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DETERMINATION OF OUTDOOR CONDITIONS FOR AIR-CONDITIONING SYSTEM DESIGN IN SUDAN

(KHARTOUM CASE STUDY)

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ABSTRACT

The main objective of this research is to set up out-door climatic conditions upon which the air-conditioning cooling load calculation, and the air-conditioning equipment selection are based to obtain an efficient and economical design .

These values have been established by statistical analysis of ten consecutive years of dry and wet-bulb temperatures hourly observations collected by Meteorological Department .

Three frequency levels of percentiles .4 , 1 and 2% for DBT with mean coincident WBT ,and WBT with mean coincident DBT has been established .The decision of selecting one of these frequency levels is referred to the design engineer.

The DBT which represents the ambient temperature upon which the air-cooled condenser capacity is based has been established .

The WBT that represents the ambient WBT upon which the cooling tower and evaporative condenser capacity is based is also established.

ملخص البحث

الهدف من هذا البحث هو استخراج قيم لظروف الطقس الخارجية تبني عليها حسابات احمال تكييف الهواء واختيار سعة مكثفات معدات التكييف سواء تم تبريدها بالهواء او الماء ، وذلك لتصميم نظام تكييف اقتصادى و ذى كفاءة عالية .

لاستخراج هذه القيم تم تحليل قراءات درجات الحرارة الجافة والرطبة المسجلة على راس كل ساعة (٢٤ ساعة فى اليوم) و لمدة ١٠ سنوات متتالية تحليلا احصائيا .

تم استخراج ٣ قيم لدرجات الحرارة الجافة مع قيمة متوسطة لدرجات الحرارة الرطبة التى تسجل معها وذلك لثلاث مستويات تكرارية لتلك القيم هى ٠,٤ ، ١,٠ و ٢,٠%.

تم استخراج ٣ قيم لدرجات الحرارة الرطبة مع قيمة متوسطة لدرجات الحرارة الجافة التى تسجل معها وذلك لثلاث مستويات تكرارية بنفس الطريقة . اختيار احدى هذه المستويات قرار يتخذه المهندس المصمم .

تم استخراج درجة الحرارة الجافة المناسبة لاختيار سعة معدات التكييف ذات المكثف المبرد بالهواء ليعطى الحمل المطلوب ، ودرجة الحرارة الرطبة لاختيار السعة للمعدات ذات المكثف التبخيرى او المكثف المبرد بالماء والذى يستخدم ابراج التبريد

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NOTATION

A = area m^2

Az = apparent solar irradiation for zero air mass W /m2

B = atmospheric extinction coefficient
Bt = transfer function coefficient W / m^2 ^oC
C = specific heat J/kg K
CFM = volume flow rate ft^3/min
CLF = cooling load factor
CLTD = cooling load temperature difference ^oC
D = sun's declination degrees
DBT =dry-bulb temperature ^oC
DR = daily range of temperature ^oC
G = Irradiation w/m2
Gt = transfer function coefficient ,W/ ^oC
H = heat transfer coefficient W/ m^2 ^oC
Hr = hour angle , degrees
Idn = direct normal solar radiation W/ m^2
Id = direct solar radiation on a surface W/ m^2
Ids = diffuse solar radiation W/ m^2
It = Total solar radiation incident on surface W/ m^2
I.S.R = Intensity of solar radiation W/ m^2
L = latitude , degrees
M.S.L.= mean sea level m
Ni = Heat transfer factor , inward flow fraction
P.V.P = partial vapor pressure Pa
Q = heat transfer W
Q_{eq} = heat gain by the space through indoor surfaces of a roof or wall W
Q_θ = cooling load W
Ro = outside surface resistance. C /W
Ri = inside surface resistance. C/W
Δ R = difference between long-wave radiation incident on surface from sky
and surroundings and radiation emitted by blackbody at outdoor air temperature W/m^2.
RTF = Room Transfer Function coefficients
Sc = shading coefficient
SHF = sensible heat factor
SHGF = solar heat gain factor W/ m^2
T = temperature ^oC
U = overall heat transfer coefficient W/ m^2 ^oC
WBT = Wet-bulb temperature ^oC
Y = Ratio of the sky diffuse on vertical surface to sky diffuse on horizontal surface.
b_n , = Conduction transfer function coefficients
C_n = Conduction transfer function coefficients
d_n = Conduction transfer function coefficients
h_o = coefficient of heat transfer by long-wave radiation and convection at outer surfaces W/ m^2
n = summation index
q_θ = heat gain W
t_o = outdoor temp. ^oC
$t_{eq\text{-}nd}$ = sol-air temperature at time $t_{\text{-}nd}$ ^{o} C
t_{rc} = constant room air temp. ^oC

t_s = surface temperature ^oC
v_0 = Room Transfer Function coefficients
v_1 = Room Transfer Function coefficients
w_1 = Room Transfer Function coefficients
w_2 = Room Transfer Function coefficients

GREEK LETTERS

α_t Angle of tilt
α absorbance of surface for solar radiation
β Altitude angle , degrees
γ Wall solar azimuth , degrees
Δ Change in quality or property
θ Angle of incidence , degrees
τ Time , second
ρ Density , kg /m^3
Σ Angle of tilt from the horizontal , degrees
ψ Solar azimuth angle , degrees
Φ Wall azimuth angle , degrees
ε Emittance
ε_h hemispherical emittance of surface

CHAPTER TWO

STATISTICAL ANALYSIS

2-1 INTRODUCTION AND PERPOSE OF STATISTICAL ANALYSIS :

The functions of statistical analysis are :

a- To construct frequency tables and calculate statistics such as the mean and standard deviation . Generally the annual cumulative frequency distribution was constructed from the relative frequency distribution compiled for each month

b- To reduce the data to values well representing the data used in the analysis and nearst future without repeating the analysis .

Analysis procedure including the measures used to ensure that the number and distribution of missing data , both by month and by hour of the day , did not introduce significant biases into the analysis is recommended by ASHRAE . Each individual month's data was included if they met screening criteria for completeness and unbiased distribution of missing data .

A station design conditions were included only if there where data from at least 8 months that met the screening criteria from the period of record for each month of the year . (2)

For instance , there had to be 8 months each for January , February , …etc. whose data met the completeness screening criteria (gaps up to 5 hrs were filled) .A month's data were included if the month was at least 85% complete after filling .(2)

2-2 DEFINITIONS:

DRY-BULB TEMPERATURE :

Is the temperature of air as registered by an ordinary thermometer .

WET –BULB TEMPERATURE :

The temperature registered by a thermometer in which it's bulb is covered by a wetted wick and exposed to a current of rapidly moving air .

DAILY TEMPERATURE RANGE :

The daily range is the difference between the maximum and the minimum temperatures recorded during the day .

2-3 DATA TYPE :

These are ten consecutive years hourly readings of the dry-bulb , and wet-bulb temperatures from the first of January 1990 to the therty first of Decemper 1999.

2-4 DATA SOURCE

The data used and manipulated is ten consecutive (hourly readings) years delivered by the Sudan Meterological Department (The readings are taken at the Airport of Khartoum)

2-5 STATISTICAL ANALYSIS DATA

The data included for statistical analysis is

1- 10 consecutive years , 120 months , 3652 days , 24 hours records for each day for dry-bulb temperature. 2 months are completely missing , these are March 1991 and Dec 1997 .

2- 10 consecutive years , 120 months , 3652 days , 24 hours records for each day

for wet-bulb temperature.Only one complete month is missing
3- 4 consecutive years , 48 months , 1461 days , 8 readings per day for wind velocity .

2-5-2 ODD VALUES AND TREATMENT

Odd values amoung the data are as follows:

a- Records like 20 , 32 , 23 ^{o} C, this is modified by common sense to 20 , 22 , 23 ^{o} C

b- Records like 23 , 28 , 24.5 ^{o} C (in particular wbt) , this modified to 23 , 23.5 , 24.5 ^{o} C as there is no any change in the behavior of the dbt at the same hours.

c- Sharp change in records from the preceding one (jump of values) such as 14 ,26 ^{o} C , as these are observed for dbt and wbt at the same time , this is can be justified by a sudden change in the climate, and therefore these values are taken as they are .

2-5-3 MISSING DATA AND TREATMENT

Table 1 illustrates the number and percentage of the missing data

Table 1

Date	DBT	%	WBT	%
Jan 95	2	.27	0	0
Feb 94	4	.5	4	.5
Feb 98	2	.25	2	.25
Mar 91	All	100	0	0
Mar 94	40	5.4	40	5.4
Apr 91	2	.3	2	.3
Apr 94	9	1.3	64	8.9
May 94	6	.8	6	.8
May 95	2	.2	2	.2
Jun 95	2	.27	0	0
Jul 90	1	.15	2	3
Jul 94	11	1.5	11	1.5
Aug 90	2	.3	0	0
Aug 94	20	2.7	20	2.7
Aug 99	1	.13	1	.13
Sep 90	0	0	1	.13
Sep 94	2	.3	2	.3
Sep 96	2	.3	2	.3
Sep 99	20	2.7	20	2.7
Dec 97	All	100	All	100

The maximum percentage of missing data for a single month is 8.9 % of that month , and this is only for one month throughout the 10 years of records .The percentage for other month is more less than that as seen from the table .

The total missing record for DBT is 127 , represent 0.2% of the total data , and that of WBT is 178 which is also equal to 0. 2% .

The total missing data included the complete missing months is amounted to 1.8 % for DBT , and 1% for WBT .

Missing data has been treated according to the following

i- One or two records is completed by interpolation .

ii- Three and more records are completed using the temperature tendency of the previous day .

iii- A complete month records (March 1991 dbt, and December 1997 dbt and wbt) , these months are replaced by the mean values of the other nine months.

2-5-4 SIGNIFICANCE OF MISSING DATA:

1- For the data completed by interpolation , the values resulted are expected to be close to the missing values.

2- for the complete months :

i- Dec .has no contribtion in records of high Dry and WBT . For the DBTs , Dec. is one of the winter months during which the DBTs are low , and regarding the WBT , the records high values are in order of 21^{o} C and did not go over 23 ^{o} C during all nine years .

ii- For march , the number of records exceeding or equal to 42 ^{o} C during the nine years is only 10 , one record per year , and therefore these records didn’t influence the final result of the design values.

2-6 SOFTWARE USED;

All records were entered in the computer using MICROSOFT EXCEL SOFTWARE . Statistical analysis , mathematical formulas and results were done using EXCEL software as well.

2-7 DEFINITIONS OF THE STATISTICS USED:

1- Average : is a value that is typical or representative of a set of data

2- The Mean : or the arithmatic mean of a set of N nombers is denoted by X and defined as

$$X = (x_1+x_2+x_3+\ldots\ldots+x_n) / N = \sum X / N$$

The algebraic sum of the deviations of a set of numbers from their arithmatic mean is zero .

3- Percentiles : the values which divide the data into 100 equal parts

4-The mode : the most frequent value .

5- Standard Deviation: is a measure of dispersion of the data , it is the root mean square of the deviations from the mean . defined as

$$s = \sqrt{}\sum (X- X)^2$$

2-8 ANALYSIS:

2-8-1 THE FREQUENCY OCCURRENCE :

a- DBT:

Maximum temperature genarally occurs between hours 2 to 4 pm. , 3 pm is the most hour at which the maximum temperature usually recorded.

Minimum temperature occurs between 4 to 6 am , it is normally recorded at 5am . Some times the maximum and minimum are recorded at two or even three hours of the period with no temperature change during the period.

In months of Dec. through feb. (the winter months) , the high frequency occurrence seen in the range 19 to 29 ^{o}C , beyond these figures , the occurrence diminishes gradually up to 35 ^{o}C after which the occurrence is very rare throughout the ten years of the analysis .

In March , the significant occurrence lies between 20 to 37 , 25-27 ^{o}C represent the top of the graph . The figures beyond 37 are less significant .

During April , the significant frequency occurrence range begins at 25 ^{o}C and ends at 43 ^{o}C , while in May the range starts with 27 ^{o}C and ends at 43 ^{o}C .Values under 27 ^{o}C are rarely recorded , and that over 43 ^{o}C are recorded for less numbers of hours . 39 ^{o}C represents the most frequent temperature during May .

June is most similar to May , 28 ^{o}C is the lowest degree in the significant region and the most frequent degrees are 31 and 32 ^{o}C .These values indicate very hot climate during these three months .

During July through October , the lowest value of significant occurrence is 26 ^{o}C which is slightely smaller than that of the preceding months , and the highest is about 40 ^{o}C which is less . 30 ^{o}C represents the mode for all these months .

In November , the range begins with 22 ^{o}C through 37 , the most frequent values are 26 – 30 ^{o}C , 28 ^{o}C is the mode . Figs. (1-12)

Fig. 13a represents the whole year distribution . The significant occurrence ranges between 23-40 ^{o}C , 30 ^{o}C represents the mode approximately at the middle of the range.

b- WBT .

From Dec. up to March , the WBTs are distributed in a wide range from 9 ^{o}C to 22 ^{o}C, with significant occurrence range from 9 to 19 ^{o}C . 14 ^{o}C and 15 ^{o}C are the modes for these months .

During April and Nov. , the distribution is generally resemples that of March . The significant values range from 12 to 22 ^{o}C , 18 ^{o}C represent the mode .

In May and June , the significant occurrence begins at 15 ^{o}C and ends at 24 ^{o}C , 22 is the mode of these two months . July and Oct. are similar in their frequency occurrence distribution , the significant range begins at 16 and increases gradually up to 22 and 23 ^{o}C , and then decreases . 24 ^{o}C is significant in these two months.

The distribution during Aug. ans Sep. tends to be quite unique and confined in a narrow range of 20 ^{o}C to 25 ^{o}C , 23 ^{o}C is the most frequent value and 22,24 and 25 ^{o}C have also quite significant frequency occurrence . Figs. (14-25)

Fig. 13b represents the whole year distribution . The significant occurrence ranges between 10-25 ^{o}C , 23 ^{o}C represents the mode at the right side of the range .

2-8-2 THE HOURLY MEAN TEMPERATURE FOR SUMMER MONTHS

a- DBT

The DBT generally reaches the minimum value at 5 am early morning and gradualy increases to reach the highest at 3 pm and then gradualy decreases to the minimum value , the increments are greater during April . The maximum temperatures at hours 2-4 pm exceed 40 ^{o}C in May and June ,while during the other months these temperatures are generally exceed 35 ^{o}C .The range of the hours exceeding 35 ^{o}C in April is greater than in Sep. and Oct. July and Aug. having the lowest range . Figs. (33-39)

b- WBT

The WBT values through the 24 hrs , have the same behavior as the DBT . The difference is that for the WBT the range between the minimum and the maximum is less than that for DBT , the maximum values occur at 2 and 3 pm for May , June , Sep. and Oct. ,and at 1and 2 pm for July and Aug. For April this value is noted at 11 and 12 mid-day . Figs. (40-47)

For all the months except April , the minimum value occures at 4 and 5 am . In April this minimum value occures at mid night .

c- Daily range :

The percentage of the daily range that any record differ from the maximum or the minimum is not equal for the same hours in different days . To evaluate this percentage , days in which the minimum is recorded at 5 am, and the maximum is recorded at 3 pm are put for statistical analysis for the average .

The number of days included is 1571 , out of 3655 days represent the whole number of the data . Table 13 is the result.

Generally , the highest daily range occures during April and gradualy decreases to reach the minimum during July and Aug. and then increases again . Although the highest DBT in April is almost the same as that of in May and June , the minimum is less and hence the daily range is greater .

Comparing the result with the ASHRAE values , the hours 5 am and 3pm are same since these hours represent the minimum and maximum temperature record

The values about the minimum and maximum are slightly different , but they are different in the other values , however , the result of the analysis is appear more regular , and consequently are adopted for the construction of the representative day air cycle temperatures for calculation of the CLTDs .

2-9 CHARACTERISTICS OF COOLING AND DEHUMIDIFICATION DESIGN CONDITIONS

The sets of design values in this chapter represent different psychrometric

conditions

Design data based on DBT represent peak occurrences of the sensible component of ambient outdoor conditions .

The .4%,1%and 2% dbt and mean coincident wbt (Table 6) represent conditions on hot mostly sunny days , and these are useful for cooling applications in air conditioning .

Design values based on WBT (Table 7) are related to the enthalpy of the outdoor air .It represent extremes of the total sensible and latent heat of outdoor air .These information are useful for cooling towers , evaporative condensers , evaporative coolers and fresh air ventilation system design and selection .

The design engineer must decide which sets of conditions and probability of occurrence to apply to the design situation.

2-10 ANNUAL EXTREME TEMPERATURES :

Table 6 provides the mean and standard deviation of the annual extreme maximum and minimum dbt .The probability of occurrence of very extreme conditions can be required for the operation design of equipment to ensure continuous operation and serviceability , regardless of whether the heating or cooling loads are being met .(2)

CHAPTR THREE

SOLAR RADIATION

3-1 INTRODUCTION

Solar energy is energy in the form of electromagnetic waves. The electromagnetic spectrum includes the types of radiation commonly known as radio waves, micro waves …., etc.

Solar energy consists primarily of infra-red rays , the visible spectrum(light) , and ultra-violet rays , in the wave lengths range from about 10^{-4} to 10^{-8} μm .

Solar radiation forms the greatest single factor of cooling load in buildings. It is therefore necessary to study the subject not only for the purpose of load calculations, but also from the point of view of load reduction. (6)

3-2 INTENSITY OF SOLAR RADIATION

The average intensity of solar radiation normal to the sun's rays at the outer edge of the earth's atmosphere is 1373 W/m^2 , and is subject to a seasonal variation of 3.5% in January and –3.5% in July .

In passing through the atmosphere , part of this heat is absorbed , part scattered into space and part scattered back to the earth by atmosphere .

For cloudless skies, the intensities of direct and diffuse radiation depend on the thickness of the layer of atmosphere traversed and thus on the solar altitude and height above sea level. They also depend on the proportion of water vapor, dust and ozone in the atmosphere, constituents that scatter and absorb radiation.(15)

Solar radiation received at the earth's surface on a plane perpendicular to the sun's rays may amount to more than 860 W/m^2 of surface on a clear day at sea level . This maximum figure occurs when the sun is directly over head, and the amount becomes progressively less as the sun's angle of altitude above a horizontal plane decreases. (5)

3-3 DEFINITIONS:

From the outside the earth's atmosphere, the solar radiation reaches any part of the earth's surface in two ways: direct and diffuse.

3-3-1 DIRECT SOLAR RADIATION

The part of the solar radiation travels through the atmosphere and reaches the earth's surface directly.

3-3-2 DIFFUSE (SKY) RADIATION:

The re-radiated part of that part of sun radiation which is scattered , reflected back into space and absorbed by the earth's atmosphere. It is small in a clear sky , large in a hazy or cloudy sky.

3-3-3 SOLAR CONSTANT

The intensity of solar radiation at the upper limit of the earth's atmosphere on a unit area of surface normal to the sun's rays is called the solar constant Io . (8)

However, the earth's atmosphere , clouds, winds and air contaminants block a good deal of the sun's radiant energy from reaching the earth's surface , and therefore the solar constant is not a value that can be used to calculate the solar energy received at the ground level.

3-4 SOLAR DATA :

Solar heat gains during sunny periods are required for calculating maximum cooling loads in air conditioning and peak summer time temperatures in naturally ventilated buildings . (8)

Solar gains to buildings are determined by :

1- Sun's angles , and the corresponding radiation intensities on the different building surfaces
2- Proportion of glass , and provision of blinds or other sun protective devices.
3- Solar absorption and transmission properties of blinds and also of opaque parts of the structure.

3-5 SOLAR RADIATION CALCULATION.

3-5-1 SOLAR ANGLES:

a- Latitude L is the angular location north or south of the Equator, north positive $-90 \le L \le 90$.

b- Declination : δ is the angular position of the sun at solar noon with respect to the plane of the equator ,north positive

$$\delta = 23.45 * \sin(360(284+n)/365)$$

n = day of the year.

c- Hour angle w : is the angular displacement of the sun east or west of local meridian due to rotation of the earth on it's axis at 15 per hour , morning negative , afternoon positive.

d-SOLAR TIME

Time based on the apparent angular motion of the sun across the sky , with solar noon the time the sun crosses the meridian of the observer .

CORRECTION OF SOLAR TIME TO LOCAL TIME :

1- Correction for the difference in longitude between the observers meridian location and the meridian on which the local standard time is based
2- Equation of time : correction to solar time which takes into account the perturbations in the earth's rate of rotation , which affect the time the sun crosses the observer's meridian

Solar time = standard time + 4(Lst – L loc) +E

E = equation of time

$E = 9.87 \sin 2B - 7.53 \cos B - 1.5 \sin B$

$B = 360(n-81)/364$

n = day of the year

e- **Solar altitude** : β is the angle on a vertical plane between the sun's rays and the horizontal plane on the earth's surface .

$$\sin \beta = \cos \delta \cos L \cos w + \sin \delta \sin L$$

f- **Solar azimuth angle** : Φ is the angle on a horizontal plane between the due –south direction line and the horizontal projection of the sun's rays .

$$\cos \Phi = (\sin \beta * \sin L - \sin \delta)/(\cos \beta * \cos L)$$

g- **Surface azimuth angle** : ψ is the deviation of the projection on a horizontal plane of the normal to the surface from the local meridian , with zero due South , east negative -180<= ψ =180

h- **Surface solar azimuth angle** : γ is the angle on a horizontal plane between the normal to a vertical surface and the horizontal projection of the sun's rays

i- **Slope (Tilt angle)** Σ is the angle between the plane surface in question and the horizontal $0 <= \Sigma <= 180$ (Σ >90 means that the surface has a downward facing component)

j- **Angle of incidence** θ : is the angle between the beam radiation on a surface and the normal to that surface

For any surface :

$$\cos \theta = \cos \beta * \cos s * \sin \Sigma + \sin \beta * \cos \Sigma \qquad (1)$$

For horizontal surface

$\Sigma = 0$

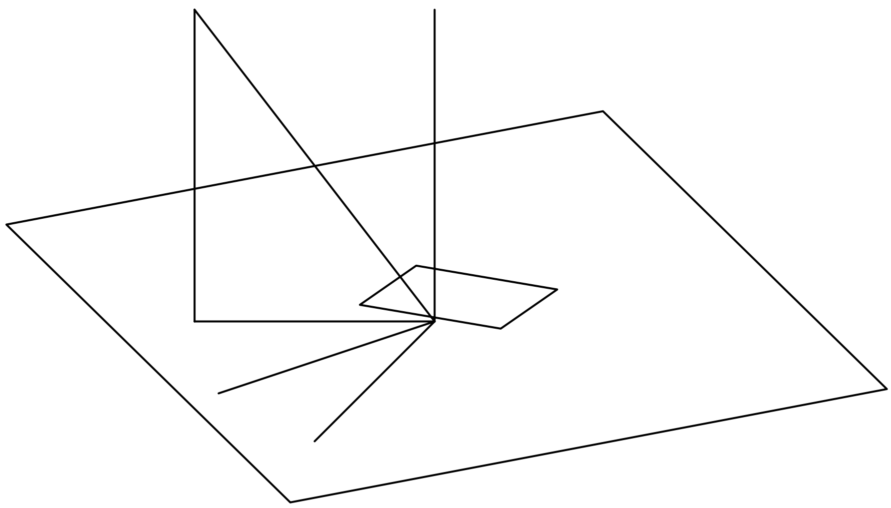
$\cos \theta = \sin \beta$

For vertical surfaces

$\Sigma = 90$

$\cos \theta = \cos \beta * \cos \psi$

Figure 51 Solar angles



3-5-2 EXTRATERRESTRIAL RADIATION ON HORIZONTAL SURFACE

At any point in time, the solar radiation outside the atmosphere incident on a horizontal plane is

$$Go = Gsc\ \{1+.033*\cos(360n/365)\}*\cos\theta z$$

Where: Gsc = Solar constant

n = day of year

3-5-3 DIRECT NORMAL SOLAR RADIATION

At earth's surface on a clear day

I_{dn} is represented by

$$I_{dn} = A_z *\exp(-B/\sin\beta)\ W/m^2$$

Where

A_z= apparent solar irradiation at air mass = 0

B = atmospheric extinction coefficient .

3-5-4 DIRECT INTENSITY Id

$$I_d = I_{dn} *\cos\theta$$

3-5-5 DIFFUSE SOLAR RADIATION

A simplified general relation for the diffuse solar radiation I_{ds} that falls on any surface from a clear sky is given by ;

$$I_{ds} = C\ I_{dn}\ Fss \qquad W/m^2$$

Ratio of the sky diffuse on vertical surface to sky diffuse on horizontal:

If $\cos\theta > -0.2$

$Y = 0.55+0.437\cos\theta+0.313\cos^2\theta$

Otherwise $Y = 0.45$

Diffuse Intensity

For horizontal surface $\quad I_{ds} = C * I_{dn}$

For vertical surface $\quad Ids = \{C*Y+0.5\ \rho g\ (C+\sin\beta)\}\ Idn$

Appendix A contains a sample of calculations .

Appendix B gives the calculated clear sky solar radiation for Khartoum on horizontal and 8 vertical orientations

3-5 HEAT GAIN THROUGH FENESTRATION

Fenestration is the term used to designate any light transmitting opening in a building wall or roof . The opening may be glazed with single or multiple sheet , plate or float glass , pattern glass plastic panels or glass block . Interior or exterior shading devices are usually employed, and more glazing systems incorporates integral sun control devices. (2)

Heat admission or loss through fenestration areas is affected by many factors including:

1- Solar radiation intensity and incident angle

2- Outdoor-indoor temperature difference
3- Velocity and direction of air flow across the exterior and interior fenestration surfaces .
4- Low-temperature radiation exchange between the surfaces of the fenestration and the surrounding .
5- Exterior and interior shading

When solar radiation strikes an un shaded window , part of the radiant energy (8% for uncoated clear glass) is reflected back outdoor , part is absorbed within the glass (from 5 to 50% , depending upon the composition and thickness of the glass) , and the remainder is transmitted directly indoors to become part of the cooling load . The solar heat gain is the sum of the transmitted radiation and the portion of the absorbed radiation that flows inward . (2)

The total instantaneous rate of heat gain through a glazing material can be obtained from the heat balance between a unit area of fenestration and its thermal environment .

Total heat admission		Heat flow due to		Radiation		Inward flow
Through glass	=	outdoor-indoor	+	transmitted	+	of absorbed
		Temperature difference		through glass		solar radiation

In this equation , the last two terms on the right hand side are present only when the fenestration is irradiated and are therefore related to the incident radiation . The first term occurs whether or not the sun is shining , since it represents the heat flow through fenestration by thermal conduction .

Combining the last two terms

Total heat transmission		conduction		solar heat
through glass	=	heat gain	+	gain

In this way , heat gain is divided into two components :

1- The conduction heat gain due to difference in outdoor-indoor air temperature .
2- The solar heat gain SHG due to transmitted and absorbed solar energy

The total load through fenestration is the sum of load due to conductive heat gain and the load due to solar heat gain .

3-6-1 CONDUCTION HEAT GAIN :

Whether or not sun light is present , heat flows through fenestration by thermal conduction as expressed by :

$$Q/A = U\,(t_0 - t_i)$$

Where

Q/A = instantaneous rate of heat transfer through fenestration
U = overall coefficient of heat transfer for the glazing
T_0 = outside air temperature
T_i = inside air temperature

3-6-2 SOLAR HEAT GAIN :

The solar heat gain is much more complex because the sun's apparent motion across the sky causes the irradiation of a surface to change minute by minute . (2)

Heat due to transmitted and absorbed solar energy or solar heat gain SHG is present in cooling load calculations only when fenestration is irradiated .

The instantaneous rate of heat gain through fenestration is given by

$$Q = \{ SHGF + U(to - ti) \} A$$

However , this instantaneous heat rate will not be the cooling requirement at that instant due to thermal storage in the space .

The solar heat gain factor for each exposure is determined by the latitude , date , and time of day . The outdoor air temperature is a matter of local climate .

To account for the different types of fenestration and shading devices used , the shading coefficient SC , which relates the solar heat gain through a glazing system under specific set of conditions to solar heat gain through the reference glazing material under the same conditions , is defined as

$$SC = \frac{\text{solar heat gain of actual fenestration}}{\text{solar heat gain of standard double strength glass}}$$

SC is unique for each type of fenestration or each combination of glazing and shading device.

The reference glass is a double strength glass with a transmittance of 0.87 , a reflectance of 0.08 and an 0.05 absorptance . (2)

The instantaneous rate of heat gain through fenestration with shading then become :

$$Q = \{ SHGF (SC) + U(to - ti) \} A$$

3-6-3 SOLAR HEAT GAIN FACTOR CALCULATION

1- **Transmitted component :**

$$= I_d \; \Sigma \; t_j \cos^j \theta + I_{ds} \, 2 \, \Sigma \, (j=0 \text{ to } 5) \; t_j /(j+2)$$

I_d = direct component

I_{ds} = diffuse component

2- Absorbed component :

$$= I_d \, \Sigma \, (j=0 \text{ to } 5) \; a_j \cos^j \theta + I_{ds} \, 2 \, \Sigma \, (j=0 \text{ to } 5) \; a_j /(j+2)$$

Where

a_j = absorption coefficient

t_j = transmission coefficient

Table 12 in Appendix A gives values of a_j and t_j .

Solar Heat Gain Factor (SHGF)

= energy transmitted + Ni * energy absorbed

Ni =Heat transfer factor , inward flow fraction.

Appendix A contains a sample of calculations

Appendix C gives SHGFs

CHAPTER FOUR
COOLING LOAD TEMPERATURE DIFFERENCES (CLTD)

4-1 INTRODUCTION :

In calculating cooling loads for air-conditioning , the heat balance approach is a fundamental concept , but it is cumbersome and difficult to use for routine using .

To simplify and reduce the time required for computations, transform methods have been applied to this problem .
Design cooling load for one day as well as long term energy calculations may be done using the transfer function method (TFM).
It is not always practical to compute the cooling load using TFM ; therefore CLTD method (a hand calculation) has been developed from TFM . The method involves extensive use of tables and charts and various factors to express the dynamic nature of the problem and predicts cooling load within about 5 percent of the values given by the TFM . (1)

4-1 TRANSFER FUNCTION METHOD (TFM)

The transfer function method is the cooling load calculation procedure most closely approximating the heat balance concept .
The transfer function method (TFM) applies a series of weighting factors , or conduction transfer function (CTF) coefficients to the various exterior opaque surfaces and to differences between sol-air temperature and inside space temperature to determine heat gain with appropriate reflection of thermal inertia of such surfaces .(1)

The coefficients relate an output function at a given time to value of one or more driving function at a given time at a set period immediately preceding .

While the TFM is preferable to other methods and technically sound for specific cooling load analysis , its computational complexity requires computer use . (7)

4-2 COOLING LOAD TEMPERATURE DIFFERENCES (CLTD)

Is a simplified version of the TFM to calculate cooling loads as an approximation of the TFM . Where applicable , this method may be suitable for hand calculations.

Data obtained by using TFM on a group of applications considered representative were then used to generate CLTD data for direct one-step calculation of cooling load from conduction heat gain through sunlit walls and roofs and conduction through glass exposure.

The CLTD method makes use of a temperature difference in the case of walls and roofs , and cooling load factors CLF in the case of solar gain through windows and internal heat sources .

The CLTD and CLF vary with time and are a function of environmental conditions and building parameters .
CLTD include the effect of :
1- Time lag in conductive heat gain through opaque surfaces .
2- Time delay by thermal storage in converting radiant heat to cooling load.

4-4 CALCULATION PROCEDURE

4-4-1 SOL-AIR TEMPERATURE

Is the temperature of the out door air that , in the absence of all radiation changes , gives the same rate of heat entry into the surface as would the combination of incident solar radiation , radiant energy exchange with the sky and other outdoor surroundings , and convective heat exchange with the outdoor air.

4-4-2 HEAT FLUX INTO EXTERIOR SUNLIT SURFACES

The heat balance at a sunlit surfaces gives the heat flux into the surface q/A as:

$$Q/A = \alpha\, It + h_o (t_o - t_s) - \varepsilon_h \Delta R$$

Where:

α = absorbance of surface for solar radiation

It = Total solar radiation incident on surface W/m^2

h_o = coefficient of heat transfer by long-wave radiation and convection at outer surfaces W/m^2 k.

t_o = outdoor temperature. ^{o} C

t_s = surface temperature. ^{o} C

ε_h = hemispherical emittance of surface.

ΔR = difference between long-wave radiation incident on surface from sky and surroundings and radiation emitted by blackbody at outdoor air temperature W/m^2.

Assuming the rate of heat transfer can be expressed in terms of the sol-air temperature

$$Q/A = h_o (t_e - t_s)$$

And therefore:

$$t_e = t_o + \alpha\, It /h_o - \varepsilon_h \Delta R$$

For horizontal surfaces that receive long-wave radiation from the sky only , an appropriate value of ΔR is about 63 W/m^2 , so if ε_h =1 and h_o = 17 $W/m^2.K$, the long wave correction term is about –3.9 ^{o} C. (3)

Because vertical surfaces receive long-wave radiation from the ground and surrounding buildings as well as from the sky , accurate ΔR values are difficult to determine. (3)

When solar intensity is high , surfaces of terrestrial objects usually have a higher temperature than outdoor air , thus their long-wave radiation compensate to some extent for the skies low emittance .Therefore it is common practice to assume ΔR = 0 for vertical surfaces. (3)

Values of It incorporate diffuse radiation from a clear sky and ground reflection , but make no allowance for reflection from adjacent walls.

Surface color:

Sol-air temperature values are given for two values of the parameter α /h_o ,

.026 for a light color.

. 052 for dark color (max).

Therefore

For horizontal surfaces:

$t_e = t_o + .026\ It + 3.9$ (light color)

$t_e = t_o + .052\ It + 3.9$ (dark color)

For vertical surfaces:

$t_e = t_o + .026\ It$ (light color)

$t_e = t_o + .052\ It$ (dark color)

4-4-3 HEAT GAIN BY CONDUCTION THROUGH EXTERIOR ROOFS AND WALLS

ASSUMPTIONS:

a- The indoor air temperature is constant.

b- Both indoor and outdoor surface heat transfer coefficients are constant.

$$Q = A\ \{\Sigma\ b_n(t_{eq\text{-}nd}) - \Sigma\ d_n(Q_{eq\text{-}nd}) / A - t_{rc}\ \Sigma\ C_n\}$$

Where:

A = indoor surface area of a roof or wall m^2

Q_{eq} = heat gain by the space through indoor surfaces of a roof or wall , Watt.

θ = time ,hours.

d = time interval , hours.

n = summation index (each summation has many terms as there are non negligible values of the coefficients)

$t_{eq\text{-}nd}$ = sol-air temperature at time $t_{\text{-}nd}$ ^oC

t_{rc} = constant room air temp. ^oC

b_n , C_n and d_n = Conduction transfer function coefficients.

Conduction transfer function coefficients (CTF).: are usually calculated using combined outdoor heat transfer coefficient of 17.0 $W/m^2.K$, indoor coefficient of 8.3 $W/m^2.K$, and the wall or roof constructions. as may be appropriate. (1)

Approximate values can be obtained by selecting a set of transfer function coefficients from tables for a roof or wall construction nearly the same as the roof or wall under consideration and multiplying b and C_n by the ratio of the U-value of the roof or wall under consideration to the U-value of the selected roof or wall from the table.(ASHRAE tables).

4-4-4 CONVERSION OF HEAT GAIN TO COOLING LOAD BY ROOM TRANSFER FUNCTION COEFFICIENTS (RTF) :

The cooling load of a space depends on the magnitude and nature of the sensible heat gain and on the location and mass of the room objects that absorb the radiant heat. Thus each component of the room heat gain gives rise to a distinct component of the cooling load , and the some of these various components at any time is the total cooling load at that time .(1)

The heat gain is related to the corresponding cooling load by a room transfer function coefficients , which depend on the nature of the heat gain and on the heat storage characteristics of the space .

Where the heat gain q_θ is given at equal time intervals , the corresponding cooling load Q_θ at time q can be related to the current value of q_θ and the preceding values of cooling load and heat gain

$$Q_\theta = \sum_{i=1} (v_0 * q_\theta + v_1 q_{\theta-\delta} + v_2 * q_{\theta-2\delta} + \ldots\ldots) - (w_1 * Q_{\theta-\delta} + w_2 * Q_{\theta-2\delta} + \ldots\ldots)$$

Where i is taken from 1 to the number of the heat components

δ = time interval

$v_0, v_1 \ldots, w_1, w_2 \ldots$ = RTF coefficients

4-4-5 **WALLS AND ROOFS CLTD** :

These are commonly used walls and roofs (in Khartoum Buildings) to calculate CLTD for :

Table 9 **COMMON WALLS AND ROOFS** IN KHARTOUM BUILDINGS

Wall	Construction	U-value	ASHRAE wall	U-value
W1	200 mm common brick + plaster (two sides) A0+ E1+C9+ E1+ +E0	1.957	G10	.881051
W2	100 mm face brick +200 mm common brick + plaster A0+A2+C9+E1+E0	1.783	G25	.341588
R1	150 mm h.d. concrete	2.174	G7	.784431
R2	Steel siding +insulation	.739	G1	.455689

The calculation of the CLDT is based on :

1- ASHRAE conduction transfer function coefficients tables .
2- dark flat surface
3- indoor temperature of 25 ^{o} C
4- outdoor maximum temperature of 43 ^{o} C
5- mean temperature of 36.9 C
6- daily range of 12 ^{o} C
7- solar radiation of 15.6 ^{o} N latitude , 16th day of May .
8- outside surface resistance Ro = .059 m^2 C /W
9- inside surface resistance Ri = .121 m^2 C/W
10- roof without suspended ceiling , no attic fan or return duct in suspended ceiling space .

Appendix A contains a sample of calculations

Tables 14 - 20 contain summary of the calculations and calculated values .

CHAPTER FIVE

COOLING LOAD CALCULATION

5-1 INTRODUCTION:

The cooling load is the rate at which energy must be removed from a space to maintain the temperature and humidity at the design values .

The cooling load will generally differ from the heat gain for any instance of time .

In design for cooling load, however, transient analysis must be used if satisfactory results are to be obtained. This is because the instantaneous heat gain into a conditioned space is quite variable with time primarily because of the strong transient effect created by the hourly variation in solar radiation. There may be an appreciable difference between the heat gain of the structure and the heat removed by the cooling equipment at a particular time. (4) . This difference is caused by the strong and subsequent transfer of energy from the structure and contents to the circulated air. If this is not taken into account the cooling and dehumidifying equipment will usually be grossly oversized , and estimates of energy requirements meaningless.

5-2 OUTDOOR DESIGN CONDITIONS :

Weather conditions vary considerably from year to year . however , the purpose of cooling load calculation is to obtain data on which the cooling system components are sized .

It is not reasonable to design for the worst conditions on records because a great excess of capacity will result (4)

The heat storage capacity of the structure also plays an important role in this regard . a massive structure will reduce the effect of overload from short intervals of outdoor temperatures above the design value . (4)

A proper cooling load calculation gives values adequate for proper performance .

5-3 INDOOR DESIGN CONDITIONS :

The perception of comfort, temperature and thermal acceptability is related to one's rate of metabolic heat production, its rate of transfer to the environment, and the resulting physiological adjustment and body temperature. The heat transfer rate is influenced by the environmental factors of air temperature , thermal radiation , air movement and humidity and by the personal factors of activity and clothing .The main purpose of HVAC system is to maintain comfortable indoor conditions for occupants . In most cases the system will rarely be set to operate at design conditions , therefore , the use and occupants of the space is a general consideration from the design temperature point of view . 25 ^{o}C , and 50% relative humidity is recommended for summer comfort . (4)

5-4 WEATHER ORIENTED DESIGN FACTORS

The usual approach in air conditioning system design involves computation of peak design load at specific hour of a design day using one of the frequency levels of design conditions in the table . (2) . Table (6)

A design day is defined as :

1- A day on which the dry and wet-bulb temperature are peaking simultaneously .
2- A day when there is little or no haze in the air to reduce the solar heat .
3- All of internal loads are normal .

Using the design dbt with the design wbt gives computed loads significantly greater than actual load .Typically the design dbt with mean coincident wbt should be used in computing building cooling loads. (2)

Minimum temperature usually occur between 6:00 am and 8:00 am sun time on clear days when the daily range is greatest . For residential and other applications where occupancy is continuous throughout the day , the recommended design temperatures apply . For commercial or other applications where occupancy is only during hours near the middle of the day , design temperatures above the recommended minimum might apply. (2)

Maximum temperature usually occur between 2:00 pm. And 5:00 pm. Sun time with deviations on cloudy days when the daily range is less .Typically the design dry-bulb temperatures should be used with the coincident wet-bulb temperatures in computing building cooling loads . For residential and other applications where occupancy is continuous throughout the day , the recommended design temperatures apply . For commercial or other applications where occupancy is only during hours near the middle of the day , design temperatures below the recommended maximum might apply. (2)

In some cases , the peak occupancy load occurs before the effect of the outdoor maximum temperature has reached the space by conduction through the building mass. In other cases the peak occupancy loads may be in months other than the three or four summer months when the maximum outdoor temperature is expected , design temperature from other months will apply .

The following details should be considered to properly design an air-conditioning system ;

1- The location , elevation and orientation of the structure so that the effects of the weather (wind, sun and precipitation) on the building air-conditioning load can be anticipated .
2- Building size and shape .
3- Space characteristics (office , bank ,… etc.)
4- Material , fenestration , doors , occupancy and lighting
5- Ventilation requirements .
6- Outside design temperatures and wind velocities

5-5 MODES OF HEAT GAIN:

1- Solar radiation through transparent surfaces
2- Heat conduction through exterior walls and roof .
3- Heat conduction through interior partitions , ceilings , and floors .
4- Heat generated within the space by occupants , lights , and appliances .
5- Energy transfer as a result of ventilation and infiltration of outdoor air .

6- Miscellaneous heat gain.

5-6 THE CLTD METHOD

5-6-1 Heat transfer through walls and roofs :

$$Q = U\ A\ CLTD$$

Where U = overall heat transfer coefficient $W/m^2\ C$

A = area m^2

CLTD = temperature difference which gives cooling load at time q , C

5-6-2 Solar heat gain through glass :

The principles of calculating heat gain for fenestration consists of two components :

1- Cooling load from conduction :

For conduction heat gain , the overall heat transfer coefficient accounts for the transfer processes of :

a- convection and long wave radiation exchange outside and inside the conditioned space and

b- conduction through the fenestration

To calculate cooling load for this component , the conduction heat gain is treated in similar to that through walls and roofs , the coefficient used to convert heat gain to cooling load are the same as those for walls and roofs .

The cooling load from conduction and convection heat gain are calculated by :

$$q = UA(CLTD)$$

Where A is the net area of the fenestration m^2

2- Solar heat gain :

The basic principle of evaluating heat gain from transmitted and absorbed solar energy through fenestration , including the primary terms solar heat gain factor (SHGF) , and shading coefficient (SC), are the same for the CLTD, CLF procedure as previously described .

Cooling load due to solar heat gain is expressed in terms of a shading coefficient , a solar heat gain factor , and a cooling load factor :

$$q = A\ (Sc)\ SHGF\ (\ CLF)q$$

where

A = area , m^2

Sc = shading factor (internal shade)

SHGF = Solar heat gain factor W/m^2

CLF = cooling load factor for time q

The CLF accounts for the variation of the solar heat gain factor with time , the massiveness of the structure , internal shade , and thermal response of the radiant part of the solar input .

5-7 OUTDOOR AIR HEAT GAIN :

5-7-1 General :

The dominating function of outdoor air is to control air quality, and spaces that are less continuously occupied require some outdoor air. The required outdoor air is dependent on the rate of contaminant generation and the maximum acceptable contaminant level. (4)

The outside air flow being introduced to a building or zone by an air handling unit can also be described by the outside air fraction X_{oa} which is the ratio of the flow rate outside air brought in by the air handler to the total supply air flow rate . When expressed by the percentage , the outside air fraction is called the percent outside air .The supply air flow rate is that required to meet the thermal load . The outside air fraction and percent outside air then described the degree of recirculation , where a low value indicates a high rate of recirculation , and a high value shows a little recirculation . Conventional all-air handling systems for commercial and institutional buildings have approximately 10 to 40% outside air . 100% outside air means no circulation of return air through the air handling system . Instead , all the supply air is treated outside air , also known as make up air , and all return air is discharged directly to the outside as relief air . (4)

Outside air introduced into a building constitute a large part of the total space conditioning load , which is one reason to limit air exchange rates in buildings to a minimum required . Air exchange typically represents 20 to 40% of a shell-dominated building's thermal load .

First , the incoming air must be cooled from the outdoor air temperature to the indoor air temperature . The rate of energy consumption due to this sensible cooling is given by :

$$qs = Q \rho \, cp \, \Delta t$$

Where :

Qs = sensible heat load

Q = airflow rate

ρ = air density

Cp = specific heat of air

Δt = outdoor – indoor temperature difference

Second , air exchange modifies the moisture content of the air in the building . This is particular important in the summer when the outdoor air must be dehumidified . The rate of energy consumption associated with these latent loads is given by :

$$q_1 = Q \rho h_{fg} \Delta W$$

Where :

q_1 = latent heat load

h_{fg} = latent heat of water

Δw = humidity ratio of outside air minus humidity ratio of indoor air

Air exchange of outdoor air with the air already in a building can be divided into :

1- Ventilation :

Is the intentional introduction of air from the outside into a building . This ventilation can be subdivided into natural and forced ventilation .

Natural ventilation is the intentional flow of air through open windows , doors , grilles , and other planned building envelope penetrations , and it is derived by natural and or artificially produced pressure differentials .

Forced or mechanical ventilation is the intentional movement of air into or out of the building using fans ,and intake and exhaust vents .

2- Infiltration or air leakage :

Is the uncontrolled flow of outside air into a building through cracks and other unintentional openings and through the normal use of exterior doors for entrance and egress .

Exfiltration is the leakage of indoor air out of the building .

Infiltration and exfiltration are driven by natural and or artificial pressure differences .

A conditioned space may be ventilated by natural infiltration alone or in combination with intentional mechanical ventilation

Natural infiltration varies with indoor – outdoor temperature difference , wind velocity and the tightness of the construction .

The magnitude of the outdoor airflow into the building must be known for proper sizing of the HVAC equipment and evaluation of energy consumption .It can generally be stated that in most cases more outdoor air than necessary is supplied , and attempts to save energy through reduction of outdoor air will cause poor quality indoor air. ASHRAE 62(10) defines acceptable air quality as ambient air in which there are no known contaminants at harmful concentrations and with which a substantial majority of people exposed do not express dissatisfaction.

Outdoor air should be held to a minimum consistent with the health and comfort of occupants and energy conservation considerations. The outdoor air should be reduced to allow for the expected infiltration. (4).

5-7-2 Enthalpy:

The enthalpies corresponding to the design values based on dbt are (from the psychrometric chart)

Table 10

Per. %	Dbt C	wbt C	H1 KJ/Kg K
.4%	43	22	63.74
1%	42	22	63.78
2%	41.5	22	63.8

The enthalpies corresponding to the design values based on wbt are (from the psychrometric chart)

Table 11

Per. %	Wbt C	dbt C	H3
.4%	25.5	35	78
1%	25	35	76
2%	24	34	71

The enthalpy of the air based on 25 C , and 50% relative humidity (the normal inside design conditions used in calculations) is 50.36 KJ/kg K .

The difference in enthalpies is as follows :

63.74-50.36 = 13.38 KJ/kg K

78-50.36 = 27.64 KJ/kg K

Therefore , the cooling load of outside air based on wbt design values is two times that based on dbt design values .

The designer should decide if the outdoor air load is dominant to base his design on wbt design values as the difference of enthalpy per kg of outdoor air is greater than that based on dbts design values , and hence , the dbts to be used are those corresponding to the wbts design values , which are relatively lower than that based on the dbts design values.

Computation of the heat gain from outdoor air may be computed separately to obtain an estimate of the total load .

$$Qs = m_a\, c_p\, (to-ti) = Q_o\, c_p / v_o\, (to-ti)$$

$$Ql = m_a\, (w_0-w_1) = Q_o/v_o\, (w_0-w_1)$$

These heat gains may also be computed using the psychrometric chart

CHAPTER SIX

SELECTION OF CONDENSER TYPE

6-1 INTRODUCTION :

The condenser removes the heat from refrigerant carried from evaporator and added by compressor , and convert the vapor refrigerant into liquid refrigerant

It is a heat exchanger in which heat transfer takes place from high temperature vapor to low temperature air or water, which is used as cooling medium.

Two considerations are necessary in design for effective functioning of the condenser:

1- Effective temperature differential
2- High heat transfer coefficient.

6-2 COOLING MEDIUM :

The cooling mediums provided by the nature are air and water . Both can be used as cooling medium either separately or combined as per the availability and requirement. (10)

Air cooled condensers are designed for condensing temperatures of 15 to 20 ^{o}C above the temperature of the entering air , and 6 to 12 ^{o}C above if water is used as cooling medium . (10)

Few more factors considered with air or water-cooled condensers:

1- Quantity of cooling medium: the quantity of air required in air-cooled condensers is 30 to 35 cum/min/ton of refrigeration, where as quantity of water is 7 to 20 lit/min /ton depends upon the source of water. (10)
2- Amount of condensing surface : lower heat transfer surface can be provided with higher heat transfer surface coefficient and greater temperature differential between the cooling medium and condensing gas .The area required with air cooled condensers is 4 to 5 times greater than the water cooled condensers . Water cooled condensers required 0.5 to 1 m^2 area per ton of refrigeration capacity .
3- Velocity of cooling medium : The heat transfer rates increase with an increase in velocity of the cooling medium . The recommended velocities for air cooled condensers are 270 to 300 m/min , and with water cooled condensers are 120 to 180 m/min . (10)
4- Type of refrigerant ; The heat transfer coefficient of condensing side mostly depends upon the properties of the refrigerant as density , conductivity ,viscosity and latent heat .
5- Purity of the refrigerant : the heat transfer coefficient also depends upon the purity of the refrigerant as free from oil and air .

6-3 CONDENSING TEMPERATURE :

Is the refrigerant temperature required to reject heat to the condensing medium. Increasing the condensing temperature 15 degrees from a base of 40 F and 105 F reduces the capacity about 13% , and at the same time increases the compressor horse power per ton 27% . (10)

The condensing temperature adopted with air as cooling medium is 55 to 60 ^{o} C , where with water it is 31 to 37 ^{o} C . The condensing temperature with water is also depends upon the purity of water

6-4 ECONOMIC OPERATION OF CONDENSERS :

The compressor power cost increases continuously with the increase in condenser pressure as the work done per Kg of the refrigerant increases, but the cost of cooling water decreases with an increase of condensing pressure.

6-5 AIR-COOLED CONDENSER

An air-cooled condenser consists of a coil , casing , fan and motor . It condenses the refrigerant gas by means of a transfer of sensible heat to air passed over the coil .The circulation of air may be natural or forced convection . The area required for natural convection is considerably large compared with forced convection due to its low heat transfer coefficient .

For a given surface and air quantity , the capacity of an air-cooled condenser varies , for practical purposes , in direct proportion to the difference TD between the condensing temperature and the entering air dbt . Therefore , assuming the heat rejection requirement is constant , a fall or rise in entering air temperature results in an equal decrease or increase in condensing temperature.

Ratings of air-cooled condensers are based on the temperature difference (TD) between the dry-bulb temperature of the air entering the coil and the saturated condensing temperature corresponding to the pressure at the inlet.

Values for TD range from 15-35 degrees , with condensing temperatures between 110-135 F . (10).

Typical TD value is 14-17 K for air-conditioning systems at a 7 ^{o} C evaporator temperature.(3)

Condenser is considered proportional to the TD . The capacity at 16 K TD is about 50% greater than the same condenser selected for 8 K TD. (10) .(3)

The advantages of air-cooled condensers compared to water-cooled condensers are ;

1- Simplicity of construction
2- .No handling problems .
3- Piping arrangement for carrying the air are not required
4- No problem of disposal of used air
5- Fouling effects are very low
6- Installation and maintenance cost are considerably less
7- High flexibility

The disadvantages are :

1- Poor heat carrying capacity
2- It requires very large quantity of air per ton of refrigeration .
3- The condenser gives lower capacity when the outside air temperature is high and this is usually the time when greatest capacity is required .

6-5-1 UNIT SELECTION :

Condenser first cost advantages may be realized with higher condensing temperature. However, it must be recognized that as the chosen condensing temperature is increased, compressor power input increases also. Higher compressor power requirements may be partially or fully offset by decreased condenser fan motor KW .

The 2.5% occurrence value is suggested for summer operations (10).The statistical analysis gives 42 ^{o} C as 2.5% occurrence value during summer months .

An entering air temperature that is higher than expected quickly necessitates compressor pressure and power that is higher than design .Both these factors may cause unexpected system shutdown , usually when it can least be tolerated.

The increased condensing pressure increases power consumption and reduces capacity.

6-6 EVAPORATIVE CONDENSER:

These condensers are more preferable where acute water shortage exists , drain facilities are not available , high water cost and uneconomical cooling tower installation . It uses the combined principles of water cooled condensers and cooling towers . Water is sprayed on the surface of the coil , air is drawn crossing the refrigerant coil and water spray , the air current carries the water from the surface of the coil in the form of vapor . This mode of heat transfer in the form of evaporation of water reduces the water required to be circulated in the condenser .

Most of the heat given by the refrigerant vapor is carried by the air in the form of sensible and latent heat , therefore , the effectiveness of this type of condensers depends upon the wet bulb temperature of the incoming air . Lower is the WBT of the incoming air , higher will be the effectiveness of the condenser .

The capacity of the condenser also depends upon the quantity of air circulated through the condenser , but this is limited by the maximum air velocity permitted through the eliminators without the carryover of water particles . (10)

This type of condensers gives better performance in dry weather (low wbt) compared with wet weather (high wbt)

The capacity of an evaporative condenser may be increased either by lowering the entering air wbt or by increasing the condensing temperature. Condensing temperature normally range between 100-110^{o} F .

6-6-1 UNIT SELECTION :

The normal design wbt of the location should be the basis of an evaporative condenser selection since high wbt seldom occur. The 2.5% occurrence value is 25 ^{o} C. If designed conditions must be maintained at all times the maximum wbt should be used to insure adequate equipment capacity . (10)

6-7 WATER- COOLED CONDENSERS

A water-cooled condenser consists of heat transfer tubes mounted within a steel shell . Condenser water passes through the tubes , and the condensing refrigerant occupies the shell surrounding the tubes .The shell is equipped with a hot gas inlet , a liquid sump , purge connections , water regulating valve connection , and a pressure relief device .

Water-cooled condensers are always preferred where adequate supply of clean and inexpensive means of water disposal are available .

6-7-1 ECONOMICAL WATER RATE THROUGH THE CONDENSER:

The lower the water quantity used, the higher is the condensing temperature and greater the power cost . Conversely, the higher the water quantity used, the lower will be the condensing temperature and smaller the power cost .

Condensing temperature usually range from 100-110 ^{o} F , but may be as low as 80 ^{o} F .The water –cooled condenser and the cooling tower should be considered as a single heat rejection device for the purpose of selection and application . (10)

6-7-2 COOLING TOWER

A cooling tower consists of a casing, basin and sump , water distribution system , fill , fan , motor and drive .

The rate of heat transfer from the water to the air depends on the enthalpy of the air which is represented by wbt. This rate is independent of the air dbt .

For a given air and water quantity through a tower the rate of heat transfer , or rated tower capacity , is increased by lowering the entering air wbt requirement , or by raising the temperature of the water entering the tower .

Tower performance is specified in terms of water range and approach .

Cooling range is the difference between water entering and leaving temperatures. And is equal to the temperature rise through the condenser .

Except for relatively small installation on which the spray- filled atmospheric tower may be used, the mechanical draft tower is the most widely used for air-conditioning applications.

The difference between the leaving water temperature and entering air wet bulb temperature is the approach of the cooling tower.

The approach is a function of cooling tower capability, and a larger cooling tower produces a closer approach (colder leaving water for a given heat load , flow-rate and entering conditions).

Thus the amount of heat transferred to the atmosphere by a cooling tower is always equal to the heat load imposed on the tower, while the temperature level at which the heat is transferred is determined by the thermal capability of cooling tower and the entering air wet-bulb temperature.

The thermal performance of a cooling tower depends principally on the entering air wet-bulb temperature.

The entering air dry-bulb temperature and relative humidity taken independently have an insignificant effect on the thermal performance of mechanical draft cooling tower, but they do affect the rate of water evaporation within the cooling tower.

The amount of heat transferred from the water to the air is proportional to the difference in enthalpy of the air between the entering and leaving conditions (Hb-Ha).

Because lines of enthalpy correspond almost exactly to lines of constant wet-bulb temperature lines the change in enthalpy of the air may be determined by the change in wet-bulb temperature of the air .

The thermal capability of any cooling tower may be defined by the following parameters:

1-Entering and leaving water temperature.
2-Entering air wet-bulb or entering air wet and dry-bulb temperature.
3-Water flow rate.

The thermal capability of a cooling tower used for air-conditioning applications may be expressed in nominal capacity which is based on heat dissipation of 1.25 KW per KW of evaporator cooling.

Nominal cooling capacity is defined as cooling 54ml/s of water from 35 C to 29.4 C at 25.6 C entering wet-bulb temperature. At these conditions the cooling tower rejects 1.25 KW per KW of evaporator capacity. (10)

6-7-3 UNIT SELECTION

A cooling tower should be selected for the normal design wbt of the location

The 2.5% design value is 25 ^{o} C. If designed conditions must be maintained at all times, the maximum wbt should be used to insure adequate equipment capacity .

CHAPTER SEVEN

EVAPORATIVE COOLING

7-1 INTRODUCTION

It is a process of adiabatic saturation of air when a spray of water is made to evaporate into it without transfer of heat from or to the surrounding.

Simple evaporative cooling is achieved by direct contact of water particles and moving air stream . If the water is circulated without a source of heat and cooling , dry air will become more humid and will drop in temperature .

In a complete contact process , the air would become saturated at WBT of the entering air .The air may be sufficiently cooled by evaporative processes to result a considerable degree of summer comfort in climates of high dry-bulb temperatures associated with low relative humidity .The comfort given by evaporative cooling always depends upon the outdoor temperatures and relative humidity . High DBT and low WBT always give more comfort with evaporative cooling .

In climates where high dry-bulb temperatures are associated , over time with relatively low wet-bulb temperatures , air may be sufficiently cooled by evaporative (adiabatic) process to produce a considerable degree of sensible cooling , perhaps resulting in an air condition within the summer comfort zone .A rather higher relative humidity within the space usually result.

The first requirement for successful evaporative cooling is a high dry-bulb temperature , usually in excess of 95 ^{o} F . (11) . The dry-bulb temperature must be so uncomfortably high that any amount and kind of cooling will be considered a welcome relief, and that is the case in KHARTOUM in summer months.

The second requirement – a relatively low wet-bulb temperature determines the amount of cooling which can be accomplished.

The wbt prevailing in the area is then one of the important limitations on evaporative cooler performance.

It is generally considered that evaporative cooling is satisfactory only in areas where DBT in excess of 90 ^{o} F combine with WBT lower than 75 ^{o} F. (11)

Although the evaporative cooling does not perform all the functions of true air-conditioning, but it provides comfort by filtering and circulating the cooled air This system does not dehumidify the air , but on the contrary further humidify the air .

The evaporative cooling has the following major drawbacks:

1- Its capacity is limited by the WBT of the ambient air , maximum cooling achieved is the WBT .
2- It is not useful for high DBT and high relative humidity summer weather.
3- During the cooling process, a moisture is added resulting in an increase in the moisture content of the space being cooled contrary to the requirement.

7-2 Thermodynamics of the evaporative cooling ;

Evaporative cooling is considered as an adiabatic system , therefore , the process occurs at constant enthalpy .

The cooling of air (reducing DBT) is the result of transferring the sensible heat of air to the evaporated water carried with the air , and this sensible heat is transferred in form of latent heat . The latent heat required to evaporate water carried with the air when it is passing through a spray of water is taken from the sensible heat of air .

Sensible heat loss by the air = latent heat gain by the air

$$C_p (T_1-T_2) = h_{fg} (w_2- w_1)$$

Where :

C_p = sensible heat of moist air W

T_1 and T_2 = cooler entering and leaving air dry-bulb temperatures ^oC

Hfg = latent heat of vaporization of water at T_2 W

w_1 and w_2 = specific humidity of air entering and leaving the evaporative cooler Kg/Kg of dry air.

Higher drop in DBT (T_1-T_2) is possible with greater increase in specific humidity (w_2-w_1) .

The evaporative cooling of the same DBT of outdoor air is more effective if the relative humidity of the air is lower .

7-3 Effective temperature and evaporative cooling :

The concept of effective temperature ET , which combines the effects of temperature and humidity and air motion is useful in design of evaporative cooling system .

The DBT reduction brought about by evaporative cooling always results in a lower ET regardless of the relative humidity . This is one of the plus factors for evaporative cooling . (11)

It is evident that much less air is required with evaporative cooling than with untreated outside air . High air velocity and more air changes are necessary for better performance of evaporative cooling . (11)

7-4 ANALYSIS:

Based on the foregoing and looking to the temperature records , it is seen that in April the wbt is rarely exceed 23 ^oC , with average value of 19 ^oC during the hottest hours (10-17),while the dbt normally exceeds 35 ^oC with average value of 37.6 ^oC , and that means the climate in April is strongly calling for evaporative cooling as the conditions of the climate prevails is suitable. In May and June the wbt is slightly higher than April , it is rarely exceed 23.5 ^oC , (with mean values of 21.3^o for May and 21.8^o for June) during the hottest period of the day, of which the dbt is also a bit higher than April with mean values of 39.4 ^oC for May and 38.9 ^oC for June respectively , and therefore the situation is still adequate for evaporative cooling .

During July , August , September and October , the DBTs are generally lower than the preceding months , and on the other hand the WBTs are generally higher as these months are the rainy season .This situation decreases the benefit of the evaporative cooling , although the DBTs are still adequately high as seen from the table below , the WBTs will limit the drop in temperature and hence reduce the cooling effect .

October is the best of these , August is the worst .

The following table illustrate the mean temperature values of summer months during the hottest hours of the day:

Table 12 Summer months mean temperatures (D and WBTs)

Month	DBT C	WBT C
Apr	37.6	19
May	39.4	21.3
Jun	38.9	21.8
July	35.7	23.2
Aug.	34.6	23.5
Sep	36.3	23.3
Oct	36.4	21.9

CHAPTER EIGHT

DISCUSSION, CONCLUSION AND RECOMMENDATIONS

8-1 DISCUSSION :

8-1-1 SYSTEM DESIGN VALUES:

The design values are provided and used as an estimate of the annual cumulative frequency occurrence of the weather conditions at Khartoum for several years in the future

The DBT design values represent the peak sensible load that can be expected from the transmission through walls, roofs , glazing , and the sensible component of the outdoor air load , whereas the mean coincident WBTs are the values upon which the latent load of the outdoor air is based when the DBTs of the peak sensible load are practically occurred . This conditions are usually occurred during the most hot months of April , May and June , during which the high DBTs values of the whole year are recorded .

During the months of July , August and September. , the DBTs are lower than that of the preceding period , but on the other hand the WBTs are increasing significantly. These values are the highest during the whole year, and hence should be considered as design values when the outdoor air is the dominant source of heat since the latent load of the air is large due to the high enthalpy of the air .

The occurrence of high DBTs with high WBTs has no existence .

The 43 ^{o}C (0.4% value) represents the maximum design value , equals the maximum design value for June . It is 0.5 ^{o} C less than that of April and 1 ^{o} C of May . This value will result in oversized system in the rest of the year .

If for any reason , the building will not be used during these months ,or if it is possible to reduce the load during these months , using this value will lead to an uneconomical design .

The value of 1% which is 42 ^{o} C , is less than the design value of May (the hottest month),and 0.5 ^{o} C less than the intermediate design values (2.5%) of April and June , and 0.5 ^{o} C more than the third design values of these months , but it exceeds the maximum design values of July , August , September and October . This value represent a suitable design value throughout the year if the load during the hottest period of April to June is not maximum or the indoor design conditions can be accepted.

The value of 2% (41.5 ^{o} C), coincident with the third frequency level (5%) design value of April and June and the maximum design value of September, but exceeds the maximum values of the other summer months .

For all of the above dry-bulb design values , the analysis gave the same value of the mean coincident wet-bulb temperature (22 ^{o}C), and therefore the contribution of the load due to wet-bulb temperature will be the same irrespective of the dry-bulb temperature value selected .

Generally, the wet-bulb temperature design values are much higher in months of July through September (the humid period) than the values of April to June (the hottest period) , and this lead to a large difference in outdoor air enthalpy when comparing the two periods , ie the enthalpy in humid period is much greater . These values can not be used unless the outdoor load is the dominant source of load compared with other sources, and that the indoor conditions have to be maintained.

A clear sky solar radiation intensity is now available to be used in calculation of sol-air temperatures for each hour during any month.

The CLTD calculations can be done for every month to any type of walls or roofs by the same procedure.

The CLTDs calculated and tabulated in tables (in the Appendix) can be used in a one-step calculation procedure of cooling load . It can be entered in a computer program for quick and precise calculations as this done for some places such as USA cities.

The solar heat gain factors to calculate solar heat gain through fenestration, has been generated from the solar radiation intensity for each hour of the year.

The outdoor dry-bulb temperature design value selected from the table of design values can be used for calculation of the conduction heat portion of the total heat gain through fenestration.

8-1-2 CONDENSER SELECTION :

It is obvious from the temperature profiles that for months of April to June , when the dbt is high accompanied by relatively low wbt and hence low relative humidity , the water-cooled condenser will be expected to be the choice . This situation is reversed during the wet months , when the wbt is high , and the dbt is relatively low , and therefore the relative humidity is high , and the situation is not suitable for operation of cooling towers , and accordingly air-cooled condensers emerge as the suitable suggestion to fit the situation .

Selecting a cooling tower based on Sep. wet-bulb temperature. data will result in a larger tower cooling capacity during the preceding months as the water will leave colder , and this affect the condenser capacity which give larger capacity . If the entire design is based on Sep. data , the selection is satisfactory , if not , the over capacity resulted from the tower performance has no meaning and can be saved as money if proper data is considered from the beginning .

The air-cooled condenser selection should be based on the warmest months temperature data to maintain the designed cooling load , as during these months the condenser normally gives its lowest capacity .

The condenser will give more capacity when the dry-bulb temperature decreases during the rainy season . If the designed load is based on the rainy season parameters , the condenser can be selected considering the same parameters for economical and energy saving reasons .

8-2 CONCLUSION :

8-2-1 The outdoor design temperature should generally be 2.5% value as recommended by ASHRAE standard 90 A(3) . This value is 42 ^{o} C from the analysis . Should the outdoor temperature rise above the design value for some extended period, the indoor temperature may do likewise, and the structure will reduce the effect of overload from this period. The failure of the system to maintain design conditions during brief periods of severe weather is not critical .

However close regulation of indoor temperature may be critical for some industrial processes.

If the structure is of light construction (low heat capacity) , poor insulated , has considerable glass and space temperature control is critical , the 0.4 % value should be considered . This value is 43 ^{o} C from the analysis .

8-2-2 For the air-cooled condenser type selection , 42 ^{o} C out-door dry-bulb temperature represents the adequate value on which the selection can be based to operate at to give the designed load .

8-2-3 For cooling towers and evaporative condensers , 25 ^{o} C outdoor wet-bulb temperature is found suitable from the statistical analysis .

8-3 RECOMMENDATIONS :

1- Weather conditions vary from year to year , and to some extent from decade to decade , due to the inherent variability of climate , and therefore design values vary depending on the period of records , and hence up dating of the design conditions is recommended from time to time .

2- Investigation of similar data including dew-point records (hourly readings) to set design values based on dew points for humidity ratio peaks.

3- Similar study is recommended for the other towns in Sudan , such as Portsudan , Dongla , …. , etc. , to prepare values for proper design of air-conditioning system of their buildings .

4- Cooling load temperature differences are recommended to be calculated for the common building walls and roofs in Sudan .

5- A study using statistical analysis is recommended to prepare design values for a complete year

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