THEORETICAL AND EXPERIMENTAL STUDY ON THERMAL PERFORMANCE OF CLOSED WET COOLING TOWER

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Abstract

Thermal performance of closed wet cooling tower has been investigated experimentally and theoretically in this work. The theoretical model based on heat and mass transfer equations and heat and mass transfer balance equations which are established for steady state case. A new small indirect cooling tower was used for conducting experiments. The cooling capacity of cooling tower is 1 kW for an inlet water temperature of 38^{o}C, a water mass velocity 2.3 kg/m^{2}.s and an air wet bulb temperature of 26^{o}C. This study investigates the relationship between saturation efficiency, cooling capacity and coefficient of performance of closed wet cooling tower versus different operating parameters such wet-bulb temperature, variable air-spray water flow ratio and cooling water inlet temperature. Results indicate that the capacity and saturation efficiency was found close to the related experimental results. Good agreement was obtained between the theoretical results and experimental measurements for the performance of small cooling tower.

الخلاصة

تم التحقق من الاداء الحراري لبرج التبريد المغلق والرطب. النموذج رياضي مبني على اساس معادلات انتقال الكتلة والطاقة ومعادلات موازنة الكتلة والطاقة للحالة المستقره. استعمل برج تبريد غير مباشر في اجراء التجارب. السعة التبريدية لبرج التبريد هي ا كيلو واط عند درجة حرارة مقدارها ٣٨°م لماء التدوير، سرعة كتلة لماء تدوير مقدارها ٢,٣ كيلو غرام/ م٢.ثا و درجة حرارة هواء رطبة مقدارها ٢٦° م. تم التحقق في هذه الدراسة العلاقة بين كفاءة الاشباع ، السعة التبريدية ومعامل الاداء الحراري لبرج التبريد المغلق والرطب وعوامل مختلفة كدرجة هواء البصلة الرطبة عند الدخول، نسبة جريان الهواء- الماء المرشوش المتغيرة، درجة حرارة ماء التبريد عند الدخول. بينت النتائج بان سعة التبريد وكفاءة الاشباع مقاربة الى النتائج العملية ذات العلاقة. تم الحصول على تقارب جيد بين نتائج النظرية والنتائج القياسات العملية لاداء ابراج التبريد ذات السعات الصغيرة.

Keywords: Theoretical and Computational Analysis, Closed wet cooling tower, heat and mass transfer balance, Wet operation mode, variable air-spray water flow ratio.

1: Introduction

Cooling towers are increasingly used in buildings as a component of environmental cooling systems. The use of cooling towers to reject heat, cool buildings and reduce the temperature of water circulated through various heat rejection equipment have been also increased considerably in recent days. A closed circuit cooling tower maintains an indirect contact between the fluid and the atmosphere. Indirect contact cooling towers have been traditionally used to remove excess heat from various industrial process with hot water temperatures between 32 and 40C^{o} and a typical cooling capacities above 40 kW [1,2].

Cooling towers respectively are called wet tower when evaporative cooling is used, dry cooling tower when air blast cooling is utilized and the hybrid closed circuit cooling tower which is capable of working in wet-dry mode which has the simultaneous characteristics of both dry and wet towers[3,4]. The closed wet cooling tower is one applications of evaporative cooling and can be used to replace the vapour compression system in cooling of buildings. The cooling tower can be combined with chilled ceiling or beams. Highest potential for this concept is cool and dry climates but warm and dry maritime temperate climates offer a significant potential as well [5].

A number of numerical simulation and mathematical modes of cooling tower have been reported [5]. Most of the simplified models are built the basis of merkel's theory [6] assuming a Lewis number equal to unity and neglecting the losses due to water evaporation. The most important coefficient used in models are the mass transfer coefficient between spray water interface and air, and the heat transfer coefficient between tubes and spray water which are built on the basis of enthalpy potential [1]. Existing simplified models allow the prediction of cooling tower performance but using as input heat and mass transfer correlations which were experimentally obtained for large-size cooling towers [7,8]. Experimental studies have been carried on the wet cooling tower but the experimental results on hybrid closed circuit cooling tower is lacking on the relevant literature [9,10,11].

The aim of this paper is to adapt a simplified model for analyzing the combined heat and mass transfer in indirect cooling tower to evaluate the tower performing condition. The analysis of energy and mass balance for the tube element of bare type heat exchanger will define the mathematical equations for tower performance. Performance characteristics will compare with the experimental measurements.

2: Experimental setup

A new indirect cooling tower was modulated in order to be used for conducting experiments. Design conditions were a cooling capacity of 1 kW, for an inlet water temperature of 38 ^{o}C, a water mass velocity 2.3 kg/m^2.s and an air wet bulb temperature of 26 ^{o}C. The tower has a section of 0.15*0.15m and height of 0.8 m. The tube bundle has 72 inline tubes of 8 mm outside diameter and 73 mm inside diameter, with a horizontal pitch of 20 mm and vertical pitch of 25 mm, and with a total transfer area of 0.24 m^2. This corresponds to a much smaller size than usual towers.

The tower was manufactured by Gaunt company (Germany). A forced draft configuration was chosen with a cross flow fan located at air entrance. This arrangement has a lower noise level, and also leads to a lower pressure drop. It was also chosen to facilitate air flow measurements. **Figures (1)** and **(2)** shows schematically the cooling tower and main variables involved. A test facility was assembled at air conditioning lab of Mechanical Engineering Department, University of Baghdad to test this cooling tower. The thermal load was modeled with an electric heater located at a water tank. Tower inlet water temperature was controlled by varying heating power. Fan speed was also controlled by varying power supply, which allowed changing air flow rate. Spray and cooling water flow rates could be changed manually by using regulating valves.

The tower water inlet and outlet temperatures were measured with PT100 probe. Air flow rate was measured with Orifice meter at tower outlet section. In order to measure cooling water temperature evolution, thermocouples were connected to the tubes. The data acquisition system used a data logger and its software.

(1) Constant temperature bath (2) heater (3) cooling water circulation pump (4) cooling water flow meter (5) heat exchanger (6) spray nozzle (7) fan (8) spray water flow meter (9) spray water circulation pump
(10) data acquisition.

Table (1): Experiment condition

Cooling water mass velocity	:	1.1 to 2.3 kg/m^2.s
Inlet temperature	:	34 to 46 ^{o}C.
Spray water mass velocity	:	0.66 to 2.1 kg/m^2.s
Air mass velocity	:	0.47 to 0.84 kg/m^2.s
Wet bulb temperature	:	21 to 26 ^{o}C.
Dry bulb temperature	:	33 to 45 ^{o}C.

3: Theoretical and computational modeling

In this section of the paper, the basic equations of heat and mass transfer which occurs in this type of cooling tower. The main assumptions are:

- The heat exchange between the cooling tower and the surroundings is negligible.
- The specific heats of the fluids are assumed to be constant.
- The heat and mass transfer take place only in the direction normal to the flow.
- The air and water flows are uniformly distributed in the test column.
- The water film covers the entire wall separating the air from the water.
- The interface temperature between water film and air is assumed to be equal to the water-film temperature.

3.1: Mass balance

The rate at which spray water is transferred between two phases using the overall mass transfer coefficient (ka) and referring to the humidity ratio of saturated air and water-vapor at the bulk water temperature ($H^{/}$) is:

$$GdH = dL = Ka\,(H^{/} - H)\; dz \tag{1}$$

or

$$\frac{Ka.dz}{G} = \frac{dH}{H^{/} - H} \tag{2}$$

Multiplying both sides by $\frac{G}{L}$ gives:

$$\frac{Ka.dz}{L} = \frac{G}{L}\frac{dH}{H^{/} - H} \tag{3}$$

Let del = $\frac{Ka.dz}{L}$ (4)

Applying the backward finite deference method approximation, substituting equation (4) and rearranging equation (3) gives:

$$H(n+1) = H(n) + \frac{del \quad L(n)}{G}\left[H'(n) - H(n)\right] \tag{5}$$

3.2: Energy balance

3.2.1: Enthalpy balance of air-water film

The rate of heat transfer from the inter face to the air stream using the over all mass transfer coefficient (ka) and referring to the enthalpy of saturated air and water-vapor at the bulk water temperature (h'), neglected air and variation in humid heat is:

$$Gdh_a = Ka(h' - h_a)dz \tag{6}$$

Which is known as Merkel equation [6].

Or $\frac{Ka.dz}{G} = \frac{dh_a}{h' - h_a}$ (7)

The integration of equation (7) gives the NTU based on the air flow rate:

$$(NTU)_G = \frac{Ka.z}{G}\int_{ha_1}^{ha_2}\frac{dh_a}{h' - h_a} \tag{8}$$

If the evaporation rate is ignored, then substituting for Gdh_a from equation (8) gives:

$$\frac{Ka.dz}{L} = \frac{C_L.dt_f}{h' - h_a} \tag{9}$$

The integration of equation (9) gives the NTU based the spray water flow rate:

$$(NTU)_L = \frac{Ka.z}{L}\int_{t_{f1}}^{t_{f2}}\frac{C_L dt_f}{h' - h_a} \tag{10}$$

Multiplying equation (7) by $\frac{G}{L}$ we have:

$$\frac{Ka.dz}{L} = \frac{G}{L}\frac{dh_a}{h' - h_a} \tag{11}$$

Applying the backward finite difference method approximation, substituting equation (4) and, rearranging equation (11) gives:

$$h_a(n+1) = h_a(n) + \frac{delL(n)}{G}\left[h'(n) - h_a(n)\right] \tag{12}$$

From definition of the enthalpy of air-water vapor mix true, the bulk air temperature, to and reference temperature = 273K can be written as:

$$t_a = \frac{h_a - h_{fg}H}{Cp_a + Cp_v H} \tag{13}$$

Using finite deference substituting we have:

$$t_a(n+1) = \frac{h_a(n+1) - h_{fg}H(n+1)}{Cp_a + Cp_v H(n+1)} \tag{14}$$

From the equation of conservation of mass equation (1) using the finite deference method we have:

$$L(n+1) - L(n) = G\left[H(n+1) - (H(n)\right] \tag{15}$$

Rearranging equation (15) we have:

$$L(n+1) = L(n) + G\left[H(n+1) - (H(n)\right] \tag{16}$$

Introducing the spray water enthalpy h_f in the enthalpy balance gives:

$$d\lfloor Lh_f \rfloor = d\left[Gh_a\right] \quad (17)$$

Or in the finite difference approximation:

$$L(n+1)h_f(n+1) - L(n)h_f(n) = G\left[h_a(n+1) - h_a(n)\right] \quad (18)$$

Rearranging equation (18) we have:

$$h_f(n+1) = L(n)h_f(n) + G\left[h_a(n+1) - h_a(n)\right] / L(n+1) \quad (19)$$

The spray water enthalpy is defined as:

$$h_f = C_L[t_f - 273] \quad (20)$$

Therefore

$$t_f = \frac{h_f}{C_L} + 273 \quad (21)$$

Or in the finite difference approximation:

$$t_f(n+1) = \frac{h_f(n+1)}{C_L} + 273 \quad (22)$$

3.2.2: Overall enthalpy balance

The rate of heat transfer from cooling water through the outside surface of heat exchanger to the interface of spray water-vapor film to the stream can be written as:

$$d[Wh_w] + d[Gh_a] = d[Lh_f] - dLdh_f \quad (23)$$

or

$$Wdh_w = d[Lh_f] - Gdh_a - dLdh_f \quad (24)$$

Or in the finite difference approximation:

$$Wh_w(n+1) - Wh_w(n) = [L(n+1)h_f(n+1) - L(n)h_f(n)] - [L(n+1) - L(n)] [h_f(n+1) - h_f(n)] - G\left[h_a(n+1) - h_a(n)\right] \quad (25)$$

Rearranging equation (25) we have:

$$h_w(n+1) = h_w(n) + \{[L(n+1)h_f(n+1) - L(n) h_f(n)] - [L(n+1) - L(n)][h_f(n+1) - h_f(n)] - G\left[h_a(n+1) - h_a(n)\right]\} / W \quad (26)$$

The cooling water enthalpy is defined as:

$$h_w = C_L[t_w - 273] \quad (27)$$

$$t_w = \frac{h_c}{C_L} + 273 \quad (28)$$

Or in the finite difference approximation

$$t_w(n+1) = \frac{h_c(n+1)}{C_L} + 273 \quad (29)$$

Interface humidity ratio and enthalpy, the interface humidity ratio H' and enthalpy, h' should be know at the bottom boundary of the incremental volume (n). The bulk water temperature at this boundary is considered equal the interface temperature, a computer sub routine is prepared to evaluate H' and h'. This is done by, first using the Keenan-Keyes formula [1] to find the saturation water-vapor pressure (P_s) (in atmosphere).

$$\log_{10}\left[\frac{P_s}{218.167}\right] = -\frac{\beta}{T}\left[\frac{a + b\beta + C\beta^3}{1 + d\beta}\right] \quad (30)$$

Where: a, b, c and d are constants:
a = 3.2437814
b = $5.86826*10^{-3}$
c = $1.1702379*10^{-8}$
d = $2.1878462*10^{-3}$
T = absolute temperature, K
β = 647.27–T

From the perfect gas low, the saturation humidity ration (H_s) is defined by [1]:

$$H_s = 0.62198\frac{P_s}{1 - P_s} \quad (31)$$

But the interface enthalpy is defined as:

$$h^{/} = [Cp_a + Cp_v H_s]t_f + h_{fg} H_s$$

Substituting for H_s from equation (31), then:

$$h^{/} = Cp_a t_f + 0.62198\frac{P_s}{1 - P_s}[Cp_v t_f + h_{fg}] \quad (32)$$

Thus h' could be found and $H^{/}$ is H_s.

A basic computer program was written, all the measured parameters taken from experimental runs, such as: air inlet dry and wet bulb temperatures, air outlet dry and wet bulb temperatures, spray water inlet and outlet temperatures, cooling water inlet and outlet temperatures, air velocity, spray and cooling water flow rates, and heat exchanger dimensions. These data were fed the program as input data. All computed parameters taken from the output from the computer which are: The water to air ratio, performance coefficient, outlet air dry-bulb temperature and air humidity ratio, inlet and outlet cooling water temperature, air and spray water flux, mass and heat transfer coefficients and the rejected heat. The flow chart of this program is presented in **Figures (13)**.

3.3: Results and Discussions

Under the standard experimental conditions given in **table (1)**, the experiment and computer program of computational model was repeated over and over changing the air and spray water mass velocities, air dry and wet bulb temperatures and cooling water inlet temperature.

The variation of saturation and cooling capacity efficiency with respect to air wet bulb temperature for different air mass velocities is shown in **figs.(1) and (2)**. The latent heat transfer between the air and the tube surface of heat exchanger is determinates by the density difference between them. Thus, when the wet bulb temperature of air increases the temperature difference between the cooling water and air at inlet decreases, therefore the rate of evaporation of spray water flowing at the outer surface of tubes decreases so that the temperature decrease of cooling water flowing inside the tubes deceases causing the cooling capacity to be significantly reduced. However, the saturation efficiency of the tower is increased slightly with the increasing of air wet bulb temperature, thus increase become linear at temperature higher than 23. Saturation efficiency increases with wet bulb temperature about 8 % for temperature between 22 and 26 ^{o}C.

The range of cooling water and the coefficient of performance behaves similarly versus air wet bulb temperature as shown in **figs.(3) and (4)**. The rate of increase of coefficient of performance is looks identical to the cooling capacity because the wet bulb temperature has no effects on the power consumption of both fan and spray water pump.

Fig.(5) shows almost linearity and a little increase of efficiency with the air dry bulb temperature due to the decrease in overall heat transfer coefficient sequencing by the less amount decrease in the approach of tower.

The air mass velocity affects the cooling tower characteristics represented by saturation efficiency, cooling capacity and coefficient of performance for different spray water mass velocities as shown in **figs (6), (7)** and **(8)**. Higher saturation efficiency and higher coefficient of performance can be obtained with higher air mass velocity. This mostly because when air mass velocity increases it becomes sufficient to accomplish the same transfer of heat and mass especially by evaporation, thus causing an increase of the range of cooling water leading to increase these characteristics.

Also these figures show the effect of spray water mass velocity on tower characteristics. It is clear that the cooling capacity is influenced slightly by the spray water mass velocity, that is mainly because when the spray water mass velocity increases, the rate of evaporation is augmented causing more heat transferred from the cooling water because the heat and mass transfer coefficients are affected by the spray water mass velocity. If **fig.(6)** compared with **fig.(7)** it can concludes that the effect of spray water mass velocity is relatively small compared to the effect of air mass velocity, this conforms well to Yeon Yoo et al [12]. Also it is clear that the coefficient of performance is increased slightly with increasing of spray water mass velocity. When the spray water mass velocity increases, both cooling capacity and power consumption of spray water and supply air fan increase which overcomes by the cooling capacity, thus conforms well to Riffat et al[13].

The saturation efficiency and cooling capacity while varying the cooling water inlet temperature for different air mass velocities are shown in **figs.(9), (10)** and **(11)**. When the temperature and density differences between the tube surface and the air increase according to the increase in water inlet temperature, the heat and mass transfer coefficients increase, thus the efficiency and cooling capacity is increased. However, as it could be noticed in this figure, the increase in efficiency is not significant compared to the increase in cooling capacity.

Finally **Fig.(12)** show a very little influence of spray water mass velocity on the cooling capacity.

3.4: Concluding Remarks

The performance characteristics of wet closed circuit cooling tower having a rated capacity of 1kw were investigated experimentally and theoretically in the present study. The following conclusion can be abstracted from results:

1- All the characteristics of cooling tower increase significantly with the increasing of air mass velocity and slightly increase with the increasing of spray water mass velocity.

2- Air wet bulb temperature has a significant influence on the cooling capacity and has no significant influence on the saturation efficiency.

3- The cooling water inlet temperature has a very little influence on the performance of cooling tower. The same the behavior for the effect of air dry bulb temperature.

4- There is a good agreement in cooling tower performance characteristics between the experimental and theoretical results.

5- Thermal computational model investigated during theoretical analysis is useful for designing and predicting cooling tower performance.

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Nomenclatures

Symbol	Definitions
a	Surface area per unit volume, m^2/m^3
A	Cross section area, m^2
C_L	Specific heat of water, kJ/kg.K
Cp_a	Specific heat of air, kJ/kg.K
C_V	Specific heat of moist air, kJ/kg.K
C_S	Specific heat of water vapor, kJ/kg.K
H	Humidity ratio, kg_a/kg_v
$H^/$	Humidity ratio at sat. mixture, kg_a/kg_v
h_a	Enthalpy of moist air, kJ/kg
$h^/$	Enthalpy of moist air at sat. mixture, kJ/kg
h_{fg}	Latent heat of evaporation, kJ/kg.K
G	Air flux, $kg/s.m^2$
K	Mass transfer coefficient, $kg/s.m^2$
$K.a$	Volumetric mass transfer coeff., $kg/s.m^2$
$K.az/G$	Performance coefficient based on G
$K.az/L$	Performance coefficient (NTU) based on L
L	Air flux, $kg/s.m^2$
m	Mass flow rate, kg/s
Q	Heat load, kW
t	Temperature, K
V	Coil volume / cross section, m^3/kg
Z	Coil height, m
W	Cooling water flux, $kg/s.m^2$

Subscript

f (L)	Spray water
a (G)	Air
w	Cooling water
v (m)	Humid air
1	Down tower section
2	Up tower section
i	Interface

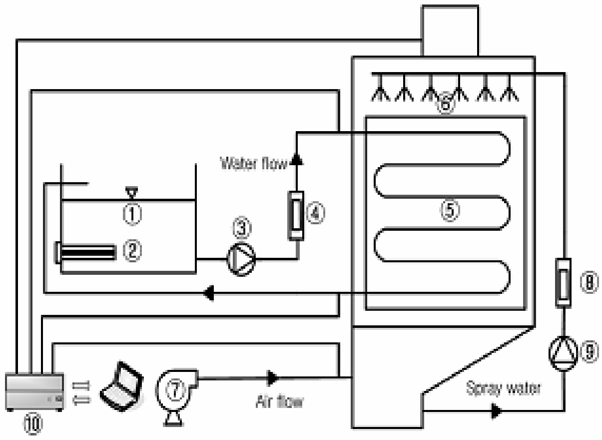


Fig.1. Schematic of closed wet cooling tower

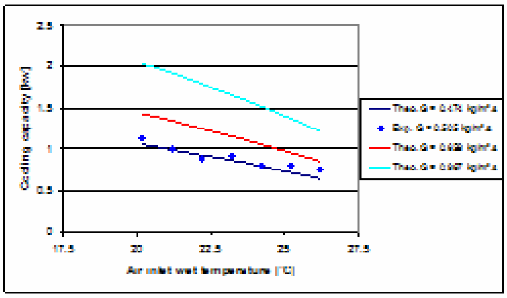


Fig 1: Cooling capacity w.r.t. air wet bulb temperature for different air mass velocity.

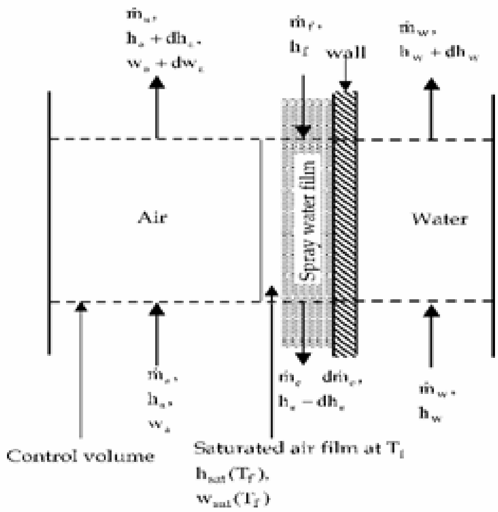


Fig. 2. Flow steams inside an element

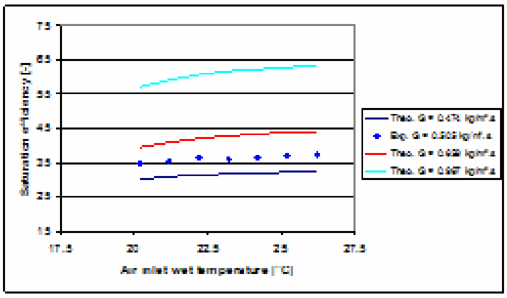


Fig 2: Saturation efficiency w.r.t. air wet bulb temperature for different air mass velocity.

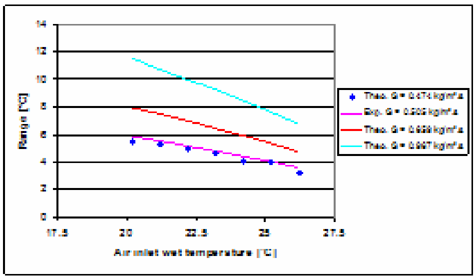


Fig 3: Range of cooling tower w.r.t. air wet bulb temperature for different air mass velocity.

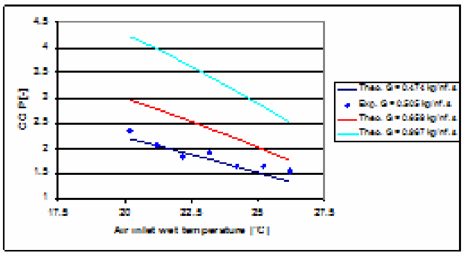


Fig 4: Coefficient of performance w.r.t. air wet bulb temperature for different air mass velocity.

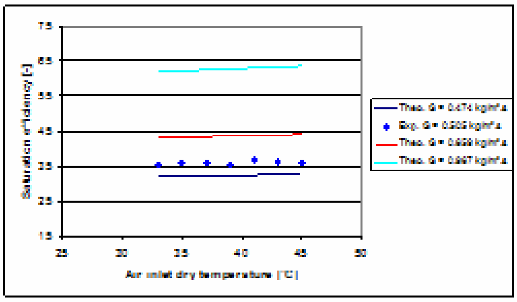


Fig 5: Saturation efficiency w.r.t. air dry bulb temperature for different air mass velocity.

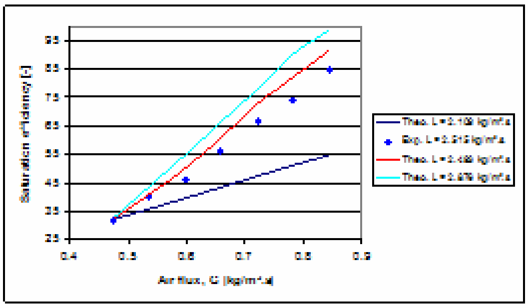


Fig 6: Saturation efficiency w.r.t. air mass velocity for different air mass velocity.

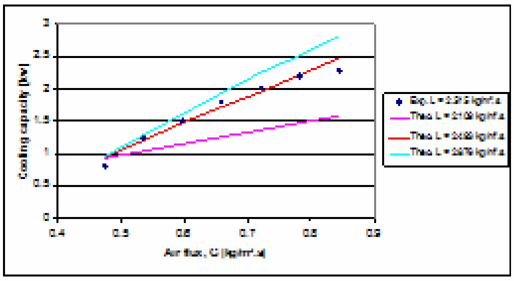


Fig 7: Cooling capacity w.r.t. air mass velocity for different air mass velocity.

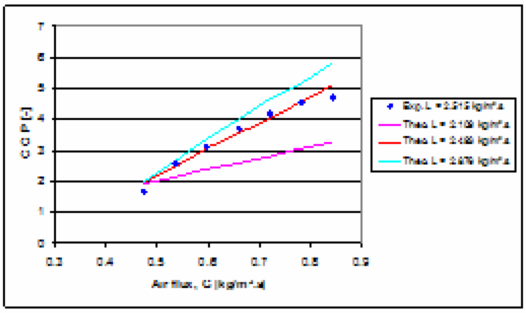


Fig 8: Coefficient of performance w.r.t. air mass velocity for different air mass velocity.

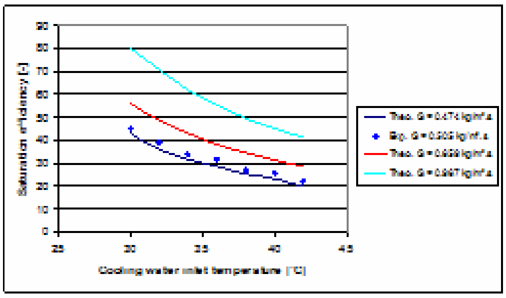


Fig 9: Saturation efficiency w.r.t. cooling water inlet temperature for different air mass velocity.

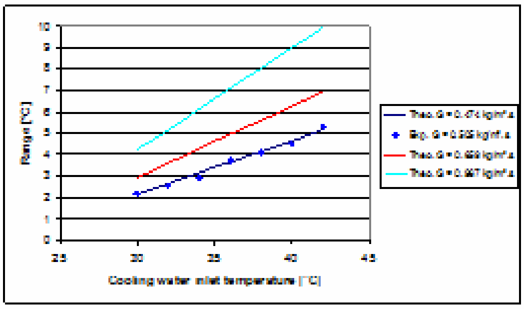


Fig 11: Range of cooling tower w.r.t. cooling water inlet temperature for different air mass velocity.

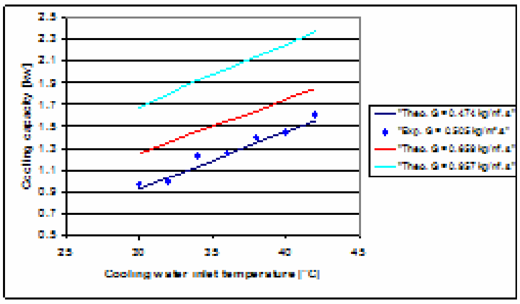


Fig 10: Cooling capacity w.r.t. cooling water inlet temperature for different air mass velocity.

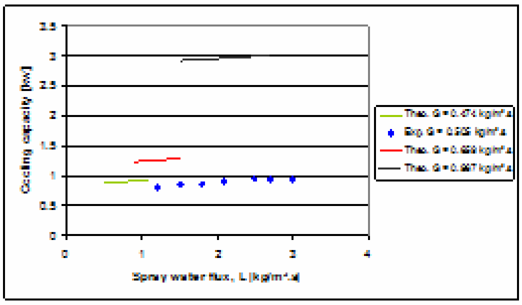
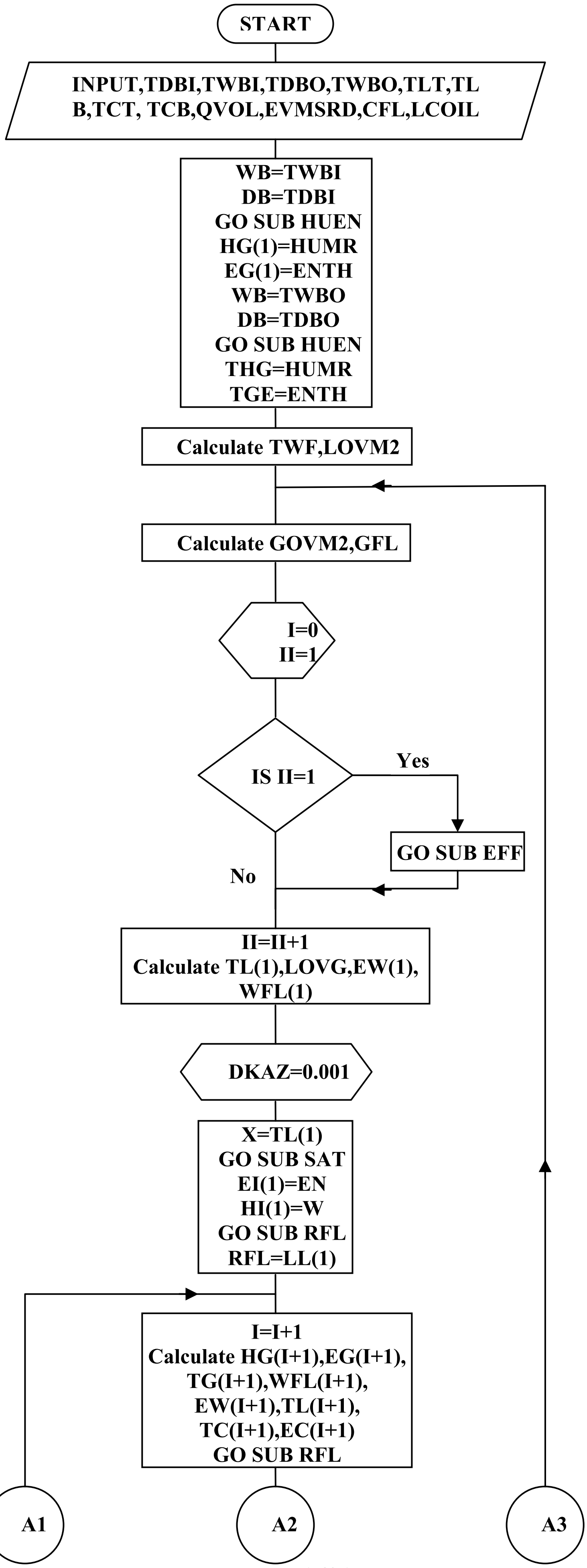
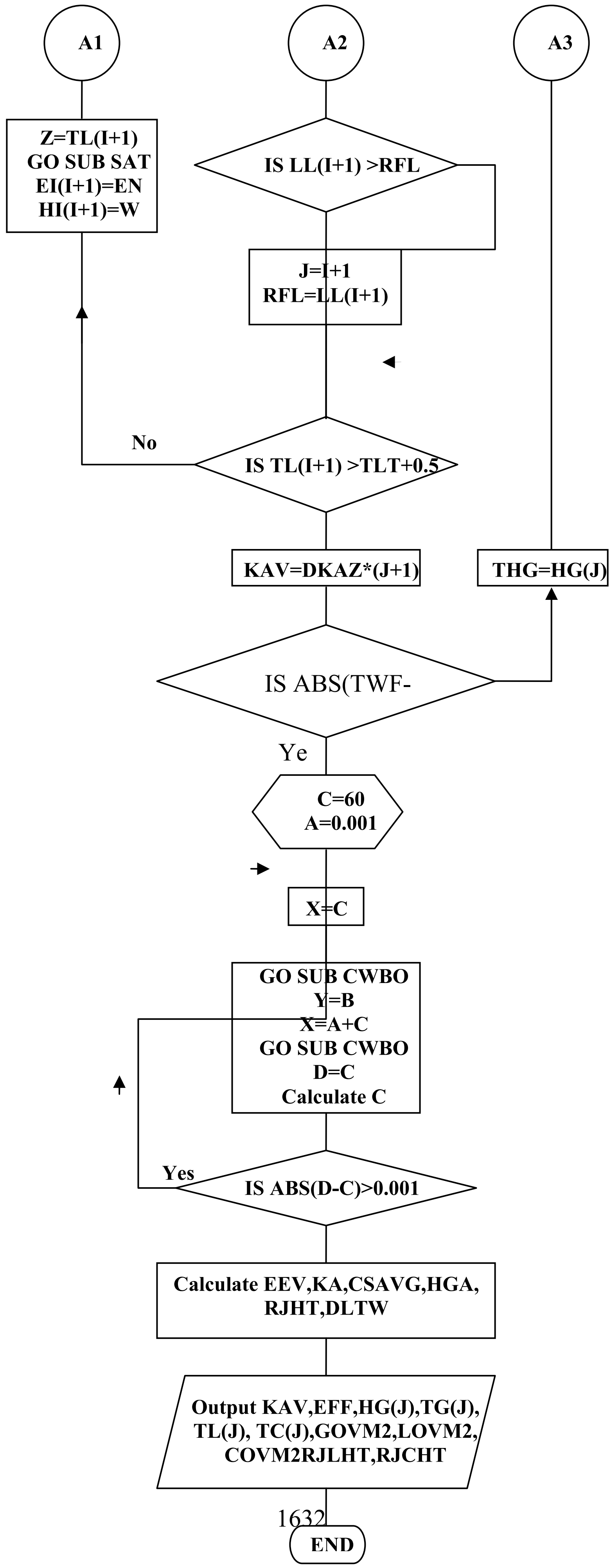


Fig 12: Cooling capacity w.r.t. spray water mass velocity for different air mass velocity.


START
INPUT,TDBI,TWBI,TDBO,TWBO,TLT,TL B,TCT, TCB,QVOL,EVMSRD,CFL,LCOIL
WB=TWBI
DB=TDBI
GO SUB HUEN
HG(1)=HUMR
EG(1)=ENTH
WB=TWBO
DB=TDBO
GO SUB HUEN
THG=HUMR
TGE=ENTH
Calculate TWF,LOVM2
Calculate GOVM2,GFL
I=0
II=1
IS II=1
Yes
GO SUB EFF
No
II=II+1
Calculate TL(1),LOVG,EW(1), WFL(1)
DKAZ=0.001
X=TL(1)
GO SUB SAT
EI(1)=EN
HI(1)=W
GO SUB RFL
RFL=LL(1)
I=I+1
Calculate HG(I+1),EG(I+1), TG(I+1),WFL(I+1), EW(I+1),TL(I+1), TC(I+1),EC(I+1)
GO SUB RFL
A1
A2
A3


A1
A2
A3
Z=TL(I+1)
GO SUB SAT
EI(I+1)=EN
HI(I+1)=W
IS LL(I+1) >RFL
J=I+1
RFL=LL(I+1)
No
IS TL(I+1) >TLT+0.5
KAV=DKAZ*(J+1)
THG=HG(J)
IS ABS(TWF-
Ye
C=60
A=0.001
X=C
GO SUB CWBO
Y=B
X=A+C
GO SUB CWBO
D=C
Calculate C
Yes
IS ABS(D-C)>0.001
Calculate EEV,KA,CSAVG,HGA,
RJHT,DLTW
Output KAV,EFF,HG(J),TG(J),
TL(J), TC(J),GOVM2,LOVM2,
COVM2RJLHT,RJCHT
END