discussion

Ansys fluent 15.0 was employed to analyze the tower figure influence on thermal behavior. There are three configurations that studied as following:

1- Cubic cross flow wet cooling tower Fig.(3):- Three different cases of data is surveyed .

2- The cylindrical cross flow wet cooling tower design as shown in Fig.(20) :-

3 - The pyramid cross flow wet cooling tower design which shown in Fig.(31):-

All the previous cases were analyzed under the same conditions such dry bulb temperature and pressure for air and water at inlet and outlet . The contours of static temperature for figs.(4), (21), and (32) for this conditions appear that the cylindrical tower is the best because the cylindrical design tower has less exit water temperature. Two cooling ranges were used, several water inlet pressure were employed in present study. When the inlet water pressure was minimized ,the water and air heat exchanger is increased, vice versa, that the same physical concept for thermodynamic fluid temperature loss is grew if the pressure of this liquid minimized directly. The rise of cooling range from $3^{O}C$ to $13\ ^{O}C$ that increased the water loss by evaporation with dragged air by fan to atmosphere by 192.93 m^3/hr, that means to minimize waste water with evaporation ought to reduce cooling range.

Figs. (4-9) are different contours for Temperature of cubic shape sections cooling tower. The lower values on the bar of magnitude distribution represent the ambient air temperature.

Figs.(10-14) illustrated pressure distribution. The lower values on the bar of magnitude distribution represent the draught fan pressure hence it in minus values.

Figs.(15-19) show the velocity distribution for the cubic design of cooling tower for the five different cases.

Figs. (21-24) represent the static pressure magnitudes for cylindrical shape design with X,Y,Z direction respectively while show that the inlet water pressure has no influence on thermal behavior in the X direction of tower and it is quasi steady at the same temperature for all cases with position except in forth case, the temperature value is high with position (0.2- 0.4) m because this distance represents the metal thickness who separates the fill and air vacuum chamber. This metal has high temperature because of thermal convection.

Figs. (25-27) indicates when the inlet water pressure increase the thermal behavior decreased because the exit water temperature is increased and other parameter in equation (21) is constant and the reduction of the inlet water pressure leads to increase the thermal behavior. This result is agreed with thermodynamic concept which state that the fluid pressure and is temperature are proportioned directly[18].

Figs. (28- 30) show that the velocity increase at the ends of cylinder because of air induced and water pushed. The velocity and its area of flow is represented the flow rate of liquid according to equation (16) the ratio ($\frac{m_w}{m_{air}}$) has directly proportioned with exit water temperature.

Figs. (32- 35) reveal the behavior of parameters of pyramid cooling tower sections.

Figs.(36,37) represent the static temperature magnitudes with X direction of the cubic tower design show that the inlet water pressure has no influence on thermal conduction in the X direction of tower and it is quasi steady at the same temperature for all cases with position except forth case, the maximum temperature value is in position (0.2- 0.4) m because this distance represents the metal thickness who separate the fill and air vacuum chamber. This metal has high temperature because of thermal convection.

Fig. (38) shows the static temperature behavior for three cases of study to indicate the same manner of curves for cylindrical tower considered for X- axis. This case means the increase of air temperature leads to increase exit water temperature. This result agreed with equation (16), the increase in air temperature leads to increase its enthalpy. The third case of this figure is the same properties of fifth case of cubic shape with increasing inlet water pressure. These curves findings are the difference among the values at the center of tower that the second case is more effective than others because of lower temperature of mixture.

Figs. (38,39) explain the thermal behavior in X,Z direction of the cylindrical design respectively. All cases are similar in conduct, all curves has different temperature value because the cooling range is different. The sides of all curve represent the inlet water temperature. Thermal behavior is agreed with equation (16).

Fig. (40) presents the static temperature of pyramid shape with X- axis direction. From (0-0.1) m is the distance between the upper surface side of tower (fill) which the temperature remain constant but the other distance (0.1-0.3) m is the center of fan, the curve decrease to minimum value at the tower center.

Fig.(41) implies when the inlet water pressure increase the thermal behavior decreased and the reduction of the inlet water pressure leads to increase the temperature of the space. The result is agreed with thermodynamic concept [8].

Fig.(42) coincides with the fact "the temperature values are constant with the tower height for the second and third case because of the low inlet water pressure compared with first case that agree with thermodynamic concept [15] which state the fluid is desired to loss temperature with its pressure decrease", means the inlet water cooled in higher point of tower and has the same magnitude at the bottom of device. Any case has different magnitude of cooling range. The high inlet water pressure in the first case leads to variation of their curves manner compared to other cases.

The comparison for all cooling tower designs in this paper appeared through the figs.(41,42,43) conclude the thermal behavior of cylindrical design of tower is the best, the cubic with inclined walls (pyramid shape) is better than cubic shape. The discussion of the pressures as well as velocities of cylindrical cooling tower design with X,Y,Z directions illustrated the change cooling range has no influence on the cooling tower characteristics as mentioned [16].

Fig.(44) the temperature is steady with position. The influence of inlet water pressure is inversely with thermal behavior in tower thickness.

Figs.(45, 46, 47) show that the inlet water pressure compared with pressure of inlet and outlet of air , the maximum value of pressure occurs at the first third of distance of X direction and when the inlet water pressure is nearby the inlet and outlet air pressure, the increase and decrease in pressure value will be regular in the horizontal direction of tower and steady at the same magnitude of pressure with the height of device. High inlet water pressure relative to air pressure at entrance and exit causes irregular pressure with Y direction. Inlet water pressure has no significant influence in the tower thickness and it is quasi steady. But Fig. (48) for velocity of pyramid shape shows the same conditions of first case of cubic shape mentioned in Fig. (49) with different manner of curve because of different fill.

Figs.(50,51,52) represent the velocity values with X,Y,Z directions respectively that illustrate the speed grow up with the increase of inlet water temperature and vice versa.

Figs.(53,54) explain the static pressure with position of X,Z direction .The second and third case indicate that the cooling range not affect the static pressure when compared with first and second case observe the inlet water pressure has significant effect on the static pressure.

Fig. (54) shows that the first case conduct differs from the second and third with cooling tower height because of different inlet water pressure while Fig. (55) observe (from curves) the second case is the same as third case conduct, it concluded the cooling range has no effect on the static pressure but the first case differs from second and third cases because its inlet water pressure is the highest. Figs .(56,52) indicate with the second and third case that the cooling range has no effect on velocity in X, Z directions, inlet water pressure has no influence on the velocity behavior in the mentioned directions.

The second and third curves appear the cooling range has no effect on velocity in the tower height which demonstrated by Fig. (57). The first case behavior is different from the other cases because the divergence in the inlet water velocity.

7- Validity of study:

The comparison between the present study with reference R. UmeshSundar et**.** al**.** **[15]** shows that the static temperature is the same because the boundary conditions are the same too approximately, where the input data of Ansys 15 were shown in table (1) as well as the results shown in Fig. (58). The result of exit water temperature was 308 K.

8- Conclusions

The study concluded from simulation and figures some issues as following :-

1- The inlet water pressure magnitude is prefers to be moderate high than the air pressure at the inlet and outlet because the tower will be good thermal behavior and the pressure will be more stability in the device.

2-The inlet water pressure has inversely influence on the thermal behavior and directly effect on the velocity in the tower height .

3 – The change in water cooling range has no effect on tower characteristics .

4- The cylindrical design is the best, the pyramid design is better than cubic in the thermal behavior.

5- The increase in the ratio of inlet air mass flow rate to inlet water mass flow rate ($\frac{m_{air}}{m_{water}}$) leads to increase cooling tower performance.

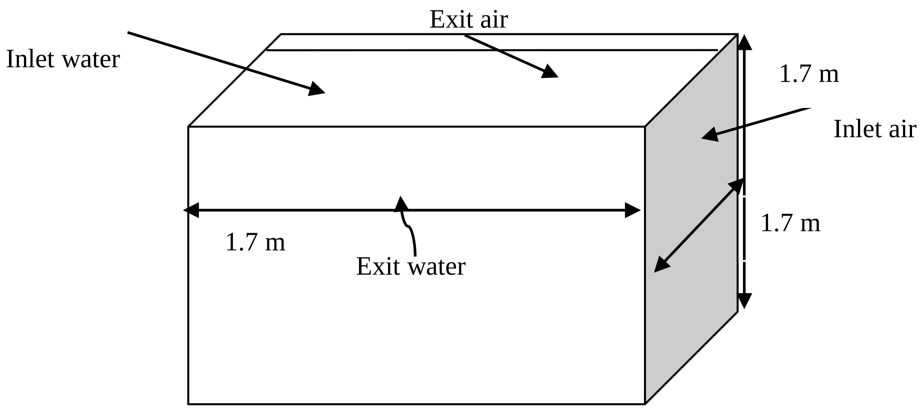


Fig .(3) cubic design for cooling tower

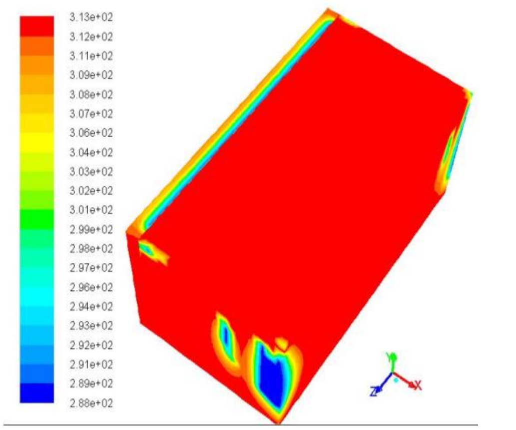


Fig .(4) temperature contour of First case

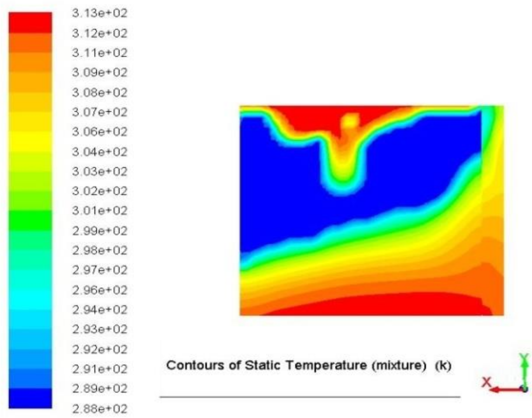


Fig.(5) static temperature contour of first case

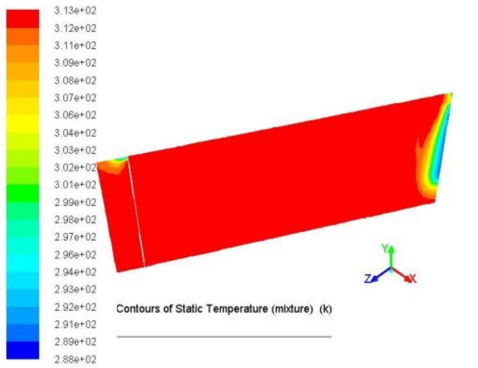


Fig.(6) static temperature contour of second case

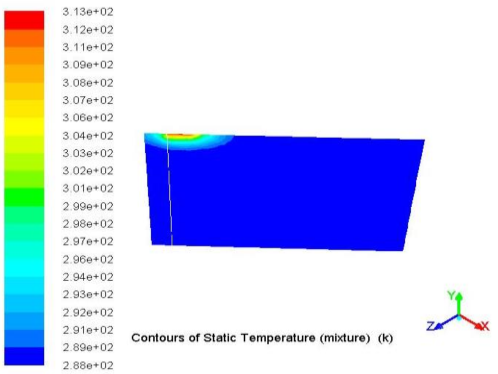


Fig.(7) static temperature contour of third case

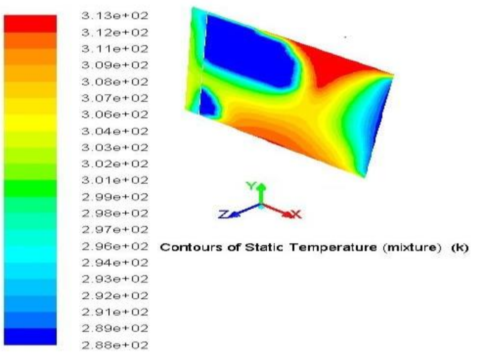


Fig.(8) static temperature contour of forth case

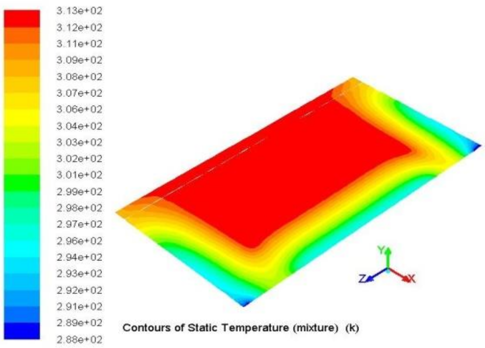


Fig.(9) static temperature contour of fifth case

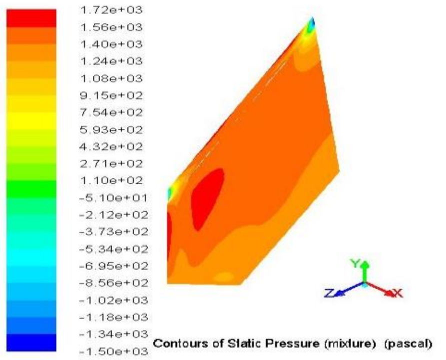
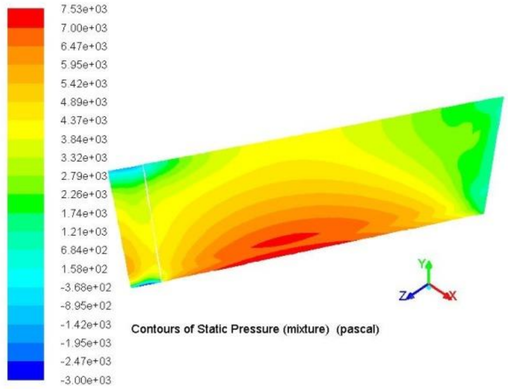


Fig (10) static pressure contour of first case



Fig(11) static pressure contour of second case

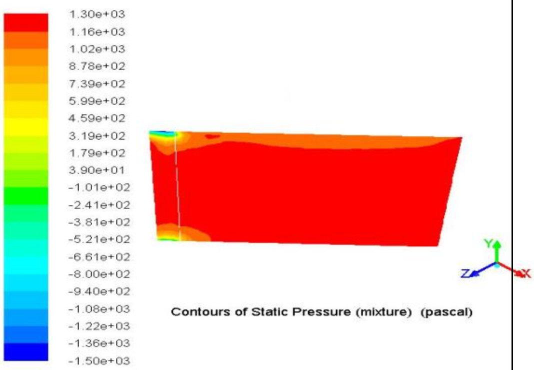


Fig.(12) static pressure contour of third case

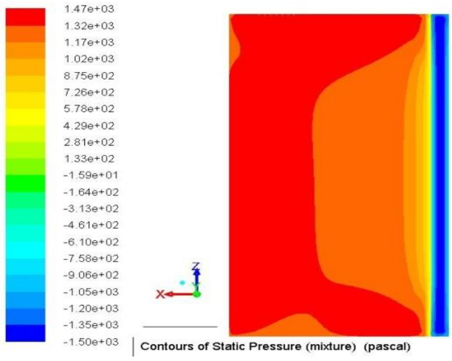


Fig.(13) static pressure contour of forth case

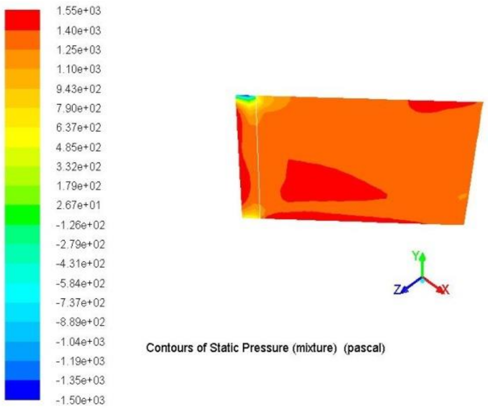


Fig.(14) static pressure contour of fifth case

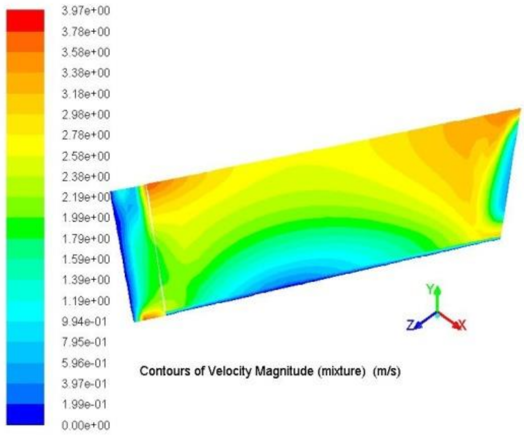


Fig.(15) velocity contour of first case

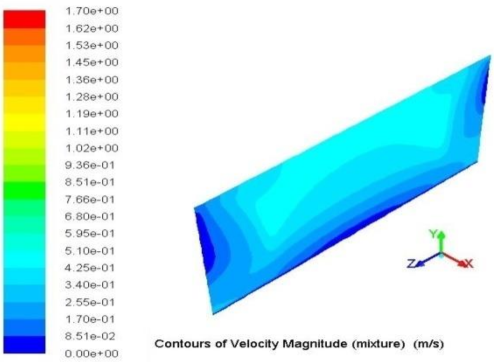


Fig.(16) velocity contour of second case

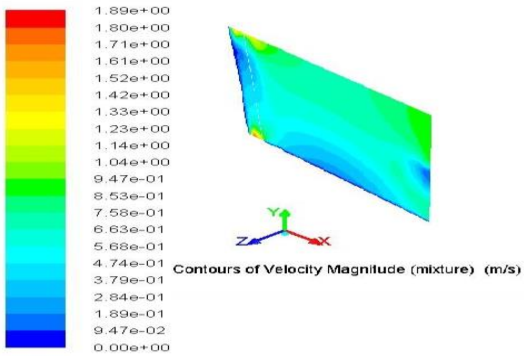
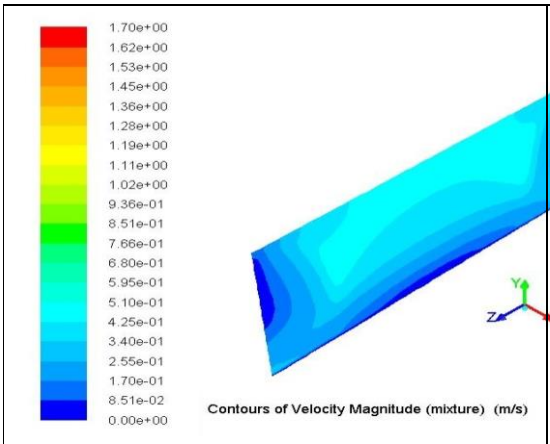


Fig.(17)velocity contour of third case



contour velocity of third case Fig.(18)

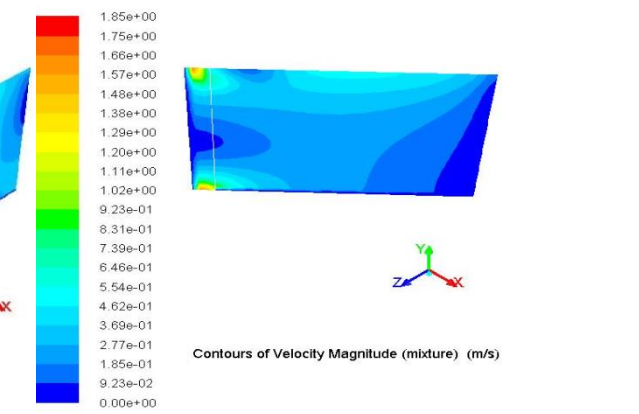
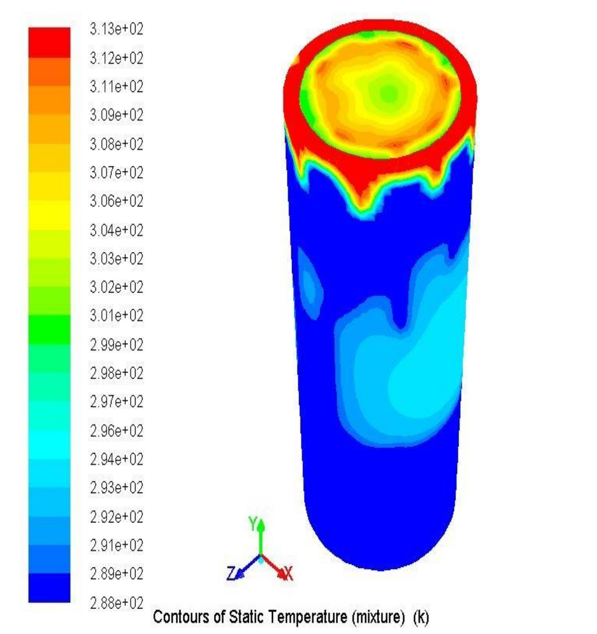


Fig.(19) velocity contour of fifth case



Fig(21) temperature contour of third case

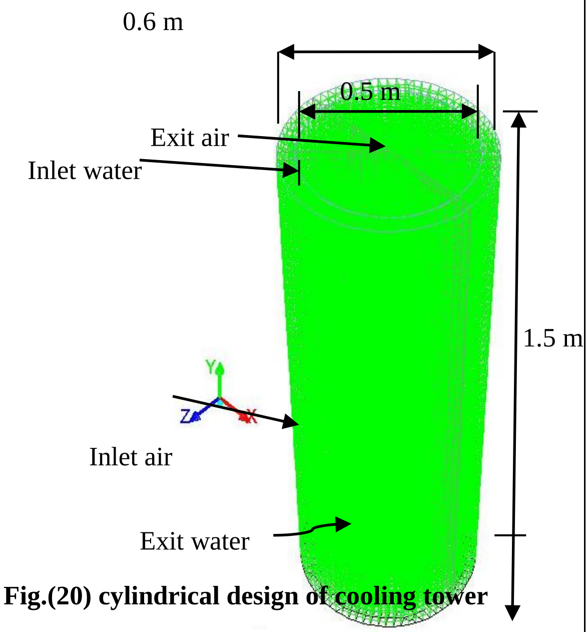


Fig.(20) cylindrical design of cooling tower

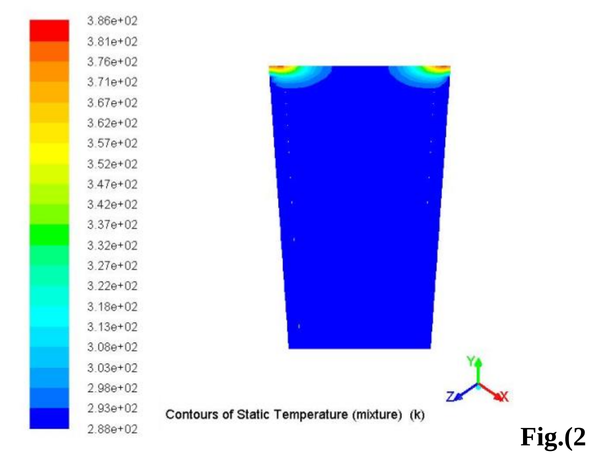


Fig.(23) temperature contour of second case

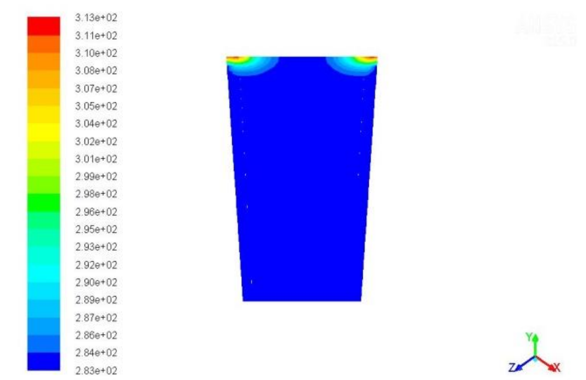


Fig.(22) temperature contour of first case

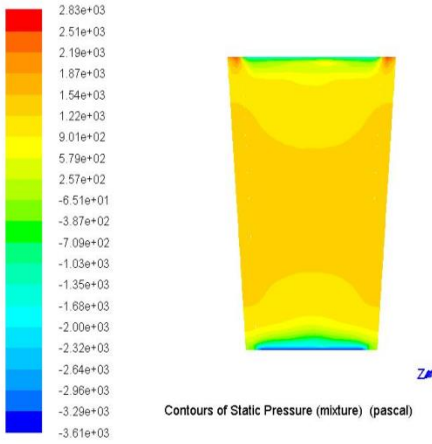


Fig.(25) contour pressure of first case

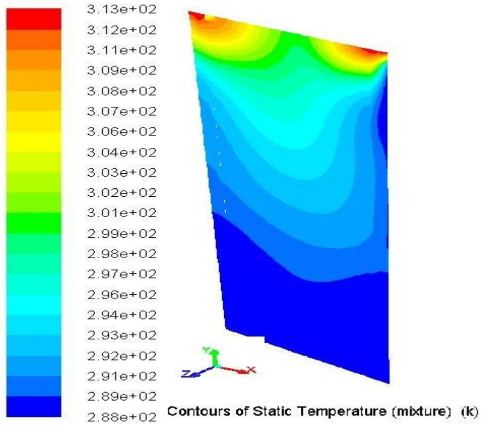


Fig.(24) temperature contour of third case

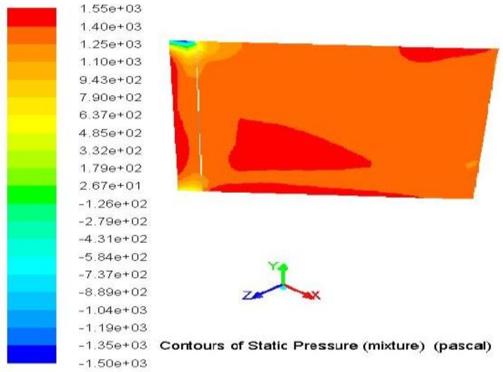


Fig.(27) contour pressure of third case

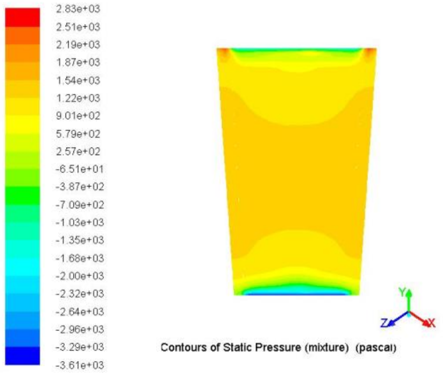


Fig.(26) contour pressure of second case

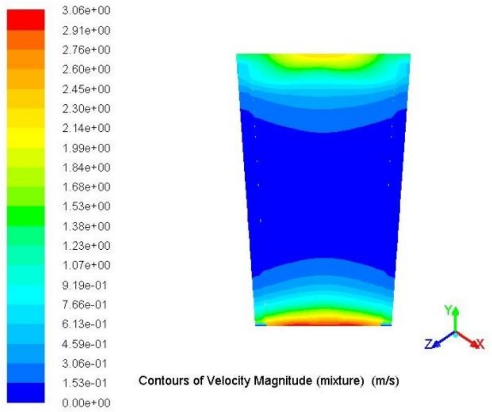


Fig.(29) velocity contour of second case

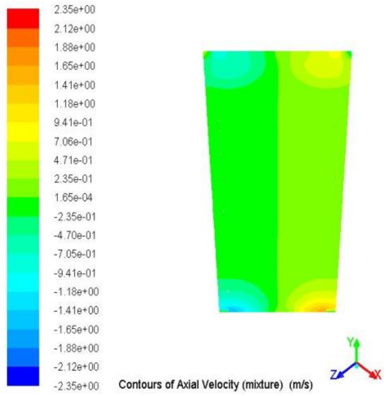


Fig.(28) Velocity contour of First case

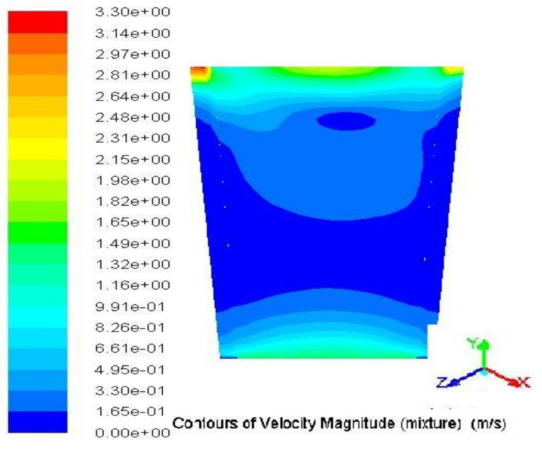


Fig.(30) velocity contour of third case

Pyramid shape

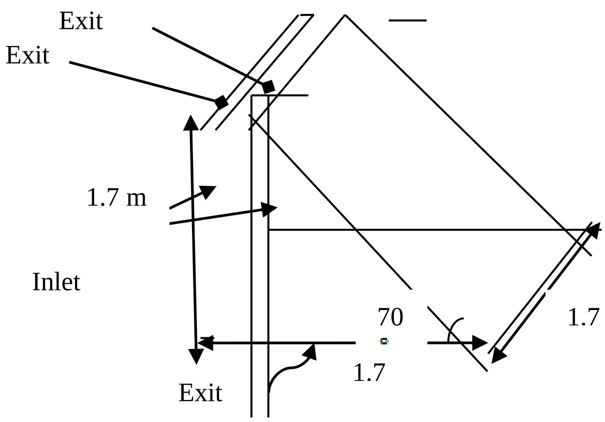


Fig.(31) Pyramid cooling tower design

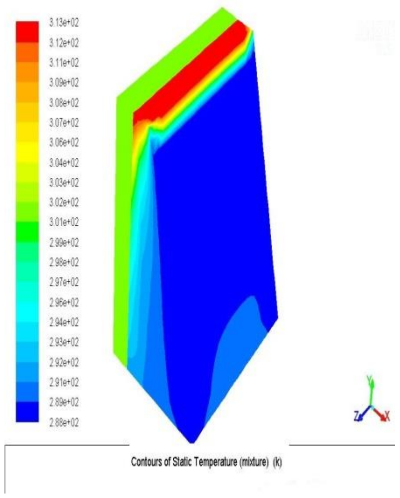


Fig.(32) temperature contour

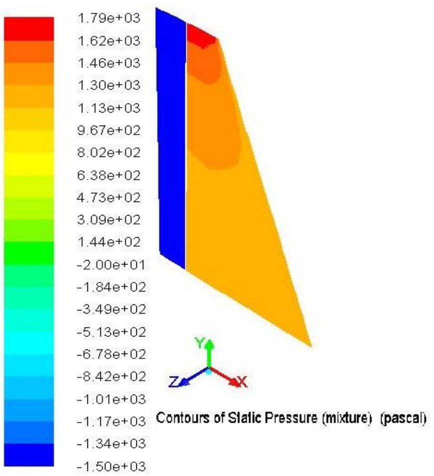


Fig.(33) pressure contour

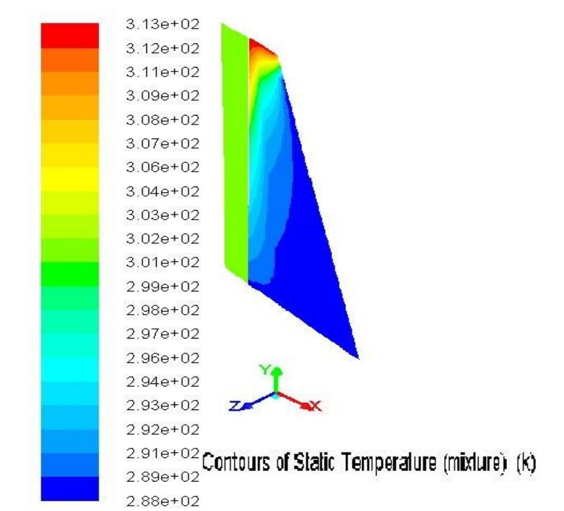


Fig.(34) temperature contour

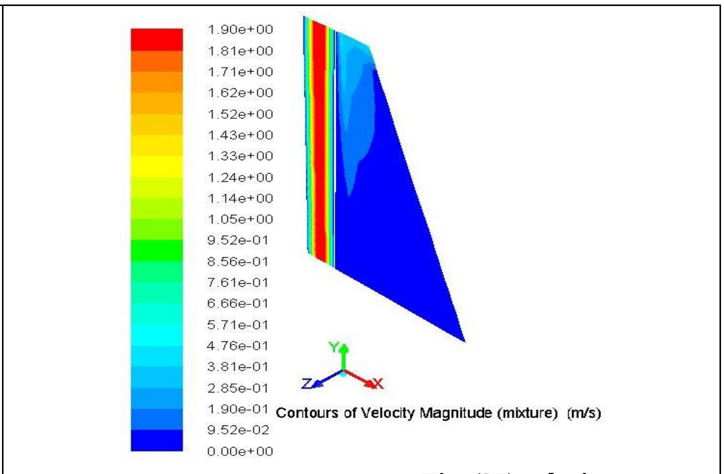


Fig. (35) velocity contour

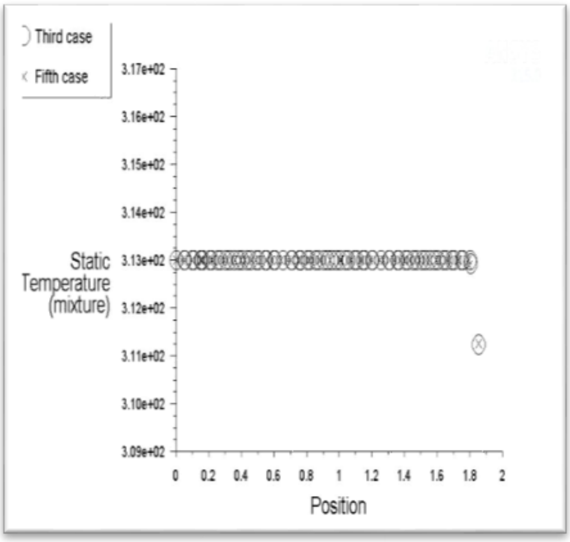


Fig.(36) static temperature with X axis of cubic design

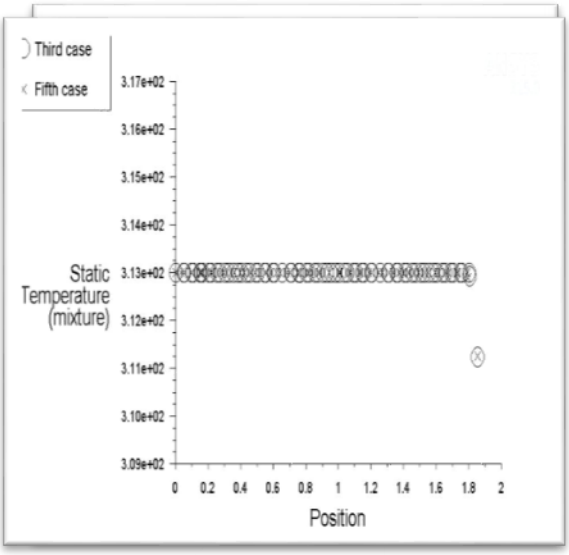


Fig.(37) static temperature with X axis of cubic design

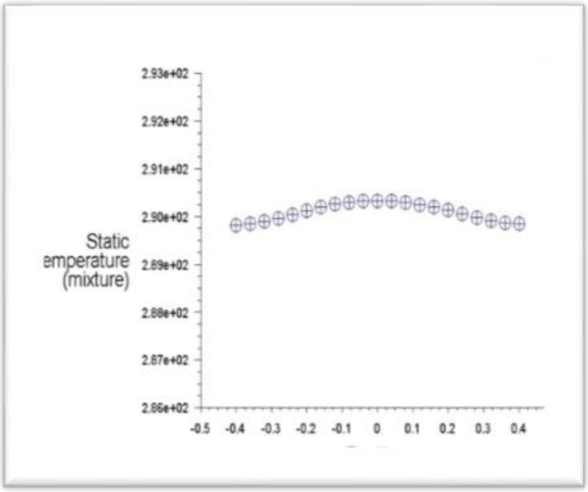


Fig.(38)static temperature with X axis of cylindrical design

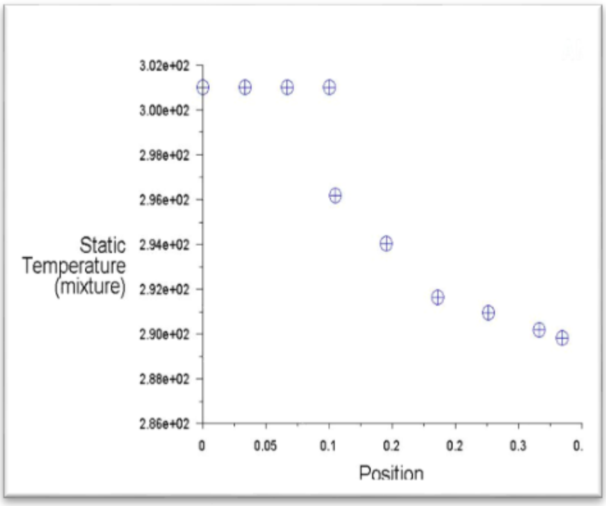


Fig.(39)static temperature with Z- axis of cylinder design

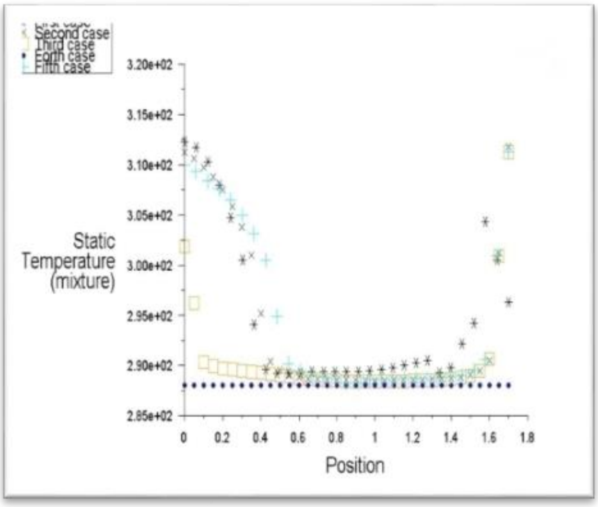


Fig.(40)static temperature with X axis of pyramid design

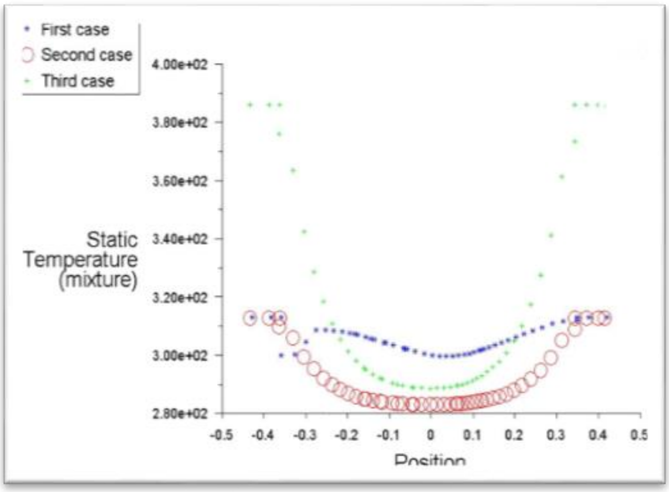


Fig.(41)static temperature with y axis of cubic design

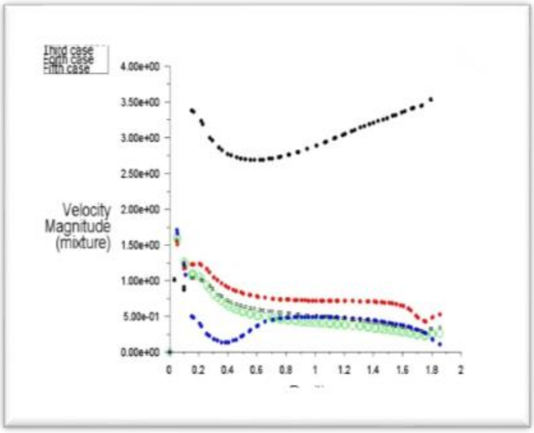


Fig.(42)static temperature with y axis of cylindrical design

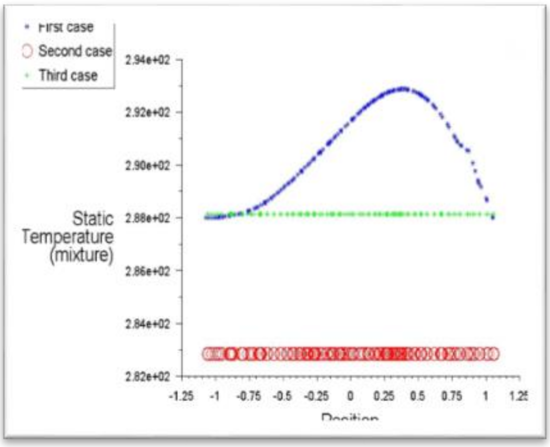


Fig.(43) static temperature with Y axis of pyramid design

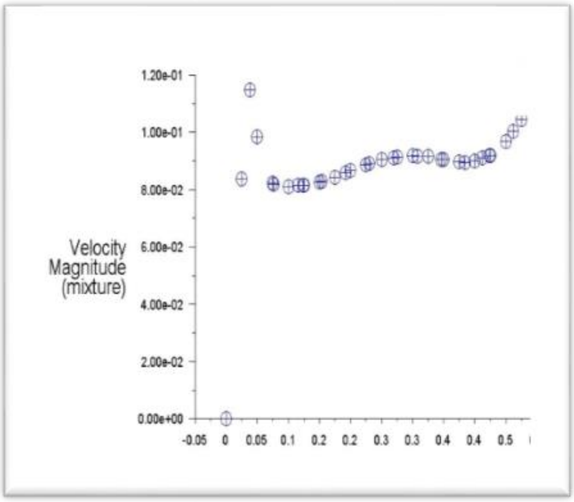


Fig.(44) static temperature with Z- axis of cubic design

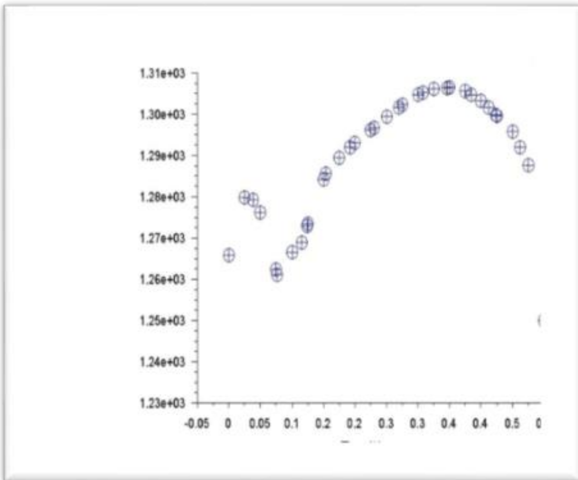
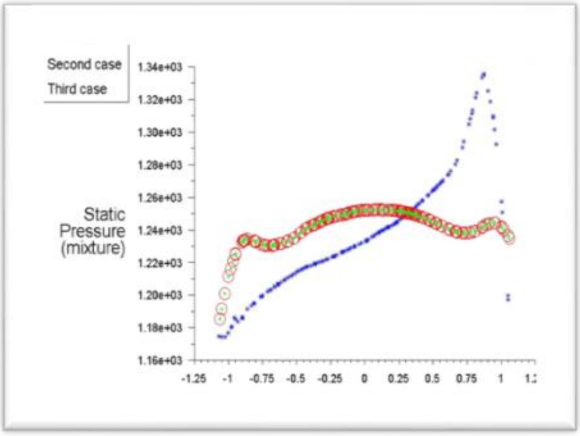


Fig.(45)static pressure with X-axis of cubic design



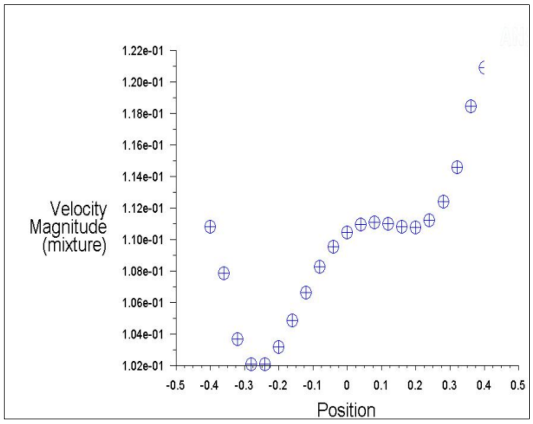
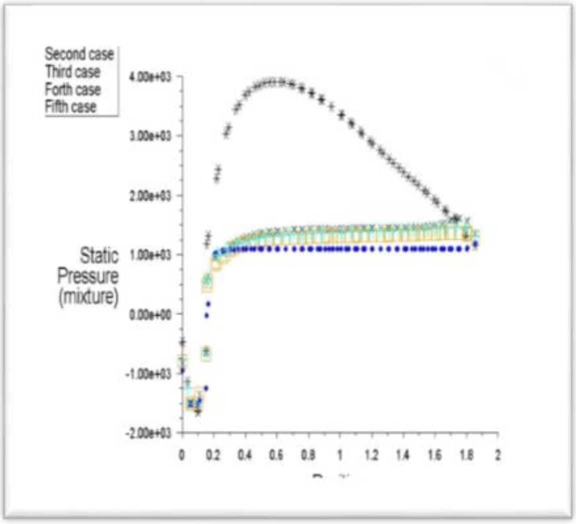


Fig.(46) static pressure with Y-axis of cubic design Fig.(47) static pressure with Z-axis of cubic design



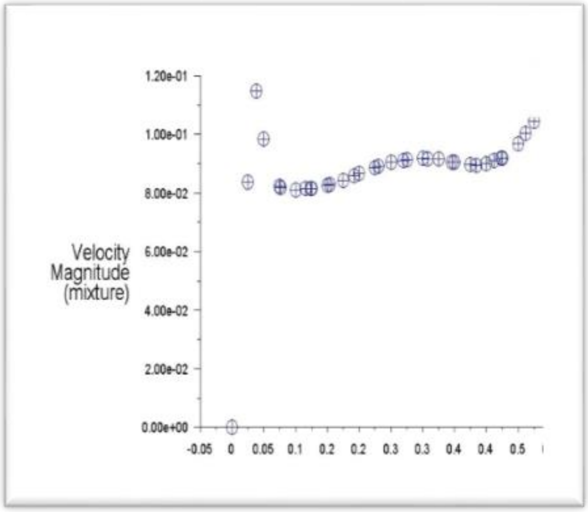
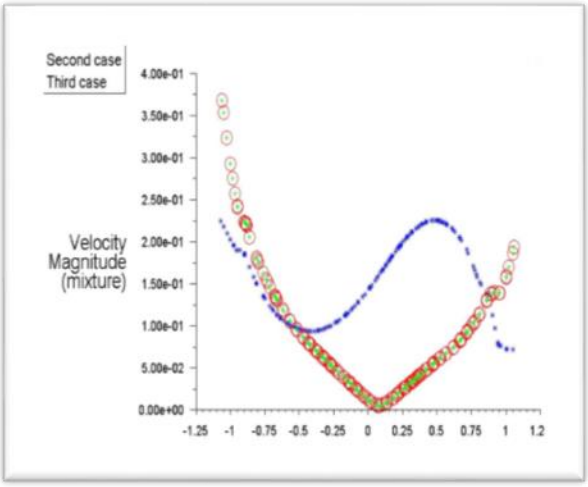


Fig.(49) velocity with Y-axis of pyramid design Fig.(48)static pressure with X-axis of cubic design



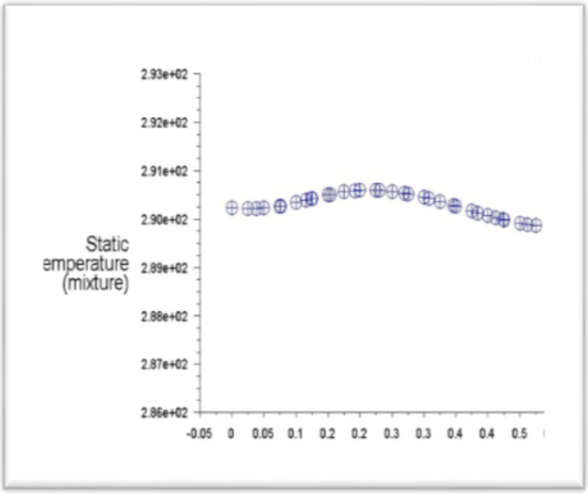


Fig.(50) velocity with X axis of cubic design Fig.(51) Vel0city with Y axis of cubic design

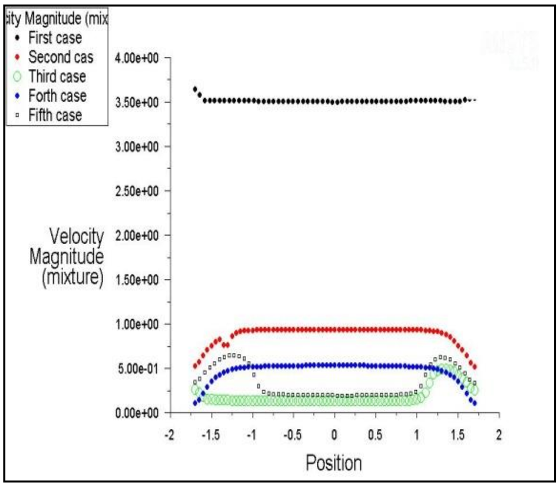


Fig.(52) velocity with Z-axis of cylindrical design

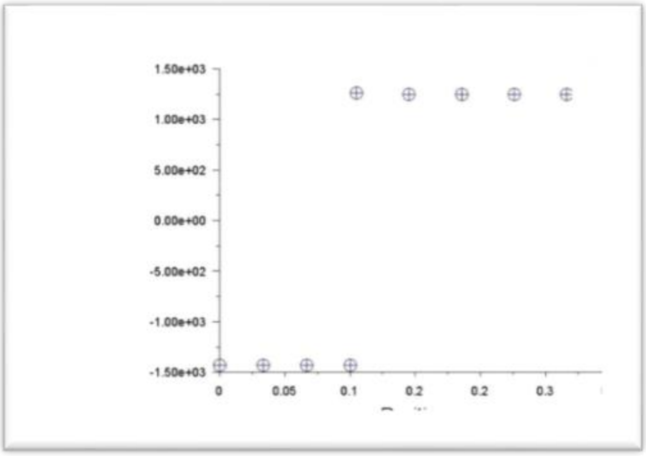


Fig.(53)Static pressure with X-axis of cylindrical design

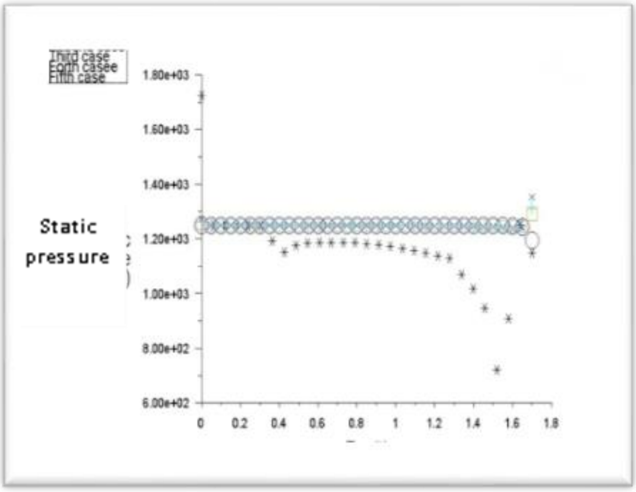


Fig.(54) static pressure with Z-axis of cylindrical design

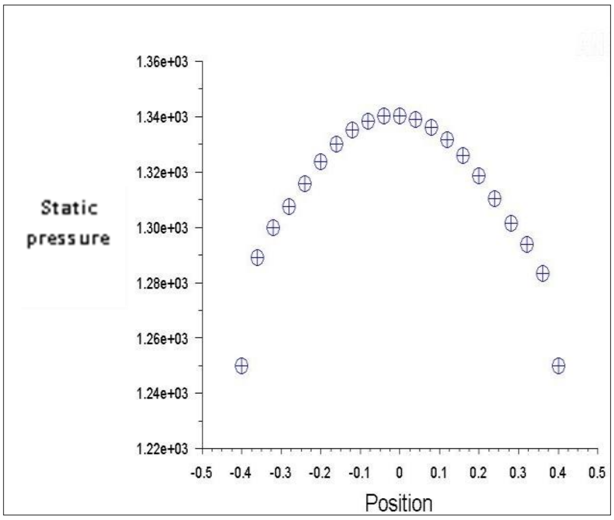


Fig.(55)static pressure with Y-axis of cylindrical design

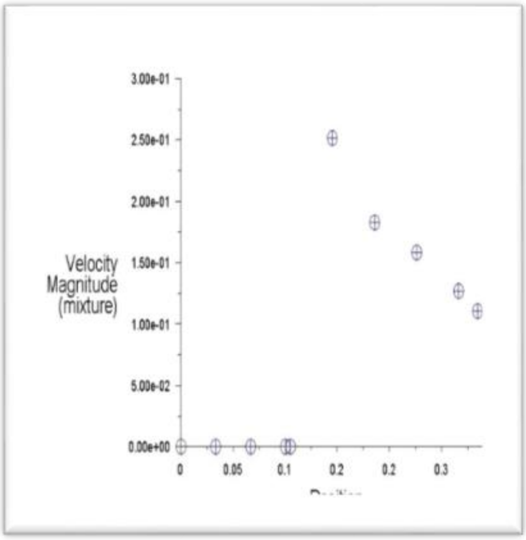


Fig.(56) velocity with X axis of cylindrical design

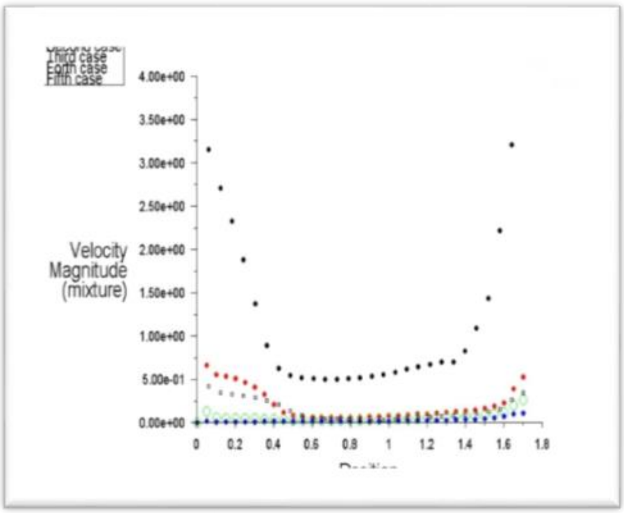
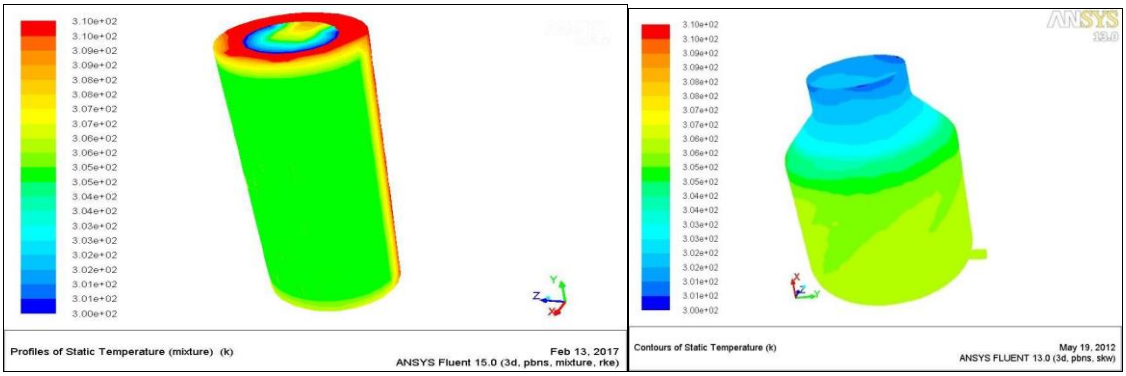


Fig.(57) velocity with Y- axis of cylindrical design



(a) (b)

Fig. (58) The validity of the study (a) present study, (b) R. Umesh Sundarand G.Mohan Kumar [15]

Table (1) the boundary conditions used for validation

	Exit Air	Inlet Air	Inlet water	Exit water
Dry bulb temperature K		305	310	
Velocity m/s		72.6	2652.58	2652
Pressure pascal	-4160			

6. References

[1]K. Bourouni, M.M. Bassem, M.T. Chaı¨bi, "Numerical study of coupled heat and mass transfer in geothermal water cooling tower", Energy conversion and management, ELSEVIER, 49 (2008) PP. 988–994.

[2] Ebrahim Hajidavalloo, Reza Shakeri, Mozaffar A. Mehrabian, "Thermal performance of cross flow cooling towers in variable wet bulb temperature, "Energy conversion and management, ELSEVIER, 51 (2010) PP. 1298–1303.

[3] Yngvi Gudmundsson, "Performance Evaluation of Wet-Cooling Tower Fills with Computational Fluid Dynamics", MSc. Thesis, Stellenbosch University, South Africa, 2012.

[4] B.Kiran Naik and P. Muthukumar, "Performance Investigations of a Cross-Flow Induced Draft Cooling Tower Employed in a Water Cooled Condenser of 900 TR A/C Plant", International Journal of Scientific & Engineering Research, Volume 4, Issue 12, December-2013, ISSN 2229-5518.

[5] PipatJuangjandee, ThawanSucharitakul, "Performance Evaluation of Induced-Draft Cross-FlowCooling Tower",Chiang Mai University, Chiang Mai 50200, Thailand,
2004,https://www.google.iq/url?sa=t&rct=j&q=&esrc=s&source=web&cd=1&cad=rja&uact=8&ved=0ahUKEwjnhIHv0pLSAhXsHpoKHWuPCe8QFgglMAA&url=http%3A%2F%2Fwww.thaiscience.info%2FArticle%2520for%2520ThaiScience%2FArticle%2F3%2F10004598.pdf&usg=AFQjCNFcXY.

[6] R. Umesh Sundar ,Dr.G. Mohan Kumar, "Application of QFD and CFD for Increasing theEfficiency of 30TR FRP Cooling Tower-A Review and Case Studies", International Journal of Mechanical & Mechatronics Engineering IJMME-IJENS Vol:14 No:01, India.

[7] R. Immanual, Dr. S. Raja kumar, "Performance Evaluation and Optimization of Cross Flow Cooling Tower", International Journal of Research in Science and Technology, (IJRST) 2016, Vol. No. 6, Issue No. I, Jan-Mar.

[8] Marcia H. B. Mantelli, "New Passive Vapor Recovering Technologies for Wet Cross-Current Cooling Towers", Applied Thermal Engineering, Volume 108, 5 September 2016, PP.398-413.

[9] Mushtaq Esmael Hassen and Kadhim Hassan Gubban, "A comparison between the counter and cross flow for thermal power plant", Journal of university of Thi- Qar, No.1, Vol.2, June/2006.

[10] S.P. Fisenko, A.A. Brin, "Simulation of a cross-flow cooling tower performance", International Journal of Heat and mass transfer, ELSEVIER, 50 (2007) PP.3216–3223.

[11] A.A. Zhukauskas, High-Performance Single-Phase Heat Exchangers, Hemisphere Publishing Corporation, New York, Washington, Philadelphia, London, 1989.

[12] S.P. Fisenko, A.A. Brin, A.I. Petruchik, Evaporative cooling of water in a mechanical draft cooling tower, Int. J. Heat Mass Transfer 47 (2004) 167–177.

[13] G.B. Hill, E.J. Pring, Peter D. Osborn,"Cooling Towers- principles and practice" Third edition, Butterworth- Heinemann Ltd. copyright, British library, 1990.

[14] Electrical Energy Equipment (Cooling Tower Chapter), India, Energy Efficiency Guide for Industry in Asia – www.energyefficiencyasia.org, 2004.

[15] A.K. Raja, Amit Brakash Srivastava, Manish Dwivedi, "Power Plant Engineering", New Age International Publishers, 2006, India.

[16] Chris Hsligo, "Cooling Towers: Design and Operation Considerations, 2010, Cheresources.com,https://www.google.iq/url?sa=t&rct=j&q=&esrc=s&source=web&cd=1&cad=rja&uact=8&ved=0ahUKEwjPs53s55LSAhVDSZoKHRADAPoQFgglMAA&url=http%3A%2F%2Fwww.cheresources.com%2Fcontent%2Farticles%2Fheattransfer%2Fcooling-towers-design-and-operation.

[17] R. Umesh Sundar, G. Mohan Kumar, "Application of QFD and CFD for Increasing the Efficiency of 30TR FRP Cooling Tower-A Review and Case Studies", International Journal of Mechanical & Mechatronics Engineering, Vol:14 No:01 142401-9696-IJMME-IJENS © February 2014 IJENS, India.

[18] R.K. Rajput, "Engineering Thermodynamics", Laximi Publications 113, Golden House, Daryaganj, 2007, India.

Nomenclature

A : interfacial contact area m2

COC Cyclic of concentration dimensionless

dT : water temperature difference

h1 : enthalpy of saturated air at the water temperature (kJ/kg)

h:enthalpy of main air stream (kJ/kg)

K : overall heat transfer coefficient dimensionless

P pressure (Pascal)

T Temperature (K)

Twi inlet water Temperature (K)

Two outlet water Temperature (K)

Tair wet bulb inlet air wet bulb temperature (K)

v : planar volume

Vxvelocity (m/s) in X direction

Vy velocity (m/s) in y direction

Vz velocity (m/s) in Z direction

Vaxial velocity (m/s)in axial direction

Vtangential velocity (m/s) in tangential direction

Tds Total dissolved solid in the water dimensionless