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Graduation Project

Energy Conservation Study for the building of "Children Happiness Center"

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Abstract

Energy Conservation Study for a building of
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The project aims to study the ability of conserving the energy consumed in a building of "Children Happiness Center" that belongs to Hebron Municipality by counting the energy consuming devices, monitoring there operation at the entire conditions, calculate the efficiency of them; try to install modifications to reduce the energy loss and study economical benefits for these modifications.

يهدف المشروع الى دراسة امكانية توفير الطاقة المصروفة في مبنى مركز اسعاد الطفولة التابع لبلدية الخليل وذلك من خلال عمل احصاء شامل لجميع الاجهزة المستهلكة للطاقة بانواعها ومراقبة عملها عند الظروف المحلية وحساب كفائتها ومحاولة ادخال تعديلات تعمل على تقليل نسبة الضياع في الطاقة ودراسة جدوى اقتصادية لهذه التعديلات.

Table of Contents

CHAPTER ONE: INTRODUCTION

CHAPTER TWO: SURVEY AND BUILDING DESCRIPTION

CHAPTER THREE: METHODOLEGY OF ENERGY AUDITING

CHAPTER FOUR: ENERGY ANALYSIS FOR LIGHTING &BOILARS

CHAPTER FIVE: ENERGY ANAALYSIS FOR THE AIR-COONDITIONING SYSTEMS HVAC

CHAPTER SIX: ENERGY ANAALYSIS FOR HUDROLIC MACHINES

CHAPTER SEVEN: CONCLUSION

List of tables

List of figures

Chapter 1

Introduction

1.1 Overview:

Energy is the common approach for the experts and scientists in all over the world because it's the basic operating agent for any service producing help & for any comportment of humans.

It's the underlying "currency" of our lives so it is necessary for everything that human beings can do. Also, it can be critical to our economy and our environment so it should be used wisely. Yet for most of us, our only experience with the vast network that supplies our energy is flipping the switch that turns on the lights (or other appliances).

Energy literate citizens have knowledge and skills to make informed choices on using energy so that they could understand local, state, national, and international energy issues. Energy literate citizens are essential for the economic and environmental future. Energy education is fundamental to ensure the future generations and to literate energy consumers and decision makers. As a result it important to learn energy in our schools, universities and colleges.

Energy is considered as the working agent in buildings. It means that electricity, fuel, gas should be used smartly in order to operate any power producing machine, so we must keep the consumption of these sources to avoid energy run out. This initiate our attention to calculate the amount of energy consumed to produce output power which is familiar to the user , and calculate the actual value of energy consumes effectively .

In other words an Energy Audit is an examination of energy consuming such as equipment and systems to ensure that energy is being used efficiently. For example, in financial accounting, Building manager examines the energy account of energy consuming such as equipments and systems. He checks the way energy is used in its various components .He also checks for areas of inefficiency or that less energy can be used and identifies the means for improvement.

1.2 Objectives of Energy Audit Process:

Energy audit is a top-down initiative. It's important for:

a) Commitment on energy conservation and environmental protection.
b) Anticipation on the energy savings achievable.
c) Improving to corporate image by promoting energy efficiency and conservation.

It is important that the building management should be provided with the right perception of the benefits of the energy audit.

Energy consumption audit is a great interest to study since we have rapidly increasing in the cost of petroleum nowadays. This makes a macroeconomic problem for the governments through the world.

We perceive that common buildings pay huge amount of energy (electricity, fuel, gas ...etc) this forms burden on the organization which have the responsibility of these buildings.

Hebron municipality have many public utilities (as one of the most important establishments in Hebron city), which forms a charge of energy cost on its Balance Sheet, so we are enthusiastic to try saving energy in the building under consideration.

We intend to conduct the energy conservation process in "Children happiness ". This building has a theatre, auditoriums, and other conference facilities and rooms for the children activities, it is located behind "Al-Hussein Ibn Ali Boys School". Many activities are held in this place so it is interactive with the community.

In order to execute the energy audit in a building, we must test the consumption of the equipment and systems which are used and differentiate between the input / output power in each system.

We ought to get data and measurements that describe the systems behavior and check energy losses in the equipment which include: motors, boilers, (HVAC Heating and Ventilation Air-conditioning unit). This helps us to precede our modifications.

1.3 The Auditing procedure includes

1.3.1 Scope of Energy Audit

The target is to calculate the total power that is consumed by the building and find the opportunities for savings in order to avoid high payments.

The process needs calculation the input/output power for each system in the building so that we could calculate the efficiency i.e. calculating the heating and cooling loads that desired and the required input power to accomplish them and so for other systems.

To accomplish the analysis, walls insulation, intensity of lighting, plumping system and fans distribution must be checked.

1.3.2 Energy Audit Team

In order to make harmony on this project, energy audit team consists of two students from the following branches in the mechanical engineering department:

- Mechatronics Engineering.
- Refrigeration and Air-conditioning Engineering.

1.3.3 Estimation of Time frame for the First Semester

Table 1.1: Time frame for the first semester

Week / activity	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Connection with the municipality																
Selection the desired building																
Visiting the building to make the survey process																
Calculation the amount of electrical consumption																
Printing the final report																
Documentation																

1.3.4 Estimation of Time frame for the Second Semester

Table 1.2: Time frame for the second semester

Week / activity	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Thermal Loads Calculations																
Conducting Site Inspection and Measurements																
Analysis of Collected Data																
Comparison between theoretical & actual data																
Finding the opportunities to save energy																
Strategies of energy conservation																
Results and conclusion																
Printing the final report																
Documentation																

1.3.5 The Budget:

Table 1.3: The Budget

Activity	Cost ($)
Telecommunications	80
Transportation	100
Printing	60
counter	20
Total	**260**

1.3.6 Collection of Building Information

1.3.6.1 Conduction of Site Inspection and Measurements

- Strategic points for measurements.
- Instrumentation.

1.3.6.2 Analysis the Collected Data

- Identification of energy management opportunities.
- Costing.
- Normalization of data.
- Maintain thermal and lighting comfort.
- Scheduling maintenance and refurbishment works.

Chapter 2

Survey and Building Description

2.1 Introduction

"Children Happiness Center" is considered one of the cultural, educational and entertaining places for children, families and for who visits this center. It had been built by Hebron Municipality to raise the cultural, technical and educational level of Hebron's children.

Hebron Municipality builds many centers in Hebron. These buildings present local services for the biggest sector of the community such as training courses in summer camps.

2.2 General Description for the Building

"Children Happiness Center" belongs to Hebron Municipality, which responsible for these buildings. Moreover, it is the side that pays for the electrical usage supported to these buildings.

"Children happiness" is located at Ein Sarah Street near Al Hussein School, which is considered the main street in the city. The building view is illustrated in figures 2.1 and 2.2. The basement floor, ground floor, first floor each has an area 845.4, 1189.6, 344.3 m^2 respectively. All details are illustrated on layouts A1, A2, A3 in appendix B.

The building is constructed at 6000 m^2 land area and it consists of:

1- A main theater for 400 seated persons with an area of 208.3 m^2.
2- Two multi-purpose halls with an area of 243, 413 m^2
3- Children library which serves children from (6-18) years old with area of 102 m^2.
4- Computer center with area of 102 m^2.
5- Outside amphitheater of 300 seated persons with an area of 120 m^2.

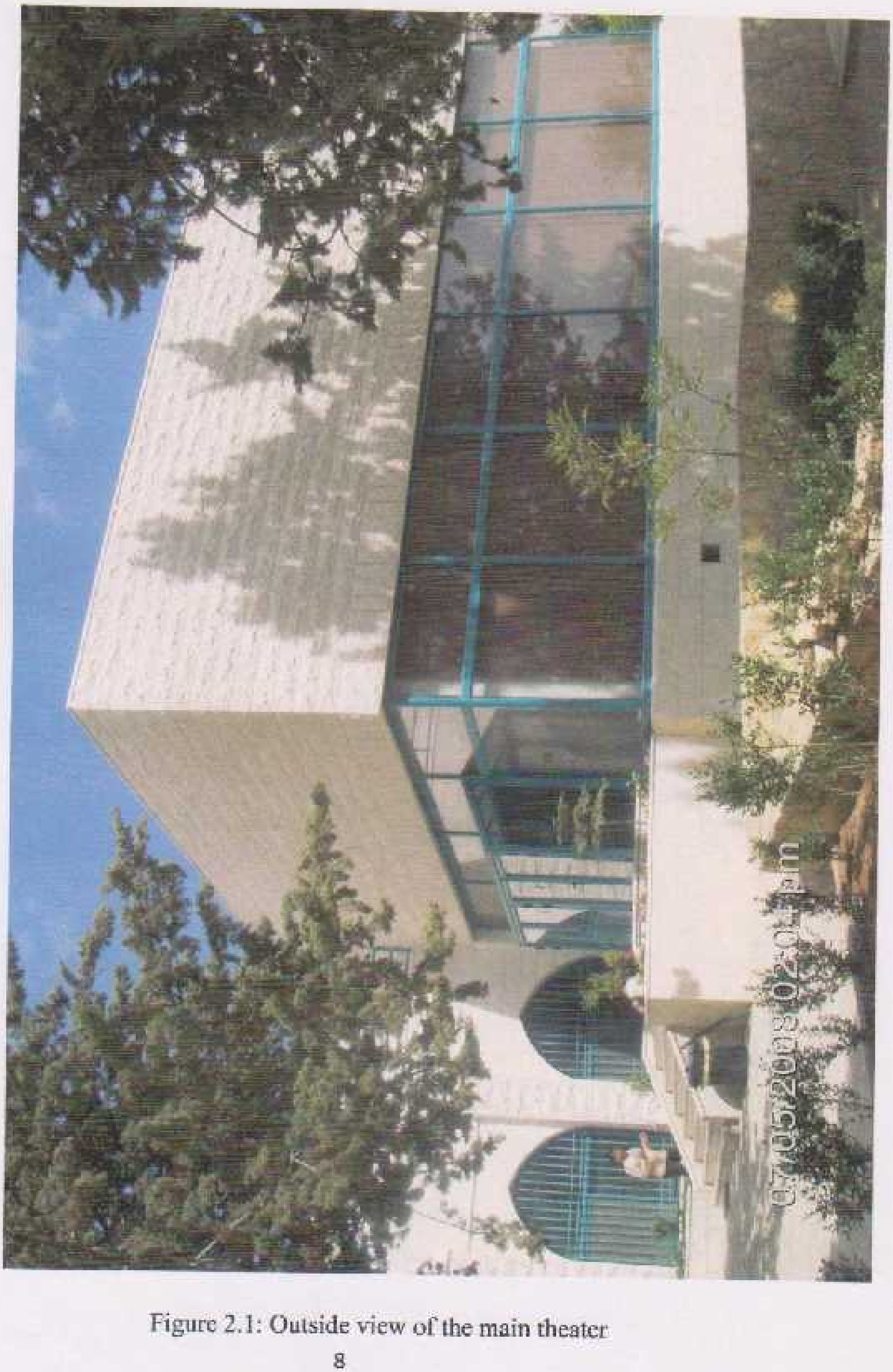


Figure 2.1: Outside view of the main theater

Figure 2.2: Outside view of the building

2.3 Buildings' Equipment Description

This building consists of many electrical systems that are considered the main consuming parts in the building such as, Packaged units, ducted split units, local boilers, lighting systems, plumping and fire suppression systems. Also, there are other electrical devices such as computers, televisions, data show, printers and scanners.

In this section, each system will be explained separately.

2.3.1 Packaged air conditioning unit (PPH 450 Petra Packaged Hermatic)

Figure 2.3: Package unit

Two packaging air conditioning units are shown in figure 2.3, they are used to provide the common human comfort to the spectators who set inside the theater and two halls in the building; every PPH completed units is single packaged unit which are factory assembled, tested and shipped fully charged with compressors, evaporators, condenser coils, fans and controls.

These packages are designed for outdoor installation, and they can be used for cooling only or cooling and heat pump application (optional).

PPH units are ideal for residential, commercial and industrial applications and they are available from (38-625) nominal MPH (Million British Thermal Unit) at 50 Hz and from (42-700) MPH at 60 Hz.

PPH units are compact in design, supplied as a complete package and ready for operation with no extra controls or additional items for installation. The PPH unit has a single power entry with simple connection. All units are designed to ensure maximum compliance with European standard.

Quick start-up is assured once completed installation; it is the same as each PPH unit is manufactured in an ISO 9001 listed facility to guarantee quality. General data are described in table 2.1.In addition; all units are tested at the factory to provide reliable start-up.

2.3.1.1 Operation

2.3.1.1.1 Cooling Mode

The package operates at a refrigeration cycle in this mode. The air passes on the evaporator coils, its temperature decreases to the set point temperature before reaching to the mixing box in order to be mixed with the return air in a specific ratio.

When the controller is powered on, it starts its' self – diagnostic check after time delay (EFM Evaporator Fan Motor) is energized.

The controller reads the return air temperature and compares it to the set point. If the return air temperature is greater than ["cooling (summer) temperature set point" + "cooling proportion band"/No of compressors], the first compressor will start after time delay (programmed in the controller). Then the second starts after the time delay between two different compressors (Programmed in the controller), and so on for third, forth compressor.

The controller continues operation the unit until it reaches the set point temperature.

During the unloading stages, the controller will start switching OFF while the compressors keep (EFM Evaporator Fan Motor) running, the (CFM Condenser fan motor) will be ON if any compressor is enabled.

2.3.1.1.2 Heating Mode

In this mode, the package operates at a heat pump cycle i.e. converting the flow direction of the refrigerant by reverse valve in order to transfer amount of heat to the air that passes on the condensing coil. The heating process is completed using support heaters to bring the air to the set point temperature in winter.

The controller reads the return air temperature and compares it to the set point if the return air temperature is less than ["heating (winter) temperature set point" - "heating proportion band"/No of heaters]. As a result, the controller will operate the electric heaters steps just running (EFM), and they will continue operating electric heaters until the air reaches "heating set point".

During the unloading stage, the controller will start switching OFF electric heaters and keeping (EFM) running.

All areas that supported by the PPH units are illustrated on layouts A4, A5 in appendix B.

Table 2.1: General Data Description

Model PPH 450	Quantity	P (kW)
Compressor motor	4	7.460
CFM (condenser fan motor)	2	0.75
Total for package		**43.84**
Total for both packages = 2*43.84		**87.68**

2.3.2 Ducted Split Unit (DSP)

Figure 2.4: Ducted Split Unit (DSP)-(outdoor unit)

Figure 2.4 shows that DSP units are designed specifically for easy maintenance and installation for medium and large size air conditioning systems. Also they are designed for commercial and residential applications such as homes, stores, shops, villas, restaurants, commercial complex, offices ...etc.

These ducted split units are used for cooling or heating (heat pump optional).They are also used in a part of the basement area that is illustrated on the layout where two similar units are used there.

The principle of operation is dependent on the refrigeration cycle, it consists of two main parts: Outdoor unit which is illustrated in table 2.2 that contains the compressor, condenser, expansion (throttling device) and Indoor unit which is shown in figure 2.5 that contains the evaporator which makes the heat transfer between the refrigerant and the required air (i.e. cooling the air and superheating the refrigerant). Table 2.2 illustrates the general data for the outdoor unit. The area that supported by the DSP units are illustrated on layout A4 in appendix B.

2.3.2.1 Outdoor Unit (Condensing Unit)

Table 2.2: General Data Description

Type	Quantity	Power (kW)
CFM (Condenser Fan Motor)	1	0.75
Compressor	2	6.118
Total		**12.986**

2.3.2.2 Indoor Unit (Air Handling Unit AHU)

It consists of evaporator coils that takes the responsibility for making the heat transfer between the inside air in the space and the refrigerant flow in the pipe of the coil. Air is moved using a single fan. It has heaters to support the heating process in winter when the cycle operates in the reverse mode (heating mode).

Figure 2.5 shows the indoor unit and table 2.3 illustrates the general data for the indoor unit.

Figure 2.5 Indoor unit

Table 2.3 General data for indoor unit.

Electrical elements	No. of elements	Power for single element (kW)	Total power for elements (kW)
Heater	3	0.450	1.350
Fan	1	0.750	0.750
Total Power			**2.100**

2.3.3 Exhaust Fans (EF)

These fans are used to exhaust the inside air which is an agent for the ventilation process and air changes that are required for the human comfort. Each fan power is illustrated in table 2.4.

Table 2.4: General Data Description for exhaust fans

Floors	No. of fans	Power of single fan (kW)	Total power in each floor (kW)
Basement	4	0.05	0.2
Ground	3	0.05	0.15
Roof	2	0.35	0.9
		0.55	
Total power of fans.			1.25

2.3.4 Local Boilers

As mentioned later, the domestic hot water is supplied to the fixtures by local boilers; two of them are located in the underground floor in the bathrooms. The third one in the first floor locates in kitchen. Table 2.5 illustrates the power for the boilers.

Table 2.5: Power for boilers

No. of boilers	Power of each boiler (kW)	Total power = No. * $P_{single\ boiler}$ (kW)
3	2.5	7.5
1	3	3
Total power of boilers		10.500

2.3.5 Plumping and Fire Suppression Systems

Two parallel pumps are used to supply the required amount of water from suction tank to the roof tanks (6 tanks, each one is 2 m^3), which supply the needed water for the building fixtures. Both pumps alternate in operation.

Another two parallel pumps are used for supplying fire suppression system inside the building and they are standing to support this process. Moreover, these both pumps are activated by smoke sensors that send an electronic signal to operate pumps which will pump water to the sprinklers in order to suppress the fire. These four pumps have the same specification and required power. Table 2.6 illustrates the specifications for the pumps which are used in plumping and fire suppression. See appendix B layout A7, A9.

Table 2.6 data table for water pumps and fire suppression pumps.

Power	5.5 (hp) 4.103 (kW)
Voltage (V)	230-400
Current (A)	9.4-16.3
Frequency (Hz)	50
Flow rate (m^3/hr)	6-20
Minimum head (m)	40
Maximum head (m)	42-54
Speed (rpm)	2880

2.3.6 Lighting system

The building has a number of activities running during the day; these activities need a good lighting intensity and distribution that must be suitable for the area of the floors.

Many types of lighting systems are used to achieve the comfortable lighting for the human; these types are distributed according to a specific usage in each area. Each floor has special board to supply the lights.

The lights in the theatre are controllable type i.e. one can vary the lighting intensity by control switch in the control room. Other lights are uncontrollable but can be switched ON/OFF from different distribution boards. And there are other different lights that are used to make lighting for the outside areas. Tables 2.7, 2.8, 2.9, 2.10 illustrate the amount of lights and power in each floor. Lighting distribution is illustrated on layouts A6, A8, A10 in appendix B.

Table 2.7: Lights in the basement floor

Type of light	**No. $_{of\ lights}$= No. $_{Total}$ *No. $_{of\ lamps\ in\ each\ unit}$**	**P $_{of\ single\ Light.}$ (W)**	**P $_{Total}$= No $_{of\ lights}$*P $_{of\ single\ light}$ (kW)**
Florescent lighting fixture	28*4	18	2.016
Recessed mounted lighting fixture	9*2	36	0.648
Surface mounted lighting fixture	15*1	36	0.540
Spot light recessed mounted lighting fixture	18*1	60	1.080
Exit lighting	6*2	8	0.096
Emergency lighting	8*2	8	0.128
Ceiling mounted lighting fixture	10*1	100	1.000
Wall mounted lighting fixture for mirror lighting	6*1	18	0.108
Total power in the floor			**5.616**

Table 2.8: Lights in the ground floor.

Type of light	No. $_{of\ lights}$= No. $_{Total}$ *No. $_{of\ lamps\ in\ each\ unit}$	P $_{of\ single\ Light.}$ (W)	P $_{Total}$ = No $_{of\ lights}$ *P $_{of\ single\ light}$ (kW)
Lighting fixture type par 80	6*1	120	0.720
Spot light recessed mounted lighting fixture	17*1	60	1.020
Down lighting fixture recessed mounted with reflector	70*1	100	7.000
Ceiling mounted lighting fixture	12*1	100	1.200
Exit lighting	5*2	8	0.080
Florescent lighting fixture	12*4	18	0.864
Surface mounted lighting fixture	8*1	36	0.288
Recessed mounted lighting fixture	35*2	36	2.520
Wall mounted lighting fixture for mirror lighting	6*1	18	0.108
Total power in the floor			**13.8**

Table 2.9: Lights in the first floor.

Type of light	No. $_{of\ lights}$= No. $_{Total}$ *No. $_{of\ lamps\ in\ each\ unit}$	P $_{of\ single\ Light.}$ (W)	P $_{Total}$ = No $_{of\ lights}$ *P $_{of\ single\ light}$ (kW)
Recessed mounted lighting fixture	33*2	36	2.376
Total power in the floor			**2.376**

Table 2.10: Lights of the outside area.

Type of light	No. of lights	P for each one (W)	P Total for each type (kW)
Florescent Lamps	61	26	1.586
Medium type	6	150	0.9
High intensity	7	250	1.750
Total for all types			**4.236**

2.3.7 Other electrical devices (accessories)

Table 2.11 illustrates a lot of electrical devices in the building, such as computers in computer center and offices, scanners, printers, televisions and data show. All these devices are taken in consideration in calculation electrical load in building.

Table 2.11: Accessories (other electrical devices)

Application	No.	P (for each element) (W)			P Total for the entire elements (kW)
Computers	30	1060			31.8
Scanners	2	150			0.3
Printers	5	150			0.75
Televisions	3	130	130	75	0.335
Data show	2	650			1.3
Video conference	1	110			0.11
Water cooler	2	400			0.8
Local water heater	3	1800			5.4
Video	1	18			0.018
Small fans	10	54			0.54
Projector	1	540			0.54
Total Power					41.893

2.4 Electrical Consumption

Energy consumption of the building is referred to the electrical energy only, since there are no devices that depend on fuel-oil or other sources of energy. For 2007 the electrical consumption has increased by 31.8% relative to 2006, this increase in energy consumed is due to the human activities (increasing the activity hours) in the building.

Table 2.13 and figure 2.6 show the monthly electrical consumption over the first year (2006) and table 2.14, figure 2.7 show the monthly electrical consumption over the second year (2007).

2.4.1 Statistics for the electrical consumption data

Table 2.12 shows the total electrical power (Energy index) in the building:

Table 2.12: Energy Index.

Type	P_{Total} (kW)
HVAC	104.016
Lighting	26.028
Pumps	16.412
Fans	4.400
Boilers	10.500
Accessories	41.893
Total	**202**

Table 2.13: Data for the electrical usage in 2006

Month	Power (kW.h)	Cost (NSI)
January +February	1800	1170
March + April	2040	1326
May +June	9480	6162
July + August	12180	7917
September + October	6780	4407
November + December	6780	4407
Total	**39060**	**25389**

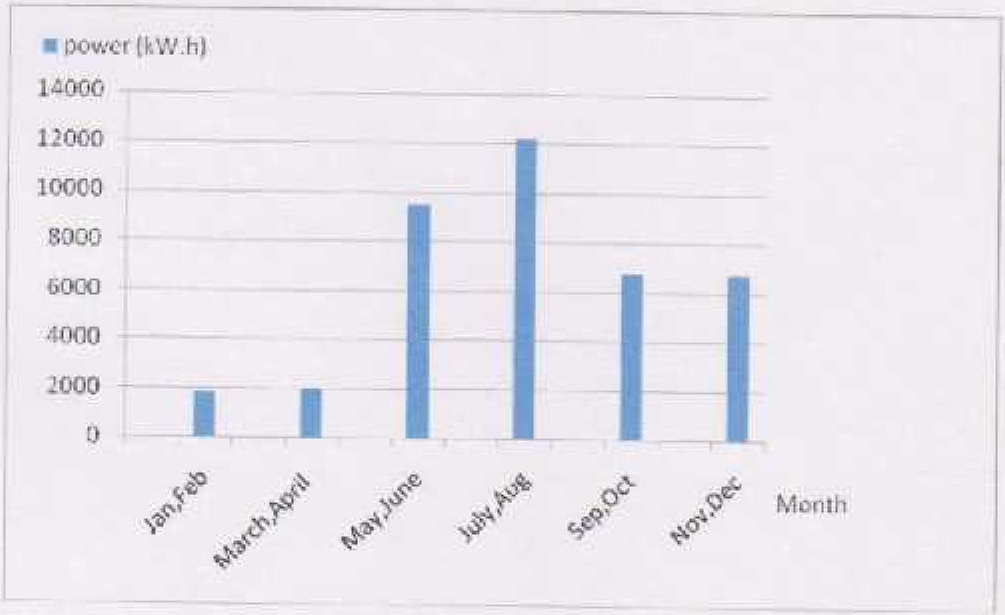


Figure 2.6: Energy consumption in 2006

Table 2.14: Data for the electrical usage in 2007

Month	Power (kW.h)	Cost (NSI)
January +February	10440	6786
March + April	8040	5226
May +June	9540	6201
July + August	15240	9906
September + October	8880	5772
November + December	5100	3315
Total	**57240**	**37206**

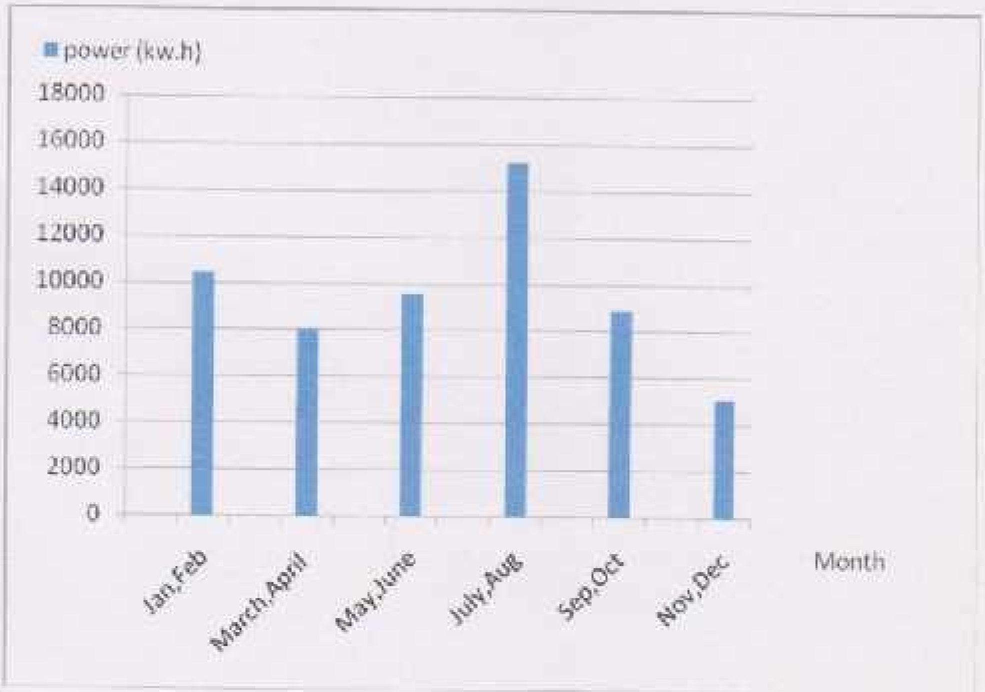


Figure 2.7: Energy consumption in 2007

Figures 2.6, 2.7 illustrate that the peak value of the energy usage in the building occurs at summer season, since the number of activities is more than other seasons because of the large number of operation hours for the electric devices. In winter the energy usage is the smallest value due to the number of activities.

The energy consumption curve in a building relates to the operating hours, the maximum number of operation occurs at the period where the peak value of activities can be obtained.

Chapter 3

Methodology of energy Auditing

3.1 Methodology of Energy Study

3.1.1 Monitoring of Electrical devices

Monitoring equipment can be useful to measure the actual operating parameters of various energy equipments and compare them with the design parameters to determine if energy efficiency can be improved.

The most common Parameters that are often monitored during an energy assessment are:

Basic electrical parameters in AC & DC systems: voltage (V), current (I), power factor, active power (kW), maximum demand (kVA), reactive power (kVAr), energy consumption (kWh), frequency (Hz), harmonics.

This module provides information for various monitoring equipment that are often used during Energy assessments in industry:

1. Electrical measuring instruments
2. Thermometers , Hygro Thermometer
3. Lux meters

For each type of monitoring equipment the following information is given:

- What the monitoring equipment does.
- Where the monitoring equipment is used.
- How to operate the monitoring equipment.
- Precautions and safety measures necessary for the monitoring equipment.

3.1.2 Electrical Measuring Instruments

3.1.2.1 What electrical measuring instruments do

Electrical measuring instruments include clamp-on or power analyzers and are used to measure main electrical parameters such as KVA, kW, PF, Hertz, KVAr, Amps and Volts. Some of these instruments also measure harmonics. Instant measurements can be taken with hand- held meters, while more advanced ones facilitates cumulative readings with print outs at specified intervals.

There are several models available in the market from different companies. One such instrument is the MAGNELAB- Clamp-on Power Hitester.

Electrical specifications

- Output 0.333V at rated current.
- Accuracy ±0.01.
- Phase angle < 2degrees (valid for 150A or higher).
- Rated accuracy at 10% to 130% of rated current.

3.1.2.2 Where electrical measuring instruments are used

These instruments are applied on-line to measure various electrical parameters of motors, transformers, and electrical heaters. There are no needs to stop the equipment while taking the measurements.

3.1.2.3 How to operate electrical measuring instruments

The instrument has three leads (wires), which are connected to the crocodile clips at the end. The three leads are yellow, black and red. Figures (3-1, 2) give illustrate the measurement method for various conditions. However, operating procedures may vary for different types of clamp-on or power analyzers. For the correct operation procedure the operator should always check the instruction manual supplied with the instrument.

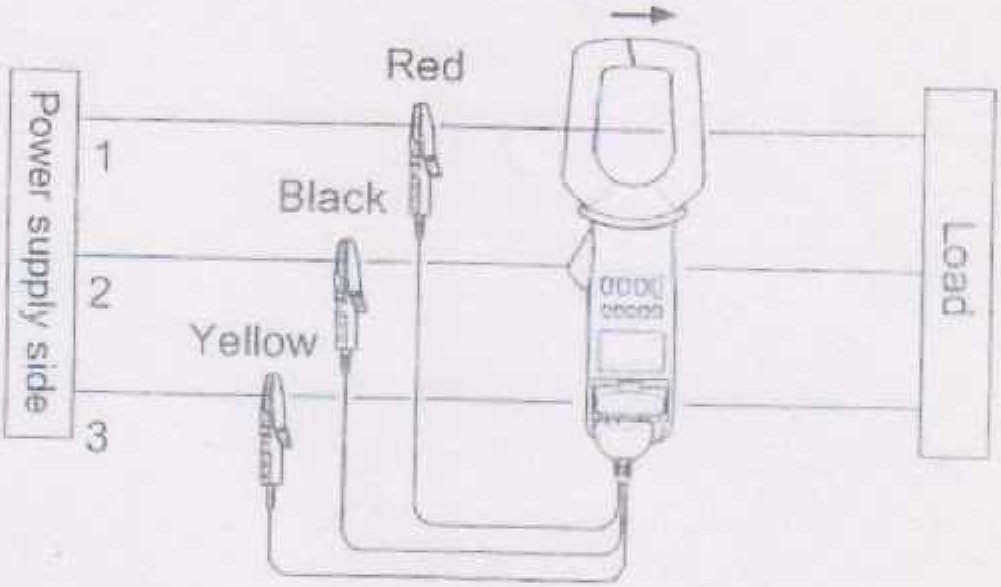


Figure (3-1) switching on clamp-meter for 3 phase

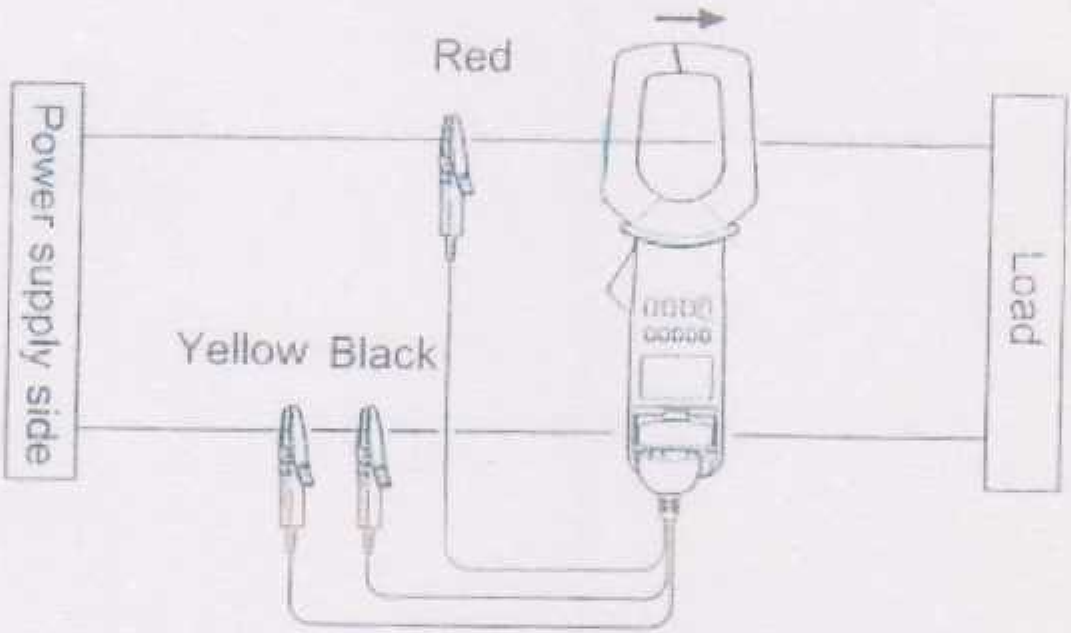


Figure (3-2) switching on clamp-meter for single phase

3.1.2.4 Precautions and safety measures

Some precautions and safety measures to be taken while using clamp-on and power analyzers are:

- To avoid short circuits and potentially life-threatening hazards, never attach the clamp to a circuit that operates at more than the maximum rated voltage, or over bare conductors.
- The clamp-on probe should be connected to the secondary side of a breaker, so the breaker can prevent an accident if a short circuit occurs.
- While using the instrument, use rubber hand gloves, boots, and a safety helmet, to avoid electrical shocks, and do not use the instrument when hands are wet.
- Check the operating manual of the monitoring equipment for more detailed instructions on safety and precautions before using the equipment.

3.1.3 Thermometer

3.1.3.1 What a thermometer does

Thermometers are instruments used to measure the temperature of fluids, surfaces or gases, for example of the flue gases after combustion has taken place. Thermometers are classified as Contact thermometers or non-contact or infrared thermometers and are described below.

3.1.3.1.1 Contact thermometer

There are many types of contact thermometers. A simple clinical thermometer is the best known example of a contact thermometer. However, for the purpose of energy audits in an industrial plant we generally use thermocouples for measuring temperatures with a high accuracy. It consists of two dissimilar metals, joined together at one end. The thermocouple metal alloys are commonly available as wire. A thermocouple is available in different combinations of metals or calibrations. The four most common calibrations are J, K, T and E. There are high temperature calibrations R, S, C and GB. Each calibration has a different temperature range and environment, although the maximum temperature varies with the diameter of the wire used in the thermocouple. Although the thermocouple calibration dictates the temperature range, the maximum range is also limited by the diameter of the thermocouple wire.

3.1.3.1.2 Non-contact or infrared thermometer

A non-contact or infrared thermometer allows the measurement of temperatures without physical contact between the thermometer and the object of which the temperature is determined. The thermometer is directed at the surface and immediately gives a temperature reading. This instrument is useful for measuring hot spots in furnaces, surface temperatures etc. Infrared thermometer allows users to measure temperature in applications where conventional sensors cannot be used or cannot produce accurate temperature readings.

3.1.3.2 Where the thermometer is used

In energy audits, the temperature is one of the most important parameters to be measured in order to determine the thermal energy loss or to make a thermal energy balance. Temperature measurements are taken for the audit of air conditioning units, boilers, furnaces, steam systems, waste heat recovery systems, heat exchangers, etc. During the audits, the temperature can be measured of the:

- Ambient air
- Chilled water in refrigeration plants
- Inlet air into the Air Handling unit of AC plant
- Cooling water inlet and out let at the Cooling Tower
- Surfaces of steam pipelines, boilers, kilns
- Input water to the boiler
- Exhaust gases

- Condensate returned
- Pre heated air supply for combustion
- Temperature of the fuel oil

3.1.3.3 How to operate a thermometer

The thermocouple (contact thermometer) consists of two dissimilar metals, joined together at one end. When the junction of the two metals is heated or cooled a voltage is produced that can be correlated back to the temperature. A probe is inserted into a liquid or gaseous stream to measure the temperature of, for example, flue gas, hot air, or water. A leaf type probe is used to measure surface temperatures. In most of the cases the thermocouple directly gives the reading in the desired units (Centigrade or Fahrenheit) on a digital panel. The operation of a non-contact or infrared thermometer is simple. The infrared thermometer (gun) is pointed towards the surface where the temperature must be measured. The measurement result is read directly from the panel.

3.1.3.4 Precautions and safety measures

The following precautions and safety measures apply when using a thermometer:

- The probe must be immersed in the fluid and the measurement must be taken after 1-2 minutes, i.e. after the stabilization of the readings.
- Before using the thermocouple, the temperature range for which the thermocouple is designed for should be checked.
- The probe of the thermocouple should never touch the bare flame.
- Before using a non-contact thermometer the emissivity should be set in accordance with the surface where the temperature is to be measured.
- Check the operating manual of the monitoring equipment for more detailed instructions on safety and precautions before using the equipment.

3.1.4 Door opening counter

Infiltration is one of the most common cooling load combinations since the infiltrated air enters to the conditioned spaces by the pressure difference between the inside and outside air, the amount of air is to be calculated by measuring the crackage length doors and windows.

But there is an amount of air leakages infiltrated by opening the doors of the spaces along the time of operation of the HVAC unit.

Special time behavior have been designed to count the number of door openings in a given time period. The principle of the counting process is based on a digital signal comes from a limit switch, send the signal to the 555 timer then comparing it with the time consumed after opens the door , the signal then transfer to the BCD counter then to the decoder to display the result number on (7- segment) . If the activation of the limit switch continues more than four seconds the 555 timer send a pulse in every time period (4 seconds), since the door is still open and the air infiltrates to or from the conditioned space. Each display count range is between

(0 - 9), and the total counting capacity of the counter is up to 999. The components of the circuit are illustrated in the table (3-1) , Figure (3.3) shows door opening circuit .

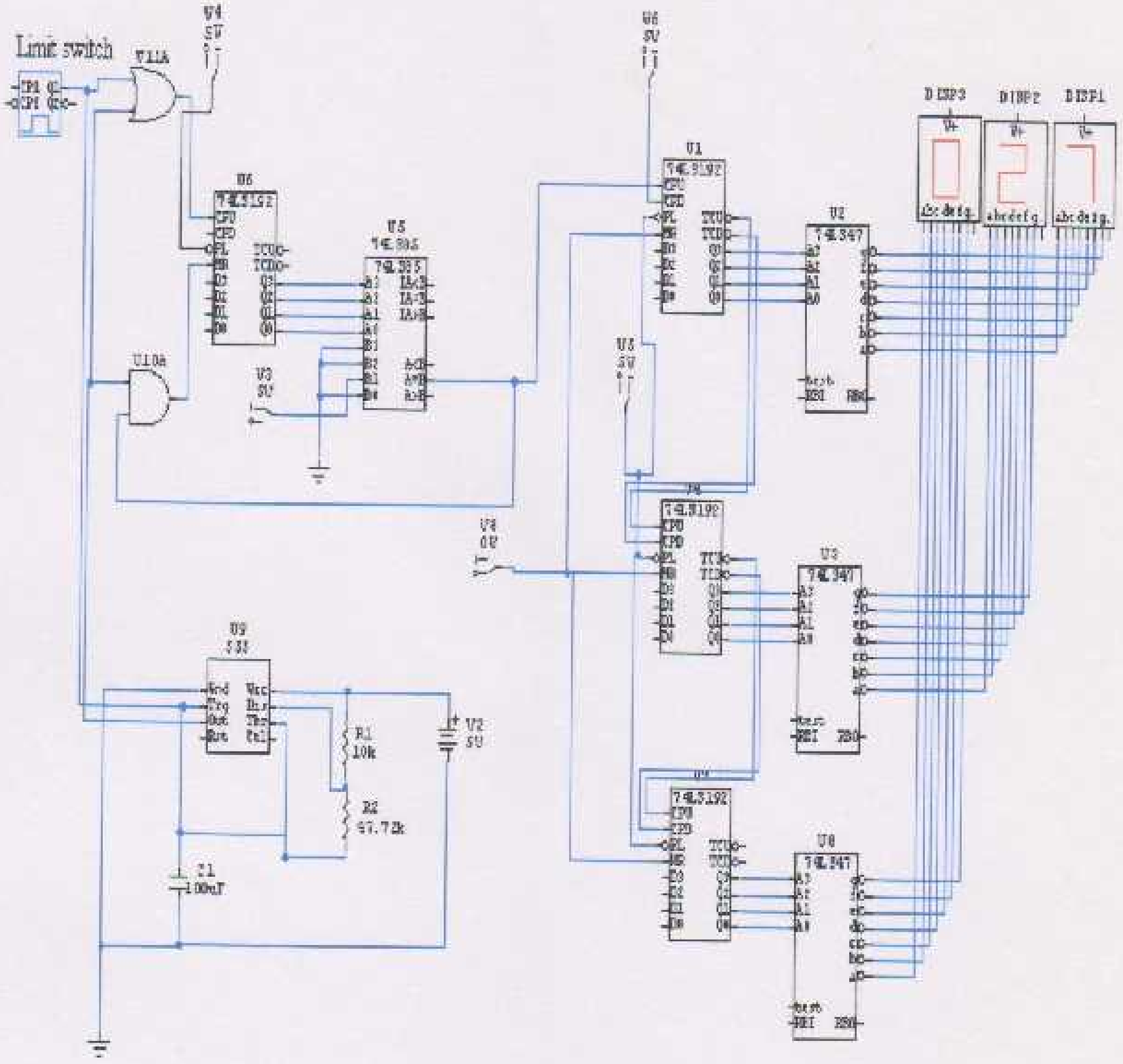
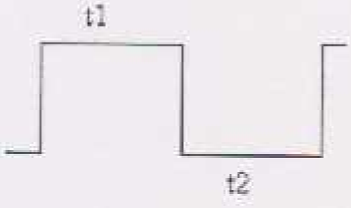


Figure (3.3) Door opening counter

Table (3.1): parts description of counter

Part name	Description
74LS192	BCD counter: used to count the number of pulse comes from Limit switch or from the comparator
74LS85	Comparator :used to compare the output of the timer and counter with 4 second
74LS47	Decoder : is an interfacing parts used to connecting counter with 7-segment
74LS08	AND gate : output is 1 if the two inputs is 1 other wise output is 0
74LS32	OR-Gate : output is 1 if there is at least one input is 1 ,other wise output is 0
7-Segment	Display : used to display the number of door opening
555 Timers	Timer : used to controlling the time which door opened, If the door opened larger than 4 sec the timer send a high pulse to increase counter by one.
Resistance	R_1, R_2 and C_1 is a function of t_1(charging time of capacitor) and t_2 (discharging time of capacitor)
Capacitor	

The 555 timers consist of resistance (R_1, R_2), capacitor C_1 and supply voltage 5v, to estimate the value of these unknown parameter follow these steps:



1- Assume that R_1=10 kΩ ,C_1=100μF, t_1=4sec

2- Calculate R_2 by equation $t1=0.693(R_1+R_2)C_1$ charging, output HIGH.

4sec=0.693(10000+R_2)100*10^-6

This yield $R_2 = 47.72k\Omega$

$t_2 = 0.693R_2C_1$ discharging, output LOW

$t_2 = 3.31sec$

3.1.5 Lux meter

3.1.5.1 What lux meters do

Lux meters are used to measure illumination (light) levels. Most lux meters consist of a body, a sensor with a photo cell, and a display panel as shown in figure (3-4). The sensor is placed under the light source. The light that falls on the photo cell has energy, which is transferred by the photo cell into electric current. The more light is absorbed by the cell, the higher the generated current. The meter reads the electrical current and calculates the appropriate value of either Lux or Foot candles. This value is shown on the display panel.

A key thing to remember about light is that it is usually made up of many different types (colors) of light at different wavelengths. The reading, therefore, is a result of the combined effects of all the wavelengths. A standard color can be referred to as color temperature and is expressed in degrees Kelvin. The standard color temperature for calibration of most light meters is 2856 degrees Kelvin, which is more yellow than pure white. Different types of light bulbs burn at different color temperatures. Lux meter readings will, therefore, vary with different light sources of the same intensity. This is why some lights seem "harsher" or “softer” than others.

Figure (3-4): Lux meter

3.1.5.2 Where lux meters are used

Lux meters are used to measure illumination levels in offices, factories etc.

3.1.5.3 How to operate a lux meters

This instrument is very simple to operate. The sensor is to be placed at the work station or at the place where intensity of the light is to be measured, and the instrument will directly give the reading on the display panel.

3.1.5.4 Precautions and safety measures

The following measures should be taken when working with lux meters:

- The sensor is to be properly placed on the work station to obtain an accurate reading
- Due to the high sensitivity of sensor it should be stored safely
- Check the operating manual of the monitoring equipment for more detailed instructions on safety and precautions before using the equipments.

3.1.6 Pumps and Fans

$$\zeta = \frac{PQ}{IV} \tag{3-1}$$

Where

P: pressure supported by the pump.(Pa)

Q: Flow rate ($\frac{m^3}{s}$).

I: current inter the pump (A).

V: voltage across the terminals of the pump (V).

3.1.7 Air conditioning Units

3.1.7.1 Package unit

The efficiency of a thermal device or a thermo electrical device such as the package unit is termed by COP (Coefficient of Performance) it gives the ratio of the output thermal load relative to the input electrical load.

$$COP_{PPH\ cooling} = \frac{Q_L}{P_{compressors} + P_{EFM} + P_{CFM}} \tag{3-2}$$

$$COP_{PPH\ Heating} = \frac{Q_H}{P_{compressors} + P_{EFM} + P_{CFM} + P_{Heaters}} \tag{3-3}$$

3.1.8 Split unit

$$COP_{DSP\ cooling} = \frac{Q_L}{P_{compressors} + P_{EFM} + P_{CFM}} \quad (3\text{-}4)$$

$$COP_{DSP\ Heating} = \frac{Q_H}{P_{compressors} + P_{EFM} + P_{CFM} + P_{Heaters}} \quad (3\text{-}5)$$

In order to calculate the COP coefficient of performance cooling, heating loads must be estimated to determine the required output thermal power in heating and cooling mode.

3.1.9 Boiler

The domestic hot water is generally estimated by assuming a constant daily consumption of hot water. The following equation is used:

$$q_W = M_w C_p \ (T_h - T_c) \quad (3\text{-}6)$$

Where:

Q_{HW} = Amount of heat for the domestic hot water. (kJ)

M_w: is the daily consumption of domestic hot water (Kg).

T_h: the hot water supply temperature.

T_c: is the temperature of cold water that must be heated to T_h.

A widely used value for the daily consumption of domestic hot water is 280 liter per family. This consumption rate can be reduced to 200 liter per day per family if a clothes washer is not used. However, if the number of the family members is known a daily consumption

Chapter 4

Energy Analysis for Lighting and Boilers

4.1 lighting system

4.1.1 Background

From the dawn of civilization until recent times, human beings created light solely from fire, though it is more a source of heat than light. We are still using the same principle in the 21st century to produce light and heat through incandescent lamps. Only in the past few decades have lighting products become much more sophisticated and varied. Estimates indicate that energy consumption by lighting is about 20 - 45% of a commercial building's total energy consumption and about 3 - 10% in an industrial plant's total energy consumption. Most industrial and commercial energy users are aware of energy savings in lighting systems. Often significant energy savings can be realized with a minimal investment of capital and common sense. Replacing mercury vapor or incandescent sources with metal halide or high pressure sodium will generally result in reduced energy costs and increased visibility. Installing and maintaining photo-controls, time clocks, and energy management systems can also achieve extraordinary savings. However, in some cases it may be necessary to consider modifications of the lighting design in order to achieve the desired energy savings. It is important to understand that efficient lamps alone would not ensure efficient lighting systems.

4.1.2Basic Theory of Light

Light is just one portion of the various electromagnetic waves flying through space. These waves have both a frequency and a length, the values of which distinguish light from other forms of energy on the electromagnetic spectrum.

4.1.3 The Importance of Lighting

Lighting consumes a tremendous amount of energy and financial resources. Lighting accounts for approximately 17 percent of all electricity sold in the United States. Energy star estimates that if efficient lighting were used in all locations where it has been shown to be profitable throughout the country, the nation's demand for electricity would be cut by more than 10 percent. This could save nearly $17 billion in ratepayer bills and result in the following annual pollution reductions:

• 202 million metric tons of carbon dioxide, the primary cause of global climate change. This would be the equivalent of taking 15 million cars off the road.

• More than 1.3 million metric tons of sulfur dioxide, which contributes to acid rain.

• 600,000 metric tons of nitrogen oxides, which contribute to smog.

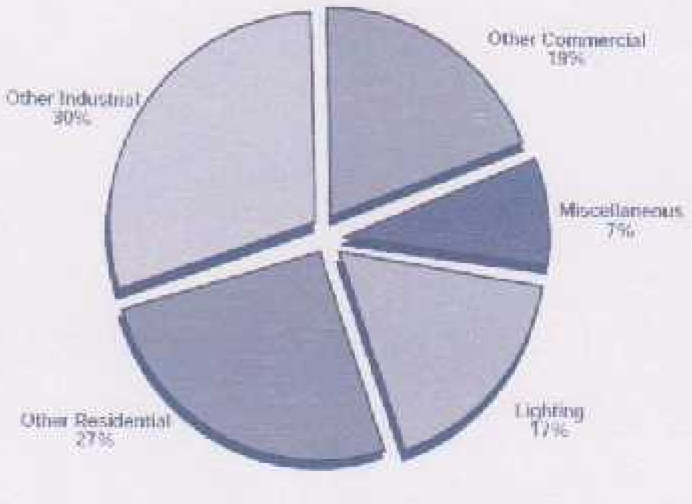


Figure (4-1): Lighting Share of All Electric Energy Use

Lighting is also a significant expense in operating buildings. Lighting is the largest cost component of a commercial building's electricity bill (see Figure (4-1)) and a significant portion of its total energy bill.

4.1.4 Lighting and the Building

4.1.4.1 Reduce Heat Gain

In addition to visible light, all lighting systems produce heat. Lighting is typically the largest source of waste heat, often called "heat gain," inside commercial buildings. Improving lighting efficiency reduces heat gain, which affects your buildings in two ways. Waste heat is a useful supplement when the building requires heat, it must be removed by the HVAC system when the building needs to be cooled. The impact of this tradeoff the penalty for increased heating costs versus the bonus for reduced cooling costs depends

on your building type, its geographic location, and its HVAC system. Although heating costs may rise, they will rarely exceed the resultant cooling savings, even in buildings in northern climates that use electric resistance heat.
By reducing internal heat gain, efficient lighting also reduces your building's cooling requirements. Consequently, your existing cooling system may be able to serve future added loads, or may be appropriate for "rightsizing". Given the large impact lighting upgrades can have on your HVAC system requirements and the high cost of cooling equipment, you should always quantify HVAC and lighting interactions. There are simplified methods available for calculating the impacts of lighting upgrades on heating and cooling systems.

Lighting also affects the power quality of your building's electrical distribution system. Poor power quality is a concern because it wastes energy, reduces electrical capacity, and can harm equipment and the electrical distribution system itself. Upgrading to lighting equipment with clean power quality (high power factor and low harmonic distortion) can improve the power quality in your building's electrical system. Furthermore, upgrading with higher efficiency and higher power factor lighting equipment can also free up valuable electrical capacity. This benefit alone may justify the cost of a lighting upgrade.

The relationship of lighting to task performance and visibility is well understood. Improved lighting enhances visual comfort, reduces eye fatigue, and improves performance on visual tasks. Well-designed lighting is likely to improve performance, increase productivity, and reduce absenteeism. Because costs associated with your employees greatly outweigh the other building costs, any lighting changes that improve your occupants' workspaces are worth investigating.

Lighting also contributes to the safety of occupants and the security of buildings. Emergency lighting must be available during power outages, and minimum levels of light must be available at night when most lighting is turned off. In addition, safety codes require exit signs to highlight escape routes during fires or other emergencies. Outside lighting and indoor night lighting deters crime by exposing intruders' movements and permitting occupants to move safely through the building or to cars. Although such effects are difficult to quantify, comfort, mood, productivity, health, safety, and other impacts on people should be considered as part of every lighting upgrade.

Light is emitted from the body due to any of the following phenomena:

1- **Incandescence** Solids and liquids emit visible radiation when they are heated to temperatures about 1000K. The intensity increases and the appearance becomes whiter as the temperature increases.

2- **Electric Discharge**: When an electric current is passed through a gas the atoms and molecules emit radiation whose spectrum is characteristic of the elements present.

3- **Electro luminescence**: Light is generated when electric current is passed through certain solids such as semiconductor or phosphor materials.

4- **Photoluminescence:** Radiation at one wavelength is absorbed, usually by a solid, and re-emitted at a different wavelength. When the re-emitted radiation is visible the phenomenon may be termed either *fluorescence* or *phosphorescence*.

4.1.5 Definitions and Commonly Used Terms

Luminaire: A luminaire is a complete lighting unit, consisting of a lamp or lamps together with the parts designed to distribute the light, position and protect the lamps, and connect the lamps to the power supply.

Lumen: Unit of luminous flux; the flux emitted within a unit solid angle by a point source with a uniform luminous intensity of one candela. One lux is one lumen per square meter. The lumen (lm) is the photometric equivalent of the watt, weighted to match the eye response of the "standard observer". 1 watt = 683 lumens at 555 nm wavelength.

Lux: This is the metric unit of measure for illuminance of a surface. Average maintained illuminance is the average of lux levels measured at various points in a defined area. One lux is equal to one lumen per square meter. The difference between the lux and the lumen is that the lux takes into account the area over which the luminous flux is spread. 1000 lumens, concentrated into an area of one square meter, lights up that square meter with an illuminance of 1000 lux. The same 1000 lumens, spread out over ten square meters, produce a dimmer illuminance of only 100 lux.

Rated luminous efficacy: The ratio of rated lumen output of the lamp and the rated power consumption expressed in lumens per watt.

Room Index: This is a ratio, which relates the plan dimensions of the whole room to the height between the working plane and the plane of the fittings.

Utilization factor (UF): This is the proportion of the luminous flux emitted by the lamps, reaching the working plane. It is a measure of the effectiveness of the lighting scheme.

4.2 Methodology of Lighting System Energy Efficiency Study

A step by step approach to assessment of improvement options in lighting at any facility would involve the following likely steps.

Step 1: Inventory the lighting system elements & transformers in the facility as per following typical format.

Step 2: With the aid of a lux meter, measure and document the lux levels at various plant locations at working level, as daytime lux and night time lux values alongside the number of lamps "ON" during measurement.

Step 3: With the aid of portable load analyzer, measure and document the voltage and power consumption at various input points, namely the distribution boards or the lighting voltage transformers at the same as that of the lighting level audit.

Step 4: Compare the measured lux values with the standard. Use the values as a reference and identify locations of under lit and over lit areas.

Step 5: Analyze the failure rates of lamps, ballasts and the actual life expectancy levels from the past data.

4.3 Designing of Lighting System

4.3.1 How much light is needed

Every task requires some lighting level on the surface of the body. Good lighting is essential to perform visual tasks. Better lighting permits people to work with more productivity. Typical book reading can be done with 100 to 200 lux.

These recommended values have since made their way into national and international standards for lighting design see Table (4.1).

Table 4.1 Illuminance Levels for different Areas of Activity

Building/Space Type	Guideline Illuminance Range (footcandles)
Commercial interiors	
Art galleries	30-100
Banks	50-150
Hotels (rooms and lobbies)	10-50
Offices	30-100
-Average reading and writing	50-75
-Hallways	10-20
-Rooms with computers	20-50
Restaurants (dining areas)	20-50
Stores (general)	20-50
Merchandise	100-200
Institutional interiors	
Auditoriums/assembly places	15-30
Hospitals (general areas)	10-15
Labs/treatment areas	50-100
Libraries	30-100
Schools	30-150
Industrial interiors	
Ordinary tasks	50
Stockroom storage	30
Loading and unloading	20
Difficult tasks	100
Highly difficult tasks	200
Very difficult tasks	300-500
Most difficult tasks	500-1000
Exterior	
Building security	1-5
Floodlighting (low/high brightness or surroundings)	5-30
Parking	1-5

4.3.2 Lighting design for interiors

The step by step process of lighting design is illustrated below with the help of an example. Figure (4.2) shows the parameters of a typical space.

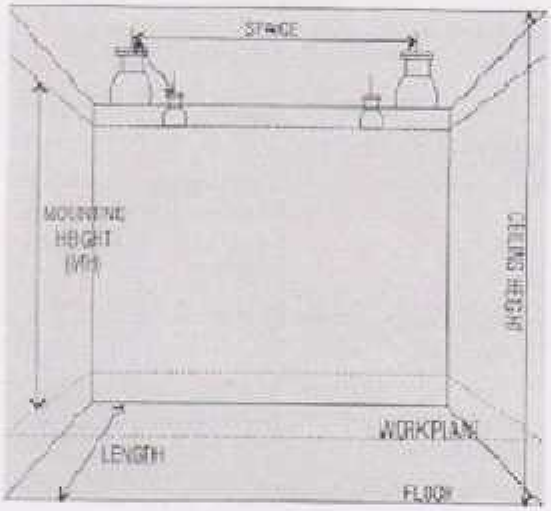


Figure (4.2) Room with dimensions

Step 1: Decide the required Illuminance on work plane, the type of lamp and luminaries.

A preliminary assessment must be made of the type of lighting required, a decision most often made as a function of both aesthetics and economics. For normal office work, illuminance of 200 lux is desired.

For an air-conditioned office space under consideration, we choose 36 W fluorescent tube lights with twin tube fittings. The luminaries are porcelain-enameled, suitable for the above lamp. It is necessary to procure utilization factor tables for these luminaries from the manufacturer for further calculations.

Step 2: Collect the room data in the format given in table (4.2).

Region	Length (m)	Width (m)	High (m)	Area (m^2)	Room Index
Library	13.95	7.30	3.15	101.835	1.52

Table (4.2) Data for library room

Step 3: Calculate room index.

$$\text{Room Index} = \frac{\text{Length} \times \text{Width}}{\text{Hight} \times (\text{Length} + \text{Width})} \tag{4-1}$$

Step 4: Calculating the Utilization factor.

Utilization factor is defined as the percent of rated bare-lamp lumens that exit the luminaries and reach the work plane. It accounts for light directly from the luminaries as well as light reflected off the room surfaces. Manufacturers will supply each luminaries with its own CU table derived from a photometric test report. Using tables available from manufacturers, it is possible to determine the utilization factor for different light fittings if the reflectance of both the walls and ceiling is known, the room index has been determined and the type of luminaries is known.

Step 5: Calculate the number of fittings required by applying the following formula.

$$N = \frac{E \times A}{F \times UF \times LLF} \tag{4-2}$$

Where:

N = Number of fittings

E = Lux level required on working plane

A = Area of room (L x W)

F = Total flux (Lumens) from all the lamps in one fitting

UF = Utilization factor from the table for the fitting to be used

LLF = Light loss factor. This takes account of the depreciation over time of lamp output and dirt accumulation on the fitting and walls of the building.

4.4 Sample of calculations

Library in the ground floor in the old sector

$$\text{Room index} = \frac{13.95\times7.3}{3.15\times(13.95+7.3)} = 1.52$$

From table (C-1) in appendix (C), with general reflectance's for walls, ceiling and surface of the floor then:

UF (Utilization Factor) = 0.8236 .... (Using interpolation from table C-3)

LLF = 0.8 from table (C-2) in appendix (C).

Flux = 5000 (Lumen)

After that, number of fittings can be calculated by:

$$N = \frac{322.92\times101.835}{5000\times0.8236\times0.8} = 9.9819$$

Table (4.3) Difference between the actual and measured values.

Calculated Number of Fittings	Actual Number of Fittings	Recommended flux (fc)	Measured flux (fc)
9.982	10	30	34.36

The difference between the actual and recommended number of fittings forms the amount of energy that can be achieved in the space, the previous note confirmed by instrumentation using lux meter as illustrated in table (4.3).

4.5 Maximizing Efficiency and Quality

A comprehensive lighting upgrade achieves your qualitative lighting objectives while maximizing efficiency and profitability. With rewards beyond the sum of its parts, this process integrates equipment replacement with deliberate design, operation, maintenance and disposal practices. This whole-system approach takes what is frequently regarded as a complex system of individual decisions and unites them into a strategic approach that ensures that each opportunity is addressed and balanced with other objectives (see Figure (4.3)).

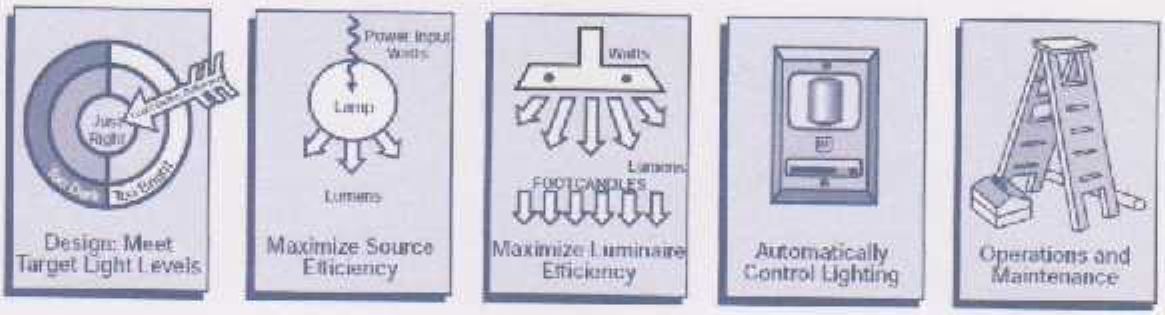


Figure (4.3) Different opportunity

4.5.1 Automatically Control Lighting

Reducing the connected load (wattage) of the lighting system represents only half of the potential for maximizing energy savings. The other half is minimizing the use of that load through automatic controls. Automatic controls switch or dim lighting based on time, occupancy, lighting-level strategies, or a combination of all three. In situations where lighting may be on longer than needed, left on in unoccupied areas, or used when sufficient daylight exists, you should consider installing automatic controls as a supplement or replacement for manual controls.

4.5.2 Time-Based Controls

The most basic controlling strategies involve time-based controls, best suited for spaces where lighting needs are predictable and predetermined. Time-based controls can be used in both indoor and outdoor situations. Common outdoor applications include automatically switching parking lot or security lighting based on the sunset and sunrise times. Typical indoor situations include switching lighting in production, manufacturing, and retail facilities that operate on fixed, predefined operating schedules. Time-based control systems for indoor lighting typically include a manual override option for situations when lighting is needed beyond the scheduled period. Simple equipment, such as mechanical and electronic time clocks and electromechanical and electronic photocells, can be independent or part of a larger centralized energy-management system.

4.5.3 Occupancy-Based Controls

Occupancy-based strategies are best suited to spaces that have highly variable and unpredictable occupancy patterns. Occupancy or motion sensors are used to detect occupant motion, lighting the space only when it is occupied. For both initial and sustained success in using occupancy sensors, the sensor must be able to see the range of motion in the entire space while avoiding either on or off false triggering. This requires proper product selection, positioning, and testing. Occupancy sensors should first be selected based on the range of body motion expected to occur throughout the entire lighted space. Controls for hallways, for example, need only be sensitive to a person walking down a narrow area, while sensors for offices need to detect smaller upper body motion, such as typing or reaching for a telephone. Once sensitivity and coverage area is established, sensors are selected from two predominant technology types. Passive infrared sensors detect the motion of heat between vertical and horizontal fan pattern detection zones. This technology requires a direct line of sight and is more sensitive to lateral motion, but it requires larger motion as distance from the sensor increases. The coverage pattern and field of view can also be precisely controlled. It typically finds its best application in smaller spaces with a direct line of sight, warehouses, and aisles as shown in Figure (4.4).

Figure (4.4): Infrared Sensor Coverage Patterns

Ultrasonic sensors detect movement by sensing disturbances in high-frequency ultrasonic patterns. Because this technology emits ultrasonic waves that are reflected around the room surfaces, it does not require a direct line of sight, is more sensitive to motion toward and away from the sensor, and its sensitivity decreases relative to its distance from the sensor (see Figure (4.5)). It also does not have a definable coverage pattern or field of view. These characteristics make it suitable for use in larger enclosed areas that may have cabinets, shelving, partitions, or other obstructions. If necessary, these technologies can also be combined into one product to improve detection and reduce the likelihood of false on or off triggering. To achieve cost-effective, user-friendly occupancy sensor installations, both types of technologies need to be carefully commissioned at installation to make sure that their position, time delay, and sensitivity are properly adjusted for the space and tasks. To ensure proper performance, the position of both wall- and ceiling-mounted sensors needs to be evaluated carefully. Ultrasonic sensors, for example will respond to strong air movement and need to be located away from ventilation diffusers. Infrared sensors should have their line of sight checked to ensure that it is not blocked by room furnishings. Both types of technologies should be positioned and adjusted so that their coverage area is not allowed to stray outside of the intended control area. All sensors have an adjustable time delay to prevent the lights from switching off when the space is occupied but there is little activity. Some infrared and all ultrasonic sensors also have an adjustable sensitivity setting. Customizing these settings to the application is necessary to balance energy savings with occupant satisfaction.

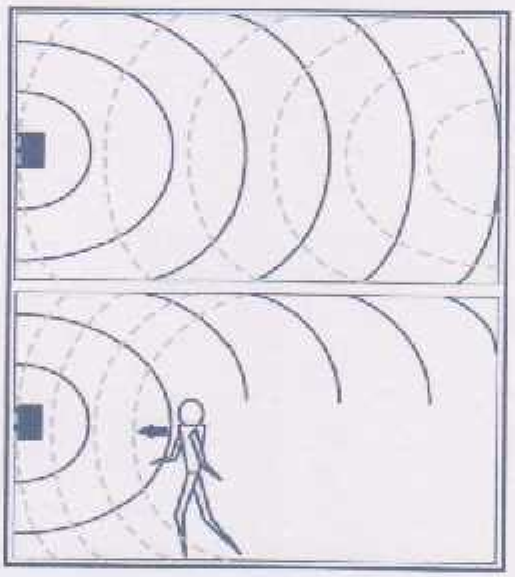


Figure (4.5): Ultrasonic Sensor Coverage Patterns

Although increasing time delays will reduce the possibility of the lighting being switched off while the space is occupied, it will also reduce the energy savings. Setting the sensitivity too high may turn the lighting on when the room is unoccupied, wasting energy. Similarly, setting the sensitivity too low will create occupant complaints, as the lighting may turn off when the room is occupied. Evaluating the potential savings from an occupancy sensor installation should, and can, go beyond guesswork or speculation. Although sensors primarily affect energy use, they also affect energy demand, load on HVAC system, and lamp life. Evaluating the economic feasibility of an installation is best done by monitoring lighting and occupancy patterns. The use of inexpensive loggers will indicate the total amount of time the lights are on when the space is vacant, the time of day the savings take place, and the frequency of lamp cycling.

4.5.4 Lighting Level-Based Controls

Lighting level-based strategies take advantage of any available daylight and supply only the necessary amount of electric light to provide target lighting levels. In addition to saving energy, lighting level controls can minimize over lighting and glare and help reduce electricity demand charges. The two main strategies for controlling perimeter fixtures in daylighted areas are daylight switching or daylight dimming.

Daylight switching involves switching fixtures off when the target lighting levels can be achieved by utilizing daylight. To avoid frequent cycling of the lamps and to minimize distraction to occupants, a time delay, provided by a dead band, is necessary. Several levels of switching are commonly used to provide for flexibility and a smooth transition between natural and electric lighting.

Daylight dimming involves continuously varying the electric lighting level to maintain a constant target level of illumination. Dimming systems save energy by dimming fluorescent lights down to as low as 10 to 20 percent of full output, with the added benefit of maintaining consistent lighting levels. Because HID sources cannot be frequently switched on and off, they are instead dimmed for time, occupancy, and lighting level-based control strategies.

4.5.5 Calculating the amount of energy saving

When an occupancy sensor based on ultrasonic or infrared principle is used, the power consumed with non occupied space is saved since the lamp is turned off in this time interval, so the lighting use will be more controllable than the previous state i.e. without using this sensor.

Let a Library as an example for using occupancy sensor then the amount of energy saved can be calculated sing equation (4-3):

Energy saving per hour = total amount of lighting Power × Time light turns off per day

(4-3)

$$\text{Energy saving per hour} = 2\times36\text{w}\times10\times1\text{ hour} = 0.72\text{ kWh}$$

Since the hours of operation is 7 hours in the day, we can calculate the energy consumed per year. And the energy saved by the occupancy sensor can be calculate as the difference between 6 and 7 hours of operation since we have at least one hour of losing energy in the missing turning on of lights in a space.

Then, the total energy saving per year can calculated using equation (4-4):

$$\text{Energy saved per year} = P\,saved\ (\text{kW}) \times \frac{No.hours}{day} \times (26)\frac{day}{mounth} \times (12)\frac{mounth}{year} \qquad (4\text{-}4)$$

$$= 224.64\ (\text{kWh/year})$$

$$= \frac{224.64\times0.65}{3.9} = 37.44(\$/year)$$

This cost of energy saving is due to the Library region as a sample for the whole study.

This study will be adapted to the old part of the building to accomplish the amount of energy cost saving in a year.

$$\text{The Cost of Energy saving} = \frac{1617.4\times0.65}{3.9} = 269.6(\$/year)$$

$$\text{Cost-recovery time} = \frac{50\$}{37.44\$} = 1.33\text{ year} = 16\text{ months}$$

4.2 Boilers systems

4.2.1 Domestic hot water

The domestic hot water load is generally estimated by assuming a constant daily consumption of hot water.

$$Q_{HW} = M_{HW} C_P \left(T_h - T_c\right) \tag{4-5}$$

Where:

M_w: the daily consumption of domestic hot water.

C_P =Specific heat for hot water. (kJ/kg.k)

T_h : the hot water supply temperature.

T_c : the temperature of cold water that must be heated to T_h.

Domestic hot water is water available for hand washing, general kitchen use, and showering and small-scale laundry activities. There is potential for substantial money and energy savings even for small businesses if domestic hot water is produced and used efficiently.

Most water heaters consume a great proportion of their energy just to keep a supply of hot water ready and waiting in the tank. The heat gradually leaks out of the tank until the heater turns on again to heat the water back up. Reducing hot water consumption and replacing an old water heater with a new, more efficient model are simple but effective changes. You can further reduce your hot water use by buying appliances with low water usage, such as front-loading (horizontal-axis) washing machines, and by installing water-conserving plumbing fixtures.

Although it is not often feasible to eliminate water heating entirely, it is possible to substantially reduce the need for water heating without making large-scale changes to installations. It highlights simple opportunities to improve the energy efficiency of the hot water supply, consequently saving on running costs.

Each question in the energy audit checklist has a space where you can write your energy efficiency improvement ideas. You should refer to the explanatory notes when considering what can be done to improve energy efficiency. You may need to take additional notes and attach them to the checklist, or attach other relevant documentation (such as instruction sheets and site plans) in order to support the improvement ideas and completely document the audit process.

4.2.2 Design of a programmable circulation system for the domestic hot water based on PLC (Programmable Logic Controllers)

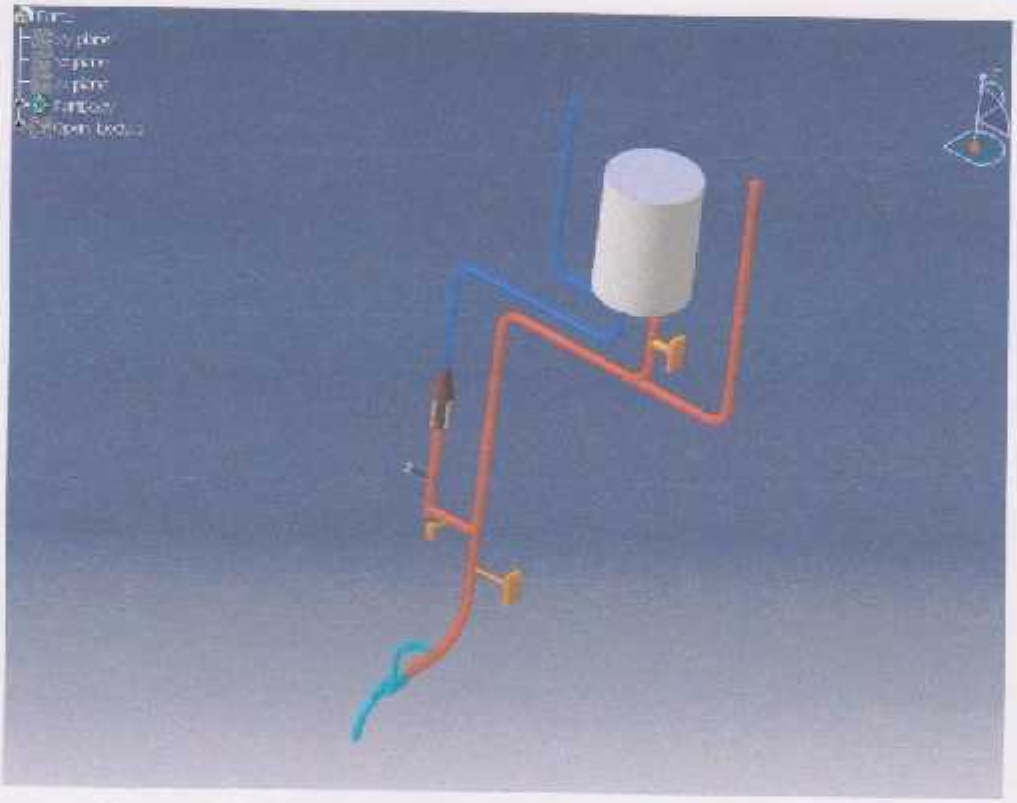


Figure (4-6): programmable circulation system for the hot water.

Heat and waste water conservation can be done by design a controlling unit which recirculates the unused hot water that escaped away from the boiler outlet as illustrated in figure (4-6). This process can be done by installing valves and small pump to move the water again to the boiler according to the specific head required , all these electrical devices controlled by a PLC program.

When the person use an amount of hot water, the remaining amount of water in the pipes has an amount of heat that will be transmitted through the pipe surface away from the control volume, this amount of waste heat can be pumped again to the boiler tank using small pump.

The procedure of the regulation process for the system can occurred by installing three solenoid values (Y1 ,Y2, Y3) at the fixture outlet , outlet of the boiler, suction pump line , respectively . Starting switch (S1) must be put at the fixture outlet and flow switch (S2) before the pump (at the suction line) to detect the water flow in the pipe. Y1 can open to give the required amount of water to the user, after S1 has closed, then Y1 and Y2 are closing together to forbid the flow, and Y3 is opining at this state to reflect the direction of flow to the pump suction line, this step initiate the flow switch to operate the pump directly to move the water to the boiler. The figure (4-7) below describes the process occurs.

In the state graph method, the starting state is always (F0), each state is activated by the previous state and the conditions of it. Then the same state is deactivated with the next state of operation. In the first state of the process (S1) is activated to get the hot waterfrom the boiler to the user, after (S1) deactivated (Y1 and Y2) are to shut off but latest amount of water stay in the pipe so (Y3) opens directly to convert the floc direction to the pump suction line, water activates the flow switch (S2) to turn the pump on. The pump stays in operation while (S1) activated again or deactivation of the flow switch (S2).

Each state can be explained by S-R Flip Flop as shown in Figure (4 - 8), S-R Flip Flop contains two pins for input (Set and Reset) and one output (Q) it is equal to 1 when the S=1 and equal to zero when R=1.

The output of the system described by OR gate, the activation of these output is controlled by the state itself as shown in figure (4 - 9).

Where:

Signals detection by:

S1: Normally open switch when S1 is activated, then S1 = 1.

S2: Normally open switch when S2 is activated, then S2 = 1.

Output:

Y1: Solenoid valve at the fixture outlet. When it is activated, then Y1=1

Y2: Solenoid valve at the boiler outlet. When it is activated, then Y2=1

Y3: Solenoid valve at the pump's suction line. When it is activated, then Y3=1

P: contactor to operate the pump. When it is activated, then P=1

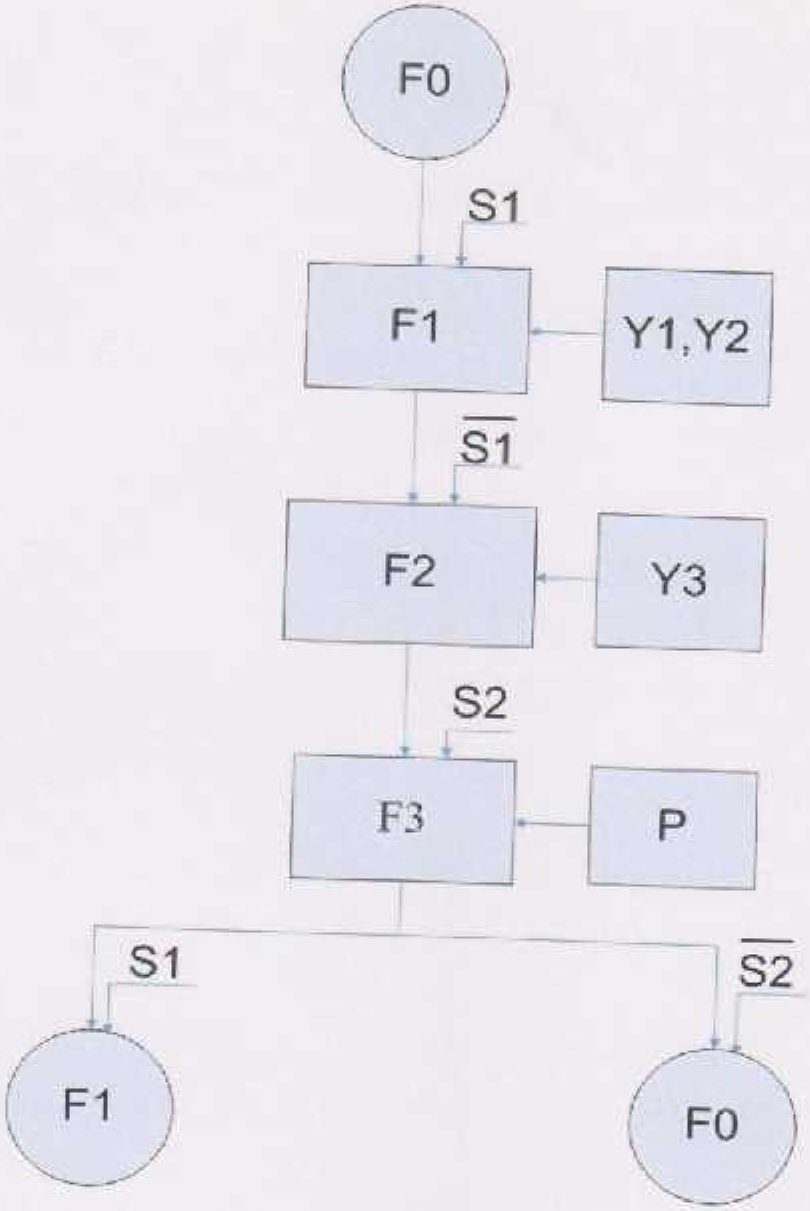


Figure (4-7) state graph representation

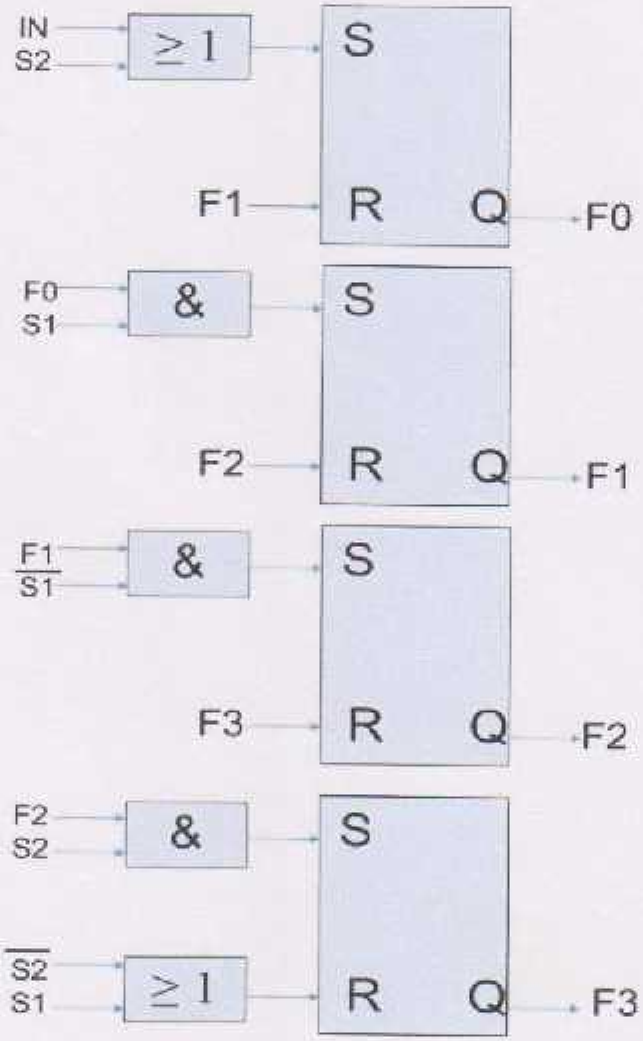


Figure (4 - 8): description of state using (S - R) Flip Flop.

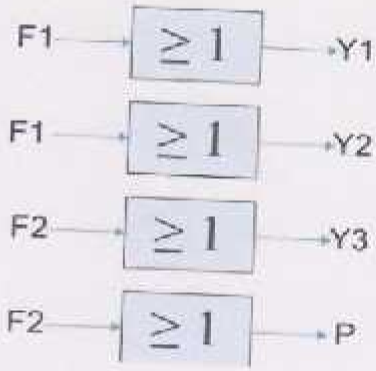


Figure (4 - 9) description of the output using OR gate.

The pump is turned off if the switch S1 is switch on again or the flow switch is turned off with the discontinuity of water flow, i.e. the system return to the initial state again.

Plumping problem rise up with this procedure contains the vacuum pressure will exerted by the pumping process, especially with the poor venting in the piping system. A venting line goes from the outlet boiler to the high level of the supplying tank as a solution of the plumping problem this will make balancing in the pressure inside the pipes.

4.2.3 Energy saving exerted by using the programmable circulation system

The pipe has an amount of heat that can be calculated using the following equation:

$$q = M\, C_p (T_h - T_c) \; ; \qquad (4\text{-}6)$$

But $M_{\text{water inside the pipe}} = \rho v_{\text{Cylinder}} L.$ (4-7)

Then equation (4-6) will at the form:

$$q = \rho\, v\, L\; C_p (T_h - T_c)$$

$$= \rho \frac{\pi}{4} D^2 L\; C_p (T_h - T_c)$$

Where:

q: amount of heat that is carried by the water in the pipe. (kJ)

M: mass of water exists inside the pipe. (kg)

T_h: water outlet temperature from the boiler. (C°)

T_c : water outlet temperature from the pipe exit. (C°)

ρ = Density of saturated water and its value is 990.6 (kg/m^3)

v = Volume of water inside the pipe. (m^3)

L= Pipe length. (m)

Then:

$$q = 990.6 \times \frac{\pi}{4}(0.622 \times 0.02547)^2 \times 6 \times 1.174 \times (43.3 - 7)$$

$$= 199.614 \text{ (kJ)}$$

This amount of heat can be saved by recirculate the water to the boiler but when this amount exists inside the pipe, heat is transmitting from the water by convection , through the pipe surface by conduction and finally by convection to the outside air.

This transmitted heat can be calculated using the following equation:

$$Q_{\text{Transmitted}} = \frac{\Delta T}{R_{Th}} \qquad (4\text{-}8)$$

ΔT = Temperature difference between the water and the outside air.(C°)

R_{Th} = Thermal resistance for the heat flow.(C°/W)

$$R_{Th} = \frac{1}{h_{in}A_{in}} + \frac{\ln(R_o/R_{in})}{2\pi kL} + \frac{1}{h_o A_o} \qquad (4\text{-}9)$$

Where:

h_o = convection heat transfer coefficient of the outside air, it is equal to 9.37 (w/m^2.C°)

A_o= outside area of the pipe. (m^2)

r_o= outside diameter of the pipe. (m)

r_{in}= inside diameter of the pipe. (m)

k = thermal conductivity of the pipe metal (steel). (W/m.C°)

L = length of the pipe. (m)

A_{in} = inside area of the pipe. (m^2)

h_{in}= convection heat transfer coefficient of the water inside the pipe. ($w/m^2.C°$)

Convection heat transfer coefficient (h_{in}) can be calculated using the following equations:

Flow rate has been measured using a basic method, by measure the time (sec) of passing a specific amount of water (L).

$$Q = 198.176\times10^{-6}(m^3/s)$$

$$v = \frac{4\times Q}{\pi\times D^2} \quad (4\text{-}10)$$

$$= \frac{4\times198.176\times10^{-6}}{\pi\times(0.622\times0.02547)^2} = 1.005(m/s)$$

$$Re = \frac{\rho\ v\ D}{\mu} \quad (4\text{-}10)$$

$$= \frac{990.6\times1.005\times0.622\times0.02547}{6.16\times10^{-4}} = 2.5603\times10^4 > 4000 \text{Turbulent Flow}$$

Pr = 4.04 from appendix D table (D-15) (Pr: Prantle No ; Nu : Nuslt No)

$$N_{ud} = (0.023)\ Re^{0.8}\ Pr^{0.4} \quad (4\text{-}11)$$

$$= (0.023)\ (2.5603\times10^4)^{0.8}\ (4.04)^{0.4} = 135.176$$

$$h_{in} = \frac{k_{water}}{D_{Pipe}}\times Nud \quad (4\text{-}12)$$

$$= \frac{0.637}{(0.622\times0.02547)}\times135.176 = 5435.25(W/m^2C°)$$

$$R_{Th} = \frac{1}{h_{in}(\pi D_{in}L)} + \frac{\ln(R_o/R_{in})}{2\pi kL} + \frac{1}{h_o(\pi D_o L)}$$

$R_{th} = 0.26554\ (C^{o}/W)$

$$Q_{Transmitted} = \frac{\Delta T}{R_{Th}} = \frac{(43.3 - 10)}{0.26554} = 125.404\ (W)$$

This amount of heat that transmitted through the pipe surface can be saved when the programmable circulation system is used.

Chapter 5

Energy Analysis for the Air conditioning systems (HVAC)

5.1 INTRODUCTION

The mechanical heating or cooling load in a building is dependent upon the various heat gains and losses experienced by the building including solar and internal heat gains and heat gains or losses due to transmission through the building envelope and infiltration (or ventilation) of outside air. The primary purpose of the heating, ventilating, and air-conditioning (HVAC) system in a building is to regulate the dry-bulb air temperature, humidity and air quality by adding or removing heat energy. Due to the nature of the energy forces which play upon the building and the various types of mechanical systems which can be used in non-residential buildings, there is very little relationship between the heating or cooling load and the energy consumed by the HVAC system. This chapter outlines the reasons why energy is consumed and wasted in HVAC systems for nonresidential buildings. These reasons fall into a variety of categories, including energy conversion technologies, system type selection, the use or misuse of outside air, and control strategies. Following a review of the appropriate concerns to be addressed in analyzing an existing HVAC system, the chapter discusses the aspects of human thermal comfort. Succeeding sections deal with HVAC system types, energy conservation opportunities and domestic hot water systems.

5.2 SURVEYING EXISTING CONDITIONS

As presented in Chapter 3, the first stage of any effective energy management program is an energy audit of the facility in question. In surveying the HVAC system(s) in a facility, the first step is to find out what you have to work with: what equipment and control systems exist. It is usually beneficial to divide the HVAC systems into two categories: equipment and systems which provide heating and cooling, and equipment and systems which provide ventilation. It is essential to fully document the type and status of all equipment from major components including boilers, chillers, cooling towers and air-handling units to the various control systems: thermostats, valves and gauges, whether automated or manual; in order to later determine what elements can be replaced or improved to realize a saving in energy consumed by the system. The second step is to determine how the system is operating. This requires that someone measure the operating parameters to determine whether the system actually operates as it was specified to operate. Determine the system efficiency under realistic conditions. This may be significantly different from the theoretical or full-load efficiency. Determine how the system is operated.

What are the hours of operation? Are changes in system controls manual or automatic? Find out how the system is actually operated, which may differ from how the system was designed to be operated. It is best to talk to operators and/or users of the system who know a lot more about how the system operates than the engineers or managers. If the system is no longer operating at design conditions, it is extremely useful to determine what factors are responsible for the change. Potential causes of operational changes are modifications in the building or system and lapses in maintenance. Have there been structural or architectural changes to the building without corresponding changes to the HVAC system? Have there been changes in building operations? Is the system still properly balanced? Has routine maintenance been performed? Has scheduled preventative maintenance been performed? Finally, it is useful to determine whether the system can or should be restored to its initial design conditions. If practical, it may be beneficial to carry out the needed maintenance *before* proceeding to analyze the system for further improvements. However, some older systems are so obviously inefficient that bringing them back to original design parameters is not worth the time or expense. Before continuing with an analysis of the system, it is also useful to determine future plans for the building and the HVAC system which can seriously affect the energy efficiency of system operation. Are there plans to remodel the building or parts of the building? How extensive are proposed changes? Are changes in building operations planned? Document everything. Only when you have a full record of what the system consists of, how it is operating and how it is operated, and what changes have been made and will be made in the future, can you properly evaluate the benefit of energy conservation techniques which may be applicable to a particular building system.

Buildings generally consist of a number of rooms which may have different energy and moisture gains or losses — the loads. Loads exhibit both seasonal and diurnal variation. Adjacent rooms with similar loads are usually lumped together into one *zone* which is controlled by one thermostat. Air handlers in an HVAC system can be designed to condition one zone (called single zone systems) or multiple zones (called multiple zone systems). Residences and small commercial buildings are usually designed and operated as single zone spaces.

Figure (5-1) indicates the total cooling terms exists in a space.

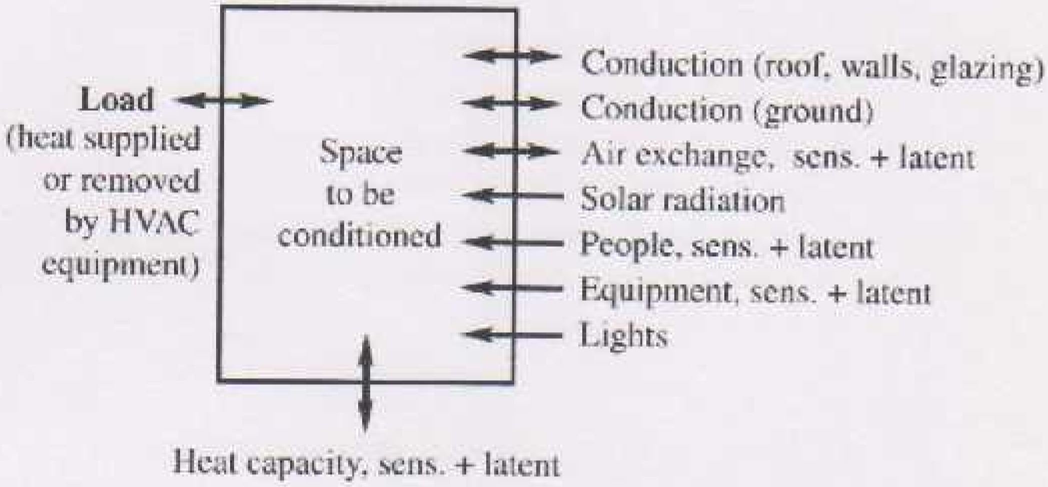


Figure (5.1): Principal terms for space to be conditioned

5.3 Heat Balance

Loads are the heat that must be supplied or removed by the HVAC equipment to maintain a space at the desired conditions. The calculations are like accounting. One considers all the heat that is generated in the space or that flows across the envelope; the total energy, including the thermal energy stored in the space, must be conserved according to the first law of thermodynamics. The principal terms are indicated in Figure (5.1). Outdoor air, occupants, and possibly certain kinds of equipment contribute both sensible and latent heat terms.

Load calculations are straightforward in the static limit, i.e., if all input is constant. That is usually an acceptable approximation for the calculation of peak heating loads. But for cooling loads, dynamic effects (i.e., heat storage) must be taken into account because some of the heat gains are absorbed by the mass of the building and do not contribute to the loads until several hours later. Dynamic effects are also important whenever the indoor temperature is allowed to float. Sometimes it is appropriate to distinguish several aspects of the load. If the indoor temperature is not constant, the instantaneous load of the space may differ from the rate at which heat is being supplied or removed by the HVAC equipment. The load for the heating or cooling plant is different from the space load if there are significant losses from the distribution system, or if part of the air is exhausted to the outside rather than being returned to the heating or cooling coil.

It is convenient to classify the terms of the static energy balance according to the following groups.

The sensible energy terms are:

- Conduction through the building envelope other than ground
- Conduction through the floor,
- Heat due to air exchange (infiltration and/or ventilation)

The latent heat gains are mainly due to air exchange, equipment (such as in the kitchen and bathroom), and occupants. The total load is the sum of the sensible and the latent loads.

During the heating season, the latent gain from air exchange is usually negative because the outdoor air is relatively dry. A negative Q_{lat} as indicated in equation (5-1) implies that the total heating load is greater than the sensible heating load alone but this is relevant only if there is humidification to maintain the specified humidity ratio Wi. For buildings without humidification, one has no control over Wi, and there is not much point in calculating the latent contribution to the heating load at a fictitious value of Wi.

$$\dot{Q}_{lat} = \dot{Q}_{lat,air} + \dot{Q}_{lat,occ} + \dot{Q}_{lat,equ} \qquad (5\text{-}1)$$

5.6 Cooling load

It is in summer and it is the rate at which heat must be removed from a space in order to maintain the desired conditions in the space, generally a dry-bulb temperature and relative humidity. It contains:

1- Heat gains that transmitted through building structures such as walls, floors and ceiling that are adjacent to unconditioned spaces .The heat transmitted is caused by temperature difference that exists on both sides of structures.

2- Heat gain due to solar effect which include:

a- Solar radiation transmitted through the glass and absorbed by inside surfaces and furniture.

b- Convection heat gain.

3- Sensible and latent heat gains brought into the space as a result of the ventilation process and infiltration of air through windows and doors.

4- Sensible heat produced in space by lights, appliances, motors and other miscellaneous heat gains.

5- Latent heat produced from hot baths, or any other moisture producing equipment.

6- Sensible and latent heat produced by occupants' .It is an important load since we have high occupancy.

7- Miscellaneous load, such as heat loss through the duct surface.

All parts combination of the cooling load are described in equation (5-2):

$$Q_{Cooling} = Q_{Tr.w} + Q_{Vent.} + Q_{Occupants} + Q_{Scr.} \qquad (5\text{-}2)$$

where:

$Q_{cooling}$: Total cooling load (kW).

$Q_{Tr.w}$ – Heat transmitted through walls, windows, doors.

$Q_{Vent.}$ = Heating load gained by ventilation.

$Q_{Occupants.}$ = Heating gain by persons.

$Q_{Ser.}$ = Heat gained by equipments.

5.6.1 Building Heat Transmission Coefficient

One of the most important terms in the heat balance of a building is the heat flow across the envelope. Heat flow can be assumed to be linear in the temperature difference when the range of temperatures is sufficiently small; this is usually a good approximation for heat flow across the envelope. Thus one can calculate the heat flow through each component of the building envelope as the product of its area *A*, its conductance *U*, and the difference $(Ti - To)$ between the interior and outdoor temperatures. The calculation of *U* (or its inverse, the R_{Th} value). Here we combine the results for the components to obtain the total heat flow.

5.6.1.1 Heat transmitted through walls & doors

This amount of heat can be calculated using the following equation:

$$\mathbf{Q_{Transmitted\ through\ walls,\ doors} = UA\ (CLTD)_{corr.}} \qquad (5\text{-}3)$$

Where:

U = Overall heat transfer coefficient. ($W/m^2.C$)

A = Surface area of heat transfer. (m^2)

$(CLTD)_{corr}$ = is called cooling load temperature difference (C) , it can be calculated using equation (5-4) .

$$\mathbf{(CLTD)\ corr = (CLTD + LM) \times K + (25.5 - Ti) + (29.4 - To,\ m)\ f} \qquad (5\text{-}4)$$

LM: Latitude correction factor for horizontal and vertical surfaces (9-2).

K: colors adjustment factor such that k=1.0 for dark colored roofs, & k=0.65 for Permanently Light colored walls.

DR: the daily temperature range which equal to the difference between the Average maximum and Average minimum temperature for warmest month of the summer season.

5.6.1.2 Heat transmitted through glass

To calculate the contribution to the cooling load, the daily maximum of the solar heat gain is multiplied by the cooling load factor. Thus the actual cooling load at time t due to solar radiation is given by the formula

$$Q_{\text{transmitted through glass}} = A \times SHG \times SC \times CLF \qquad (5\text{-}5)$$

SHG = Solar Heat Gain from table (D-20) in appendix D. (w/m^2)

SC = Shading Coefficient (D-21) in appendix D.

CLF = Cooling Load Factor.

5.6.1.3 Heat gain through glass by convection

$$Q_{\text{Convection}} = UA \times CLTD_{\text{corr}} \qquad (5\text{-}6)$$

CLTD: cooling load temperature difference. From appendix D, table (D-24)

5.6.1.4 Heat gain due to equipments: Sensible and latent heat

5.6.1.5 Heat gain due to Lights

$$Q_{\text{Lights}} = n \times (P_{LT} \times F_U\, F_b \times CLF)_{LT}. \qquad (5\text{-}7)$$

n = Number of lamps.

P_{LT} = The lamp rated power in watts.

F_U = The fraction of lumps that are in use

F_b = The ballast factor that equal to (1.2) for florescent lump and (1) for ordinary lumps

CLF_{LT} = Light cooling load factor, from table (D-27) in appendix D.

Diversity factor for selected Appl. from table (D-26) in appendix D.

5.6.1.6 Heat due to Occupants

$$Q = n \times Q_s \times CLF + (n \times Q_L) \times D_f \qquad (5\text{-}8)$$

n = number of persons.

Q_s = Sensible heat, from table (D-14) in appendix D.

Q_L = Latent heat, from table (D-14) in appendix D.

5.6.1.7 Heat gain due to ventilation

$$Q_{Ventilation,\,Total} = m \times (h_{out} - h_{in}) \qquad (5\text{-}9)$$

$$Q_{Ventilation,\,sensible} = \frac{\dot{V}f}{v} \times C_P \times (To - Ti) \qquad , m = \frac{\dot{V}f}{v}$$

V = n * recommended air (L/s) / 1000. (m^3/s)

v = specific volume at the state of outside air.

C_P = specific heat for air = 1.2 (kJ/kg.k).

Recommended air: from table (D-17) in appendix D.

$$Q_{Ventilation,\,Latent} = Q_{L,V} = m_f \times (w_i - w_o) \times h_{fg} \qquad (5\text{-}10)$$

Or $\quad Q_T = m_f \times (h_o - h_i) \qquad , Q_{L,V} = Q_{T,V} - Q_{S,V}$

5.6.1.8 Heat Gain by Infiltration

Estimation of infiltration due to door opening:

The amount of air entering each time a door is opened depends on the type of the door, number of entrance passages and weather there are doors in one wall only or more than one wall. The amount of infiltrated air through various types of doors is given in table (D-17) in appendix D for summer air conditioning. For winter heating, infiltration amount of table (D-17) are increased by 50%. Table (6-6) gives the expected number of entrance passages per occupant per hour for different commercial establishment. If such information is not provided in table (6-6) then, number of door opening for other types of establishments can be determined using the following relationship.

$$\frac{\text{Door opening}}{\text{hour}} = \frac{NF}{n\,t} \qquad (5\text{-}11)$$

Where N is number of people in the establishment, F is the factor for arrivals and departure of occupants. Its value is equal to 2 for light traffic and 1.33 for heavy traffic, t is average time of occupancy in hours, and n is the number of doors.

For many commercial application such as restaurants, banks, drug stores, etc. the infiltration of air due to door opening far exceeds the air infiltration through windows and doors cracks. In addition, heat load due to door opening may be considered as the major component of the total heating load of the space.

From table (6-5), the infiltrated air m^3/passage

For number of passage $\rightarrow \dfrac{m^3}{\text{passage}} \times \dfrac{\text{Number of passages}}{\text{hour}}$

$$\text{Number of Passages or door opening} = \frac{NF}{n\,t}$$

N = Number of people.

F = factor for arrivals and departure = 2 for light traffic or 1.33 for heavy traffic.

n = number of doors.

t = time of occupancy.

In this study the number of door opening is measured during a time interval, using a counter which illustrated in figure (3 - 3).

The other source of infiltration is due to the crack of doors and windows; it's calculated by the crack age method:

The crack age method is based on the length of the crack or the perimeter of the window or the door, and the square root of the pressure difference across the crack. The infiltration rate per unit length of crack for different style of windows and doors under different wind velocities are given in table.

It should be noted that air which enters by infiltration from the wind windward side of the building leaves the building from the lee ward side, or through vertical openings. As a result of this, only one half of the length of the crack of a room is used in the computation of infiltration. But, if the room has only one exposed side, then the total length of the crack is used. For rooms with two or more exposed walls, the wall with the longest crack is taken to estimate the infiltration rate.

5.7 Design conditions

In summer:

Outside design temperature = 33 C.

Inside design temperature = 24 C.

$$T_{\text{unconditioned spaces}} = T_{In} + \frac{2}{3} \times (T_{out} - T_{In}) = 30\ C^{o} \qquad (5\text{-}12)$$

R.H, Out = 60% , $h_{out} = 22.7\ w/m^2.C$

R.H, Out = 50% , $h_{out} = 9.37\ w/m^2.C$

$$T_{Out,m} = To - \frac{DR}{2} = 25\ C^o$$

DR = 16 C.

LM = -1.1 (N).

= 0.0 (E).

= 0.0 (W).

= 0.5 (S).

CLTD = 10 for heavy walls.

k = 0.65

f = 1

→ $CLTD_{corr} = 11.885$ (N) $CLTD_{corr} = 12.5$ (E, W)

$CLTD_{corr} = 12.825$ (S)

Sample calculation for "Foyer" space:

- $\mathbf{Q_{Transmitted\ through\ walls,\ doors} = UA\ (CLTD)_{corr.}}$

  $Q_{Ceiling} = (1.004) \times (203.7) \times (33\text{-}24) = 1840.63$ (W)

- $\mathbf{Q_{transmitted\ through\ glass} = A\ (SHG)\ (SC)\ (CLF)}$

  $Q_{Glass\ windows\ (w)} = (20.22) \times (691) \times (.57) \times (.82) = 6530.522$ (W)

- $\mathbf{Q_{Convection,\ Glass} = UA\ (CLTD)_{corr}}$

  $Q_{South\ window} = (1.842) \times (25.02) \times (14.4) = 663.65$ (W)

- $\mathbf{Q_{Equipments}}$ = Load (Power) of equipment (W)

  $Q_{Equipments} = 0$ (W) (No equipments in the space).

- $\mathbf{Q_{Lights} = n\ (P_{LT}) \times (F_U\ F_b) \times (CLF)_{LT}}$.

  $Q_{Lights} = (21) \times (100) \times (1) \times (1) \times (0.89) = 1869$ (W)

- $\mathbf{Q_{Occupancy}} = n \times Q_s \times CLF + (n \times Q_L) \times D_f$

- $Q_{Ventilation,\,sensible} = \frac{\dot{V}f}{v} \times Cp \times (To - Ti)$

The first Package unit distributes its load to the theater and preparing rooms as listed in tables (5 - 1), (5 - 2):

Cooling Load for First Package

Table (5 - 1) Cooling Load for the theater

Space :- Theater	**Area :- 432.58m**		
Load combination	**Sensible load (W)**	**Latent load (W)**	**Total load (W)**
Q **Transmitted through walls ,doors**	11502.404	0	11502.404
Q **convention glass**	0	0	0
Q **glass Transmitted**	0	0	0
Q **Equipments**	650	0	650
Q **Lights**	7205.44	0	7205.44
Q **Occupancy**	20736	10800	31536
Q **Ventilation**	36201.1	100558.6	136759.7
Q **Infiltration**	831.42	2309.5	3140.92
Total	**77126.364**	**113668.1**	**190794.5**

Table (5 - 2) Cooling Load for the preparing rooms

Space :- Preparing rooms		**Area :- 39.1 m²**	
Load combination	**Sensible load (W)**	**Latent load (W)**	**Total load (W)**
Q **Transmitted through walls ,doors**	2048.222	0	2048.222
Q **convention glass**	0	0	0
Q **glass Transmitted**	0	0	0
Q **Equipments**	0	0	0
Q **Lights**	556.25	0	556.25
Q **Occupancy**	331.76	343.2	674.96
Q **Ventilation**	603.2	1676	2279.2
Q **Infiltration**	516.951	1435.975	1952.926
Total	**4056.383**	**3455.175**	**7511.558**

The second Package unit distributes its load to the foyer space and bathrooms as listed in table (5 - 3):

Cooling Load for second Package:

Table (5 - 3) Cooling Load for the Foyer space

Space :- Foyer space		**Area :- 203.7 m^2**	
Load combination	**Sensible load (W)**	**Latent load (W)**	**Total load (W)**
$Q_{\text{Transmitted through walls, doors}}$	6163.495	0	6163.495
$Q_{\text{convention glass}}$	2239.945	0	2239.945
$Q_{\text{glass Transmitted}}$	22089.07	0	22089.07
$Q_{\text{Equipments}}$	0	0	0
Q_{Lights}	2349.6	0	2349.6
$Q_{\text{Occupancy}}$	2717	2145	4862
$Q_{\text{Ventilation}}$	4022.1	11172.5	15194.6
$Q_{\text{Infiltration}}$	2780.48	8515.4	11295.88
Total	**42361.69**	**21832.9**	**64194.59**

Both Fan coils distribute its load to the open area:

Cooling Load for Fan coils:

Table (5 - 4) Cooling Load for the Open area

Space :- Open area		**Area :- 391.335 m^2**	
Load combination	**Sensible load (W)**	**Latent load (W)**	**Total load (W)**
$Q_{\text{Transmitted through walls, doors}}$	6045.5917	0	6045.5917
$Q_{\text{convention glass}}$	0	0	0
$Q_{\text{glass Transmitted}}$	0	0	0
$Q_{\text{Equipments}}$	0	0	0
Q_{Lights}	826.66	0	826.66
$Q_{\text{Occupancy}}$	2033	2670	4703
$Q_{\text{Ventilation}}$	3268.1	9078.2	12346.3
$Q_{\text{Infiltration}}$	380.106	1139.894	1520
Total	**12553.46**	**12888.09**	**25441.55**

Table (5 - 5) Ventilation in the bathrooms

Bathroom	**Cooling ventilation sensible (W)**	**Latent (W)**	**Total (W)**
Basement	469.3	1563.731	1564.2
Ground	1810.5	5027.488	6837.988

5.8 Heating Loads

Since the coldest weather may occur during periods without solar radiation, it is advisable not to rely on the benefit of solar heat gains when calculating peak heating loads (unless the building contains long term storage). If the indoor temperature Ti is constant, a static analysis is sufficient and the calculation of the peak heating load is very simple. Find the design heat loss coefficient $Ktot$, multiply by the design temperature difference $Ti - To$, and subtract the internal heat gains on which one can count during the coldest weather to find the design heat load. However, it is also necessary to warm a space that has had night setback. In a given situation, the required extra capacity, called the *pickup load*, depends on the amount of setback $Ti - To$, the acceptable recovery time, and building construction. For reasonable accuracy, a dynamic analysis is recommended. Optimizing the capacity of the heating system involves a tradeoff between energy savings and capacity savings, with due attention to part load efficiency. As a general rule for residences, ASHRAE (1989a) recommends over sizing by about 40% for a night setback of 10°F (5.6 K), to be increased to 60% over sizing if there is additional setback during the day. In any case, some flexibility can be provided by adapting the operation of the building. If the heating capacity turns out insufficient, one can reduce the depth and duration of the setback during the very coldest periods. In commercial buildings with mechanical ventilation, the demand for extra capacity during setback recovery is reduced if the outdoor air intake is closed during unoccupied periods. In winter that should always be done for energy conservation (unless air quality problems demand high air exchange at night).

5.8.1 Heating Load Calculations

1. Heat loss through walls, doors, windows:

   $Q = UA\Delta T = UA\,(T_{In,\,design} - T_{out})$

2. Heat loss by Ventilation process:

$$Q_{S,v} = \frac{Vf}{v} \times C_P \times (T_{In,\,Design} - T_o)$$

$$Q_{T,v} = m_f \times (h_i - h_o).$$

3. Heat loss by Infiltration, using Door Opening Method. Modification of values in table (6-5), by increase 50%, must be occurred to get the true values.

5.8.2 Design conditions in winter

Outside temperature = 4 (C°).

Inside design temperature for the preparing rooms = 22 (C°).

Inside design temperature for the other spaces = 18 (C°).

Outside relative humidity = 70 %

Air velocity = 1.4 (m/s)

$$T_{Unheated\ spaces} = T_{In} + \frac{2}{3}\,(T_{Out} - T_{In})$$

$$= 22 + \frac{2}{3}(4 - 22)$$

$T_{Unheated\ spaces} = 10$ (C°).

$T_{Ground} = 10$ (C°).

$\nu_{out\,@\,(T=4\,(C^\circ)\,\&\,\Phi=70\%)} = 0.79$ (m³/Kg).

$h_{out} = 13$ (kJ/Kg)

$h_{in} = 44$ (kJ/Kg)

For the open area:

Table (5-6): Heating load quantities for the open area.

Load type	Q_s (W)	Q_L (W)	Q_T (W)
Q **walls ,doors ,windows**	7713.16	0	7713.16
Q **Ventilation**	5759.5	2879.75	8639.25
Q **Infiltration**	506.808	126.702	633.51
Q **Total**	**16985.92**		

For the small rooms:

Table (5-7): Heating load calculation for the preparing rooms.

Space	Q_{Walls} swindows ,doors	$Q_{Ventilation}$ (W)			$Q_{Infiltration}$ (W)		
		Q_s	Q_l	Q_T	Q_s	Q_l	Q_T
1	938.995	340.77	247.837	588.607	129.238	358.99	488.2315
2	1302.7	340.77	247.837	588.607	129.238	358.99	488.2315
3	745.054	340.77	247.837	588.607	129.238	358.99	488.2315
4	999.4136	340.77	247.837	588.607	129.238	358.99	488.2315
Total	**3986.162**	**1363.08**	**991.348**	**2345.428**	**459.512**	**114.878**	**574.39**
Q_{Total}	**9834.798**						

For theater space:

Table (5-8): Heating load calculation for the theater.

Space	$Q_{Walls, windows, doors}$	$Q_{Ventilation}$ (W)			$Q_{Infiltration}$ (W)		
		Q_S	Q_L	Q_T	Q_s	Q_L	Q_T
Theater	15602	63797.46	31898.74	95696.2	739.04	184.76	923.8
Total	**208842**						

For foyer space and Bathrooms in the ground floor:

Table (5-9): Heating load calculation for foyer space and the bathrooms in the ground floor.

Space	Q_{Walls} ,windows ,doors	$Q_{Ventilation}$ (W)			$Q_{Infiltration}$ (W)		
		Q_s	Q_L	Q_T	Q_s	Q_L	Q_T
Foyer space	2130.634	7088	1013.3	8101.3	4268.13	9566.25	13834.38
Bathrooms (Ground floor)	0	3189.87	531.63	3721.5	0	0	0
Total	2130.634	10277.87	1544.93	11822.8	4268.13	9566.25	13834.38
Q_{Total}	**53444.99**						

5.9 Matching Loads to Source

Comparison between the actual load for the device and the calculated load based on the design conditions of the relating region.

Table (5-10): The total actual and designed loads for each HVAC system.

HVAC Unit	Actual Load for Device (ton)	Actual Load for Device (kW)	Calculated Load for the space (kW)
First Package	37.5	132	198.306
Second Package	37.5	132	71.032578
Fan Coil No.1	10	35.2	32.097
Fan Coil No.2	10	35.2	32.097

5.10 Opportunities for energy saving in HVAC

5.10.1 Rising the inside design dry bulb temperature in summer cooling

Normal conditions inside the space is (T_{db} = 24 C°, φ = 50%), but the human comfort region let the designer to select a higher temperature than the comfort point on psychometric chart, we select (T_{db} = 26 C°, φ = 50%) and make comparison between both cooling loads which relates to each temperature respectively.

Temperature sensors must be put inside each space (Theatre, General open area, Small preparing rooms and Foyer space) to keep the temperature in the specified region. Results of this process are illustrated in the following table.

Table (5-11): The total cooling load difference in the case of rising the temperature to 26 (C°) in the theater.

Theatre							
	T_{db} = 24C			T_{db} = 26C			
Load type	S(W)	L(W)	T(W)	S(W)	L(W)	T(W)	ΔQ(W)
$Q_{\text{Transmitted through walls ,doors}}$	11502.404	0	11502.404	8338.15	0	8338.15	3164.254
$Q_{\text{convention glass}}$	0	0	0	0	0	0	0
$Q_{\text{glass Transmitted}}$	0	0	0	0	0	0	0
$Q_{\text{Equipments}}$	650	0	650	650	0	650	0
Q_{Lights}	7205.44	0	7205.44	7205.44	0	7205.44	0
$Q_{\text{Occupancy}}$	20736	10800	31536	20736	10800	31536	0
$Q_{\text{Ventilation}}$	36201.1	100558.6	136759.7	28156	84469	112625	24134.7
$Q_{\text{Infiltration}}$	831.42	2309.5	3140.92	646.66	1939.98	2586.64	554.28
Q_{Total}	77126.36	113668.1	190794.5	65732.25	97208.98	162941.2	**27853.23**

Table (5-12): The total cooling load difference in the case of rising the temperature to 26 (C°) in Foyer space.

Foyer space							
Load combination	T_{db} = 24C			T_{db} = 26C			
	S(W)	L(W)	T(W)	S(W)	L(W)	T(W)	ΔQ(W)
$Q_{\text{Transmitted through walls doors}}$	6163.495	0	6163.495	4650.328	0	4650.328	1513.167
$Q_{\text{convention glass}}$	2239.945	0	2239.945	22089.07	0	22089.07	19849.125
$Q_{\text{glass Transmitted}}$	22089.07	0	22089.07	1920.955	0	1920.955	20168.115
$Q_{\text{Equipments}}$	0	0	0	0	0	0	0
Q_{Lights}	2349.6	0	2349.6	2349.6	0	2349.6	0
$Q_{\text{Occupancy}}$	2717	2145	4862	2717	2145	4862	0
$Q_{\text{Ventilation}}$	4022.1	11172.5	15194.6	3128.3	9384.9	12513.2	2681.4
$Q_{\text{Infiltration}}$	2780.48	8515.4	11295.88	2162.53	6487.72	8650.25	2645.63
Q_{Total}	42361.7	21832.9	64194.6	39017.8	18017.6	57035.4	**46857.4**

Table (5-13): The total cooling load difference in the case of rising the temperature to 26 (C°) in Preparing Rooms.

Preparing Rooms							
	$T_{db} = 24C$			$T_{db} = 26C$			
Load type	**S(W)**	**L(W)**	**T(W)**	**S(W)**	**L(W)**	**T(W)**	**ΔQ(W)**
Q Transmitted through walls ,doors	2048.222	0	2048.222	1428.5	0	1428.5	619.722
Q convention glass	0	0	0	0	0	0	0
Q glass Transmitted	0	0	0	0	0	0	0
Q Equipments	0	0	0	0	0	0	0
Q Lights	556.25	0	556.25	556.25	0	556.25	0
Q Occupancy	331.76	343.2	674.96	331.76	343.2	674.96	0
Q Ventilation	603.2	1676	2279.2	469.2	1407.6	1876.8	402.4
Q Infiltration	516.951	1435.975	1952.926	402.073	1206.127	1608.2	344.726
Q Total	4056.383	3455.175	7511.558	3187.783	2956.927	6144.71	**1366.848**

Table (5-14): The total cooling load difference in the case of rising the temperature to 26 (C°) in Open Area.

Open Area							
	T_{db} = 24C			T_{db} = 26C			
Load type	S(W)	L(W)	T(W)	S(W)	L(W)	T(W)	ΔQ(W)
$Q_{\text{Transmitted through walls ,doors}}$	6045.5917	0	6045.5917	3445.48	0	3445.48	2600.11
$Q_{\text{convention glass}}$	0	0	0	0	0	0	0
$Q_{\text{glass Transmitted}}$	0	0	0	0	0	0	0
$Q_{\text{Equipments}}$	0	0	0	0	0	0	0
Q_{Lights}	826.66	0	826.66	826.66	0	826.66	0
$Q_{\text{Occupancy}}$	2033	2670	4703	2033	2670	4703	0
$Q_{\text{Ventilation}}$	3268.1	9078.2	12346.3	2541.89	7625.61	10167.5	2178.8
$Q_{\text{Infiltration}}$	380.106	1139.894	1520	253.404	886.914	1140.318	379.682
Q_{Total}	12553.5	12888.1	25441.6	9100.43	11182.5	20283	**5158.59**

5.10.2 Mixing of both humidified adiabatic air streams

5.10.2.1 Psychometric Properties of Moist Air

Psychometric is the study of the properties of moist air, i.e., a mixture of air and water vapor. A thorough understanding of psychometric is essential since it is fundamental to understanding the various processes related to air conditioning. Atmospheric air is never totally dry; it always contains varying degrees of water vapor. Just like relatively small amounts of trace materials drastically impact the physical properties of steel alloys, small amounts of moisture have a large influence on human comfort.

The amount of water vapor contained in air may vary from near zero (totally dry) to a maximum determined by the temperature and pressure of the mixture. Properties of moist air can be determined from tables or from equations , and steam tables, or from the psychometric chart. Moist air up to about three atmospheres pressure can be assumed to obey the perfect gas law. Assuming dry air to consist of one gas only, the total pressure pt of moist air, given by the Gibbs-Dalton Law for a mixture of perfect gases, is equal to the individual contributions of dry air and water vapor, equation (5-13).

$$p_t = p_a + p_v \tag{5-13}$$

Where pa is the partial pressure of dry air, and pv is the partial pressure of water vapor. It is because pv << pa that we can implicitly assume water vapor also follows the perfect gas law for atmospheric air. The thermodynamic state of an air-vapor mixture is fully determined if three independent intensive properties are specified. Since one can assume for most of the HVAC processes being studied that the total atmospheric pressure does not change, a chart known as the psychometric chart, applicable to a specific value of total pressure (commonly the standard atmospheric pressure), is used. The psychometric chart not only provides a quick means for determining values of moist air properties, it is also very useful in solving numerous process problems with moist air and allows quick visualization of how the process occurs. Hence, for better comprehension, we describe the manner in which it is generated along with the description of the pertinent moist air properties.

The primary moist air properties shown on a psychometric chart are described below:

- ***Dry-bulb temperature:***

  T_{db} or t is the temperature of air one would measure with an ordinary thermometer. This property is the x-axis of the psychometric chart.

- ***Saturation pressure* of water vapor:**

  p_s or $p_{v,sat}$ can be determined or obtained from steam tables.

- ***Humidity ratio* or specific humidity:**

  (ω) is defined as the ratio of the mass of water vapor to that of dry air, i.e., Using the ideal gas law under saturated air conditions, where V is an arbitrary volume of the air and water vapor mixture, R is the universal gas constant, M_{Wa} is the molecular weight of dry air, and M_{Wv} is the molecular weight of water . The above formula then reduces to :

$$\omega = \frac{\text{mass of water vapor}}{\text{mass of dry air}} \tag{5-14}$$

$$\omega = \frac{mv}{ma}$$

$$\omega = 0.622 \times \frac{P_s}{P_t - P_s} \tag{5-15}$$

5.10.2.2 Psychometric Processes

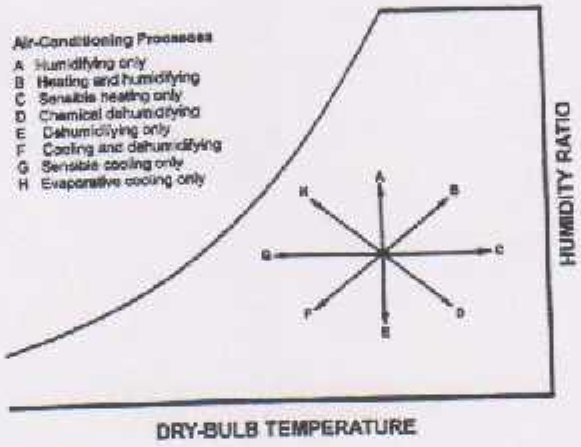


Figure (5-3) the process's direction that is possible on the psychometric chart.

Analysis of moist air processes with various HVAC devices essentially involves a few fundamental processes, discussed below. Consider a duct containing a device through which moist air is flowing. The device could be a cooling or heating coil and/or a humidifier. The analysis of moist air processes flowing through such a device is based on the laws of conservation of mass and energy. Although in actual practice the properties of the moist air may not be uniform across the duct cross section especially downstream of the device), such phenomena are neglected, and the focus is on bulk or fully mixed conditions. Further, assuming (1) steady state conditions and (2) a perfectly insulated duct.

5.10.2.3 Sensible Heating and Cooling

A process is called *sensible* (either heating or cooling) when it involves a change in dry-bulb temperature only (i.e., the moisture content specified by the specific humidity is unchanged in a sensible heating or cooling process). This could apply to either heating (an increase in T_{db}) or to cooling (a decrease in T_{db}). In such a case, mw = 0, and W1 = W2. The system equation is:

$$Q = m_a \times (h_2 - h_1) \qquad (5\text{-}16)$$

Where:

$$m_a = m_{a1} = m_{a2}$$

The process of sensible heating or cooling is represented as a straight line on the psychometric chart as shown in Figure (5-3) such a process occurs from point (G) to point (C) when moist air flows across a cooling coil when condensation does not occur.

5.10.2.4 Cooling and Dehumidification

This process occurs when conditioning outdoor air in summer or in internal spaces where heat and moisture are removed by cooling coils in a conditioned space. For this process to occur, moist air is cooled to a temperature below its dew point. Some of the water vapor condenses out of the air stream. Although the actual process path varies depending on the type of surface, surface temperature, and flow conditions, the heat and mass transfer can be expressed in terms of the initial and final states. As shown in Figure (5-4), a certain amount of moisture condenses out of the air stream. Although this condensation occurs at various temperatures ranging from the initial dew point to its final saturation temperature, it is assumed that condensed water is cooled to the final air temperature t2 before it drains out. The system equations are:

Rate of water condensation:

$$m_w = m_a(\omega_1 - \omega_2) \qquad (5\text{-}17)$$

Rate of total heat transfer:

$$Q = m_a[(h_1 - h_2) - (\omega_1 - \omega_2)h_{w2}] \qquad (5\text{-}18)$$

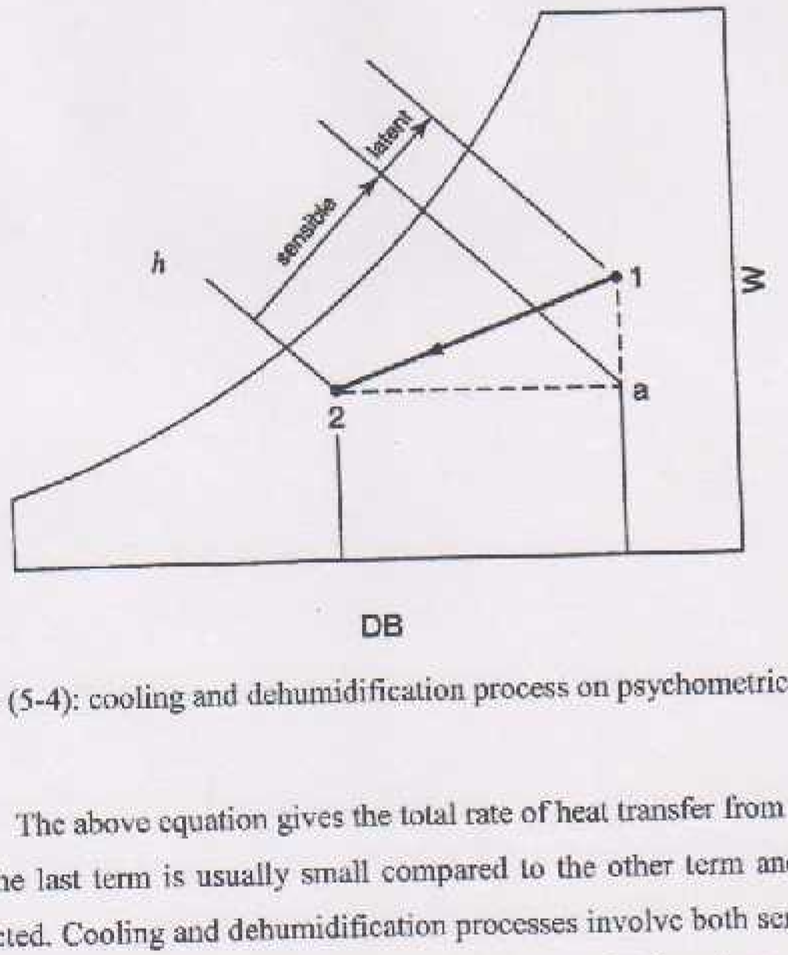


Figure (5-4): cooling and dehumidification process on psychometric chart.

The above equation gives the total rate of heat transfer from the moist air. The last term is usually small compared to the other term and is often neglected. Cooling and dehumidification processes involve both sensible and latent heat transfer. The sensible heat transfer qs results in a decrease in dry-bulb temperature, while the latent heat transfer ql is associated with the decrease in specific humidity. These quantities can be estimated as follows. Let point "a" be the intersection point between the constant dry-bulb temperature line from point 1 and the constant specific humidity line at point 2 (see Figure (5-4). Then:

Rate of sensible heat transfer equation :

$$Q_s = m_a \times (h_a - h_2) \qquad (5\text{-}19)$$

Rate of latent heat transfer:

$$Q_L = m_a (h_1 - h_a) \qquad (5\text{-}20)$$

It is customary to characterize the relative contributions of sensible and latent heat transfer rates by the sensible heat ratio (SHR) where SHR is defined as follows as indicated in equation (5-20):

$$SHR = \frac{Qs}{QT} \qquad (5\text{-}20)$$

5.10.2.5 Mixing Of Outdoor Air and Recirculated Air

A space is air conditioned to offset the heating and/or cooling loads of the space as a result of envelope heat transmission, ventilation air requirements, and internal loads due to occupants, lights, and equipment. The calculations involved in air conditioning design reduce to the determination of the mass of dry air to be circulated, its dry-bulb temperature, and its humidity level that will result in comfortable indoor conditions for the occupants. Let Q_s and Q_l be the sensible and latent loads on a space to be air conditioned. The latent load is due to the sum of all rates of moisture gain designated by mw. Assuming steady conditions, the sum of equations (5-19), and (5-20) give:

$$Q_s + Q_L = m_a \times (h_2 - h_1) \qquad (5\text{-}21)$$

Outdoor air is required continuously to ventilate occupied spaces in commercial and institutional buildings. This ventilation air is used to control the concentration of airborne pollutants indoors by diluting them with cleaner outdoor air. The minimum amount of outdoor ventilation air that is required in a space is determined by ASHRAE Standard 62, "Ventilation for Acceptable Indoor Air Quality". This voluntary standard developed within ASHRAE has been incorporated by reference into many state building codes, including the Minnesota Energy Code, where it is an enforceable document. The outdoor air is introduced into a building through an outdoor air intake. The intake should be located away from contaminant sources such as roadways, loading docks and building exhaust stacks. The intake usually has a coarse screen to prevent birds and large debris from entering.

The disadvantage of bringing outdoor air into a building is that the air is usually not at the psychometric conditions (i.e. temperature and humidity) desired in the building. In cold weather, this air requires heating and humidification. In hot weather, this air must be cooled and dehumidified. Thus there is an energy penalty associated with bringing in this outdoor ventilation air. Most of the time, the majority of the air distributed within a building is recirculated. The air returning from the building to the heating or cooling plant is at the desired indoor conditions so less energy is required to bring it to the supply air conditions than to treat outdoor air. This is not always true, however. There are certain times of the year when the outdoor air is just cool enough to provide the necessary cooling in the building without operating the cooling system. Under these conditions, the supply air can be all outdoor air. When the outdoor air conditions are near the yearly extreme values, only the minimum amount of outdoor air required is brought in to minimize the heating and cooling load on the mechanical equipment.

Most air handling systems are designed with mixed air dampers near the outdoor air intake. These dampers are controlled automatically in actual buildings to vary the amount of outdoor air brought in and the amount of air that is recirculated. The automatic control system monitors the outdoor temperature and humidity, or outdoor air enthalpy, and time of day to determine the amount of outdoor air that should be admitted. The remaining amount of air is recirculated. The damper positions are controlled by actuators that receive signals from the controller. Automatic dampers can be set to bring in any amount of return and outdoor air by adjusting the percent that the damper blades are open.

However the relationship between percent open and flow is not linear. Figure 1 is an example of a damper characterization. Therefore if code required a minimum of 25% outdoor air at all times, the outdoor air dampers would need to have a minimum set-point of approximately 60%.

The mixed air passes through a particulate air filter downstream of the mixed air dampers. Then the air can pass over a heating or cooling coil if the coils are located before the supply fan. One of the difficulties that arise in cold climates is the density difference that occurs between the outdoor air and the recirculated air. The cold, denser air tends to settle on the bottom of the mixed air duct. When its temperature is very low, this cold air can cause the lower portion of the coils to freeze. In an attempt to reduce this possibility, parallel blade dampers often direct the outdoor air and the recirculated air toward each other to reduce the possibility of stratification. If this low temperature air has a low flow rate, then it will have a minimum affect on the bulk air temperature.

The bulk air temperature is a mass average of the temperatures distributed throughout the duct cross-section. This bulk temperature may not agree with the temperature measured by the average temperature sensors because these averaging sensors do not account for air flow. These averaging sensors assume uniform flow throughout each cross section of the duct; however we know this not to be true from the Air Handling System Characterization lab.

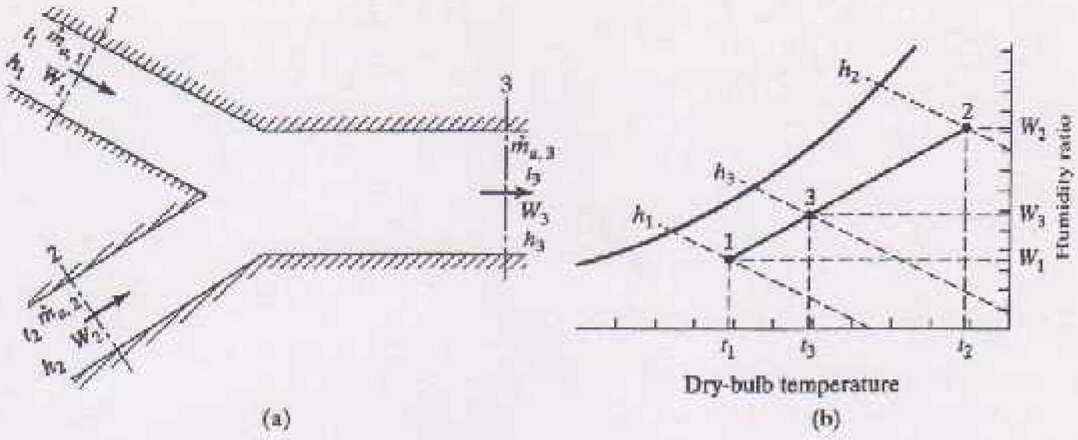


Figure (5-5): Adiabatic mixing of two streams of moist air

Mixing of two different streams of moist air in an insulated duct is a common process in air conditioning systems. The properties of the mixed streams can be determined graphically using the psychometric chart.

Figure (5-5) illustrates the adiabatic mixing of air streams. The first law of thermodynamic as applied to this process gives:

$$m_{a1}h_1 + m_{a2}h_2 = m_{a3}h_3 \qquad (5\text{-}22)$$

The mass balance as applied on the dry air gives:

$$m_{a1} + m_{a2} = m_{a3} \qquad (5\text{-}23)$$

And the mass balance as applied on the water vapor is:

$$m_{a1}\omega_1 + m_{a2}\omega_2 = m_{a3}\omega_3 \qquad (5\text{-}24)$$

By combining the previous equations and eliminating m_{a3}, the following results are obtained.

$$\frac{h_2-h_3}{h_3-h_1} = \frac{w_2-w_3}{w_3-w_1} = \frac{m_{a1}}{m_{a2}} \qquad (5\text{-}25)$$

The form of equation (5-25) shows that the mixed stream must lie on a straight line between state one and two. This is shown in figure (5-5).It may be further inferred from equation (5-25) that the lengths of the various line segments are proportional to the masses of dry air mixed i.e.

$$\frac{m_{a1}}{m_{a2}} = \frac{\overline{23}}{13}, \quad \frac{m_{a1}}{m_{a3}} = \frac{\overline{32}}{12} \quad \text{and} \quad \frac{m_{a2}}{m_{a3}} = \frac{\overline{13}}{12} \qquad (5\text{-}26)$$

This result shows that the state of the mixed stream can be determined by drawing a straight line on the psychometric chart joining the state of the first air stream to that of the second air stream and locating a point for state three, on the line such that the ratio of the mass flow rate determines the distance of that point from either end of the drawn line. This is shown in figure (5-5b). On the hand the enthalpy h_3, the humidity ratio w_3, of the mixed stream can be calculated from equation (5-25) and equation (5-24), respectively. Thus, state three can be determined.

5.10.3 Design of Smart mixing unit using Programmable Interfacing Controller (PIC18F4550)

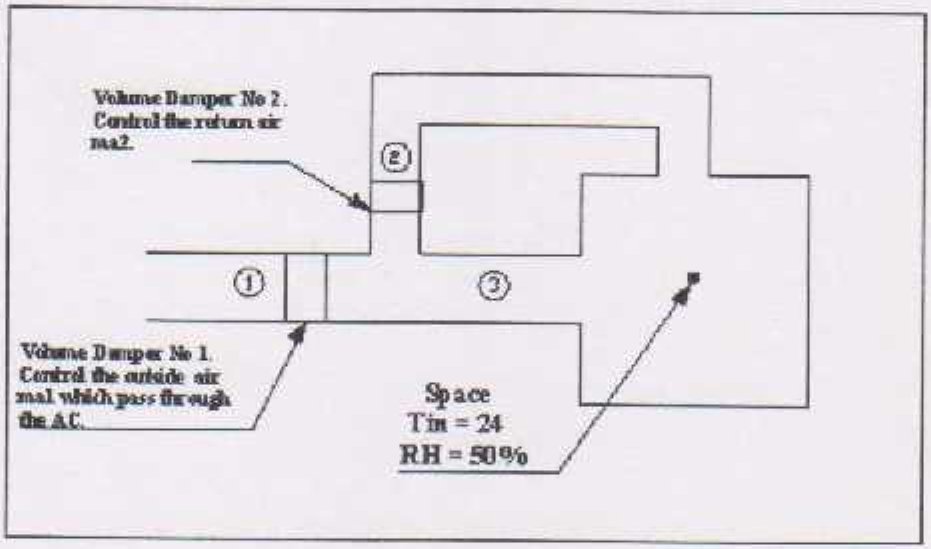


Figure (5-6): The principle of mixing.

The principle of mixing process can be monitored by measuring the outside conditions of the outside and inside humidified air, that means outside relative humidity and dry bulb temperature and so for the inside conditions of the space.

Determination of the mixing state can be calculated using the psychometric chart; it must be compared with the recommended sensible heat ratio of the conditioned space, since it must be achievable to keep the humidity of the space in the human comfort range.

Technical approach aims to relate the specified amount of humidified air that will be recirculated and mixed again with the fresh air, to the temperatures and relative humidity of the mixed streams. A group of relations to get the dependency of the input to the output values.

At the beginning of the process, the system reads the input values, and then used T_1 to calculate the pressure of saturated water vapor according to the linear equation approximated using Matlab software (5-26).

$$P_w(T) = 0.004697 \times T^2 - 0.02122 \times T + 0.8083 \qquad (5\text{-}26)$$

Where:

T : the dry bulb temperature has measured by the sensor.

Then, the saturated pressure substitutes in the following equation (5-27) to obtain the partial pressure of the water vapor in the humidified mixed air.

$$P_{v1} = \phi_1 P_{vs1} \qquad (5\text{-}27)$$

Where:

P_{v1}: The partial pressure for water vapor.

P_{vs1} : Saturation pressure for the water vapor in kPa.

Substitution the value of the partial water vapor pressure in equation (5-28) to calculate the humidity ratio of the first state of air stream, the same for the second state:

$$\omega_1 = 0.622 \frac{P_{v1}}{P - P_{v1}} \quad (5\text{-}28)$$

Where:

ω_1: The humidity ratio (kg of water vapor/ kg of dry air)

After that the value of the enthalpy for the first state of the air stream can be obtained using equation (5-29) and so for the second state

$$h_1 = C_{pa} + (C_{va}T_1 + h_g)\omega_1 \quad (5\text{-}29)$$

Where:

h_1: The enthalpy of the humidified air at state (1). (kJ/kg)

C_{pa}: Specific heat for the dry air. (kJ/kg.k)

C_{va}: Specific heat for the water vapor. (kJ/kg.k)

h_g: The enthalpy of the water vapor. (kJ/kg)

To calculate the humidity ratio for the mixed air stream, equation (5-30) must be use, by substituting the values of ω_1, ω_2.

$$\omega_3 = \frac{\dot{m}_{a1}\,\omega_1 + \dot{m}_{a2}\,\omega_2}{\dot{m}_{a3}} \quad (5\text{-}30)$$

ω_1: Humidity ratio of the humidified air at state one

ω_2: Humidity ratio of the humidified air at state two.

ω_3: Humidity ratio of the humidified air at the state of mixing.

$\dot{m}_{a1}$: Mass flow rate of the supply air comes from the HVAC.

$\dot{m}_{a2}$: Mass flow rate of the air returns from the conditioned space.

$\dot{m}_{a3}$: Mass flow rate of the mixing air stream.

And so for the enthalpy of the mixing air stream but using h_1, h_2 then substitute in equation (5-31):

$$h_3 = \frac{\dot{m}_{a1} h_1 + \dot{m}_{a2} h_2}{\dot{m}_{a3}} \tag{5-31}$$

h_1 : The enthalpy of the humidified air at state one. (kJ/kg)

h_2 : The enthalpy of the humidified air that returns from the conditioned space state two. (kJ/kg)

h_3 : The enthalpy of the humidified air at the mixing state(three) (kJ/kg)

We can obtain the temperature of the mixed air stream using equation (5- 32) using ω_3 , h_3 , h_g , C_{Pa}

$$T_3 = \frac{h_3 - h_g \omega_3}{(C_{pa} + h_g \omega_3)} \tag{5- 32}$$

Where:

T_3 : the temperature of the mixing air stream.(C$^\circ$)

Now, we have the overall properties of the mixed state of the humidified air, so sensible heat ratio can be calculated using T_{in}, h_{in}, T_3, h_3 substitution in equation (5- 33).

$$SHR1 = \frac{T_{in} - T_3}{h_{in} \quad h_3} \tag{5- 33}$$

$SHR1$: Sensible heat ratio for the primary conditions of mixing.

The value of the calculated sensible heat ratio must be compared with the designed value of sensible heat ratio using equation (5- 34) to be sure that the mixing process is recommended or not, this step consists the feedback process of the system to make the control operation:

$$SHR - SHR1 = 0 \qquad END \quad (5\text{- }34)$$

SHR : designed sensible heat ratio for the space relative to the cooling load calculations of the space.

If the equation is confirmed, then the system operates at the steady state conditions, otherwise the enthalpy will be calculated again using the designed sensible heat ratio by equation (5- 34)

$$h_3 = \frac{SHR\,h_{in} - T_{in} + T_3}{SHR} \qquad (5\text{- }34)$$

The ratio of the mass flow rate can be defined as ($Z = \frac{m_{a1}}{m_{a2}}$) and can be determined using equation (5- 35):

$$Z = \frac{\dot{m}_{a3} h_3 - \dot{m}_{a1} h_1}{\dot{m}_{a1} h_1} \qquad (5\text{- }35)$$

Using Z (the mass mixing ratio) to get the angle θ_1, θ_2 (in radians) of the first volume damper, and so for the second volume damper using equation (5- 36) and (5- 38) respectively:

$$\theta_1 = \frac{\dot{m}_{a3}}{VL_1 r_1 (1 + Z)} \qquad (5\text{- }36)$$

θ_1 : The angular displacement of the volume damper. (radians)

V : the velocity of the air in the main supply and return branches of the duct. (m/s)

L_1: Height of the main duct (supply duct). (m)

r_1: The radius of rotation of the volume damper, it equals to the distance between the centers of the duct to the end. (m)

But we must calculate the time required to run the stepper motor used for this process to give it the specific signal that accomplish the determined output value using equation (5- 37), and so for the other volume damper using equation (5- 39).

$$t_1 = \frac{\theta_1}{v_1} \tag{5- 37}$$

$$\theta_2 = \frac{Zm_{a3}}{VL_2 r_2 (1-Z)} \tag{5- 38}$$

$$t_2 = \frac{\theta_2}{v_2} \tag{5- 39}$$

5.10.3.1 PIC (Programmable Interfacing Controller)

A microcontroller is a functional computer system-on-a-chip. It contains a processor, memory, and programmable input/output peripherals. Microcontrollers include an integrated CPU, memory (a small amount of RAM, program memory, or both) and peripherals capable of input and output.

It emphasizes high integration, in contrast to a microprocessor which only contains a CPU (the kind used in a PC). In addition to the usual arithmetic and logic elements of a general purpose microprocessor, the microcontroller integrates additional elements such as read-write memory for data storage, read-only memory for program storage, Flash memory for permanent data storage, peripherals, and input/output interfaces. At clock speeds of as little as 32 KHz, microcontrollers often operate at very low speed compared to microprocessors, but this is adequate for typical applications. They consume relatively little power (milliwatts or even microwatts), and will generally have the ability to retain functionality while waiting for an event such as a button press or interrupt. Power consumption while sleeping (CPU clock and peripherals disabled) may be just nanowatts, making them ideal for low power and long lasting battery applications.

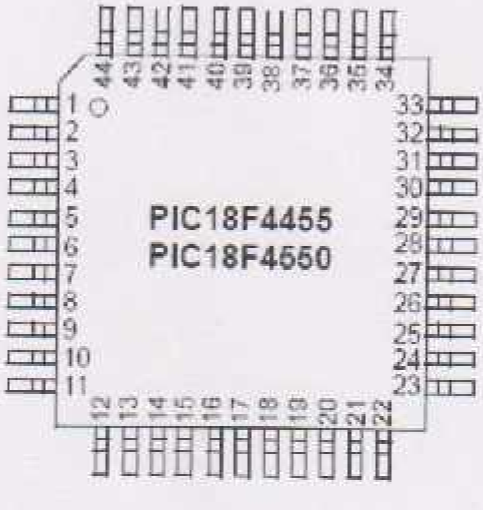


Figure (5-7): PIC18F4550 ship

5.10.3.2 Application

Microcontrollers are used in automatically controlled products and devices, such as PC USB interfacing, control systems, Temperature monitoring, remote controls, office machines, appliances, power tools, and toys. By reducing the size, cost, and power consumption compared to a design using a separate microprocessor, memory, and input/output devices, microcontrollers make it economical to electronically control many more processes.

5.10.3.3 Process Flow Chart

The sequence of the process is illustrated in the following chart, figure () step by step to be programmed and installed on the chip later.

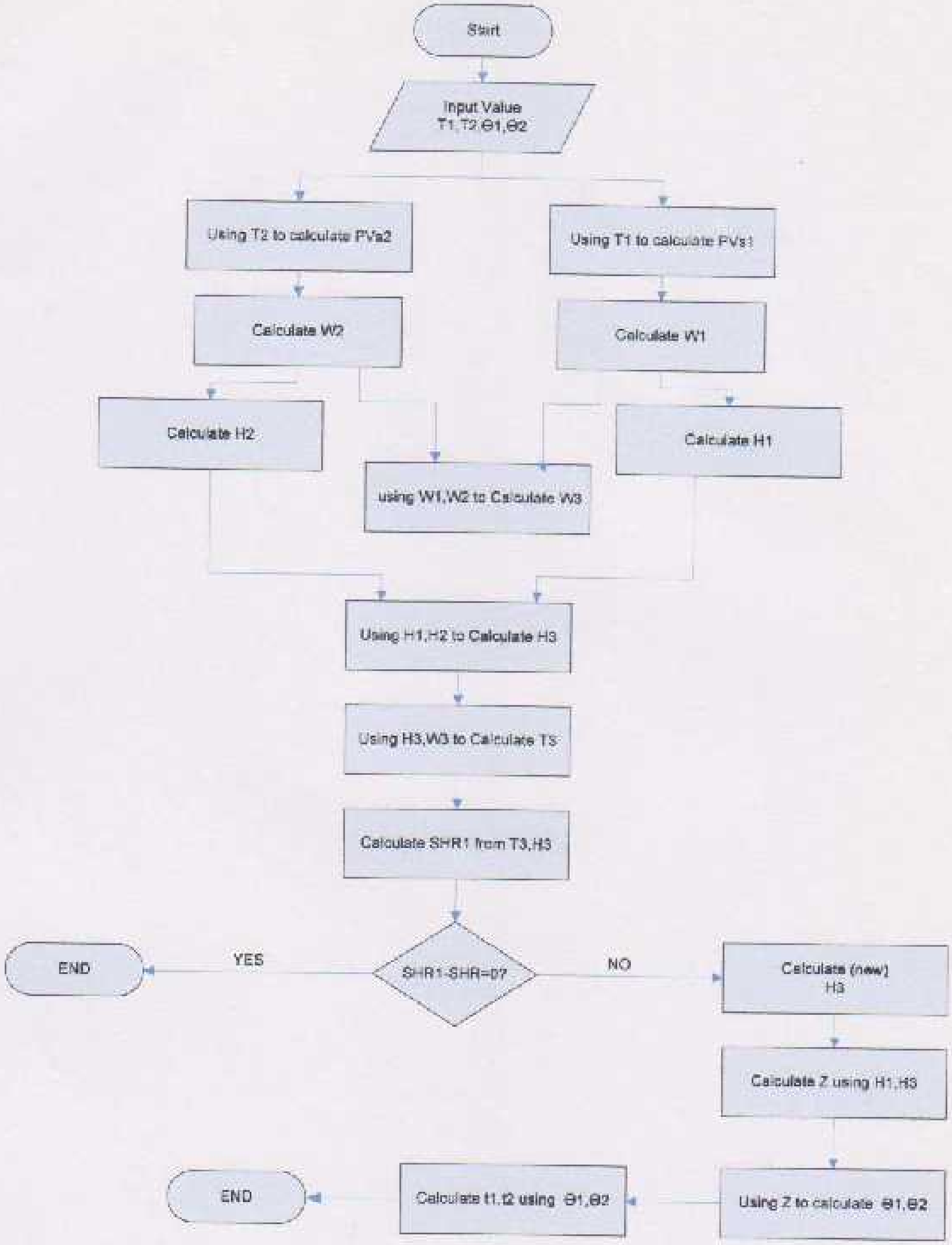


Figure (5-8): The flow chart of the process occurred on the PIC (Programmable Interfacing Controller).

5.10.3.4 Testing and evaluation of the System behavior

After applying the previous controller for mixing, testing must be done to have an indicator about the system behavior with different conditions locates on the psychometric chart in summer and winter conditions also.

Input values given to the controller determines the mixing state of both humidified air streams, but we must have the energy quantity of this state to calculate the amount of energy saving that have been kept to know the validity of installing this modification to the AC.

Testing values for the controller is:

$T_1 = 18$ C , $\varphi_1 = 45\%$

$T_1 = 26$ C , $\varphi_1 = 65\%$

$P_{vs1} = 2.08544$ kPa.

$P_{v1} = 0.45 \times 2.08544 = 0.938448$ kPa.

$\square_1 = 0.622 \times (0.938448) / (101.225 - 0.938448)$

$= 0.005820467$ (kg of water/kg of dry air)

$P_{vs2} = 2.7204$ (kPa).

$P_{v1} = 0.6 \times 2.7204 = 1.76826$ kPa.

$\square_2 = 0.622 \times (1.76826) / (101.225 - 1.76826)$

$= 0.011058654$ (kg of water/kg of dry air)

$h_1 = 1 \times 18 + (1.86 \times 18 + 2501.3) \times 0.005820467 = 32.75$ (kJ/kg of dry air).

$h_1 = 1 \times 26 + (1.86 \times 26 + 2501.3) \times 0.011058654 = 54.2$ (kJ/kg of dry air).

$h_3 = 0.5 \times 32.2 + 0.5 \times 54.2 = 43.475$ (kJ/kg of dry air).

$T_3 = 43.475 - 2501.3 \times (0.00843956) / (1 + 1.86 \times 0.00843956)$

$T_3 = 19.33$ C°.

SHR $_{Design}$ =77126.364 / 190794.5 = 0.4042

SHR $_1$ = (26 − 19.3) / (48 − 43.475) = 1.032 (Greater than the designed value)

So, the enthalpy must be calculated according to the designed value

SHR $_1$ = (26 − 19.3) / (48 − h_3) = 0.4042

h_3 = 36.372 (kJ/kg of dry air).

Z = (1 × 36.372 − 0.5 × 32.75) / (0.5 ×32.75) = 1.2219

θ_1 = 1 / (4 × 0.5 ×1.175× (1 + 1.2219)) = 0.19 rad. = 10.8917°

θ_2 = (1.2219×1) / [(4×0.75×0.57) × (1 - 1.2219)] = - 3.22019 rad. = - 184.596°

Total amount of air must be supplied to the space:

$$m_{\text{Supply Air to the space}} = \frac{Q_{\text{Cooling}}}{(h_{out} - h_{in})} = \frac{190.794464\ (kW)}{(82 - 48)(kJ/kg.k^{\circ})} = 5.611601(kg/s)$$

$$Q_{\text{Cooling without mixing}} = m_{\text{Supply Air to the space}} \times (h_{in} - h_{out})$$

$$= 5.611601 \times (48 - 32.75) = 85.576\ (kW)$$

$$Q_{\text{Cooling with mixing}} = m_{\text{Supply Air to the space}} \times (h_{In} - h_{mix})$$

$$= 5.611601 \times (48 - 36.372) = 65.251\ (kW)$$

$$Q_{\text{Saving}} = Q_{\text{Cooling without mixing}} - Q_{\text{Cooling with mixing}}$$

$$= 85.576 - 65.251 = 20.325\ (kW)$$

The amount of energy saving is equal to 20.325kW

Since the average operating hours in this space is 3 (hours per day), then the amount of energy saving in a year is:

$$\text{Energy saved per year} = P\,saved\ (\text{kW}) \times \frac{No.hours}{day} \times (26)\frac{day}{mounth} \times (12)\frac{mounth}{year}$$

$$= 20.325(kW) \times 3 \times 26 \times 12$$

$$= 19024.2(\text{kWh/year})$$

$$\text{The cost of energy saving} = \text{Energy saving(kWh/year)} \times \text{Price (\$)}$$

$$= \frac{19024.2 \times 0.65}{3.9}$$

$$= 3170.7(\$)$$

As a final evaluation for the HVAC units, the coefficient of performance must calculated, based on the cooling and heating loads calculations:

Using equations (3-2) to (3-5):

For the first package unit:

$$COP_{PPH1\ cooling} = \frac{Q_L}{P_{compressors} + P_{EFM} + P_{CFM}} = \frac{198.306}{7.460 \times 4 + 7.5 + 0.75 \times 2} = 5.105$$

$$COP_{PPH\ Heating1} = \frac{Q_H}{P_{compressors} + P_{EFM} + P_{CFM} + P_{Heaters}} = \frac{208.842}{7.460 \times 4 + 0.75 \times 2 + 7.5 + 0.5 \times 2} = 5.14$$

For the second package unit:

$$COP_{PPH2\ cooling} = \frac{Q_L}{P_{compressors} + P_{EFM} + P_{CFM}} = \frac{71.032578}{7.460 \times 4 + 7.5 + 0.75 \times 2} = 1.8288$$

$$COP_{PPH\ Heating2} = \frac{Q_H}{P_{compressors} + P_{EFM} + P_{CFM} + P_{Heaters}} = \frac{53.44499}{7.460 \times 4 + 0.75 \times 2 + 7.5 + 0.5 \times 2} = 1.34$$

The first fan coil unit and the second one have the same COP:

$$COP_{\text{Fan coil cooling}} = \frac{Q_L}{P_{\text{compressors}} + P_{\text{EFM}} + P_{\text{CFM}}} = \frac{32.097}{6.118 \times 2 + 0.75 + 0.750} = 2.336$$

$$COP_{\text{Fan coil Heating}} = \frac{Q_H}{P_{\text{compressors}} + P_{\text{EFM}} + P_{\text{CFM}} + P_{\text{Heaters}}} = \frac{84.9296}{6.118 \times 2 + 0.75 + 0.75 + 0.45 \times 3} = 5.62$$

Chapter 6

Energy Analysis for Hydraulic Machines

6.1 Pumps

6.1.1 Introduction

Hydraulics is defined as the science of the conveyance of liquids through pipes. The pump is often used to raise water from a low level to a high level where it can be stored in a tank. Most of the theory applicable to hydraulic pumps has been derived using water as the working fluid, but other liquids can also be used. In this chapter, we will assume that liquids are totally incompressible unless otherwise specified. This means that the density of liquids will be considered constant no matter how much pressure is applied. Unless the change in pressure in a particular situation is very great, this assumption will not cause a significant error in calculations. Centrifugal and axial flow pumps are very common hydraulic pumps. Both work on the principle that the energy of the liquid is increased by imparting kinetic energy to it as it flows through the pump. This energy is supplied by the impeller, which is driven by an electric motor or some other drive.

6.1.2 Pumping system characteristics

6.1.2.1 Resistance of the system: head

Pressure is needed to pump the liquid through the system at a certain rate. This pressure has to be high enough to overcome the resistance of the system, which is also called "head". The total head is the sum of static head and friction head:

a) Static head

Figure (6.1) shows the Static head is the difference in height between the source and destination of the pumped liquid. Static head is independent of flow. The static head at a certain pressure depends on the weight of the liquid and can be calculated with this equation (6-1):

$$\text{Head (feet)} = \frac{\text{Pressure (Psi)} \times 2.31}{\text{specific grafity}} \qquad (6\text{-}1)$$

Static head consists of:

- Static suction head (h_s): resulting from lifting the liquid relative to the pump center line. The h_s is positive if the liquid level is above pump centerline, and negative if the liquid level is below pump centerline (also called "suction lift)
- Static discharge head (h_d): the vertical distance between the pump centerline and the surface of the liquid in the destination tank.

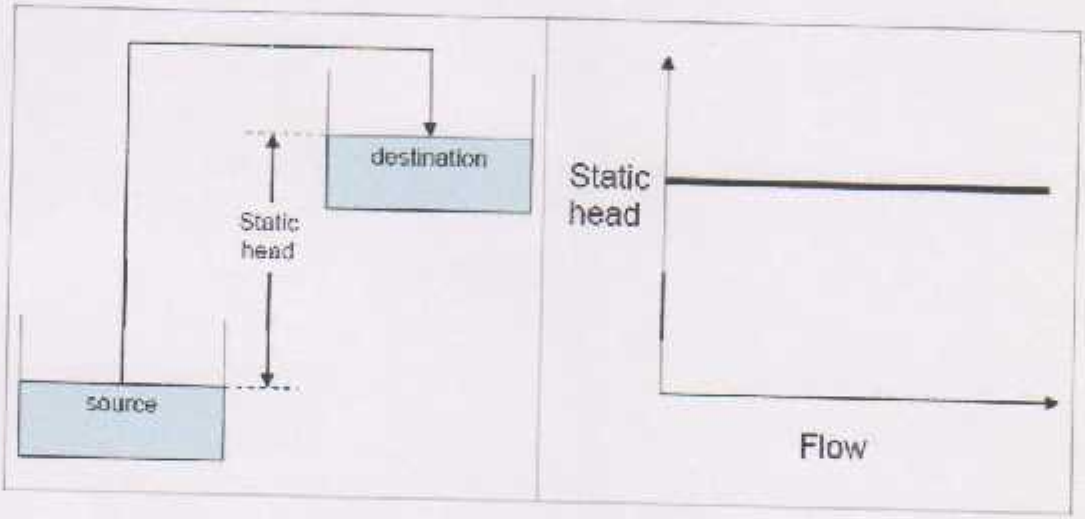


Figure (6.1) Static head

b) Friction head (h_f)

This is the loss needed to overcome that is caused by the resistance to flow in the pipe and fittings. It is dependent on size, condition and type of pipe, number and type of pipe fittings, flow rate, and nature of the liquid. The friction head is proportional to the square of the flow rate as shown in figure (6.2). A closed loop circulating system only exhibits friction head (i.e. not static head).

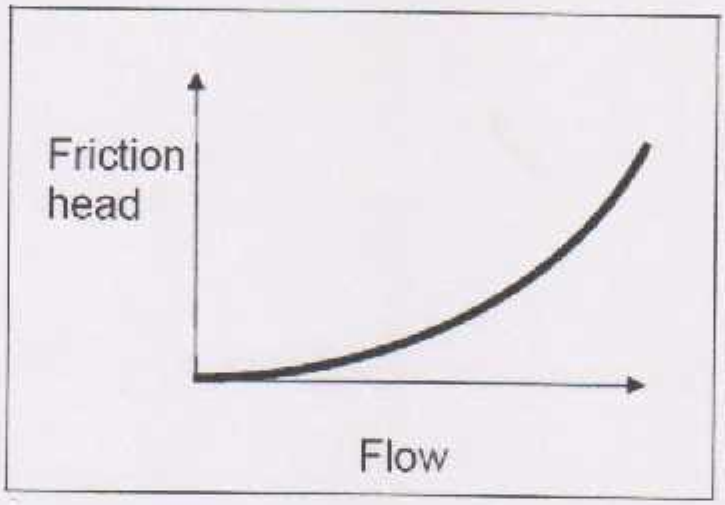


Figure (6.2) Frictional Head versus Flow

6.1.2.2 Pump performance curve

The head and flow rate determine the performance of a pump, which is graphically shown in Figure (6.3) as the performance curve or pump characteristic curve. The figure shows a typical curve of a centrifugal pump where the head gradually decreases with increasing flow. As the resistance of a system increases, the head will also increase. This in turn causes the flow rate to decrease and will eventually reach zero. A zero flow rate is only acceptable for a short period without causing to the pump to burn out.

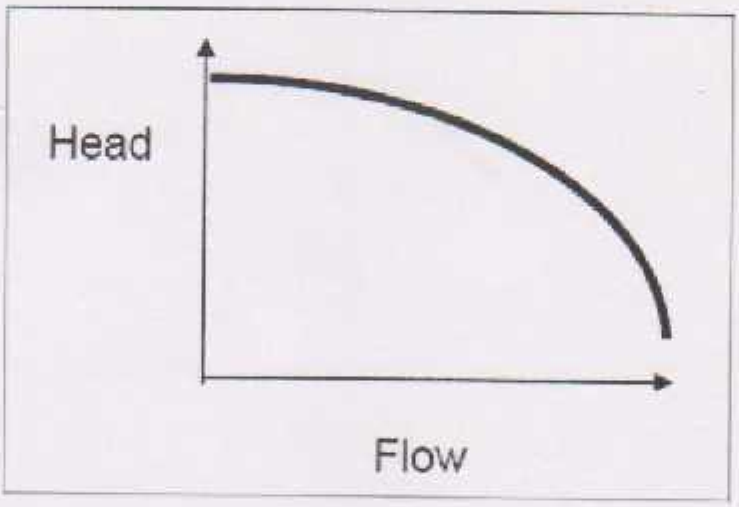


Figure (6.3) Performance curve

6.1.2.3 Pump operating point

The rate of flow at a certain head is called the duty point. The pump performance curve is made up of many duty points. The pump operating point is determined by the intersection of the system curve and the pump curve as shown in Figure (6.4).

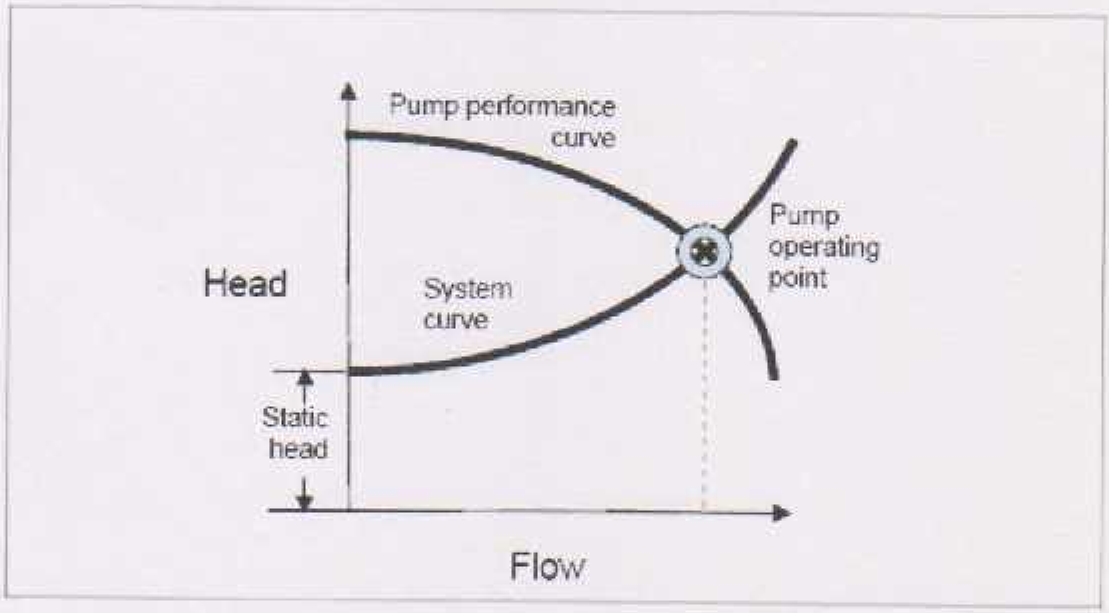


Figure (6.4) Pump operating point

6.1.2.4 Pump suction performance (NPSH)

Cavitations or vaporization is the formation of bubbles inside the pump. This may occur when at the fluid's local static pressure becomes lower than the liquid's vapor pressure (at the actual temperature). A possible cause is when the fluid accelerates in a control valve or around a pump impeller. Vaporization itself does not cause any damage. However, when the velocity is decreased and pressure increased, the vapor will evaporate and collapse. This has three undesirable effects:

- Erosion of vane surfaces, especially when pumping water-based liquids
- Increase of noise and vibration, resulting in shorter seal and bearing life

Partially choking of the impeller passages, this reduces the pump performance and can lead to loss of total head in extreme cases. The Net Positive Suction Head Available (NPSHA) indicates how much the pump suction exceeds the liquid vapor pressure, and is a characteristic of the system design. The NPSH Required (NPSHR) is the pump suction needed to avoid cavitations, and is a characteristic of the pump design.

6.1.3 TYPE OF PUMPS

This section describes the various types of pumps. Two Pumps come in a variety of sizes for a wide range of applications. They can be classified according to their basic operating principle as dynamic or positive displacement pumps as shown in Figure (6.5).

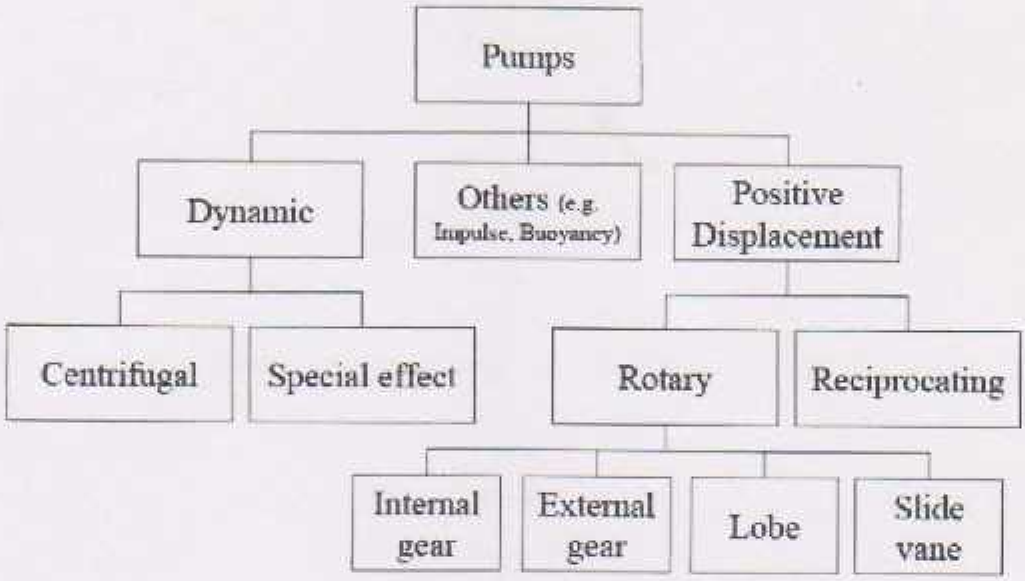


Figure (6.5) Different types of pumps

In principle, any liquid can be handled by any of the pump designs. Where different pump designs could be used, the centrifugal pump is generally the most economical followed by rotary and reciprocating pumps. Although, positive displacement pumps are generally more efficient than centrifugal pumps, the benefit of higher efficiency tends to be offset by increased maintenance costs.

6.1.3.1 Dynamic pumps

Dynamic pumps are also characterized by their mode of operation: a rotating impeller converts kinetic energy into pressure or velocity that is needed to pump the fluid. There are two types of dynamic pumps:

- Centrifugal pumps are the most common pumps used for pumping water in industrial applications. Typically, more than 75% of the pumps installed in an industry are centrifugal pumps. For this reason, this pump is further described below.

- Special effect pumps are particularly used for specialized conditions at an industrial site.

6.1.3.2 How a centrifugal pump works

- A centrifugal pump is one of the simplest pieces of equipment in any process plant. Figure (6.6) shows how this type of pump operates:
- Liquid is forced into an impeller either by atmospheric pressure or in case of a jet pumps by artificial pressure. The vanes of impeller pass kinetic energy to the liquid, thereby causing the liquid to rotate. The liquid leaves the impeller at high velocity.
- The impellers are surrounded by a volute casing or in case of a turbine pump a stationary diffuser ring. The volute or stationary diffuser ring converts the kinetic energy into pressure energy.

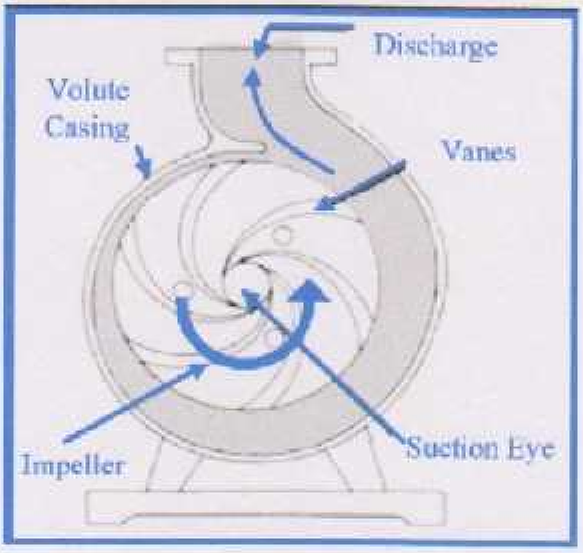


Figure (6.6) Liquid Flow Path of a Centrifugal Pump

6.1.3.3 Components of a centrifugal pump

The main components of a centrifugal pump are shown in Figure (6.7) and described below:

- Rotating components: an impeller coupled to a shaft.
- Stationary components: casing, casing cover, and bearings.

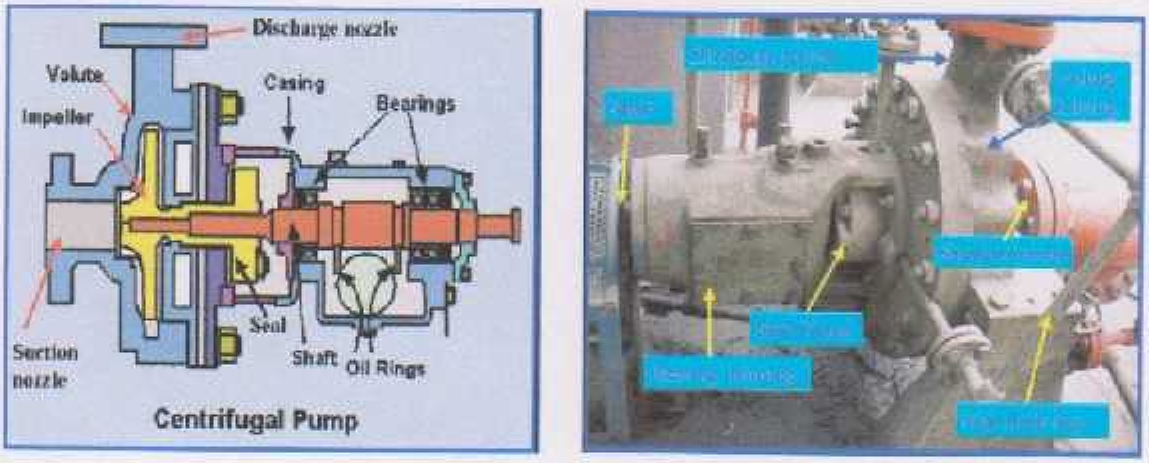


Figure (6.7) Main Components of a Centrifugal Pump

Pump output, water horsepower or hydraulic horsepower (hp) is the liquid horsepower delivered by the pump, and can be calculated as follows in equation (6-2):

Hydraulic power:

$$\text{Hydraulic Power (W)} = \rho g Q H \qquad (6\text{-}2)$$

Where:

Q = flow rate (m^3/s)

H = benefit head (m)

ρ = density of the fluid (Kg/m^3)

g = acceleration due to gravity (m/s^2)

the efficiency of the pump is calculated using the following relation (6-2):

$$\text{Pump Efficiency} = \frac{\rho g Q H}{IV} \qquad (6\text{-}3)$$

6.1.3.4 PUMP LOSSES

The following are the various losses occurring during the operation of a centrifugal pump. Losses in the suction and delivery pipe are known as hydraulic losses.

6.1.3.4.1 Theory and Analysis

In this study, it is necessary to obtain the volumetric flow rate, Q. Volumetric flow rate, put simply, is the time, t it takes for a certain volume, V to fill , and is given in units of (meters^3 / seconds). It is obtained through the use of the following equation (6-4):

$$Q = \frac{V}{t} \qquad (6\text{-}4)$$

Where:

Q: is the volumetric flow rate (m^3/s)

V: is the volume of water flow into the measuring cylinder (m^3)

t: is the time taken for the flow (sec)

The cross-sectional area, (A) area of all of the pipes is a circle, it can be obtained from the known diameter, D of each pipe using equation (6-5).

$$A = \pi \frac{D^2}{4} \qquad (6\text{-}5)$$

Where:

A: is the cross-sectional area of the pipe (m^2).

D: is the diameter of the pipe (m)

The Continuity Equation (shown below) states that the flowing velocity, V is related to the volumetric flow rate, Q and the cross-sectional area of the pipe A. The units of Velocity are in (meters / seconds) using calculation (6-6).

$$V = \frac{Q}{A} \tag{6-6}$$

Where:

V: is the flowing velocity (m/s).

Q: is the volumetric flow rate (m^3/s)

A: is the cross-sectional area of the pipe (m^2).

The Reynolds number, Re, is very useful in the determination of the friction factor, and can be determined by relating the flowing velocity, V, the fluid's density, _ (given as a lowercase Greek rho), the pipe's diameter, D, and the fluid's dynamic viscosity, μ (given by a lowercase Greek mu). The density of a fluid is a number stating how much mass can fit into a certain volume for a specific substance. In this case, the fluid is water. The dynamic viscosity explains how easily a fluid can flow when taking the friction of the walls of the pipe into account. The density and dynamic viscosity are given in the units of kilograms / meters3 and kilograms / (meters x seconds). A quick analysis of the units involved shows that they cancel each other out through division, and thusly, the Reynolds number, Re, is dimensionless. It is important to note that both density and dynamic viscosity vary with temperature. This does not apply to this case however because the temperature of the testing area was held constant substituting all values in equation (6-7).

$$Re = \frac{\rho V D}{\mu} \tag{6-7}$$

Where:

Re is the Reynolds number for the flow (non dimensional)

ρ: is the density of fluid (kg/m^3)

μ: is the dynamic viscosity of the fluid (kg/m.s)

V: is the flowing velocity (m/s)

D: is the diameter of the pipe (m).

There are two major ways to describe the flowing of a fluid. A steady, even flow is called a laminar flow, and a flow that moves violently or roughly is called a turbulent flow. The easiest way to show that a flow is laminar is to first determine the Reynolds number; if the value obtained is below 2000 then the flow is laminar, and if the value is above 4000 then the flow is turbulent. For values of the Reynolds number between 2000 and 4000, the fluid is called a transitional flow and shows characteristics of both types.

The friction factor, f, allows us to determine a variety of information regarding pipe flows. For a laminar flow only, it is determined by the Reynolds number using equation (6-8). Note that the friction factor is also dimensionless.

$$f = \frac{64}{Re} \qquad (6\text{-}8)$$

Where

f: is the friction factor.

Re: is the Reynolds number for the flow.

For turbulent flow, the friction factor, f is determined from the Moody chart (see attached at the end of this report) as a function of Reynolds number, Re and the relative roughness, r of the pipe. The relative roughness for the pipe is related to the pipe's surface roughness, _ (given by a lowercase Greek epsilon) and the diameter, D. Both values of roughness describe the ability of a fluid to flow past without "sticking" to the sides, and the friction factor, f is determined by matching the lines for relative roughness to the grid for Reynolds number and locating the value for the friction factor at the corresponding point. Although there are advanced equations that can determine the friction factor without the use of the Moody chart, it is a generally accepted method for easily determining the value.

In fluid mechanics, it is often helpful to define the head of a fluid in a system as a height of fluid in the vertical direction; this value can then be used to define more intricate values for different aspects of the flow. The head can be measured directly through the use of a *manometer*. In the case where two manometers are set up at opposing ends of a pipe, we can measure the total head loss, Δh as illustrated in equation (6-9) of the flow of the pipe. Corresponding to the experimental setup, this is merely the difference of manometer heights h1 and h2 and is measured in meters.

$$\Delta h = h_2 - h_1 \tag{6-9}$$

Where

Δh: The total head loss.

h1 is the height of the fluid in manometer 1.

h2 is the height of the fluid in manometer 2.

Although there are many reasons for the change in the size of the head, Δh, from one end of the pipe to the other, the largest factor involved is the head loss due to friction, hf. This is explained through relation with friction factor, f, the overall pipe length, L, the flow velocity, V, the acceleration due to gravity, g, and the pipe diameter, D. The units of head loss due to friction are the same as the total head loss; namely, in meters. The pipe's length is also given in meters. The acceleration due to gravity is a constant on the Earth's surface, and is given as 9.81 (meters / seconds^2). It is important to note that the pipe length is given as the length that the fluid must pass through to reach the other end; this is an important concept in analyzing pipes with bends or elbows as opposed to straight sections then the losses in the pipe due to friction can be calculated using equation (6-9).

$$h_f = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right) \tag{6-9}$$

Where:

hf :is the head loss due to friction. (m)

L: is the length of the pipe. (m)

g :is the value of acceleration due to gravity. (m/s^2)

f :is the friction factor (non dimensional)

V: is the velocity of the flow. (m/s)

D: is the diameter of the pipe. (m)

In the analysis of test sections with bends or elbows, we must add the effects of bends to the total head loss. The head loss due to bends, h_b, is also denoted in meters, and is related to the velocity, V, the acceleration due to gravity, g, and the bend resistance coefficient, K_b , which is dimensionless. The following relation (6-10) is used. Take care to remember that the velocity, V referred to in this equation is the fluid velocity determined from the volumetric flow rate, Q, and not the mean velocity, V_{mean}.

$$h_b = K_b \frac{V^2}{2g} \tag{6-10}$$

Where:

h_b :is the head loss due to the elbow/bend. (m)

K_b: is the bend resistance coefficient. (non dimensional)

V: is the velocity of the fluid. (m/s)

g: is the acceleration due to gravity. (m/s^2)

The relation used to obtain a value for bend resistance coefficient, Kb, requires several additional values, one of which is the velocity head, hv. This is another head describing the features of pipe flow, and like velocity, V it is also given in meters / seconds. This number is related to velocity and the acceleration due to gravity, g, and is given by the equation (6-11) below. Once again, the value of velocity used is the standard velocity obtained from the volumetric flow rate, Q, not the mean velocity, V_{mean}.

$$h_v = \frac{V^2}{2g} \tag{6-11}$$

Where:

hv: is the velocity head. (m)

V: is the velocity. (m/s)

g :is the acceleration due to gravity.(m/s^2).

6.1.3.5 Sample of calculation for the circular tank

$$L=2\pi r \longrightarrow r=\frac{4.67}{2\times3.14}=0.743\text{ m}$$

D=1.5 m

$$\Delta v=(\frac{\pi}{4})(1.5)^2\times0.65=1.148\ (m^3)$$

$$Q=\frac{1.148}{10\times60}=0.0019\ (m^3/sec)$$

For 10 minutes of pump operation.

$$V=\frac{4\times Q}{\pi\times D^2}=0.9383\ (m/s)$$

$$Re=\frac{\rho\, V\, D}{\mu}=\frac{1000\times0.9383\times2\times0.02547}{(1.277\times10^{-3})}=37429>4000\quad \text{Turbulent}$$

f = from Moody chart with 0.0008547 surface roughness = 0.02

$$h_f=f\times(\frac{L}{D})\times(\frac{V^2}{2g})=0.02\,(\frac{36.69}{2\times0.02547})(\frac{0.9383^2}{2\times9.81})=0.6464\ (m)$$

$$h_b=(\text{N of fittings , Elbows})\times K_b(\frac{v^2}{2g})=4\times0.9\times\frac{0.9383^2}{2\times9.81}=0.173\ (m)$$

$$h_b=(\text{N of fittings , Reducer})\times K_b(\frac{v^2}{2g})=1\times0.15\times\frac{0.9383^2}{2\times9.81}=0.00721\ (m)$$

$$h_{\text{Loss in 2'' cylindrical tank}}=0.6464+0.173+0.00721=0.82661\ (m)$$

$$PLoss=\rho gQh_{Loss}=1000\times9.81\times0.0019\times0.82661=15.4072\ (W)=0.01541\ (kW)$$

$$\text{Benefit power}=\rho gQh_d=1000\times9.81\times0.0019\times7.1234=132.773\ W$$

The pumps distribute the water to the groups of tanks according to the own head of the tank, all pipes losses discussed in Table (6.1).

Table (6.1) Pipes losses

	Diameter (m)	(Cylindrical tanks) (1 Inch diameter pipe supply line)	(Rectangular tanks) (1 Inch diameter pipe supply line)	(Main supplier) (3 Inch diameter pipe supply line)
Flow rate. (m^3/s)	**Q**	0.0019	0.0001833	0.00208
Velocity of the fluid. (m/s)	**V**	0.9383	0.09052	0.4544
Head loss due to pipe friction (m)	h_f	0.6464	0.011	0.0763
Head loss due to bend (m)	h_b	0.18021	0.00242	0.7136
Delivery head. (m)	h_d	7.1234	11.1383	1.011
Suction head. (m)	h_s	5	5	5
Power loss in pipes & elbows. (kW)	$P_{Loss\ in\ pipes}$	0.01541	0.00002413	0.016117
Benefit power .(kW)		0.13277	0.02	0.02063

Capacity of each cylindrical tank is (3) m^3

Capacity of each rectangular tank is (2) m^3

The efficiency of the pump can be determined using the characteristic curve of its operation by matching the delivered head with the flow rate as shown in Figure (6.8).

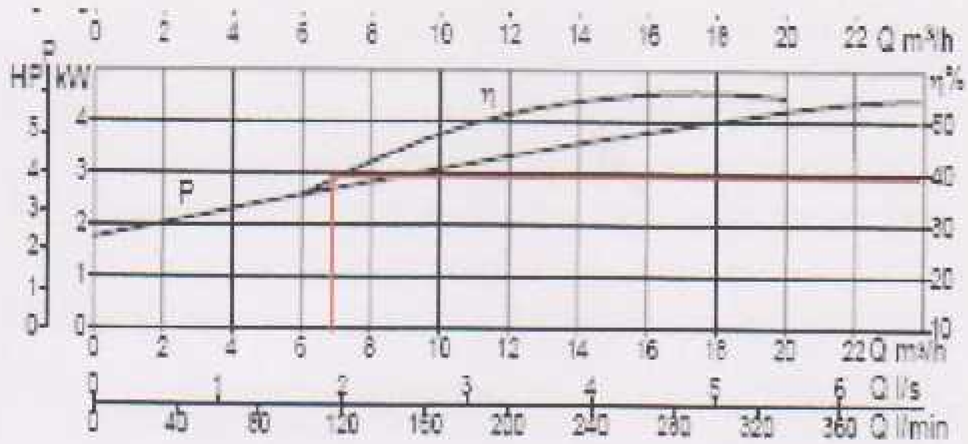


Figure (6.8) characteristic curve for Dap Pump

Flow rate = 6.84 m^3/h

Head = 11.1383 m

η= 40%

6.2 FANS AND BLOWERS

6.2.1 INTRODUCTION

Most manufacturing plants use fans and blowers for ventilation and for industrial processes that need an air flow. Fan systems are essential to keep manufacturing processes working, and consist of a fan, an electric motor, a drive system, ducts or piping, flow control devices, and air conditioning equipment (filters, cooling coils, heat exchangers, etc.). The US Department of Energy estimates that 15 percent of electricity in the US manufacturing industry is used by motors. Similarly, in the commercial sector, electricity needed to operate fan motors composes a large portion of the energy costs for space conditioning (US DOE, 1989). Fans, blowers and compressors are differentiated by the method used to move the air, and by the system pressure they must operate against. The American Society of technical Engineers (ASME) uses the specific ratio, which is the ratio of the discharge pressure over the suction pressure, to define fans, blowers and compressors .Conversely, resistance decreases as flow decreases. To determine what volume the fan will produce, it is therefore necessary to know the system resistance characteristics. In existing systems, the system resistance can be measured. In systems that have been designed, but not built, the system resistance must be calculated.

6.2.2 Important terms and definitions

Before types of fans and blowers are described it is important to first understand terms and definitions.

6.2.2.1 System characteristics

The term "system resistance" is used when referring to the static pressure. The system resistance is the sum of static pressure losses in the system. The system resistance is a function of the configuration of ducts, pickups, elbows and the pressure drops across equipment, for example bag filter or cyclone. The system resistance varies with the square of the volume of air flowing through the system. For a given volume of air, the fan in a system with narrow ducts and multiple short radius elbows is going to have to work harder to overcome a greater system resistance than it would in a system with larger ducts and a minimum number of long radius turns. Long narrow ducts with many bends and twists will require more energy to pull the air through them. Consequently, for a given fan speed, the fan will be able to pull less air through this system than through a short system with no elbows. Thus, the system resistance increases substantially as the volume of air flowing through the system increases; square of air flow.

6.2.3 Type of fans and blowers

This section briefly describes different types of fans and blowers.

6.2.3.1 Types of fans

There exist two main fan types. Centrifugal fans used a rotating impeller to move the air stream. Axial fans move the air stream along the axis of the fan.

6.2.3.1.1 Centrifugal fans

Centrifugal fans increase the speed of an air stream with a rotating impeller. The speed increases as the reaches the ends of the blades and is then converted to pressure. These fans are able to produce high pressures, which makes them suitable for harsh operating conditions, such as systems with high temperatures, moist or dirty air streams, and material handling.

6.2.3.1.2 Axial fans

Axial fans move an air stream along the axis of the fan. The way these fans work can be compared to a propeller on an airplane: the fan blades generate an aerodynamic lift that pressurizes the air. They are popular with industry because they are inexpensive, compact and light. The main types of axial flow fans (propeller, tube-axial and vane-axial).

6.2.4 Methodology of fan performance assessment

Before the fan efficiency can be calculated, a number of operating parameters must be measured, including air velocity, pressure head, temperature of air stream on the fan side and electrical motor kW input. In order to obtain correct operating figures it should be ensured that:

- Fan and its associated components are operating properly at its rated speed
- Operations are at stable condition i.e. steady temperature, densities, system resistance etc.

The calculation of fan efficiency is explained in 5 steps.

- Calculate the gas density

The first step is to calculate the air or gas density using the following equation (6-12):

$$\text{Gas Density (kg/m}^3\text{)} = \frac{273\times 1.293}{273+T\ (C^\circ)} \tag{6-12}$$

Where, t (C°) = Temperature of air or gas at site condition

- Measure the air velocity and calculate average air velocity

The air velocity can be measured with a pitot tube and a manometer, or a flow sensor (differential pressure instrument), or an accurate anemometer. The total pressure is measured using the inner tube of Pitot tube and static pressure is measured using the outer tube of Pitot tube. When the inner and outer tube ends are connected to a manometer, we get the velocity pressure (i.e. the difference between total pressure and static pressure). For measuring low velocities, it is preferable to use an inclined tube manometer instead of U-tube manometer. See the chapter on Monitoring Equipment for explanation of manometers.

- Calculate the volumetric flow

1. The third step is to calculate the volumetric flow as follows:
2. Take the duct diameter (or the circumference from which the diameter can be estimated).

Calculate the volume of air/gas in the duct by following relation (6-13):

$$\text{Volumetric Flow Rate} = \text{Velocity} \times \text{Area} \tag{6-13}$$

$$Q = V \times A$$

Where:

Q = volumetric flow rate. (m^3/s)

V = velocity of the fluid.(m/s)

A = Area of the diameter or duct (m^2)

- measure the power of the drive motor

The power of the drive motor (kW) can be measured by a load analyzer. This kW multiplied by motor efficiency gives the shaft power to the fan.

- Calculate the fan efficiency using equation (6-13).

The output power (hydraulic power) can be calculated from equation (6-2), and then the efficiency of the fan can be calculated using the characteristic curve for the fan this procedure can be done by calculating the flow rate and the dynamic head exerted by the fan.

6.2.5 Difficulties in assessing the performance of fans and blowers

In practice certain difficulties have to be faced when assessing the fan and blower performance, some of which are explained below:

- Non-availability of fan specification data: Fan specification data are essential to assess the fan performance. Most of the industries do not keep these data systematically or have none of these data available at all. In these cases, the percentage of fan loading with respect to flow or pressure can not be estimated satisfactorily. Fan specification data should be collected from the original equipment manufacturer (OEM) and kept on record.
- Difficulty in velocity measurement: Actual velocity measurement becomes a difficult task in fan performance assessment. In most cases the location of duct makes it difficult to take measurements and in other cases it becomes impossible to traverse the duct in both directions. If this is the case, then the velocity pressure can be measured in the center of the duct and corrected by multiplying it with a factor 0.9.
- Improper calibration of the pitot tube, manometer, anemometer & measuring instruments: All instruments and other power measuring instruments should be calibrated correctly to avoid an incorrect assessment of fans and blowers. Assessments should not be carried out by applying correction factors to compensate for this.
- Variation of process parameters during tests: If there is a large variation of process parameters measured during test periods, then the performance assessment becomes unreliable.

6.2.6 Determination of fans losses

The same approach that followed in studying the pumps losses are tracked by converting the non circular duct to its own equivalent diameter dimension, i.e. the duct will be processed as the pipes , using equation (6-9) to calculate the head loss h_f :

Most of the used ducts in warm air heating systems are rectangular or not circular this is due to the shape of the residences and living spaces. The following equation (6-14) can be used to collect the dimensions of the rectangular duct which can pass the same rate of flow and resulting with same pressure drop as that of the circular duct.

$$d = \frac{1.3(HW)^{0.625}}{(H+W)^{0.25}} \tag{6-14}$$

6.2.6.1 The first package unit AC

Supply line duct to the basement floor:

Sample for calculations:

Table (6-2) Sample of calculation

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.36	0.7587	6.7437	1.5303 *10^5	0.000125	0.018	0.93375	2.3179

$$d = \frac{1.3(HW)^{0.625}}{(H+W)^{0.25}} = \frac{1.3(45\times 25)^{0.625}}{(45+25)^{0.25}} = 0.36m$$

$$V = \frac{Q}{L\times W} = \frac{0.7587}{0.45\times 0.25} = 6.7437(m/s)$$

$$\mathrm{Re} = \frac{\rho\,\mathrm{V\,D}}{\mu} = \frac{1.125\times 6.7437\times 0.36}{1.7647\times 10^{-5}} = 1.5303\times 10^{5}$$

f= 0.004 from moody chart

$$h_f = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right) = 0.004\times\left(\frac{8}{0.36}\right)\left(\frac{6.7437^2}{2\times 9.81}\right) = 0.2057\text{ m}$$

$$h_v = \frac{V^2}{2g} = \frac{6.7437^2}{29.81} = 2.3179m$$

Chapter 7

Conclusion

7. Conclusion

7.1 Project Results

Auditing process includes surveying all the electrical devices exists inside the building to make the monitoring process in order to evaluate their operation, then the opportunity on the system can be checked during the efficiency calculations. After that, suitable strategy of conservation must be taken to improve the energy production of the system.

From the work plan of the project, a group of solutions have been taken to evaluate the behavior of the energized devices which is included by the building as illustrated in Table (7-1) such as:

- Setting the suitable amount of lighting levels in each space according to the recommended values based on the type of activity proceeds in it, this can save (3.6 kWh/year).
- Installing an ultrasonic occupancy sensor to turn on the light in the suitable time based on occupancy exist in the space, this procedure can save an amount of energy equal to 224.64 (kWh/year) , its cost is $37.44(\$/year)$.
- Domestic hot water used in the building can be saved by installing PLC (Programmable Logic Controller) returns the hot water again, that is usually exist inside the pipe between the boiler outlet and the fixture. Installing this system can save (199.614 kJ) of waste energy which is carried by the water inside the pipe.
- The suitable design conditions for heating and cooling requirements can save (67.173 kW) its cost in a year is approximated to.
- Depending on the acclimatization of human to the suitable conditions that is very close to the comfort point on psychometric chart i.e. rising the cooling temperature from (24 to 26 C°) .Although this temperature difference cannot be sensed directly by the human, it conserves an amount of energy equal to (81.2351kW).
- The mixing process which occurs in the theater can reduce the cooling load on the HVAC system supplies the required comfort air to the region, and keep 20.325

(kW). The process of mixing proceeds using PIC (Programmable Interfacing Controller) that depends on a special programming code to make the required mixing ratio.

- Determining the condition of operation for the pumps and fans to have their efficiencies using the characteristic cure relative to each hydraulic device.

Table (7-1) Results of Energy Conservation strategies

Strategy of conservation	Power saved (kW)	Energy saved (kWh/year)	Cost of Energy saved ($)
Setting the suitable amount of lighting levels.	0.00164835	3.6	0.6
Installing an ultrasonic occupancy sensor	0.72	1617.4	269.6
Suitable design conditions for heating and cooling requirements.	67.173	20958	3493
Rising the cooling temperature from (24 to 26 C°).	81.2351	25345.33	4224.22
The mixing process.	20.325	6341.4	1056.9
Total Conservation.	**169.455**	**54265.73**	**9044.32**
Circulation system for the boiler.	199.614 (kJ)		

7.2 Project Problems

- Missing of Data loggers as instrumentation tools to a group of variables that relates to the energy terms such as dry bulb temperature, relative humidity, electrical power consumption.
- A great number of lighting fixtures are failure in the theater. This cause incorrect lighting measurements in this region. So the auditing process cannot occur in this region as the others.
- Connection with team's center to make instrumentation for lighting in the night.

7.3 Recommendations

- Installing a control unit based on a PIC system to control and monitor the mixing operation in each unit in a single programming code on C++, since we design the mixing processor for a single HVAC unit, this system can be adopted on the other as a packaged unit.
- Installing the occupancy sensors to regulate the operation of the lighting system.
- Removing the excess number of lighting fixture in each space.
- Installing the lost and failure fixture in the suitable amount.
- Installing the PLC control system for the whole group of boilers.
- Set a temperature sensor in each conditioned space at (T_{in}= 26 C°).
- Improve the designed counter to give an alarm after a specific number of door openings. To reduce the amount of air infiltrated to the space.
- Receiving a signal from the occupancy sensor by the volume damper supplies the air to close, if they are no occupants in the conditioned space.
- Install a solar heating water system to obtain the domestic hot water, depending on the evacuated tubes technology.
- Design a special electrical board to increase the Power factor (PF).
- Install a programmable plumping system to fill the tanks regularly.

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Hyper Links

[5] http://www.arch.hku.hk/research/BEER/links.htm

[6] http://www.arch.hku.hk/research/BEER/eeba-hk.htm

[7] http://www.emsd.gov.hk/emsd/eng/pee/lceabc.shtml

Organizations and Companies

[8] Hebron Municipality, Hebron, Palestine.

[9] Petra Company, Ramallah, Palestine

Appendix A

Table (A-1) Package Unit PPH 450

Model PPH 450		**Evaporator side**	
		Power for each fan	7.5 (kW)
Cooling capacity (Ton)	450/12	Coil	Cupper tubes ,aluminum fins
Power supply (V/Ph/Hz)	(380-420)/3/50	Rows	4
Control voltage (V)	220	Fins/inch	12
Compressor	Hermetically sealed scroll	Face area (ft^2)	31.67
No. of compressors	4	Fans	Forward curve centrifugal
Power for each compressor	7.46 (kW)	No	1
		Type of drive	Belt drive
Refrigerant	R22	Total air flow (nominal) (CFM)	14400
Control	Ex. Valve	Fan motor type	Induction
Condenser side		No	1
Coil	Cupper tubes ,aluminum fins	**Air filter**	
Power for each fan	0.75 (kW)		
Rows	4	Type	Aluminum mesh
Fins/inch	12	Thickness (inch)	1
Face area (total) ft^2	60.16	Wight (approximate) lb	4610
Fans	Propeller axial		
No	2	Heaters	
Type of drive	Direct drive	No.	3 + 3
Total air flow (CFM)	31141	Power for each heater	5/6 (kW)
Total power			

Table (A-2) Ducted Split Unit

Model DSP	80
Cooling capacity	80/12 = 6.7 (Ton) = 23.33 (kW)
Power supply Ph/Hz	3 Ph/50Hz
Control voltage (V)	220-240
Compressor	Hermetically sealed type
No. of compressors	2
Condenser coil	Cupper tube, aluminum fins
Fins/in	12
No. of rows	3
Face area (total) ft^2	10.86
Tube diameter in.	3/8
Condenser fan	Propeller axial
No. of cond. Fans	1
Drive type	Direct drive
Total air f low (50Hz) CFM	5800
Refrigerant circuit	
Refrigerant type	R22
control	Expansion device

Appendix C

Table (C-1) Typical reflectance values

	Ceiling	Walls	Floor
Air Conditioned Office	0.7	0.5	0.2
Light Industrial	0.5	0.3	0.1
Heavy Industrial	0.3	0.2	0.1

Table (C-2) Typical LLF Values

Air Conditioned Office	0.8
Clean Industrial	0.7
Dirty Industrial	0.6

Table (C-3) Room Utilization Factor

Room utilisation factor										
Luminaries ceiling mounted Reflectances ρ										
Ceiling	0.8	0.8	0.8	0.5	0.5	0.8	0.8	0.5	0.5	0.3
Wall	0.8	0.5	0.3	0.5	0.3	0.8	0.3	0.5	0.3	0.3
Surface	0.3	0.3	0.3	0.3	0.3	0.1	0.1	0.1	0.1	0.1
Room factor k	Room utilisation factor in %									
0.6	73	48	37	44	36	68	38	42	35	35
0.8	82	57	47	54	46	74	45	51	44	44
1.0	91	66	56	62	54	80	53	59	52	51
1.25	98	75	65	70	62	85	61	66	60	59
1.5	103	82	73	76	69	89	67	72	66	65
2.0	109	91	82	84	78	94	75	78	73	72
2.5	114	98	90	90	84	97	81	83	79	77
3.0	117	103	96	95	90	99	86	87	83	82
4.0	120	109	103	100	95	101	91	91	88	86
5.0	122	113	107	103	98	103	93	93	91	89

Table (C-4) Room Index Tables

1- Basement floor

Region #	Length (m)	Width (m)	High (m)	Area (m^2)	Room Index (RI)
Open Area	26	11.09	2.55	288.34	3.0487
A	5.137	1.4	2.55	7.1918	0.4314
B	4.428	2.3	2.55	10.1844	0.5936
C	4.96	1.86	2.55	9.2256	0.5305
D	4.53	1.7	2.55	7.701	0.3777
E	6.38	1.417	2.15	9.0405	0.5393
G	13.64	5.491	1.75	74.8972	2.237
Path room #1	7.15	4.87	2.55	32.89	0.847
Path room #2	7.15	4.87	2.55	32.89	0.847
F	6.38	1.417	2.15	9.0405	0.5393
3	2.5	1.15	2.55	2.875	0.2712
Room #1	5.25	2.10	2.66	11.025	0.8287
Room #2	3.33	2.18	2.66	7.2594	0.7279
Room #3	4.33	3.35	2.66	14.5055	1.044
Room #4	3.34	3	2.66	10.02	0.8732
distributor	1.24	1.18	2.66	1.462	0.3338

2- Ground floor

Region #	Length (m)	Width (m)	High (m)	Area (m^2)	Room Index (RI)
A	27.34	15.9	4.8	434.766	2.094
B	21.8	4.87	2.55	106.166	1.561
C	10.82	8.03	2.55	86.8846	1.8075
D	5.6	3.46	2.55	19.376	0.8387
E1	5.89	3.475	2.35	20.4677	0.93
E2	4.94	3.33	2.35	16.452	0.846
F1	6.38	1.417	2.15	9.0405	0.5393
F2	6.38	1.417	2.15	9.0405	0.5393
G1	13.53	4.05	2.7	54.7965	1.1544
G2	6.08	3.05	2.7	18.554	0.7522
Library	13.95	7.3	3.15	101.835	1.52
Shipping room	7.95	4.85	3.15	38.5575	0.956
1	4.45	2.45	3.15	10.9025	0.5

2	3.8	2.4	3.15	9.12	0.4469
2'	2.9	2.05	3.15	5.945	0.38
Kitchen	2.4	2.05	3.15	4.92	0.351
3	4	4	3.15	16	0.635
4	3.96	3.96	3.15	7.922	0.6286
5	4.7	1.3	7.45	6.11	0.1367
5'	4.7	1.35	6.3	6.345	1.666
5''	4.7	1.35	4.15	6.345	0.2527
Supervisor Staff	7.3	4.9	3.15	35.77	0.931
Administration	7.3	5.2	3.15	37.96	0.964

3- First floor

Region #	**Length (m)**	**Width (m)**	**High (m)**	**Area (m^2)**	**Room Index (RI)**
Computer room	13.95	7.3	3.15	101.835	1.5213
Supervisor Staff	5.7	3.35	3.15	19.095	0.6698
Administration	4.95	4.5	3.15	22.275	0.7483
K1	3.45	0.95	3.15	3.2775	0.2365
K2	5.5	1.95	3.15	10.725	0.457
K3	3.7	2.3	3.15	8.51	0.45
K4	4.25	2.3	3.15	9.775	0.412
S	4.6	4.35	3.15	20.01	0.709
Multipurpose Hall	10.2	7.3	3.15	74.46	1.35
Kitchen	2.02	1.7111	3.15	3.456	0.294

Table (C-5) Information tables

1- Basement floor

Region #	Power (w) For each fitting	Utilization Factor (UF)	Light Loss Factor (LLF)	Flux (Lumen)
Open Area	4*18 2*36 1*36	1.03	0.8	4200 5000 2500
A	2*36	0.37	0.8	5000
B	2*36	0.46	0.8	5000
C	2*36	0.42	0.8	5000
D	2*36	0.3375	0.8	5000
E	2*36	0.42	0.8	5000
G	1*75	0.961	0.8	940
Path room #1	1*36 1*18	0.59115	0.8	2500 1050
Path room #2	1*36 1*18	0.59115	0.8	2500 1050
F	2*36	0.42	0.8	5000
3	1*36	0.27818	0.8	2500
Room #1	1*75	0.583	0.8	930
Room #2	1*75	0.53	0.8	930
Room #3	1*75 1*100	0.6758	0.8	930 1100
Room #4	1*75	0.6029	0.8	930
distributor	1*75	0.31359	0.8	930

2- Ground floor

Region #	Power (w) For each fitting	Utilization Factor (UF)	Light Loss Factor (LLF)	Flux (Lumen)
A	1*100 1*20 1*60	0.9156	0.8	1100 125 710
B	1*100 1*60	0.83098	0.8	1100 710
C	1*60 1*100	0.87535	0.8	710 1100
D	1*100	0.588	0.8	1100
E1	1*36 1*18	0.6285	0.8	2500 1050
E2	1*36 1*100 1*18	0.5907	0.8	2500 1100 1050
F1	1*36	0.427	0.8	2500
F2	2*36	0.427	0.8	5000
G1	4*18	0.7155	0.8	4200
G2	4*18	0.54371	0.8	4200
Library	2*36	0.8236	0.8	5000
Shipping room	2*36	0.6402	0.8	5000
1	2*36	0.405	0.8	5000
2	2*36	0.3757	0.8	5000
2'	2*36	0.339	0.8	5000
Kitchen	2*36	0.32305	0.8	5000
3	2*36	0.47925	0.8	5000
4	2*36	0.47573	0.8	5000
5	2*36	0.2052	0.8	5000
5'		0.80312	0.8	
5"		0.268985	0.8	
Supervisor Staff	2*36	0.62895	0.8	5000
Administration	2*36	0.6438	0.8	5000

3- **First floor**

Region #	Power (w) For each fitting	Utilization Factor (UF)	Light Loss Factor (LLF)	Flux (Lumen)
Computer room	2*36	0.823834	0.8	5000
Supervisor Staff	2*36	0.49839	0.8	5000
Administration	2*36	0.541565	0.8	5000
K1	2*36	0.26	0.8	5000
K4		0.3434	0.8	
K2	2*36	0.31865	0.8	5000
K3	2*36	0.3775	0.8	5000
S	2*36	0.51995	0.8	5000
Multipurpose Hall	2*36	0.778	0.8	5000
Kitchen	2*36	0.2917	0.8	5000

Table (C-6) Number of fitting in each space

1- **Basement floor**

Region #	Power (w) For each fitting	Actual No. of light	Recommended No. of Light	Recommended Flux (fc)	Measured Flux (fc)		
					Min	Max	Ave
Open Area	4*18 2*36 1*36	28 3 2	11.4 1 1.4	15	21.3	40	26.643
A	2*36	1	0.7	15	37.6	37.8	37.7
B	2*36	1	0.7846	10	29.5	30.6	30.05
C	2*36	1	0.8866	15	23.7	29.2	26.45
D	2*36	1	0.6138	10	26	28.4	27.2
E	2*36	1	0.5792	10	30.3	30.7	30.5
G	1*75	18	11.1556	10	17.5	26.1	25
Path room #1	1*36 1*18	6 3	2.9944 3.5647	15	13.4	24.3	20
Path room #2	1*36 1*18	6 3	2.9944 3.5647	15	15.2	25.1	25.7
F	2*36	1	0.5792	10	18.6	30.7	24.65

3	1*36	1	0.5562	10	9.5	22.3	15.9
Room #1	1*75	1	5.472	20	4.2	5.266	7.1
Room #2	1*75	1	3.963	20	11.7	11.7	11.7
Room #3	1*75 1*100	1 1	3.105 2.6254	20	8.2	10.75	14.4
Room #4	1*75	2	4.8089	20	8.9	10.02 5	12
distributor	1*75	1	1.349	10	10	10	10

2- **Ground floor**

Region #	**Power (w) For each fitting**	**Actual No. of light**	**Recommended No. of Light**	**Recommended Flux (fc)**	**Measured Flux (fc)**		
					Min.	**Max.**	**Ave.**
A	1*100 1*20 1*60	48 32 8	95 55.6 24.5	30	0.5	12	2.623
B	1*100 1*60	11 2	39.6699 11.174	30	1.5	11	5.05
C	1*60 1*100	7 10	23.235 21.425	30	2.1	16.3	7.988
D	1*100	3	4.8	30	12.3	18	15.62
E1	1*36 1*18	5 3	1.0954 1.5649	10	16.2	37	27.2
E2	1*36 1*100 1*18	2 1 3	0.4992 0.312 1.7844	10	19.2	25.5	22.23
F1	1*36	1	0.8237	10	14.2	14.2	14.2
F2	2*36	1	0.569	10	15	15	15
G1	4*18	10	2.45	10	20.7	28.4	24.52
G2	4*18	2	1.09	10	25.2	32.6	28.367
Library	2*36	10	9.9819	30	26	38.8	34.36
Shipping room	2*36	6	4.8621	30	30.2	34	31.92
1	2*36	1	0.7244	10	11.8	13.566	15.
2	2*36	1	0.6532	10	20.4	21	20.7
2'	2*36	1	0.4719	10	16.6	16.6	16.6
Kitchen	2*36	1	0.8197	20	11	11	11
3	2*36	1	0.8984	10	2.5	3.5	3
4	2*36	1	0.448	10	3.6	3.6	3.6
5	2*36	1	0.8013	10	3	3	3
5'			0.2126		4	4	4
5''			0.6348		1.1	1.1	1.1
Supervisor Staff	2*36	6	4.5913	30	24	36	31.6
Administration	2*36	6	4.76	30	37	45.9	42.1

3- First floor

Region #	Power (w) For each fitting	Actual No. of light	Recommended No. of Light	Recommended Flux (fc)	Measured Flux (fc)		
					Min.	Max.	Ave.
Computer room	2*36	10	6.6527	20	12.5	25	22
Supervisor Staff	2*36	3	3.093	30	17.5	29.5	24.83
Administration	2*36	2	3.3205	30	20.1	24	22.36
K1	2*36	1	0.3392	10	19.56	20.5	20.03
K4			0.766	10			
K2	2*36	1	0.9029	10	26	26	26
K3	2*36	2	0.9099	15	36	36	36
S	2*36	1	1.0356	10	9	10	9.5
Multipurpose Hall	2*36	8	7.7263	30	24.5	33.3	29.46
Kitchen	2*36	1	0.6376	20	22.8	26.7	24.75

Total power for flux measured is 13.8705 w

Total power for recommended flux is 12.2333 w

Total power saved is 1.6372

Total saved for one year is 1.6372*7*26*12=3557.6448

=3.5576448*.65 NIS

=2.32 NIS saved for one year

Appendix D

$$U = \frac{1}{(\frac{1}{h_{f,in}}) + (\sum_{i=1}^{n} \frac{X_i}{k_i}) + (\frac{1}{h_{f,out}})}$$

Calculations for overall heat transfer coefficient:

1. **Theater ceiling Construction (first floor):**

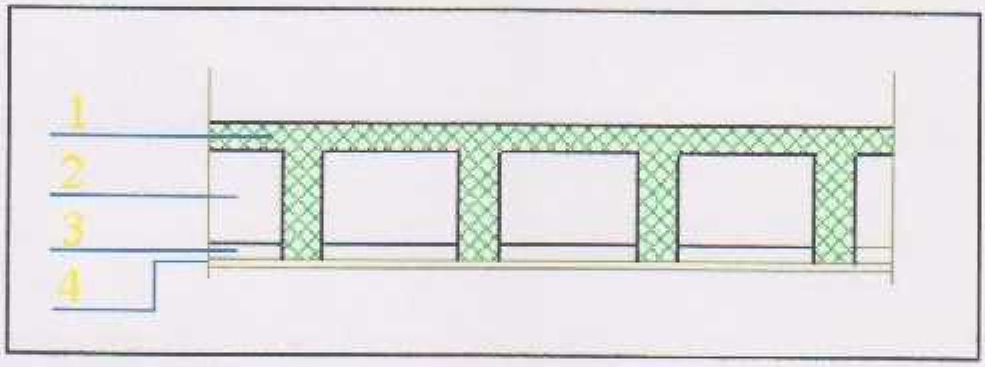

Figure (D-1) Theater ceiling Construction (first floor)

Table D-1 Theater ceiling Construction (first floor)

No.	Construction Material	t (m)	K (w/m.C°)
1	Concrete	0.08	1.48
2	طوب كلكل اسمنتي خفيف	0.27	0.38
3	Concrete block	0.05	0.95
4	plaster	0.02	0.72

$U_{Ceiling\ 1} = 1 / [(1/h_{f,in}) + (\sum(xi/k_i)) + (1/h_{f,out})]$

$= 1/ [(1/22.7)+(1/9.37)+(0.08/1.48)+(0.27/0.38)+(0.05/0.95)+(0.02/0.72)]$

$= 1.004\ (W/m^2.C°)$

$U_{Ceiling\,2} = 1 / [(1/h_{f,in}) + (\sum(xi/k_i)) + (1/h_{f,out})]$

$= 1/ [(1/22.7)+(1/9.37)+(0.08/1.48)+(0.02/0.72)]$

$= 1.9\ (W/m^2.C^\circ)$

$$U = \frac{U_1A_1 + U_2A_2}{2} = \frac{1.004\times0.4+4.299\times0.15}{0.4+0.15} = 1.9(W\ /m^2C^\circ)$$

2. **Ceiling Construction of the basement floor:**

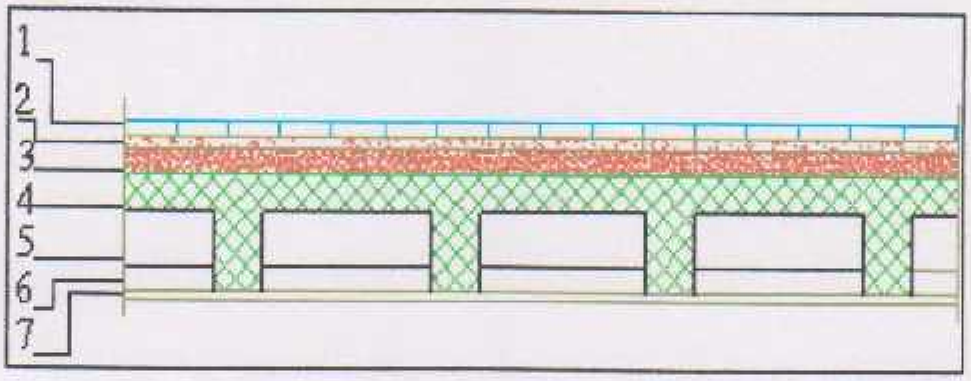

Figure (D-2) Ceiling Construction of the basement floor

Table D-2 Ceiling Construction of the basement floor

No.	Construction Material	t (m)	K (w/m.C°)
1	Terrazzo Tiles	0.03	1
2	Mortar	0.03	1.4
3	Sand fill	0.05	0.3
4	Concrete	0.08	1.48
5	طوب كلكل	0.12	0.38
6	Concrete block	0.05	0.95
7	Plaster	0.02	0.72

$U1 = 1 / [(1/h_{f,in}) + (\sum(xi/k_i)) + (1/h_{f,out})]$

$= 1/ [(2/9.37)+(0.03/1.8)+(0.05/0.3)+(0.08/1.48)+(0.12/0.38)+(0.05/0.95)+(0.02/0.75)]$

$= 1.18\ (W/m^2.C^\circ)$.

$U2 = 1 / [(1/h_{f,in})+(\sum(xi/k_i))+(1/h_{f,out})]$

$= 1/ [(2/9.37)+(0.03/1.8)+(0.05/0.3)+(0.08/1.48)+(0.02/0.75)]$

$= 2.09\ (W/m^2.C^o)$.

$$U = \frac{1.18\times0.4+2.09\times0.15}{0.4+0.15} = 1.42(W/m^2C^o)$$

3. Basement floor construction:

Table D-3 Basement floor construction

No.	Construction Material	t (m)	K (w/m.C°)
1	Terrazzo Tiles	0.03	1
2	Mortar	0.03	1.4
3	Sand fill	0.05	0.3
4	R.C slab	0.1	1.48
5	Core building	0.05	1.4
6	Hard core	0.15	1.5
7	Compacted fill	2	1.4

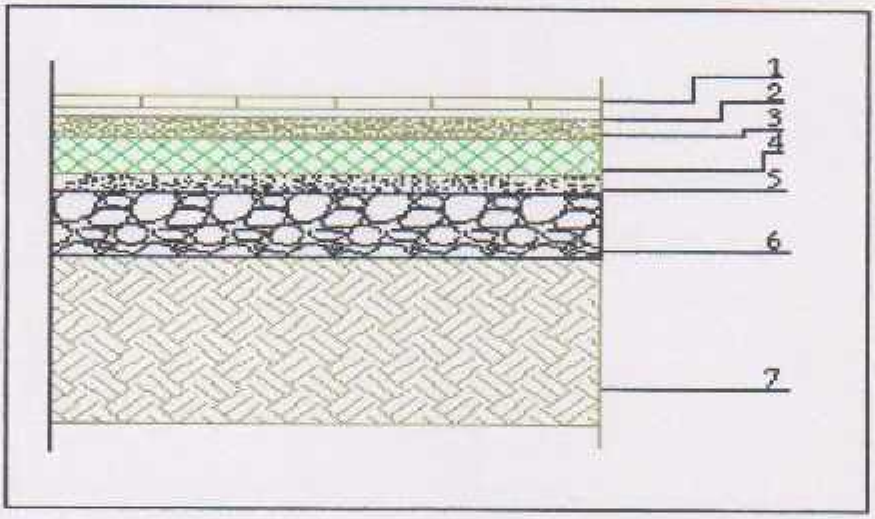

Figure (D-3) Basement floor construction

$$U_{\text{Basement Floor Construction}} = \frac{1}{(\frac{1}{9.37})+(\frac{0.03}{1})+(\frac{0.03}{1.4})+(\frac{0.05}{0.3})+(\frac{0.1}{1.48})+(\frac{0.05}{1.4})+(\frac{2}{1.4})+(\frac{0.15}{1.5})}$$

$$= 0.511\ (W/m^2.C^\circ).$$

Theater Steps:

Table D-4 Theater Steps

No.	Construction Material	t (m)	K (w/m.C°)
1	plaster	0.02	0.72
2	concrete	0.25	1.48

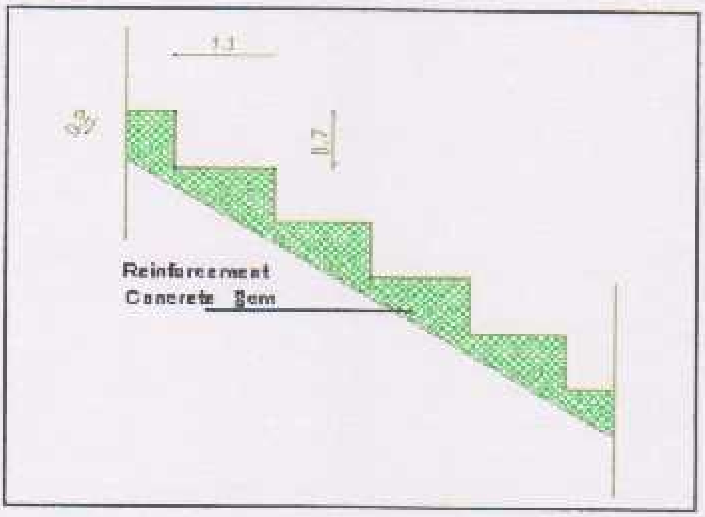

Figure (D-4) Theater Steps

$$U_{Steps} = \frac{1}{(\frac{1}{h_{f,in}}) + (\sum_{i=1}^{n} \frac{X_i}{k_i}) + (\frac{1}{h_{f,out}})}$$

$$= \frac{1}{(\frac{2}{9.37}) + (\frac{0.02}{0.72}) + (\frac{0.25}{1.48})}$$

$= 2.438\ (W/m^2.C^\circ)$.

Steps filled with concrete

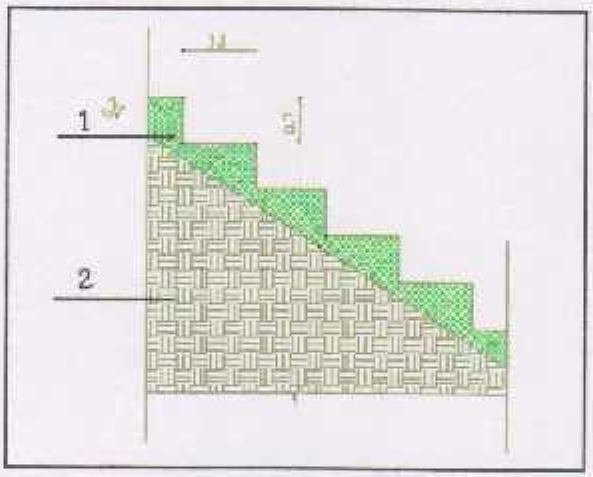

Figure (D-5) Steps filled with concrete

Table D-5 Steps filled with concrete

No.	Construction Material	t (m)	K (w/m.C°)
1	concrete	0.25	1.48
2	Compacted fill	2	1.4

$$U_{steps\ (filled\ by\ hard\ core)} = \frac{1}{(\frac{1}{h_{f,in}}) + (\sum_{i=1}^{n} \frac{X_i}{k_i})}$$

$$= \frac{1}{(\frac{1}{9.37}) + (\frac{0.25}{1.48}) + (\frac{2}{1.4})}$$

$$= 0.586\ (W/m^2.C^\circ).$$

4. Out side wall behind the street level:

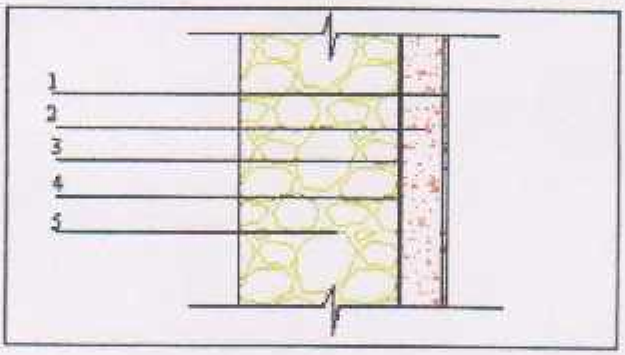

Figure (D-6) Out side wall behind the street level

Table D-6 Out side wall behind the street level

No.	Construction Material	t (m)	K (w/m.C°)
1	Plaster	0.02	0.72
2	Concrete	0.25	1.48
3	Mastic Asphalt (اسفلت) زفتة	0.002	0.1
4	Wood (قشرة خشب)	0.005	0.14
5	Compacted fill (طمم)	2	1.4

$$U_{\text{Out side wall behind the street level}} = \frac{1}{(\frac{1}{9.37}) + (\frac{0.02}{0.72}) + (\frac{0.25}{1.48}) + (\frac{0.002}{0.1}) + (\frac{0.005}{0.14}) + (\frac{2}{1.4})}$$

$$= 0.559\ (W/m^2.C^\circ).$$

5. Out side wall above the street level:

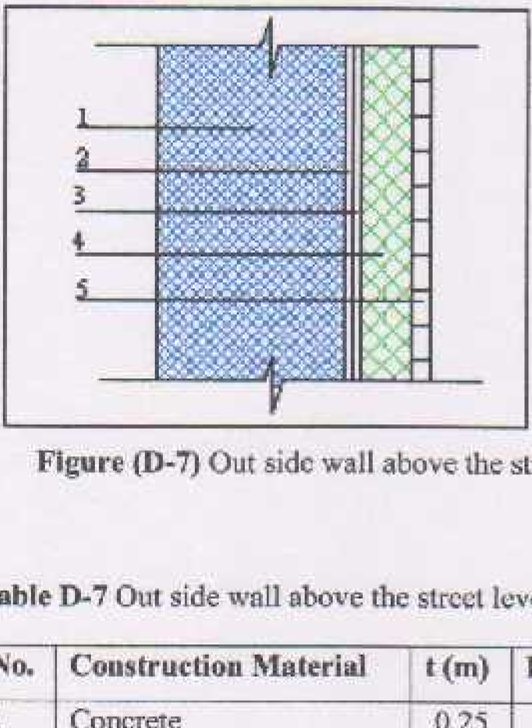

Figure (D-7) Out side wall above the street level

Table D-7 Out side wall above the street level

No.	Construction Material	t (m)	K (w/m.C°)
1	Concrete	0.25	1.48
2	Plaster	0.02	0.72
3	(سمسمية +اسمنت + رمل)	0.02	0.53
4	Concrete	0.1	1.48

5	stone	0.05	2.2

$$U_{\text{Out side wall above the street level}} = \frac{1}{(\frac{1}{9.37}) + (\frac{1}{22.7}) + (\frac{0.25}{1.48}) + (\frac{0.02}{0.72}) + (\frac{0.02}{0.53}) + (\frac{0.1}{1.48}) + (\frac{0.05}{2.2})}$$

$$= 2.103 \ (W/m^2.C^\circ).$$

6. **Internal walls between small rooms :**

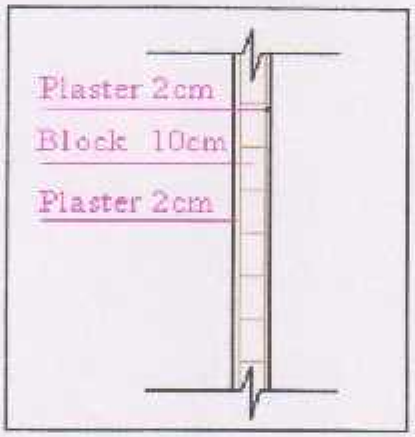

Figure (D-8) Internal walls between small rooms

Table D-8 Internal walls between small rooms

No.	Construction Material	t (m)	K (w/m.C°)
1	Block	0.1	0.77
2	Plaster	0.02	0.72

$$U_{\text{Internal walls between small rooms}} = \frac{1}{(\frac{2}{9.37}) + (\frac{0.02}{0.72}) + (\frac{0.1}{0.77})}$$

$$= 2.69 \ (W/m^2.C^\circ).$$

7. **Doors for small rooms:**

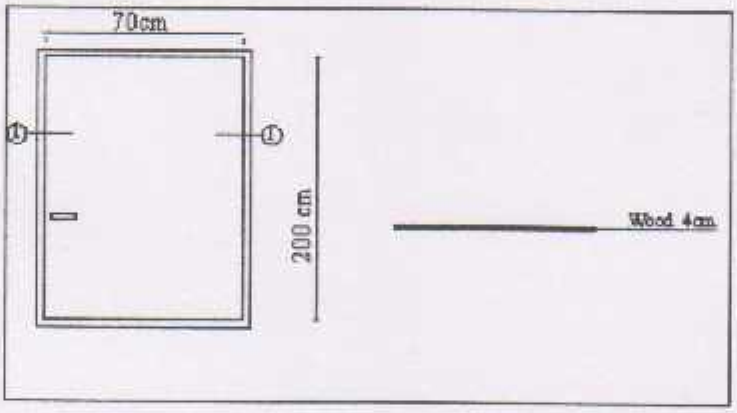

Figure (D-9) Doors for small rooms

Table D-9 Doors for small rooms

No.	Construction Material	t (m)	K (w/m.C°)
1	Wood	0.04	0.17

$$U_{\text{Doors for small rooms}} = \frac{1}{(\frac{2}{9.37}) + (\frac{0.04}{0.17})}$$

$$= 2.228 \ (W/m^2.C^\circ).$$

8. Glass windows and doors:

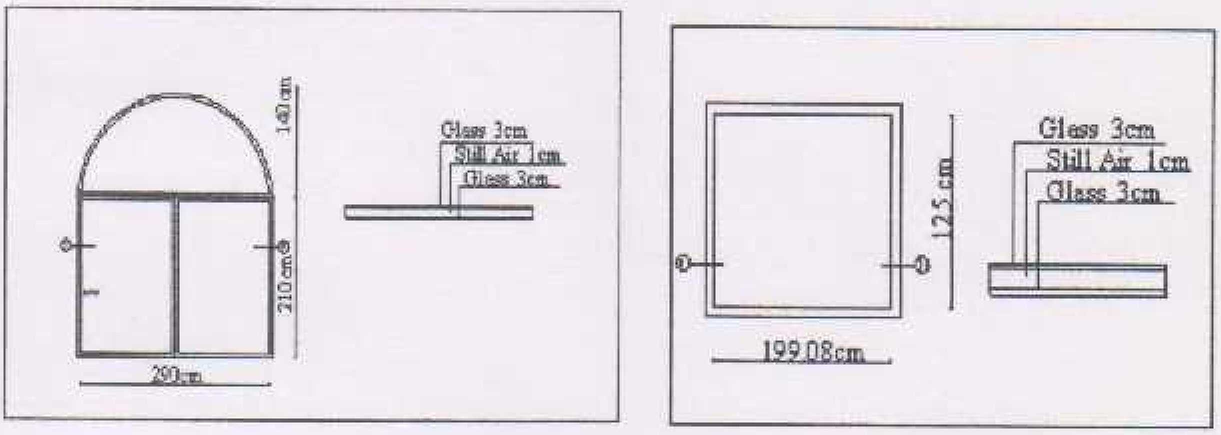

Figure (D-10) Glass windows and doors

Table D-10 Glass windows and doors

No.	Construction Material	t (m)	K (w/m.C°)
1	Glass	0.003	0.8
2	Still air	0.01	0.026

$$U_{\text{Double glass}} = \frac{1}{(\frac{1}{9.37}) + 2\times(\frac{0.003}{0.8}) + (\frac{0.01}{0.026}) + (\frac{1}{22.7})}$$

$$= 1.842 \ \ (W/m^2.C^o).$$

Columns in foyer space:

Table D-11 Columns in foyer space

No.	Construction Material	t (m)	K (w/m.C°)
1	Plaster	0.02	0.72
2	Concrete	0.6	1.48
3	Stone	0.05	2.2

$$U_{\text{Columns in foyer space}} = \frac{1}{(\frac{1}{9.37}) + (\frac{0.05}{2.2}) + (\frac{0.6}{1.48}) + (\frac{0.02}{0.72}) + (\frac{1}{22.7})}$$

$$= 1.648 \ (W/m^2.C°).$$

Theater stage (wood):

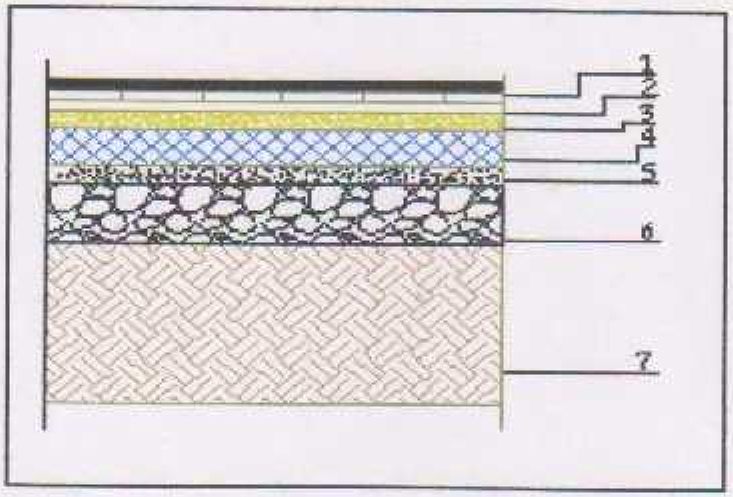

Figure (D-12) Theater stage (wood)

Table D-12 Theater stage (wood)

U_{g5}	M	t (m)	K (w/m.C)
1	Hard core	0.15	1.5
2	Compacted fill	2	1.4
3	Core building	0.05	1.4
4	R.C. Slab	0.1	1.48
5	Sand fill	0.05	0.3
6	Mortar	0.03	1.4
7	Terrazzo tiles	0.03	1
8	wood	0.05	0.065

$$U_{\text{Basement Floor Construction}} = \frac{1}{(\frac{1}{9.37})+(\frac{0.03}{1})+(\frac{0.03}{1.4})+(\frac{0.05}{0.3})+(\frac{0.1}{1.48})+(\frac{0.05}{1.4})+(\frac{2}{1.4})+(\frac{0.15}{1.5})+(\frac{0.05}{0.065})}$$

$$= 0.36685 \ (W/m^2.C^{\circ}).$$

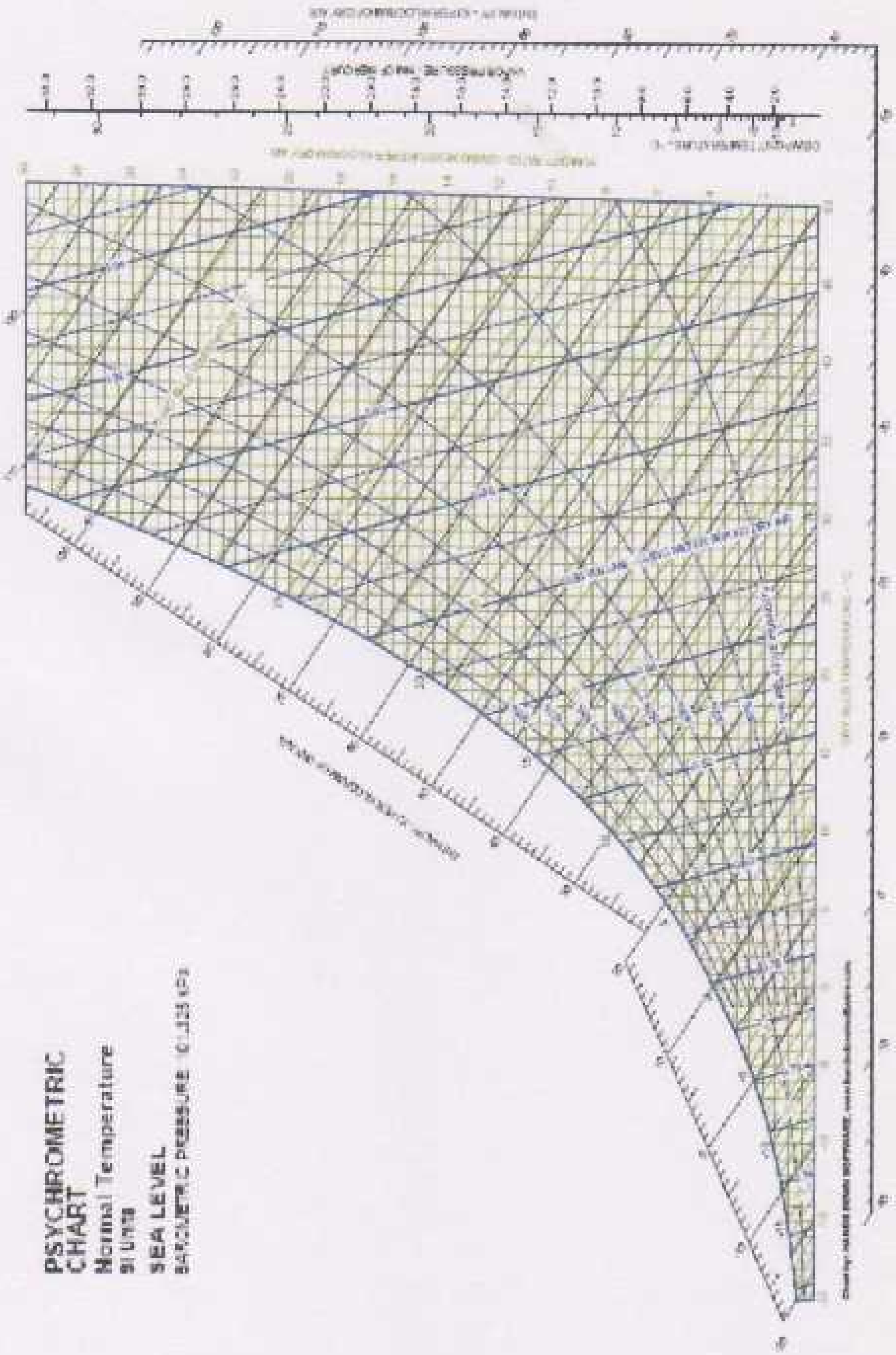

Figure (D-13) Psychometric Chart

Table D-13 Property values of metals

Metal	Properties at 20 °C				Thermal conductivity k, W/m · °C									
	ρ kg/m³	c_p kJ/kg · °C	k W/m · °C	$\alpha \times 10^5$ m²/s	−100 °C −148 °F	0 °C 32 °F	100 °C 212 °F	200 °C 392 °F	300 °C 572 °F	400 °C 752 °F	600 °C 1112 °F	800 °C 1472 °F	1000 °C 1832 °F	1200 °C 2192 °F
Aluminum														
Pure	2,707	0.896	204	8.418	215	202	206	215	228	249				
Al-Cu (Duralumin), 94–96% Al, 3–5% Cu, trace Mg	2,787	0.883	164	6.676	126	159	182	194						
Al-Si (Silumin, copper bearing), 86.5% Al, 1% Cu	2,659	0.867	137	5.933	119	137	144	152	161					
Al-Si (Alusil), 78–80% Al, 20–22% Si	2,627	0.854	161	7.172	144	157	168	175	178					
Al-Mg-Si, 97% Al, 1% Mg, 1% Si, 1% Mn	2,707	0.892	177	7.311		175	189	204						
Lead	11,373	0.130	35	2.343	36.9	35.1	33.4	31.5	29.8					
Iron														
Pure	7,897	0.452	73	2.034	87	73	67	62	55	48	40	36	35	36
Wrought iron, 0.5% C	7,849	0.46	59	1.626		59	57	52	48	45	36	33	33	33
Steel (C max ≈ 1.5%)														
Carbon steel														
C ≈ 0.5%	7,833	0.465	54	1.474		55	52	48	45	42	35	31	29	31
1.0%	7,801	0.473	43	1.172		43	43	42	40	36	33	29	28	29
1.5%	7,753	0.486	36	0.970		36	36	36	35	33	31	28	28	29

Table D-14 Heat gain from occupants in wall person

Type of Activity	Typical Application	Total Heat Dissipation Adult Male	Total Adjusted[a] Heat Dissipation	Sensible Heat	Latent Heat
Seated at rest	*Theater*				
	Matinee	111.5	94.0	64.0	30.0
	Evening	131.5	100.0	70.0	30.0
Seated, very light work	Offices, hotels, apartments, restaurants	128.5	114.0	70.0	44.0
Moderately active office work	Offices, hotels, apartments	135.5	128.5	71.5	57.0
Standing, light work, walking	Department store, retail store	157.0	143.0	71.5	71.5
Walking, seated	Drug store	157.0	143.0	71.5	71.5
Standing, walking slowly	Bank	157.0	143.0	71.5	71.5
Sedentary work	Restaurant	168.5	157.0	78.5	78.5
Light bench work	Factory	238.0	214.0	78.0	136.0
Moderate work	Small-Parts assembly	257.0	243.0	87.0	156.0
Moderate dancing	Dance hall	257.0	243.0	87.0	156.0
Walking, 1.5 m/s	Factory	286.0	285.0	107.0	178.0
Bowling (participant)	Bowling alley	428.5	414.0	166.0	248.0
Heavy work	Factory	428.5	414.0	166.0	248.0

(a) Adjusted heat dissipation is based on the percentage of men, women and children for the application.

Table D-15 Properties of water (saturated liquid)

Note: $Gr_x\, Pr = \left(\frac{g\beta\rho^2 c_p}{\mu k}\right) x^3 \Delta T$

°F	°C	c_p kJ/kg · °C	ρ kg/m³	μ kg/m · s	k W/m · °C	Pr	$\frac{g\beta\rho^2 c_p}{\mu k}$ 1/m³ · °C
32	0	4.225	999.8	1.79×10^{-3}	0.566	13.25	
40	4.44	4.208	999.8	1.55	0.575	11.35	1.91×10^{9}
50	10	4.195	999.2	1.31	0.585	9.40	6.34×10^{9}
60	15.56	4.186	998.6	1.12	0.595	7.88	1.08×10^{10}
70	21.11	4.179	997.4	9.8×10^{-4}	0.604	6.78	1.46×10^{10}
80	26.67	4.179	995.8	8.6	0.614	5.85	1.91×10^{10}
90	32.22	4.174	994.9	7.65	0.623	5.12	2.48×10^{10}
100	37.78	4.174	993.0	6.82	0.630	4.53	3.3×10^{10}
110	43.33	4.174	990.6	6.16	0.637	4.04	4.19×10^{10}
120	48.89	4.174	988.8	5.62	0.644	3.64	4.89×10^{10}
130	54.44	4.179	985.7	5.13	0.649	3.30	5.66×10^{10}
140	60	4.179	983.3	4.71	0.654	3.01	6.48×10^{10}
150	65.55	4.183	980.3	4.3	0.659	2.73	7.62×10^{10}
160	71.11	4.186	977.3	4.01	0.665	2.53	8.84×10^{10}
170	76.67	4.191	973.7	3.72	0.668	2.33	9.85×10^{10}
180	82.22	4.195	970.2	3.47	0.673	2.16	1.09×10^{11}
190	87.78	4.199	966.7	3.27	0.675	2.03	
200	93.33	4.204	963.2	3.06	0.678	1.90	
220	104.4	4.216	955.1	2.67	0.684	1.66	
240	115.6	4.229	946.7	2.44	0.685	1.51	
260	126.7	4.250	937.2	2.19	0.685	1.39	
280	137.8	4.271	928.1	1.98	0.685	1.24	
300	148.9	4.296	918.0	1.86	0.684	1.17	
350	176.7	4.371	890.4	1.57	0.677	1.02	
400	204.4	4.467	859.4	1.36	0.665	1.00	
450	232.2	4.585	825.7	1.20	0.646	0.85	
500	260	4.731	785.2	1.07	0.616	0.83	

300	148.9	4.296	918.0	1.86	0.684	1.17
350	176.7	4.371	890.4	1.57	0.677	1.02
400	204.4	4.467	859.4	1.36	0.665	1.00
450	232.2	4.585	825.7	1.20	0.646	0.85
500	260	4.731	785.2	1.07	0.616	0.83

Table D-16 Outdoor air requerments for ventilation

Application	Maximum Occupancy Per 100 m²	Outdoor Air Requirement	
		L/s/Person	L/s/m²
Offices			
Office space	7	10.0	
Reception area	60	8.0	—
Telecomm. Centers	60	10.0	
Conference rooms	50	10.0	
Public spaces			
Corridors			0.25
Public restrooms	—	25.0	—
Locker rooms	[illegible]	[illegible]	[illegible]
Smoking lounges	[illegible]	[illegible]	[illegible]
Elevators	—	—	5.00
Laundries			
Commercial laundry	10	13.0	
Commercial dry cleaner	30	15.0	
Coin-operated laundries	20	8.0	
Coin-operated dry cleaner	20	8.0	
Food and beverage services			
Dining rooms	70	10.0	—
Cafeteria	100	10.0	—
Bars	100	15.0	—
Kitchens	20	8.0	—
Garages, service stations			
Enclosed parking garage		—	7.50
Auto repair rooms			7.50
[illegible]	—		0.80
Retail stores			
Basement and street	30		1.50
Upper floors	20		1.00
Storage rooms	15		[illegible]
Dressing rooms			1.00
Malls and arcades			1.00
Warehouses	5	—	0.25
Smoking lounge	[illegible]	30.0	

* Adapted from "*ASHRAE Handbook of Fundamentals*", 1993

Continues…

Continued Application	Maximum Occupancy Per 100 m²	Outdoor Air Requirement	
		L/s/Person	L/s/m²
Specialty shops			
Barber	25	8.0	—
Beauty	25	13.0	—
Reducing salons	20	8.0	—
Florists	8	8.0	—
Supermarkets	8	8.0	—
Hardware, drugs, fabrics	8	8.0	—
Pet shops	—	—	5.00
Furniture stores	—	—	1.50
Sports and amusement			
Spectator areas	150	8.0	
Game rooms	70	13.0	
Ice arenas	—	—	2.50
Swimming pools	—	—	2.50
Gymnasium	30	10.0	
Ballrooms and discos	100	13.0	
Bowling alleys	70	13.0	
Theaters			
Ticket booths	60	10.0	
Lobbies	150	10.0	—
Auditorium	150	8.0	
Stages, studios	70	8.0	—
Transportation			
Waiting rooms	100	8.0	
Platforms	100	8.0	
Vehicles	150	8.0	
Workrooms			
Meat processing	10	8.0	—
Photo studios	10	8.0	—
Darkrooms	10		2.50
Pharmacy	20	8.0	
Bank vaults	5	8.0	
Printing, duplicating rooms		—	2.50
Correctional facilities			
Cells	20	10.0	
Dining halls	100	8.0	
Guard stations	40	8.0	
Education			
Classrooms	50	8.0	—
Laboratories	30	10.0	

Continues…

Continued Application	Maximum Occupancy Per 100 m²	Outdoor Air Requirement	
		L/s/Person	L/s/m²
Training shop	30	10.0	—
Music room	50	8.0	—
Libraries	20	8.0	—
Locker rooms	—	—	2.50
Corridors	—	—	0.50
Auditorium	150	8.0	—
Smoking areas	70	30.0	—
[illegible]			
Patient rooms	10	13.0	
Medical procedure	20	8.0	
Operating rooms	20	15.0	
Recovery and ICU	20	8.0	
Autopsy rooms	—		2.50
Physical therapy	20	8.0	
Residential facilities			
Living areas[1]	—	7.5	—
Kitchens[2]	—	12.0	—
Bath, toilets[3]	—	10.0	—
[illegible] motels, [illegible]			
Bedrooms	—	—	15 L/s/room
Living rooms	—	—	15 L/s/room
Baths		—	18 L/s/room
Lobbies	30	8.0	—
Conference rooms	50	10.0	—
Assembly rooms	120	8.0	—
Dormitory sleeping areas	20	8.0	—
Gambling casinos	120	15.0	—

[1] or 0.35 air change/hour
[2] or 50 L/s intermittent or operable window
[3] or 25 L/s intermittent or operable window.

Table D-17 Infiltration through window and door crack in cubic meter per hour per meter of crack

		Wind Speed (km/h)				
Type of Aperture	Remarks	8.0	16.0	24.0	32.0	40.0
Double-hung wood-sash windows (Unlocked)	Average; non-weather-stripped.	0.7	2.1	[illegible]	5.9	8.0
	Average; weather-stripped	0.4	[illegible]	2.4	3.6	4.9
	Poorly fitted; non-weather-stripped.	[illegible]	6.9	11.1	15.4	19.9
	Poorly fitted; weather-stripped.	0.6	1.9	3.4	5.1	7.1
	Around window frame; masonry wall uncalked.	0.3	0.8	1.4	2.0	2.7
	Around window frame; masonry wall calked.	0.1	0.2	0.3	0.3	0.5
	Around window frame; wood-frame structure.	0.2	0.6	1.1	1.7	2.3
Double-hung metal windows	Non-weather-stripped, unlocked.	2.0	4.7	7.4	10.4	13.7
	Non-weather-stripped; locked.	2.2	[illegible]	[illegible]	9.3	12.5
	Weather-stripped; unlocked.	0.6	1.9	3.2	4.6	6.0
Single-sash metal windows	Industrial, horizontally pivoted	6.2	10.8	17.6	24.4	30.4
	Residential casement	1.4	3.2	5.2	7.6	10.0
	Vertically pivoted	3.0	8.8	14.5	18.6	[illegible]
Doors	Well-fitted	2.7	6.9	11.1	15.4	19.9
	Poorly fitted	5.4	13.8	22.0	30.5	39.8

Table D-18 Infiltration rates due to door opening m per passege

№ of Passage per Hour	Doors in One Wall Only: Single Swing	Doors in One Wall Only: Vestibule Swinging Doors	Doors in One Wall Only: Revolving Doors	Doors in more than One Wall: Single Swing	Doors in more than One Wall: Vestibule Swinging Doors	Doors in more than One Wall: Revolving Doors
300	4.757	3.540	1.369	3.115	2.350	0.850
[illegible]00	4.757	3.540	1.303	3.1[illegible]	2.350	0.8[illegible]
[illegible]00	4.757	3.540	1.[illegible]18	3.115	2.322	0.755
900	4.757	3.540	1.104	3.087	2.322	0.708
1,100	4.757	3.540	0.935	3.087	2.322	0.651
1,200	4.757	3.540	0.850	3.058	2.322	0.595
1,300	4.757	3.540	0.793	3.058	2.322	0.533
1,400	4.757	3.540	0.708	3.058	2.294	0.510
1,500	4.757	3.540	0.651	3.058	2.294	0.481
1,600	4.729	3.540	0.595	3.058	2.294	0.453
1,700	4.616	3.511	0.538	3.030	2.294	0.425
1,800	4.503	3.4[illegible]	0.510	2.972	2.265	0.396
1,900	4.41[illegible]	3.368	0.481	2.94[illegible]	2.265	0.368
6,000	4.301	3.540	0.453	3.332	2.237	0.340

Table D-19 Cooling load temperature differences for calculating cooling load from sunlit roofs

No.	Roof Description of Construction	U-value W/m²·°C	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	Without Suspended Ceiling																									
1	Steel sheet with 25.4 mm (or 50.8 mm) insulation	1.209 (0.704)	0	-1	-2	-2	-3	-2	3	11	19	27	34	40	43	44	43	39	33	25	17	10	7	5	3	1
2	25 mm wood with 25.4 mm insulation	0.961	3	2	0	-1	-2	-2	-1	2	8	15	22	29	35	39	41	41	39	35	29	21	15	11	8	5
3	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
4	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
5	25.4 mm wood with 50.8 insulation	0.619	2	0	-2	-3	-4	-4	-4	-2	3	9	15	22	27	32	35	36	35	32	27	21	14	10	6	3
6	152.4 mm L.W. concrete	0.897	12	10	7	5	3	2	1	0	2	4	8	13	19	24	29	33	35	36	35	32	28	24	19	16
7	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
8	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
9	101.6 mm H.W. concrete with 25.4 mm (or 50.8 mm) insulation	1.136 (0.681)	14	12	10	8	7	5	4	4	6	8	11	15	18	22	25	28	29	30	29	27	24	21	19	16
10	63.5 mm wood with insulation	0.528	18	15	13	11	9	8	6	5	5	5	7	10	13	17	21	24	27	28	29	29	27	25	23	20
11	Roof terrace system	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
12	152.4 mm H.W. concrete with 25.4 mm (or 50.8 mm) insulation	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
13	101.6 mm wood with 25.4 mm (or 50.8 mm) insulation	0.602 (0.443)	21	20	18	17	15	14	13	11	10	9	9	9	10	12	14	16	18	20	22	23	24	24	23	22
	With Suspended Ceiling																									
1	Steel sheet with 25.4 mm (or 50.8 mm) insulation	0.561 (0.522)	1	0	-1	-2	-3	-3	0	3	13	20	28	35	40	43	43	41	37	31	22	15	10	7	5	1
2	25 mm wood with 25.4 mm insulation	0.653	11	8	6	5	3	2	1	2	4	7	12	17	22	27	31	33	35	34	32	28	24	20	17	14
3	101.6 mm L.W. concrete	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

-Continued-

No.	Roof Description of Construction	U-value W/m²·°C	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
4	50.8 mm H.W. concrete with [illegible] mm insulation	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	18
5	25.4 mm wood with 50.8 insulation	0.471	14	11	9	7	5	4	3	3	4	6	10	14	18	23	27	30	31	32	31	29	26	22	19	16
6	152.4 mm L.W. concrete	0.605	18	15	13	11	9	7	6	4	4	4	6	9	12	16	20	24	27	29	30	30	28	26	23	20
7	63.5 mm wood with 25.4 mm insulation	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	23	22	21
8	203.2 mm L.W. concrete	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	24	23
9	101.6 mm H.W. concrete with 25.4 mm (or 50.8 mm) insulation	0.727 (0.511)	17	16	15	14	13	13	12	11	11	11	12	13	15	16	18	19	20	21	21	21	21	20	19	18
10	63.5 mm wood with 50.8 mm insulation	0.409	19	18	17	16	14	13	12	11	10	10	10	11	12	14	16	18	19	21	22	23	23	22	22	21
11	Roof terrace system	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	18	18	18
12	152.4 mm H.W. concrete with 25.4 mm (or 50.8 mm) insulation	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
13	101.6 mm wood with 25.4 mm (or 50.8 mm) insulation	0.465 (0.363)	20	19	19	18	17	16	15	14	14	13	12	12	12	12	13	14	15	16	18	19	20	20	20	19

Table D-20 Solar heat gain factor (SHG), W/m², for a latitude angle of 32°

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
N	76	85	101	114	120	139	126	117	104	88	76	69
NNE/NNW	76	85	117	252	350	385	350	249	110	88	76	69
NE/NW	91	205	338	461	536	555	527	445	325	199	91	69
ENE/WNW	331	470	577	631	656	656	643	615	546	451	325	265
E/W	552	647	716	716	694	675	678	691	678	615	546	511
ESE/WSW	722	764	748	691	628	596	612	663	716	738	710	688
SE/SW	786	782	716	590	489	439	473	571	688	754	773	776
SSE/SSW	789	732	615	445	213	262	303	429	596	710	776	795
S	776	697	555	363	233	189	227	350	540	678	767	795
Horiz.	555	685	795	855	874	871	861	836	770	672	552	498

Table D-21 Shading coefficient (SC), for single, double and insulating glass without interior shading

Type of Glass	Nominal Thickness (mm)	Solar Trans.	Shading Coefficient (W/m²·K) $h_o = 22.7$	Shading Coefficient (W/m²·K) $h_o = 17.0$
Single Glass				
Clear	3	0.86	1.00	1.00
	6	0.78	0.94	0.95
	10	0.72	0.90	0.92
	12	0.67	0.87	0.88
Heat absorbing	3	0.64	0.83	0.85
	6	0.46	0.69	0.73
	10	0.33	0.60	0.64
	12	0.42	0.53	0.58
Double Glass				
Regular	3	—	0.90	—
Plate	6	—	0.83	—
Reflective	6	—	0.20-0.40	—
Insulating Glass				
Clear	3	0.71	0.88	0.88
	6	0.61	0.81	0.82
Heat absorbing	6	0.36	0.55	0.58

Table D-22 Shading coefficient for single, double and insulating glass with indoor shading by venetian blinds or roller shades.

Type of Glass	Nominal Thickness (mm)	Type Of Shading				
		Venetian Blinds		Roller Shade		
				Opaque		Transinuent
		Medium	Light	Dark	White	Light
Single Glass						
Clear	2.5-6.0					
Clear	6.0-12.0					
Clear Pattern	3.0-12.0	0.64	0.55	0.59	0.25	0.39
Heat absorbing	3	—	—			—
Pattern or Tinted	5.0-5.5					
Heat absorbing	5.0-6.0	0.57	0.53	0.45	0.30	0.36
Pattern or Tinted	3.0-5.5	—	—	—	—	—
Heat Absorbing or Pattern Heat Absorbing	10	0.54	0.52	0.40	0.82	0.32
Heat Absorbing or Pattern	—	0.42	0.40	0.36	0.28	0.31
Reflective Coated Glass		0.30	0.25	0.23		—
		0.40	0.33	0.29		—
	—	0.50	0.42	0.38	—	—
		0.60	0.50	0.44		—
Double Glass						
Regular	3	0.57	0.51	0.60	0.25	—
Plate	6	0.57	0.51	0.60	0.25	—
Reflective	6	0.20-0.40	—	—		—
Insulating Glass						
Clear	2.5-6.0	0.57	0.51	0.60	0.25	0.37
Heat Absorbing	5.0-6.0	0.39	0.36	0.40	0.22	0.30
Reflective Coated	—	0.20	0.19	0.18		—
	—	0.30	0.27	0.26	—	—
	—	0.40	0.34	0.33	—	—

Table D-23 Cooling load factors for glass without interior shading, north latitudes

Fenestration Facing	Room Construction	Solar Time h																
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
N Shaded	L	0.17	0.14	0.11	0.09	0.08	0.33	0.24	0.48	0.56	0.61	0.71	0.76	0.80	0.82	0.82	0.79	0.75
	M	0.23	0.20	0.18	0.16	0.14	0.34	0.14	0.46	0.53	0.59	0.65	0.70	0.73	0.75	0.76	0.74	0.75
	H	0.25	0.23	0.21	0.20	0.19	0.38	0.45	0.49	0.55	0.60	0.65	0.69	0.72	0.72	0.72	0.70	0.70
NNE	L	0.06	0.05	0.04	0.03	0.03	0.26	0.43	0.47	0.44	0.41	0.40	0.39	0.39	[illegible]	0.36	0.33	0.30
	M	0.09	0.08	0.07	0.06	0.06	0.24	0.38	0.42	0.39	0.37	0.37	0.36	0.36	0.36	0.36	0.33	0.30
	H	0.11	0.10	0.09	0.09	0.08	0.26	0.39	0.42	0.39	0.36	0.35	0.34	0.34	0.33	0.32	0.31	0.28
NE	L	0.04	0.04	0.03	0.02	0.02	0.23	0.41	0.51	0.51	0.45	0.39	0.36	0.33	0.31	0.28	0.26	0.23
	M	0.07	0.06	0.06	0.05	0.04	0.21	0.36	0.44	0.45	0.40	0.36	0.33	0.31	0.30	0.28	0.26	0.23
	H	0.09	0.08	0.08	0.07	0.07	0.23	0.37	0.44	0.44	0.39	0.34	0.31	0.29	0.27	0.26	0.24	0.22
ENE	L	0.04	0.03	0.03	0.02	0.02	0.21	0.40	0.52	0.57	0.53	0.45	0.39	0.34	0.31	0.28	0.25	0.22
	M	0.07	0.06	0.05	0.05	0.04	0.20	0.35	0.45	0.49	0.47	0.41	0.36	0.33	0.30	0.28	0.26	0.23
	H	0.09	0.09	0.08	0.07	0.07	0.22	0.36	0.46	0.49	0.45	0.38	0.33	0.30	0.27	0.25	0.23	0.21
E	L	0.04	0.03	0.03	0.02	0.02	0.19	0.37	0.51	0.57	0.57	0.50	0.42	0.37	0.32	0.29	0.25	0.22
	M	0.07	0.06	0.06	0.05	0.05	0.18	0.33	0.44	0.50	0.51	0.46	0.39	0.35	0.31	0.29	0.26	0.23
	H	0.09	0.08	0.08	0.08	0.07	0.20	0.34	0.45	0.49	0.49	0.43	0.36	0.32	0.29	0.26	0.24	0.22
ESE	L	0.05	0.04	0.03	0.03	0.03	0.17	0.34	0.49	0.58	0.61	0.57	0.48	0.41	0.36	0.32	[illegible]	[illegible]
	M	0.08	0.07	0.06	0.05	0.05	0.16	0.31	0.43	0.51	0.54	0.51	0.44	0.39	0.35	0.32	[illegible]	[illegible]
	H	0.10	0.09	0.09	0.08	0.08	0.19	0.32	0.43	0.50	0.52	0.49	0.41	0.36	0.33	0.31	0.29	[illegible]

–Continued–

Fenestration Facing	Room Construction	Solar Time h																
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
SE	L	0.05	0.04	0.04	0.03	0.03	[illegible]	[illegible]	0.43	0.55	0.62	0.63	[illegible]	[illegible]	[illegible]	0.37	0.33	0.29
	M	0.09	0.08	0.07	0.06	0.05	[illegible]	0.25	0.38	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	H	0.11	0.10	0.10	0.09	0.08	0.17	0.28	0.40	0.49	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
SSE	L	0.07	0.05	0.04	0.04	0.03	0.06	0.15	0.29	0.43	0.55	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	M	0.11	0.09	0.08	0.07	0.06	0.08	0.16	0.28	0.38	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	H	0.13	0.12	0.11	0.10	0.09	0.11	0.19	0.29	0.40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
S	L	0.08	0.07	0.05	0.04	0.04	0.06	0.09	0.14	0.22	0.34	0.48	0.59	0.65	0.65	0.59	0.50	0.43
	M	0.12	0.11	0.09	0.08	0.07	0.08	0.11	0.14	0.21	0.31	0.42	0.52	0.57	0.58	0.53	0.47	0.41
	H	0.13	0.12	0.12	0.11	0.10	0.11	0.14	0.17	0.24	0.33	0.43	0.51	0.56	0.55	0.50	0.43	0.37
SSW	L	0.10	0.08	0.07	0.06	0.05	0.06	0.09	0.11	0.15	0.19	0.27	0.39	0.52	0.62	0.67	0.65	0.58
	M	0.14	0.12	0.11	0.09	0.08	0.09	0.11	0.13	0.15	0.18	0.25	0.35	0.46	0.55	0.59	0.59	0.53
	H	0.15	0.14	0.13	0.12	0.11	0.12	0.14	0.16	0.18	0.21	0.27	0.37	0.46	0.53	0.57	0.55	0.49
SW	L	0.12	0.10	0.08	0.06	0.05	0.06	0.08	0.10	0.12	0.14	0.16	0.24	0.36	0.49	0.60	0.66	0.66
	M	0.15	0.14	0.12	0.10	0.09	0.09	0.10	0.12	0.13	0.15	0.17	0.23	0.33	0.44	0.53	0.58	0.59
	H	0.15	0.14	0.13	0.12	0.11	0.12	0.13	0.14	0.16	0.17	0.19	0.25	0.34	0.44	0.52	0.56	0.56
WSW	L	0.12	0.10	0.08	0.07	0.05	0.06	0.07	0.09	0.10	0.12	0.13	0.17	0.26	0.40	0.52	0.62	0.66
	M	0.15	0.13	0.12	0.10	0.09	0.09	0.10	0.11	0.12	0.13	0.14	0.17	0.24	0.35	0.46	0.54	0.58
	H	0.15	0.14	0.13	0.12	0.11	0.11	0.12	0.13	0.14	0.15	0.16	0.19	0.26	0.36	0.46	0.53	0.56

Table D-24 Cooling load temperature differences for glass convection.

Solar Time	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
CLTD (°C)	1	0	−1	−1	−1	−1	−1	0	1	2	4	5	7	7	8	8	7	7	6	4	3	2	2	1

Table D-25 heat gain rate from miscellaneous appliances (W).*

Appliance	Without Hood			With Hood
	Sensible	Latent	Total	All Sensible
Hair dryers (Blower type)	675	120	795	—
Hair dryers (Helmet type)	550	100	650	—
Coffee brewer (electrical)	225	65	290	95
Coffee brewer (gas)	490	210	700	415
Water heater	1,130	335	1,465	—
Coffee urn (electrical)	1,075	350	1,425	440
Coffee urn (gas)	1,460	625	2,085	415
Deep fat fryer (electrical)	820	1,930	2,750	730
Deep fat fryer (gas)	2,080	2,080	4,160	830
Toaster	1,055	705	1,760	440
Domestic gas oven	2,430	1200	3,630	—
Roasting oven	500	320	820	—
Food warmer (gas)	1,550	400	1,950	400
Egg boiler	335	220	555	—
Frying griddle	13,600	7,200	20,800	4,150
Hotplate	1,550	1,060	2,610	780
Neon sign, per meter length	56	—	56	—
Sterilizer	190	350	640	—
Laboratory burner	470	120	690	—
Small copy machine	1,760	—	1,760	—
Large copy machine	3,515	—	3,515	—
Motors				
400–2,000 W	1,100		1,100	—
2,000–15,000 W	2,430		880	—

Table D-26 Diversity factor for selected applications.†

Application	Diversity Factor	
	Lights	People
Peripheral areas of offices with glazing area of 20%-50%	0.70-0.85	0.7-0.8
Core areas of offices and peripheral areas with less than 20% glazing	0.90-1.00	0.7-0.8
Apartments and hotel bedrooms	0.30-0.50	0.4-0.6
Public rooms in hotels	0.90-1.00	0.4-0.6
Department stores and supermarkets	0.90-1.00	0.8-1.0

Table D-27 Cooling load factors for lighting.†

No of hours after lights are turned on	Fixture X¢ hours of operation		Fixture Y¢ hours of operation	
	10	16	10	16
[illegible]	0.08	0.19	0.01	0.05
[illegible]	0.62	0.72	0.76	0.79
[illegible]	0.66	0.75	0.81	0.83
3	0.69	0.77	0.84	0.87
4	0.73	0.80	0.88	0.89
5	0.75	0.82	0.90	0.91
6	0.78	0.83	0.92	0.93
7	0.80	0.85	0.93	0.94
8	0.82	0.87	0.95	0.95
9	0.84	0.88	0.96	0.96
10	0.85	0.89	0.97	0.97
11	0.32	0.90	0.22	0.98
12	0.29	0.91	0.18	0.98
13	0.26	0.92	0.14	0.98
14	0.23	0.93	0.12	0.99
15	0.21	0.94	0.09	0.99
16	0.19	0.94	0.08	0.99
17	0.17	0.40	0.06	0.24
18	0.15	0.36	0.05	0.20

¢ Fixture description: X, recessed lights which are not vented. The supply and return air registers are below the ceiling or through the ceiling space and grille. Y, vented or free-hanging lights. The supply air registers are below or through the ceiling with the return air registers around the fixtures and through the ceiling space.

Table D-28 Sensible heat cooling load factors for people.§

Hours after each entry into space	Total hours in space							
	2	4	6	8	10	12	14	16
[illegible]	0.49	0.49	0.50	0.51	0.53	0.55	0.58	0.62
[illegible]	0.58	0.59	0.60	0.61	0.62	0.64	0.66	0.70
[illegible]	0.17	0.66	0.67	0.67	0.69	0.70	0.72	0.75
4	0.13	0.71	0.72	0.72	0.74	0.75	0.77	0.79
5	0.10	0.27	0.76	0.76	0.77	0.79	0.80	0.82
6	0.08	0.21	0.79	0.80	0.80	0.81	0.83	0.85
7	0.07	0.16	0.81	0.82	0.83	0.84	0.85	0.87
8	0.06	0.14	0.24	0.84	0.85	0.86	0.87	0.88
9	0.05	0.12	0.21	0.38	0.87	0.88	0.89	0.90
10	0.04	0.10	0.18	0.30	0.89	0.89	0.9	0.91
11	0.04	0.08	0.15	0.25	0.42	0.91	0.91	0.92
12	0.03	0.07	0.13	0.21	0.34	0.92	0.92	0.93
13	0.03	0.06	0.12	0.18	0.28	0.45	0.93	0.93
14	0.02	0.06	0.10	0.15	0.23	0.36	0.94	0.94
15	0.02	0.05	0.08	0.13	0.20	0.30	0.47	0.95
16	0.02	0.04	0.07	0.12	0.17	0.25	0.38	0.96
17	0.02	0.04	0.06	0.10	0.15	0.21	0.31	0.49
18	0.01	0.03	0.06	0.09	0.13	0.19	0.26	0.39

Table D-29 Steel pipe dimensions

Nominal pipe size, in	OD, in	Schedule no.	Wall Thickness, in	ID, in	Metal sectional area, in²	Inside cross-sectional area, ft²
1/8	0.405	40	0.068	0.269	0.072	0.00040
		80	0.095	0.215	0.093	0.00025
1/4	0.540	40	0.088	0.364	0.125	0.00072
		80	0.119	0.302	0.157	0.00050
3/8	0.675	40	0.091	0.493	0.167	0.00133
		80	0.126	0.423	0.217	0.00098
1/2	0.840	40	0.109	0.622	0.250	0.00211
		80	0.147	0.546	0.320	0.00163
3/4	1.050	40	0.113	0.824	0.333	0.00371
		80	0.154	0.742	0.433	0.00300
1	1.315	40	0.133	1.049	0.494	0.00600
		80	0.179	0.957	0.639	0.00499
1 1/2	1.900	40	0.145	1.610	0.799	0.01414
		80	0.200	1.500	1.068	0.01225
		160	0.281	1.338	1.429	0.00976
2	2.375	40	0.154	2.067	1.075	0.02330
		80	0.218	1.939	1.477	0.02050
3	3.500	40	0.216	3.068	2.228	0.05130
		80	0.300	2.900	3.016	0.04587
4	4.500	40	0.237	4.026	3.173	0.08840
		80	0.337	3.826	4.407	0.07986
5	5.563	40	0.258	5.047	4.300	0.1390
		80	0.375	4.813	6.112	0.1263
		120	0.500	4.563	7.953	0.1136
		160	0.625	4.313	9.696	0.1015
6	6.625	40	0.280	6.065	5.581	0.2006
		80	0.432	5.761	8.405	0.1810
10	10.75	40	0.365	10.020	11.90	0.5475
		60	0.500	9.750	16.10	0.5185

Appendix E

Table (E-1): The First Package Unit AC

1- Supply line duct to the basement floor

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.36	0.7587	6.7437	$1.5303*10^5$	0.000125	0.018	0.93375	2.3179
0.34	0.569	5.0577	$1.2195*10^5$	0.0001323	0.015	0.1546	1.6501
0.3	0.1897	2.7095	$4.9531*10^4$	0.00015	0.019	0.1326	1.1459
0.29	0.1897	2.7095	$4.9531*10^4$	0.0001552	0.019	0.0552	0.3742
0.24	0.0948	1.8967	$2.8694*10^4$	0.0001875	0.022	0.0166	0.1833
0.21	0.0474	1.2644	$1.6738*10^4$	0.0002143	0.0235	0.01839	0.0815
0.73	3.07	6.39	$2.9395*10^5$	0.0001216	0.015	0.1676	2.08
0.48	2.4	10.62	$3.2127*10^5$	0.00009375	0.018	2.83636	5.75

2- Supply duct line to the theater /left hand side from entrance

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
1.04	6.21	6.7475	$4.424*10^5$	0.00004326	0.017	0.16	2.321
0.86	4.1	6.44	$3.4896*10^5$	0.00005233	0.018	0.2716	2.112
0.72	2.883	6.55	$2.9737*10^5$	0.0000625	0.017	0.162	2.1881
0.6	1.7	5.6	$2.1184*10^5$	0.000075	0.018	0.1158	1.599

3- Supply duct to the theater /right hand side

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.63	2.45	7.2	$2.8651*10^5$	0.000071428	0.0185	0.3167	2.653
0.48	2.14	10.62	$3.2127*10^5$	0.00009375	0.016	0.3684	5.7464

4- Small branches losses at the middle of the theater space

d (m)	$Q(m^3/s)$	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.49	1.2	8.5	$2.6223*10^5$	0.00009184	0.0165	0.7293	3.6736

5- Second Package unit for foyer space

d (m)	$Q(m^3/s)$	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
Main duct 1.08	8.5	9.21	$6.2689*10^5$	0.000041666	0.0155	0.1677	4.322
(right duct) 0.49	1.4	5.7	$1.7575*10^5$	0.0000918	0.0175	0.131	1.6501
0.49	1.4	5.7	$1.7575*10^5$	0.0000918	0.0175	0.131	1.65
0.29	0.38	5.1	$9.2457*10^4$	0.000155	0.018	0.3064	1.3038
0.29	0.13	1.7	$3.0819*10^4$	0.000155	0.017	0.017	0.1449
0.29	0.13	1.7	$3.0819*10^4$	0.000155	0.017	0.017	0.1449
0.29	0.13	1.7	$3.0819*10^4$	0.000155	0.017	0.017	0.1449
0.29	0.13	1.7	$3.0819*10^4$	0.000155	0.017	0.017	0.1449

6- Top supply line for foyer space (line goes to the main door side)

d (m)	$Q(m^3/s)$	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
1	7.33	8.73	$5.5034*10^5$	0.000045	0.015	0.07965	3.885
0.87	4.71	7.4	$4.036*10^5$	0.0000517	0.0154	0.1913	2.761
0.76	2.62	5.25	$2.5139*10^5$	0.0000592	0.0182	0.104	1.403
0.69	1.9	4.77	$2.0749*10^5$	0.0000652	0.0177	0.03699	1.16
0.6	1.43	4.77	$1.8042*10^5$	0.000075	0.0188	0.041	1.16
0.44	0.95	6	$1.6539*10^5$	0.0001023	0.0165	0.0313	1.8123
0.44	0.72	4.5	$1.2404*10^5$	0.0001023	0.0173	0.0939	1.02
0.6	1.43	4.77	$1.8042*10^5$	0.000075	0.0188	0.041	1.16

0.69	1.9	4.77	$2.0749*10^5$	0.0000652	0.0177	0.03699	1.16
0.76	2.62	5.25	$2.5139*10^5$	0.0000592	0.0182	0.104	1.403

d (m)	**Q(m^3/s)**	**V (m/s)**	**Re**	**roughness**	**f**	**h_f (m)**	**h_d (m)**
0.55	2.09	8.345	$2.8933*10^5$	0.0000818	0.0162	0.7228	3.55
0.55	2.09	8.345	$2.8933*10^5$	0.0000818	0.0165	0.2562	3.55
0.46	1.252	7	$2.0165*10^5$	0.0000978	0.0163	0.0881	2.465
0.43	0.8	5.3	$1.4362*10^5$	0.0001047	0.017	0.1721	1.431
0.32	0.42	4.63	$9.352*10^4$	0.000141	0.0181	0.2092	1.1

7- The last branch in foyer space.

d (m)	**Q(m^3/s)**	**V (m/s)**	**Re**	**roughness**	**f**	**h_f (m)**	**h_d (m)**
0.43	0.8	5.3	$1.4362*10^5$	0.000105	0.0174	0.0751	1.431
0.32	0.42	4.6363	$9.352*10^4$	0.0001406	0.0171	0.2098	1.1

8- Basement supply line (900 CFM Louver)

d (m)	**Q(m^3/s)**	**V (m/s)**	**Re**	**roughness**	**f**	**h_f (m)**	**h_d (m)**
0.35	0.57	5.2	$1.1412*10^5$	0.000129	0.0177	0.8233	1.3637
0.69	1.83	6.3	$2.7263*10^5$	0.0000652	0.0185	0.2199	2.0025
0.47	0.92	4.7	$1.3929*10^5$	0.0000957	0.017	0.2073	1.13

9- At the right of the open area.

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.34	5.7	5.2	$1.1086*10^5$	0.0001324	0.0178	0.5613	1.364
0.69	1.8	6.3	$2.7263*10^5$	0.00006522	0.0184	0.0523	2
0.47	0.92	4.7	$1.3928*10^5$	0.0000954	0.0178	0.2749	1.126

Table (E-2) Return duct from the basement floor to the fist Package

1- Suction line from the small rooms

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.32	0.5	6.1	$1.2243*10^5$	0.000141	0.0178	0.5145	1.8775
0.32	0.53	6.1	$1.2243*10^5$	0.000141	0.0178	0.1271	1.8775
0.29	0.57	8.13	$1.4859*10^5$	0.000155	0.0171	04275	3.37
0.27	0.354	5.9	$1.0034*10^5$	0.0001666	0.0178	0.054	1.775
0.24	0.3	5.31	$8.0342*10^4$	0.0001875	0.0176	0.2185	1.44
0.21	0.2	4.7	$6.2488*10^4$	0.0002143	0.0189	0.1436	1.14
0.16	0.1	3.9	$3.9675*10^4$	0.00028125	0.0195	0.1584	0.79

2-

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.48	1.58	8.21	$2.4843*10^5$	0.00009375	0.0172	0.4423	3.44
0.385	1.6	12	$2.9225*10^5$	0.0001169	0.0168	2.3718	7.4

3- Return duct to the first package unit from the theater from the left hand side

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.625	6	27.13	$8.8075*10^5$	0.000072	0.0182	20.174	37.52
0.7 Right side of the theater	2.3	5.5	$2.4177*10^5$	0.0000643	0.0181	0.5028	1.53
0.535 right hand side of the theater	3.4	7.7	$2.6046*10^5$	0.00008411	0.0148	1.1417	3.04
1.01 branch duct after the slope line	2.77	3.2	$2.0417*10^5$	0.00004455	0.016	0.1859	0.5242
1.01 branch duct after the slope line	7.14	8.3	$5.2688*10^5$	0.00004455	0.018	0.5586	3.5

4- Return duct to the 2nd second package

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.89	5.4	7.8	$4.3852*10^5$	0.00005056	0.0174	0.5052	3.114

5- Suction fan for the small bathrooms behind the theater

d (m)	Q(m^3/s)	V (m/s)	Re	roughness	f	h_f (m)	h_d (m)
0.33	0.57	6.3	$1.3151*10^5$	0.000136	0.0174	0.1273	2.04
0.24	0.43	8.54	$1.2912*10^5$	0.0001875	0.0177	0.3414	3.7

0.24	0.3	5.7	$8.6081*10^4$	0.0001875	0.0184	0.16	1.65
0.19	0.14	4.7	$5.679*10^4$	0.0002368	0.0192	0.02	1.15
0.19	0.1	2.4	$2.8395*10^4$	0.0002368	0.023	0.0753	0.3
Small branches							
0.21	0.07	1.9	$2.5106*10^4$	0.000214	0.0234	0.0312	0.2
0.1685	0.04	1.4	$1.511*10^4$	0.0002671	0.0251	0.0396	0.1
0.1523	0.07	3.6	$3.4149*10^4$	0.0002954	0.021	0.122	0.6446
0.24	0.07	4.74	$3.8856*10^4$	0.0002954	0.0215	0.278	1.15
0.24	0.07	4.74	$3.8856*10^4$	0.0002954	0.0215	0.278	1.15
0.24	0.07	4.74	$3.8856*10^4$	0.0002954	0.0215	0.278	1.15

6- Suction duct from the preparing rooms

d (m)	**Q(m^3/s)**	**V (m/s)**	**Re**	**roughness**	**f**	**h_f (m)**	**h_d (m)**
0.13	0.04	2.5	$2.0723*10^4$	0.000346	0.024	0.1709	0.33
0.1523	0.08	3.8	$3.6426*10^4$	0.0002955	0.0215	0.2075	0.73
0.1685	0.08	3	$3.2235*10^4$	0.0002671	0.0221	0.0749	0.5
0.17	0.11	4.6	$4.8353*10^4$	0.0002647	0.0211	0.3384	1.1
0.21	0.2	4	$5.3562*10^4$	0.0002143	0.0195	0.1613	0.83
0.21	0.2	5.1	$6.6952*10^4$	0.0002143	0.0189	0.084	1.3
d (m)	**Q(m^3/s)**	**V (m/s)**	**Re**	**roughness**	**f**	**h_f (m)**	**h_d (m)**
0.23	0.2	4.2	$6.1107*10^4$	0.0001956	0.0191	0.1033	0.91
0.23	0.2	5.1	$7.13328*10^4$	0.0001956	0.0193	0.0208	1.3

7- Suction duct from the bathrooms in the basement floor:

d (m)	**Q(m^3/s)**	**V (m/s)**	**Re**	**roughness**	**f**	**h_f (m)**	**h_d (m)**
0.27	0.3	4.2	$7.1734*10^4$	0.0001666	0.0192	0.1	0.91

0.32	0.51	5.62	$1.1336*10^5$	0.0001406	0.0178	0.2691	1.61
0.32	0.76	8.43	$1.7004*10^5$	0.0001406	0.0171	0.5875	3.62

Table (E-3) Total Supply and Return Head

Package No	**Total supply head (m)**	**Total return head (m)**
Package #1	33.876	69.2242
Package #2	41.0568	3.114
900 CFM fan coil #1	4.4962	0
900 CFM fan coil #2	4.49	0
900 CFM suction fan	0	13.2346
360 CFM suction fan	0	7
1200 CFM suction fan	0	6.14

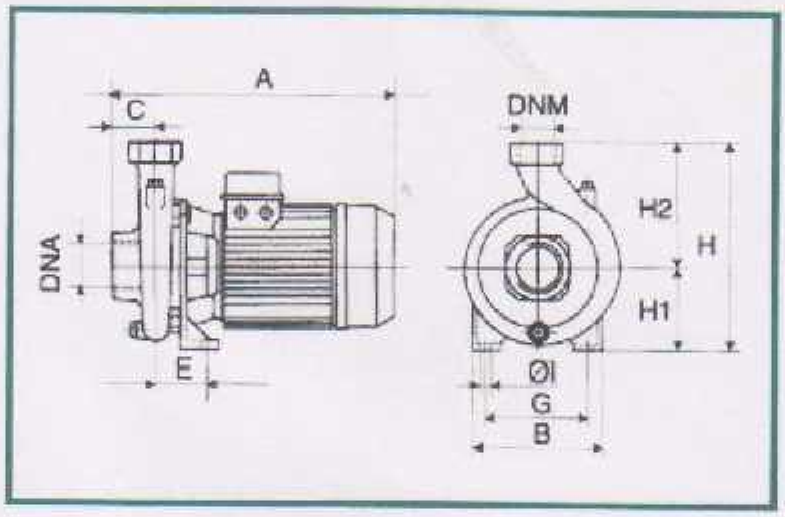

Figure (E-1) Pump Dimensions

Table (E-4) Pump Dimensions

MODEL	A	B	C	E	G	I	H	H1	H2	DNA	DNM	PACKING DIMENSIONS L/A	L/B	H	VOLUME m^3	WEIGHT Kg
K 55/200	425	250	56	96	175	14	320	135	185	2" G	1¼" G	512	278	345	0.040	39.0

Table (E-5) Pump Electrical And Hydraulic Data

MODEL	ELECTRICAL DATA									HYDRAULIC DATA (n = 2850 1/min)									
	VOLTAGE 50 Hz	P1 MAX kW	P2 NOMINAL kW	HP	In A	Icc A	1/min	η max %	cos φ	Q m^3/h	4	6	7.2	9.6	12	14.4	16.8	19.2	21.1
										l/min	4	100	120	160	200	240	[illegible]	320	355
K 55/200 T	3x230-400V ~	4.9	4	5.5	13.9-8.4	104-60	2850	81.2	0.85	H (m)	54	54	53.0	53	51.5	49.3	46.5	43.5	40

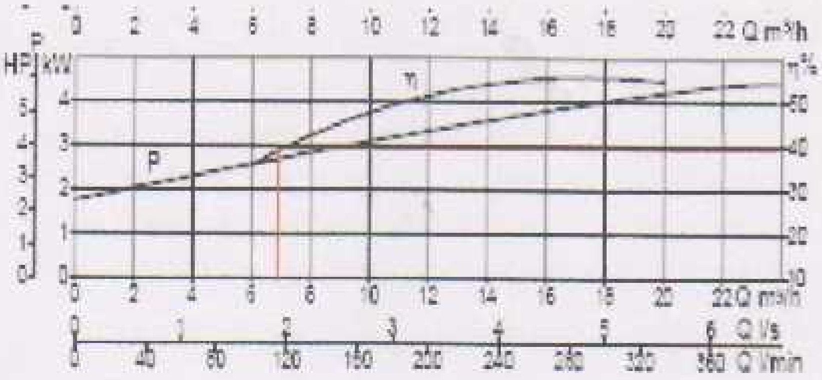

Figure (E-2) Performance Curve

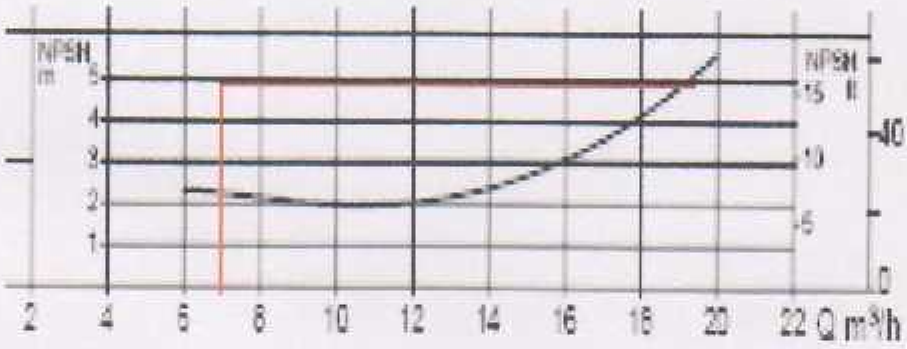

Figure (E-3) Performance Curve

Fan operating points

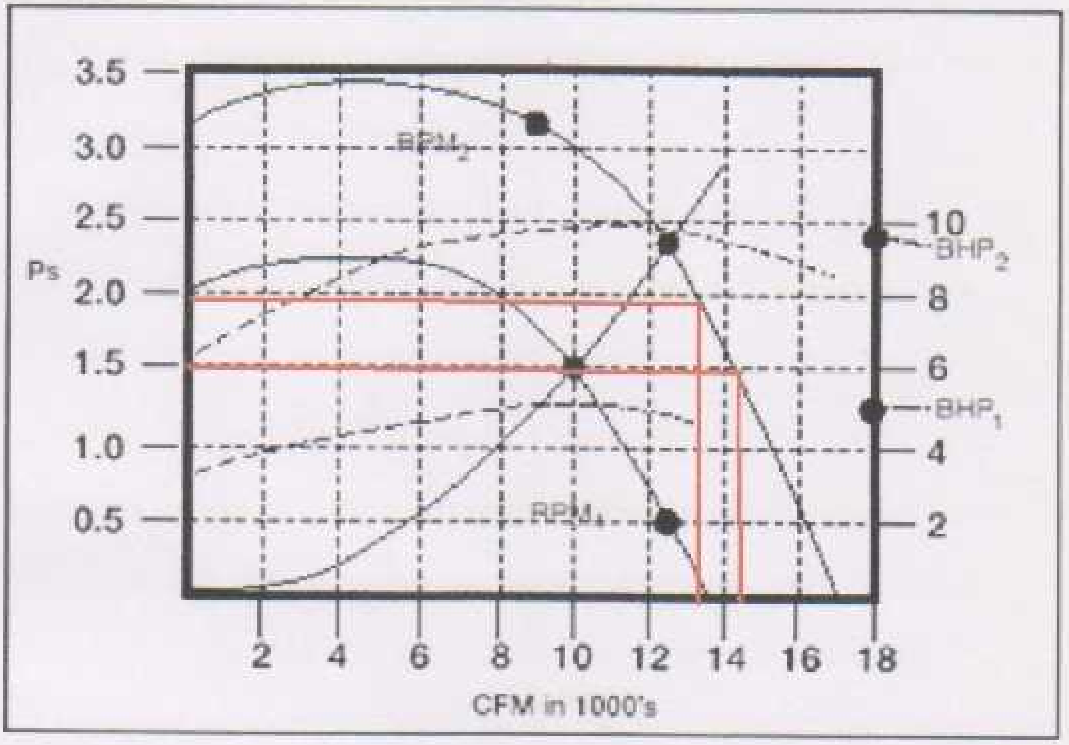

Figure (E-4) Performance Curve for the main supply fan.

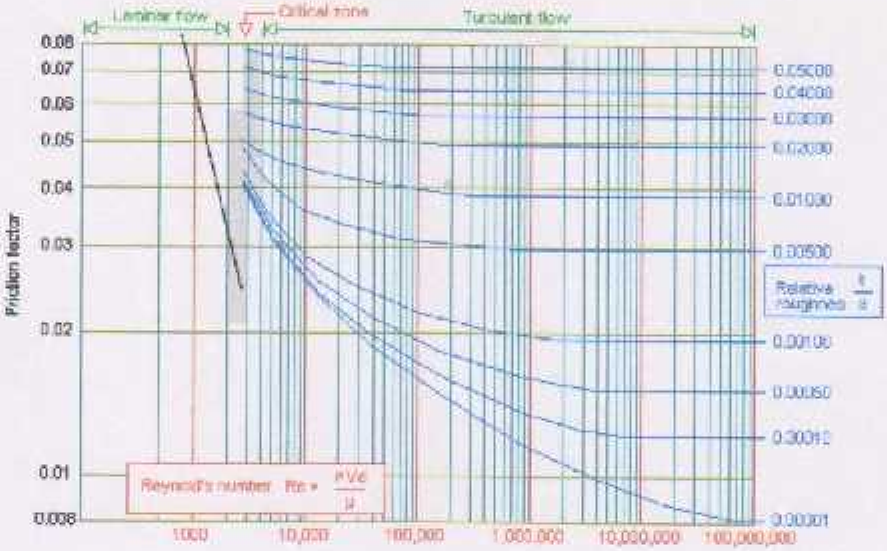

Figure (E-5) Moody chart

Fitting	Loss Coefficient, K
Gate valve (open to 75% shut)	0.25 - 25
Globe valve	10
Pump foot valve	1.5
Return bend	2.2
90° elbow	0.9
45° elbow	0.4
Large-radius 90° bend	0.6
Tee junction	1.8
Sharp pipe entry	0.5
Radiused pipe entry	0
Sharp pipe exit	0.5

Figure (E-6) Head loss coefficients for fittings.

Appendix F

Table (F-1) Description Of The Pins And Its Operations.

Pin No.	Pin Name	Pin type	Description
1	RC7/RX/DT/SDO RC7 RX DT SDO	 I/O I I/O O	Digital I/O. EUSART asynchronous receive. EUSART synchronous data (see TX/CK). SPI™ data out.
2	RD4/SPP4 RD4 SPP4	 I/O I/O	Digital I/O. Streaming Parallel Port data.
3	RD5/SPP5/P1B RD5 SPP5 P1B	 I/O I/O O	Digital I/O. Streaming Parallel Port data. Enhanced CCP1 PWM output, channel B.
4	RD6/SPP6/P1C RD6 SPP6 P1C	 I/O I/O O	Digital I/O. Streaming Parallel Port data. Enhanced CCP1 PWM output, channel C.
5	RD7/SPP7/P1D RD7 SPP7 P1D	 I/O I/O O	Digital I/O. Streaming Parallel Port data. Enhanced CCP1 PWM output, channel D.
6	VSS	P	Ground reference for logic and I/O pins.
7	VDD	P	Positive supply for logic and I/O pins
8	RB0/AN12/INT0/FLT0/SDI/SDA RB0 AN12 INT0 FLT0 SDI SDA	 I/O I I I I I/O	Digital I/O. Analog input 12. External interrupt 0. Enhanced PWM Fault input (ECCP1 module).

			SPI™ data in. I2C™ data I/O.
9	RB1/AN10/INT1/SCK/SCL RB1 AN10 INT1 SCK SCL	 I/O I I I/O I/O	Digital I/O. Analog input 10. External interrupt 1. Synchronous serial clock input/output for SPI mode. Synchronous serial clock input/output for I2C mode.
10	RB2/AN8/INT2/VMO RB2 AN8 INT2 VMO	 I/O I I O	Digital I/O. Analog input 8. External interrupt 2. External USB transceiver VMO output.
11	RB3/AN9/CCP2/VPO RB3 AN9 CCP2 VPO	 I/O I I/O O	Digital I/O. Analog input 9. Capture 2 input/Compare 2 output/PWM 2 output. External USB transceiver VPO output.
12	NC/ICCK/ICPGC ICCK ICPGC	 I/O I/O	No Connect or dedicated ICD/ICSP™ port clock.**(3)** In-Circuit Debugger clock. ICSP programming clock.
13	NC/ ICDT/ICPGD ICDT ICPGD	 I/O I/O	No Connect or dedicated ICD/ICSP port clock.**(3)** In-Circuit Debugger data. ICSP

			programming data.
14	RB4/AN11/KBI0/CSSPP RB4 AN11 KBI0 CSSPP	 I/O I I O	Digital I/O. Analog input 11. Interrupt-on-change pin. SPP chip select control output.
15	RB5/KBI1/PGM RB5 KBI1 PGM	 I/O I I/O	Digital I/O. Interrupt-on-change pin. Low-Voltage ICSP™ Programming enable pin.
16	RB6/KBI2/PGC RB6 KBI2 PGC	 I/O I I/O	Digital I/O. Interrupt-on-change pin. In-Circuit Debugger and ICSP programming clock pin.
17	RB7/KBI3/PGD RB7 KBI3 PGD	 I/O I I/O	Digital I/O. Interrupt-on-change pin. In-Circuit Debugger and ICSP programming data pin.
18	MCLR/VPP/RE3 MCLR VPP RE3	 I P I	Master Clear (input) or programming voltage (input). Master Clear (Reset) input. This pin is an active-low Reset to the device. Programming voltage input. Digital input.
19	RA0/AN0 RA0 AN0	 I/O I	 Digital I/O. Analog input 0.

20	RA1AN1 RA1 AN1	 I/O I	 Digital I/O. Analog input 1.
21	RA2/AN2/VREF-/CVREF RA2 AN2 VREF- CVREF	 I/O I I O	Digital I/O. Analog input 2. A/D reference voltage (low) input. Analog comparator reference output.
22	RA3/AN3/VREF+ RA3 AN3 VREF+	 I/O I I	Digital I/O. Analog input 3. A/D reference voltage (high) input.
23	RA4 T0CKI C1OUT RCV	I/O I O I	Digital I/O. Timer0 external clock input. Comparator 1 output. External USB transceiver RCV input.
24	RA5/AN4/SS/HLVDIN/C2OUT RA5 AN4 SS HLVDIN C2OUT	 I/O I I I O	Digital I/O. Analog input 4. SPI™ slave select input. High/Low-Voltage Detect input. Comparator 2 output.
25	RE0/AN5/CK1SPP RE0 AN5 CK1SPP	 I/O I O	Digital I/O. Analog input 5. SPP clock 1 output.
26	RE1/AN6/CK2SPP RE1 AN6 CK2SPP	 I/O I O	Digital I/O. Analog input 6. SPP clock 2 output.
27	RE2/AN7/OES RE2 AN7 OES	 I/O I O	Digital I/O. Analog input 7. SPP output enable output.
28	VDD	P	Positive supply

			for logic and I/O pins
29	VSS	P	Ground reference for logic and I/O pins.
30	OSC1/CLKI OSC1 CLKI	 I I	Oscillator crystal or external clock input. Oscillator crystal input or external clock source input. External clock source input. Always associated with pin function OSC1
31	OSC2/CLKO/RA6 OSC2 CLKO RA6	 O O I/O	Oscillator crystal or clock output. Oscillator crystal output. Connects to crystal or resonator in Crystal Oscillator mode. In RC mode, OSC2 pin outputs CLKO which has 1/4 the frequency of OSC1 and denotes the instruction cycle rate. General purpose I/O pin.
32	RC0/T1OSO/T13CKI RC0 T1OSO T13CKI	 I/O O I	Digital I/O. Timer1 oscillator output. Timer1/Timer3 external clock input.
33	NC/ ICRST/ICVPP ICRST ICVPP	 I P	No Connect or dedicated ICD/ICSP port

			Reset.(3) Master Clear (Reset) input. Programming voltage input.
34	NC/ ICPORTS ICPORTS	 P	No Connect or 28-pin device emulation.(3) Enable 28-pin device emulation when connected to VSS.
35	RC1/T1OSI/CCP2/UOE RC1 T1OSI CCP2 UOE	 I/O I I/O O	Digital I/O. Timer1 oscillator input. Capture 2 input/Compare 2 output/PWM 2 output. External USB transceiver OE output.
36	RC2/CCP1/P1A RC2 CCP1 P1A	 I/O I/O O	Digital I/O. Capture 1 input/Compare 1 output/PWM 1 output. Enhanced CCP1 PWM output, channel A.
37	VUSB	O	Internal USB 3.3V voltage regulator output.
38	RD0/SPP0 RD0 SPP0	 I/O I/O	Digital I/O. Streaming Parallel Port data.
39	RD1/SPP1 RD1 SPP1	 I/O I/O	Digital I/O. Streaming Parallel Port data.
40	RD2/SPP2 RD2 SPP2	 I/O I/O	Digital I/O. Streaming Parallel Port data

41	RD3/SPP3 RD3 SPP3	 I/O I/O	Digital I/O. Streaming Parallel Port data.
42	RC4/D-/VM RC4 D- VM	 I I/O I	Digital input. USB differential minus line (input/output). External USB transceiver VM input.
43	RC5/D+/VP RC5 D+ VP	 I I/O I	Digital input. USB differential plus line (input/output). External USB transceiver VP input.
44	RC6/TX/CK RC6 TX CK	 I/O O I/O	Digital I/O. EUSART asynchronous transmit. EUSART synchronous clock (see RX/DT).

I= input O= output P= power

- This PIC contains five ports:-

1. PORTA is a bidirectional I/O port. (Pins 19, 20, 21, 22, 23, 24)
2. PORTB is a bidirectional I/O port. PORTB can be Software programmed for internal weak pull-ups on all inputs. (Pins 8, 9, 10, 11, 14, 15, 16, 17)
3. PORTC is a bidirectional I/O port. (Pins 32, 35, 36, 42, 43, 44, 1)
4. PORTD is a bidirectional I/O port or a Streaming Parallel Port (SPP). These pins have TTL input buffers when the SPP module is enabled. (Pins 38, 39, 40, 41, 2, 3, 4, 5)
5. PORTE is a bidirectional I/O port. (Pins 25, 26, 27, 12, 13, 33, 34, 37).

The PIC has been programmed using four input pins and other two output pins by the C++ software program. Then the code has been simulated on the 18F4550 PIC SIMULATOR Program to make an indicator about the nature of the output.

F-1 The Programming Code For Mixing Process Passed On C++ Language

```
#include<adc.h>
#include<p18f4550.h>
#include<delays.h>


#pragma config FOSC = INTOSC_HS
#pragma config WDT = OFF
#pragma config LVP = OFF

void main(void)
{
int temp,rh1,rh2;
int v1,v2,i;
float t1,t2,pvs1,pvs2,pv1,pv2,w1,w2,w3,h1,h2,h3,m1,m2,m3,tt1,tt2,th1,th2,z,t3,shr1;
float ram p=101.325;
float ram cpa=1;
float ram cpv=1.86;
float ram hg=2501.3;
```

```
float ram shr=0.7;

float ram l1=0.5;

float ram l2=0.30;

float ram v=4;

int tin=24;

int hin=48;

float ram r1=0.3;

float ram r2=0.2;

float ram th11,th22;


TRISD=00;


th11=th22=0.0;

v1=v2=1;


//pvs1=5.0;

//pvs2=4.0;

ADCON1=11;


OpenADC (ADC_FOSC_2  & ADC_RIGHT_JUST & ADC_2_TAD,ADC_CH0 &
ADC_INT_OFF & ADC_REF_VDD_VSS , ADC_4ANA);


while(1)

{

ConvertADC();
```

```
while(BusyADC());
temp=ReadADC();
t1=(temp/0.0048)/0.1;



ADCON0=ADCON0 & 0b11110001;
//SetChanADC(ADC_CH1);
ConvertADC();
while(BusyADC());
temp=ReadADC();
t2=(temp/0.0048)/0.1;

ADCON0=ADCON0 & 0b11110010;
//SetChanADC(ADC_CH2);
ConvertADC();
while(BusyADC());
rh1=ReadADC();

ADCON0=ADCON0 & 0b11110011;
//SetChanADC(ADC_CH3);
ConvertADC();
while(BusyADC());
rh2=ReadADC();
```

```
pvs1= 0.004697*t1*t1-0.02122*t1+0.8083;

pvs2= 0.004697*t2*t2-0.02122*t2+0.8083;


pv1=rh1*pvs1;

pv2=rh2*pvs2;


w1=0.622* (pv1/(p-pv1));

h1=cpa+((cpv*t1+hg)*w1);


w2=0.622* (pv2/(p-pv2));

h2=cpa+((cpv*t2+hg)*w2);


w3=((m1*w1)+(m2*w2))/m3;

h3=((m1*h1)+(m2*h2))/m3;


t3=(h3-(hg*w3))/(cpa+(hg*w3));

shr1=(tin-t3)/(hin-h3);


if(shr1!=shr)

{

h3=((shr*hin)-tin+t3)/shr;

z=((m3*h3)-(m1*h1))/(m1*h1);

th1=(m3)/(v*l1*r1*(1+z));
```

```
th2=(z*m3)/(v*l2*r2*(1-z));




if(th1>th11)
{
PORTDbits.RD0=1; //forward
PORTDbits.RD1=1;

tt1=(th1-th11)/v1;
temp=(int)tt1*10;
for( i=0;i<temp;++i)
 Delay10KTCYx(20);
//delay
PORTDbits.RD1=0;
}
else if(th1<th11)
{
PORTDbits.RD0=0; //backward
PORTDbits.RD1=1;

tt1=(th11-th1)/v1;
temp=(int)tt1*10;
for( i=0;i<temp;++i)
 Delay10KTCYx(20);
```

```
//delay
PORTDbits.RD1=0;


}
if(th2>th22)
{
PORTDbits.RD2=1; //forward
PORTDbits.RD3=1;

tt2=(th2-th22)/v2;
temp=(int)tt2*10;
for( i=0;i<temp;++i)
 Delay10KTCYx(20);

//delay
PORTDbits.RD3=0;


}
else if(th2<th22)
{
PORTDbits.RD2=0; //backward
PORTDbits.RD3=1;

tt2=(th22-th2)/v2;
```

```
temp=(int)tt2*10;
for( i=0;i<temp;++i)
 Delay10KTCYx(20);

//delay
PORTDbits.RD3=0;

}

th11=th1;
th22=th2;

}

}
        }
```