بسم الله الرحمن الرحيم

Palestine Polytechnic University

College of Engineering & Technology

Mechanical Engineering Department

Graduation Project

Air Conditioning of "C" Building in PPU Using VRV System

Project Team

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Palestine Polytechnic University
(PPU)

Hebron-Palestine

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According to the project supervisor and according to the agreement of the Testing committee members, this project is submitted to the Department of Mechanical Engineering at college of engineering and technology in partial fulfillment of the requirements of (B.A) the bachelor's degree.

Department Head Signature Supervisor Signature

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Examination committee

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May- 2011

Dedication

To my friends...
To my parents...
To who love me...

To whom help me...
To my parents...
To who love me...

To all who find of his death a way of life to others.

To all martyrs who were killed by no fault of their own except wishing a flourishing future to their home

To all who is troubled by his conscience and loyalty.

To all who violently love their homes and whom swords was broken without touching their determination.

To all the mothers who bring, raise, and present obliged us to have an ever-increasing recognition of their greatness.

To our supervisor Eng. Mohammed Awad

To our country
To the souls of Palestine martyrs
To the freedom fighters
To whom their guidance and support made this work possible

Fadi zamarah

Wisam Hrebat

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And, finally, our ultimate thanks go to all lecturers and doctors, engineers, and laboratories supervisors. Their efforts and their nice dealing with us improved our characters to become successful Engineers in the future.

Abstract

A Veriable Refrigerant Volume (VRV) is a new system in air conditioning and refrigeration, and more recntly began to be used widely in our country, where ther are many companys in the world produce thies system such as Daikin company.

This system is considerd as one of the most efficient systems compared with the well known systems regarding specifications,control, installation and cost. In this project a comprehensive analysis and application of this system in one of the main buildings,C-building, in PPU is to be performed.

ملخص:

نظام التكييف الذكي أو ما يعرف ب (VRV)هو نظام جديد في التكييف والتبريد وحديثا بدا تطبيقه في بلادنا حيث انه يوجد عدة شركات في العالم تنتج هذا النظام ومن أهم هذه الشركات شركة دايكن . يعتبر هذا النظام مقارنة مع الأنظمة المعروفة الأخرى من أكثر الأنظمة فعالية من حيث التحكم والأداء والتكلفة. و وفي هذا المشروع سيتم دراسة وتحليل هذا النظام وتطبيقه على إحدى البنايات الرئيسة في الجامعة وبالتحديد مبنى C .

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List of Symbols

Q	The cooling load .
A	Area.
U	Overall heat transfer coefficient.
Φ	Relative humidity
T	Temperature.
$\dot{V}$	The volumetric flow rate.
v	Velocity
C_p	Specific heat at constant pressure.
h	The convection heat transfer coefficient.
CLF	Cooling load factor
SHG	Solar heat gain factor
SC	Shading coefficient
$\dot{m}$	Mass flow rate
ρ	Density
LHG	Latent Heat Gain
SHG	Sensible Heat Gain
JD	Jordan Dinar

Chapter One

INTRODUCTION

1.1 Introduction

The air conditioning has become one of the most important engineering branches that directly affect human being life. The applications of this engineering branch reach every house and organization. Air conditioning is used in many wide applications mainly in human comfort ,electronic industry and other wide applications.

The air conditioning is a branch of the thermo-mechanical engineering that is specialized in the engineering methodology of obtaining a suitable environment as needed. And applying this methodology on their designs and maintenance of the air conditioning systems.

1.2 General Overview of Project

We will study and anylise the variable refregeration volume system and apply this system in "C" building in palestine polytechnic university,takeing into account the disign considerations applied in heating and cooling systems .And we will selection the componentes of the system .

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Objectives of this project can be as:

- Studying and analyzing the variable refrigeration volume (VRV OR VRF) system .
- Comparison of VRV with other systems .
- Applying this system to "C" building.
- Making a cost analysis of this system.

1.3 Importance of project.

Air conditioning is important in the general life aspects, the importance of this project comes from the direct relation between the air conditioning engineering and the human being life.

The air conditioning is used to create an environment of conditions need to be fit. In this project, the importance comes from designing air conditioning system that create more human comfort ,high Efficiency operation, design versatility ,high Reliability, easy Installation, convenience ,easy service and maintenance.

1.4 Budget

Table 1.1 Budget

TASK	COST(NIS)
USING INTERNET	200
TRANSPORTATION	100
COPY FROM LIBRARY	50
PRINTING PAPER	300
TOTAL	650

1.5 Project Outline

The proposal composed of five chapters as follows:-

Chapter One:- Introduction

This chapter includes an overview main objectives, budget, and the oroject outline.

Chapter Two:- VRF System

This chapter includes a detailed review ,study and analysis of (VRV) system.

Chapter Three: General Design Parameters

This chapter includes the main climatic design parameters of Hebron city, the inside and outside conditions of "C" building , and ASHRE comfort level.

Chapter Four: Calculations

This chapter includes the calculation of the overall heat transfer coefficients, cooling and heating loads calculations for the building .

Chapter Five: Equipment Selection

This chapter includes the detail drawings (plans) using AutoCAD program, and the selection of system equipment.

Chapter Six: Equipment Selection

This chapter includes the cost of the VRV system of C building and conclusion

1.6 Time Planning

The project plan follows the following time schedule, which includes the related task of study and system analysis.

1.6.1 The First Semester Time Plan

Task/week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Collecting information																
General study about the project																
human comfort																
VRV system																
Cooling load calculation																
Project documentation																

Figure 1.1: The First Time Plan

1.6.2 The Second Semester Time Plan

Task/week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Heating load calculations																
Pipes and duct calculation																
Equipment selection																
Planning by AutoCAD																
Project documentation																

Figure 1.2: The second time plan

Chapter Two

VARIABLE REFRIGERANT

FLOW SYSTEM

VRV

2.1 Introduction

Variable refrigerant volume (VRV) systems, which were introduced in Japan more than 20 years ago, have become popular in many countries, The technology has gradually expanded its market presence, reaching European markets in 1987, and steadily gaining markets hare throughout the world. VRV systems are used in approximately 50% of medium sized commercial buildings (up to 6,500 m²).

VRV systems are larger in capacity, more complex versions of the ductless multisplit systems, with the additional capability of connecting ducted style fan coil units. They require many evaporators and complex oil and refrigerant management and control system. Also, they need a separate ventilation system.

The term variable refrigerant flow refers to the ability of the system to control the amount of refrigerant flowing to each of the evaporators. This enables the use of many evaporators of different capacities and configurations, individualized comfort control, simultaneous heating and cooling in different zones, and heat recovery from one zone to another. This refrigerant flow control lied at the heart of VRV systems and is the major technical challenge as well as the source of many of the systems advantages.

Many HVAC professionals are familiar with ductless mini split products. A variation of this product, often referred to as a multi split, includes multiple indoor evaporators connected to a single condensing unit see figure (2-1). Ductless products are fundamentally different from ducted systems. VRV air-conditioning system is in power air-conditioning system, by controlling the amount of circulating refrigerant compressor and access to indoor heat exchanger refrigerant flow in a timely manner to meet the requirements of indoor hot and cold load .

The main advantage of a VRV system is its ability to respond to fluctuations in space load conditions by allowing each individual thermostat to modulate its corresponding electronic expansion valve to maintain its space temperature set point.

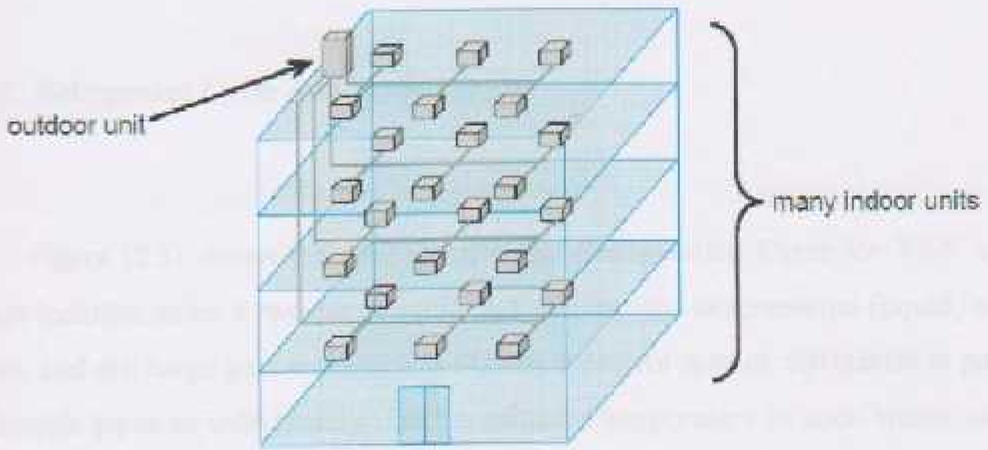


Figure 2.1: Distribution units, internal and external

Figure (2-2) shows main components of VRV system that includes outdoor unit that contains inverter compressor, heat exchanger and multiple indoor units that contain fan coil and electronic expansion valve.

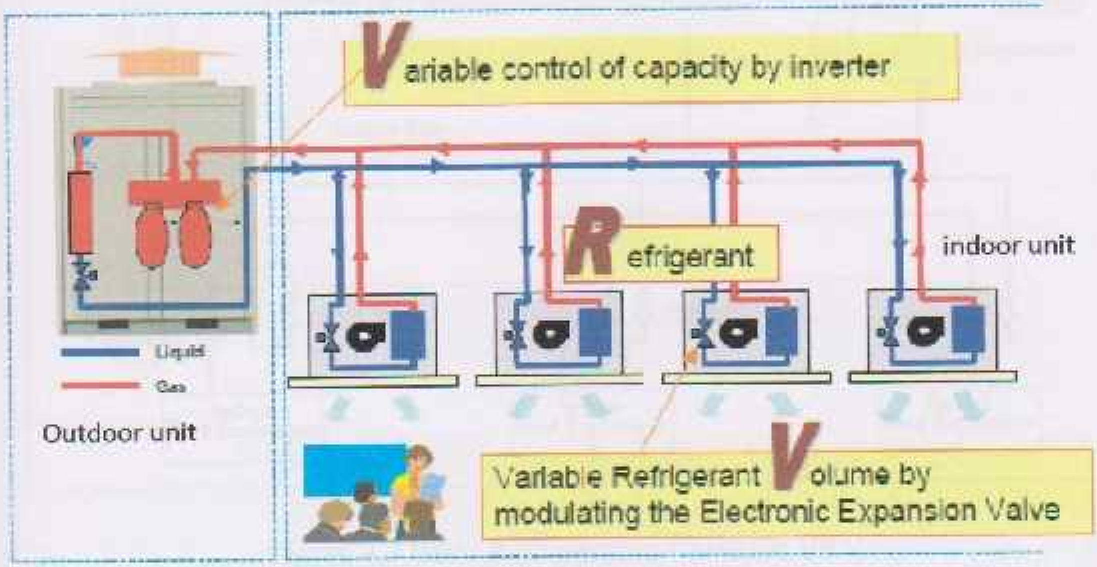


Figure 2.2: main components of VRV system

2.2 Refrigerant Cycle And Components

Figure (2.3) shows the concept of Basic Refrigeration Cycle for VRV system that includes either a two-pipe (liquid and suction gas) or three-pipe (liquid, suction gas, and discharge gas) configuration Using a control system, refrigerant is pumped through pipes to individually sized configured evaporators in each space, each of which can have its own thermostat. It's a closed-loop system so refrigerant is continuously circulated, VRV air conditioning systems need to use inverter compressor to achieve the combination of compressor capacity control and electronic expansion valve to regulate access to indoor unit refrigerant flow.

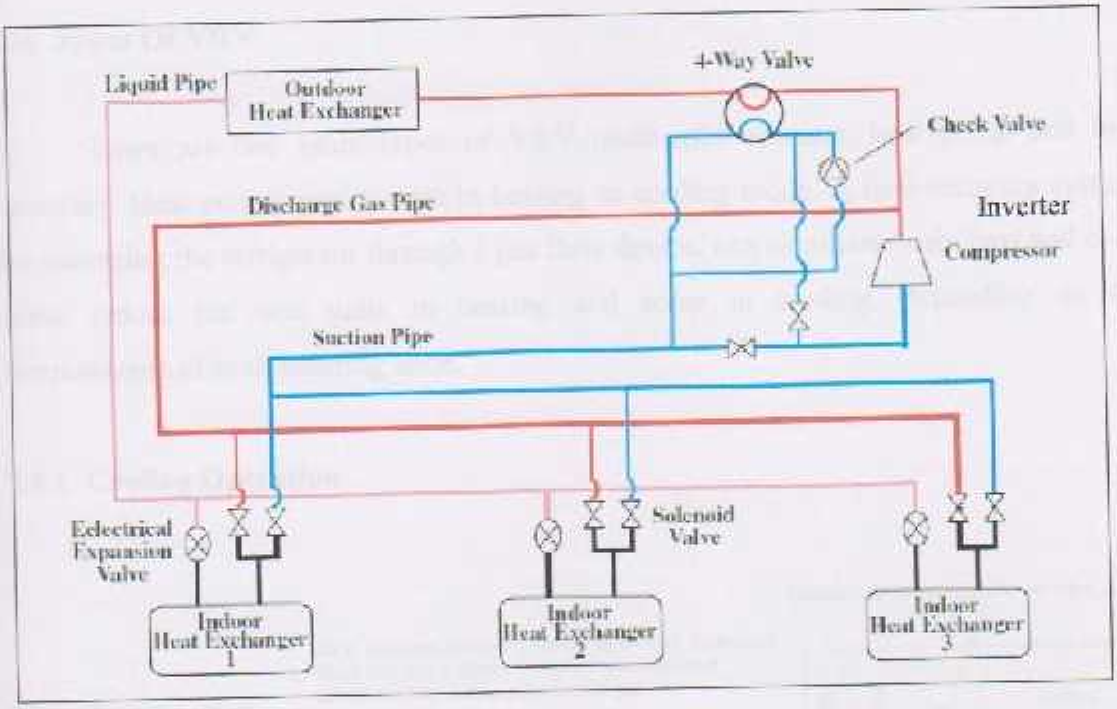


Figure 2.3 Basic Refrigeration Cycle for VRV

2.3 Inverter Compressor

VRV system use inverter controller that enables compressor speed to vary according to the cooling/heating load and therefore consume only the power necessary to match that load. The 50 Hz frequency of the power supply is inverted to a higher or lower frequency according to the required capacity to heat or cool the room. If a lower capacity is needed, the frequency is decreased and less energy is used. Under partial load conditions, the energy efficiency is higher. If the compressor rotates more slowly because less capacity is needed, the coil becomes virtually oversized. Improved efficiencies can therefore be achieved than are possible with non inverter compressors, which always run at the same speed.

2.4 Types Of VRV

There are two basic types of VRV multi-split systems: heat pump and heat recovery. Heat pumps can operate in heating or cooling mode. A heat-recovery system, by managing the refrigerant through a gas flow device, can simultaneously heat and cool, some indoor fan coil units in heating and some in cooling, depending on the requirements of each building zone.

2.4.1 Cooling Operation

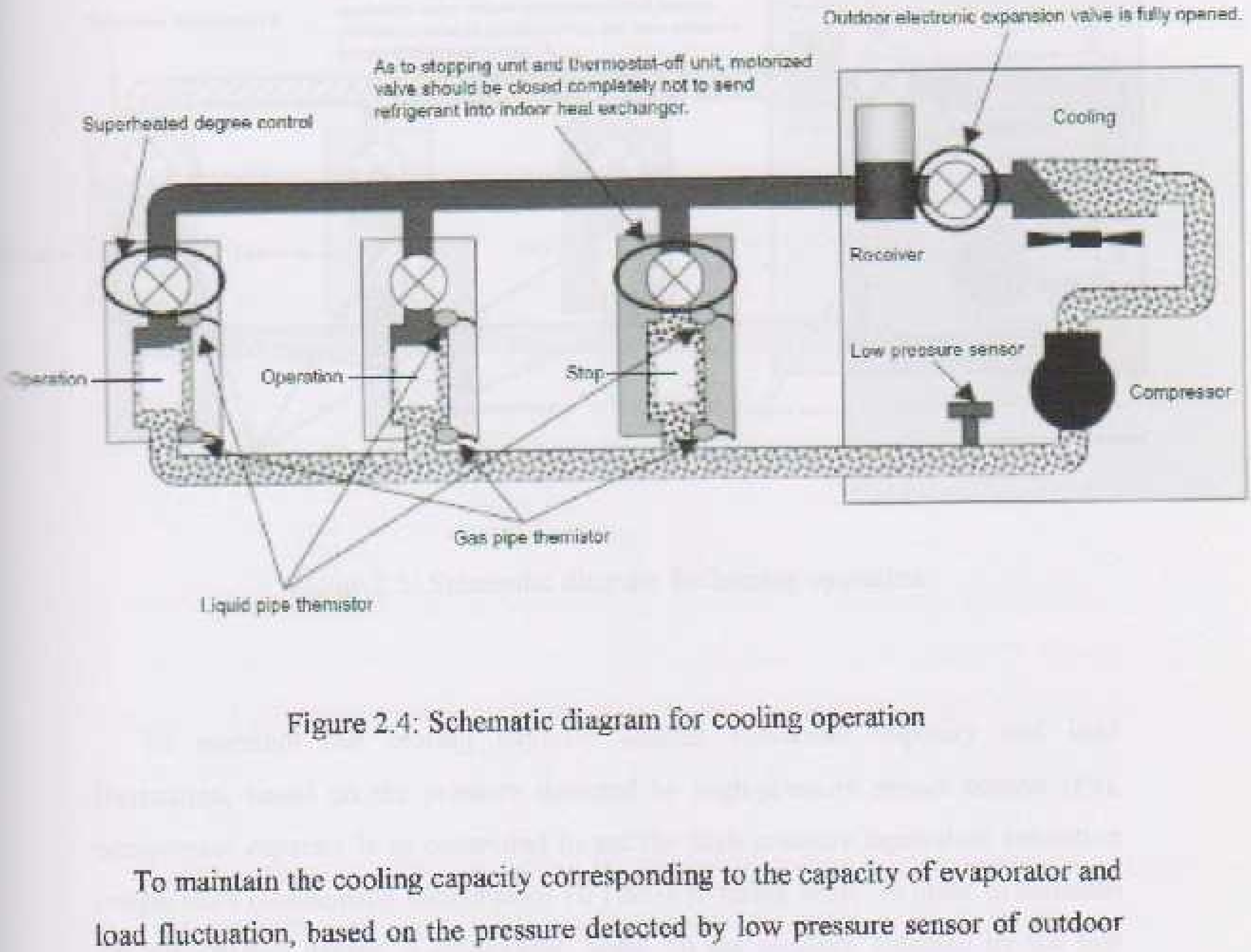


Figure 2.4: Schematic diagram for cooling operation

To maintain the cooling capacity corresponding to the capacity of evaporator and load fluctuation, based on the pressure detected by low pressure sensor of outdoor unit (Pe), the compressor capacity is so controlled to put the low pressure equivalent saturation temperatures (evaporation temperature = Te) close to target value. In order to maintain the superheated degree in evaporator and to distribute proper refrigerant flow rate in spite of different loads on every indoor unit, based on the temperature detected by thermostats of liquid pipes and gas pipes, indoor electronic expansion valve is so regulated as to put superheated degree at evaporator outlet close to target value. Superheated degree equal (indoor gas pipe temperature – indoor liquid pipe temperature).

2.4.2 Heating Operation

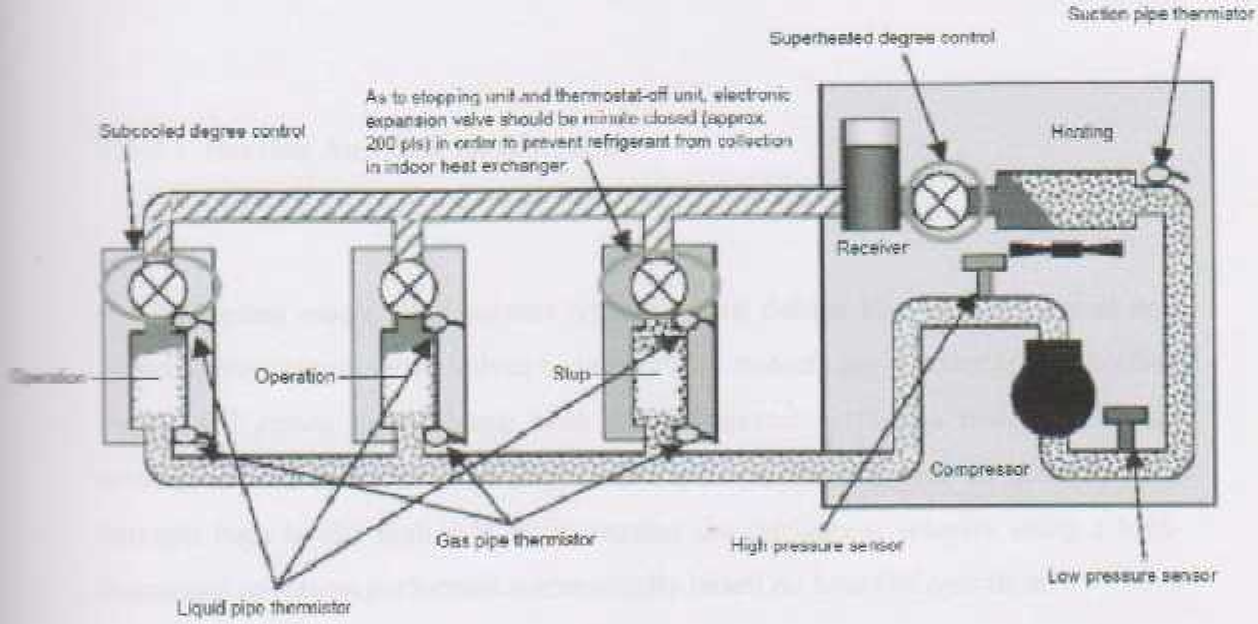


Figure 2.5: Schematic diagram for heating operation

To maintain the heating capacity against condenser capacity and load fluctuation, based on the pressure detected by high-pressure sensor control (Pc), compressor capacity is so controlled to put the high pressure equivalent saturation temperature (condensing temperature Tc) close to target value. In order to maintain the superheated degree in evaporator, based on the pressure detected by the low pressure sensor (Te) and the temperature detected by the thermostat of suction pipe, outdoor electronic expansion valve is so controlled as to put superheated degree at evaporator outlet close to target value. Superheated degree equal (outdoor suction pipe temperature – outdoor evaporating temperature).

In order to distribute proper refrigerant flow rate in spite of different loads on every indoor unit, based on the pressure detected by the high pressure sensor of outdoor unit (Tc) and the temperature detected by the thermistor of indoor liquid pipes, indoor electronic expansion valve is so controlled as to put subcooled degree at condenser outlet close to target value. Subcooled degree equal (outdoor condensing temperature – indoor liquid pipe temperature).

2.4.2.1 Heating And Defrost Operation

In heating mode, VRVsystems typically must defrost like any mechanical heat pump, using reverse cycle valves to temporarily operate the outdoor coil in cooling mode. Oil return and balance with the refrigerant circuit is managed by the microprocessor to ensure that any oil entrained in the low side of the system is brought back to the high side by increasing the refrigerant velocity using a high-frequency operation performed automatically based on hours of operation.

The fan coils are constant air volume, but use variable refrigerant flow through an electronic expansion valve. The electronic expansion valve reacts to several temperature-sensing devices such as return air, inlet and outlet refrigerant temperatures, or suction pressure. The electronic expansion valve modulates to maintain the desired set point.

2.4.2.2 Heat Recovery Operation

Heat recovery operation is achieved using either 3 pipes or 2 pipes (depending on manufacturer). See figure. 2.6 and 2.7 . The 2 pipe heat recovery system has a central branch controller with two pipes from the outdoor unit and 2 pipes to each indoor unit . For mixed mode operation the branch controller separates a mixture of saturated liquid and vapor delivered by the outdoor unit so that each indoor can receive high pressure liquid or vapor. In both cases the liquid produced by indoor units in heating mode is then used to serve indoor units in cooling mode and improved energy saving is possible.

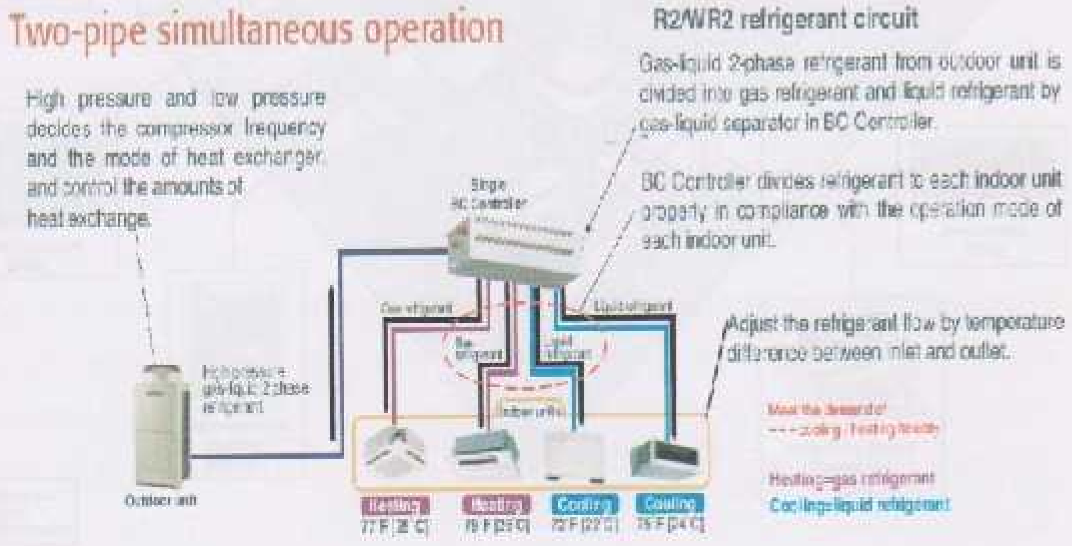


Figure 2.6 Two pipe heat recovery system

The three pipe heat recovery system has a liquid line, a hot gas line and a suction line from the outdoor unit. Each indoor unit is branched off from the 3 pipes using solenoid valves. An indoor unit requiring cooling will open its liquid line and suction line valves and act as an evaporator. An indoor unit requiring heating will open its hot gas and liquid line valves and will act as a condenser.

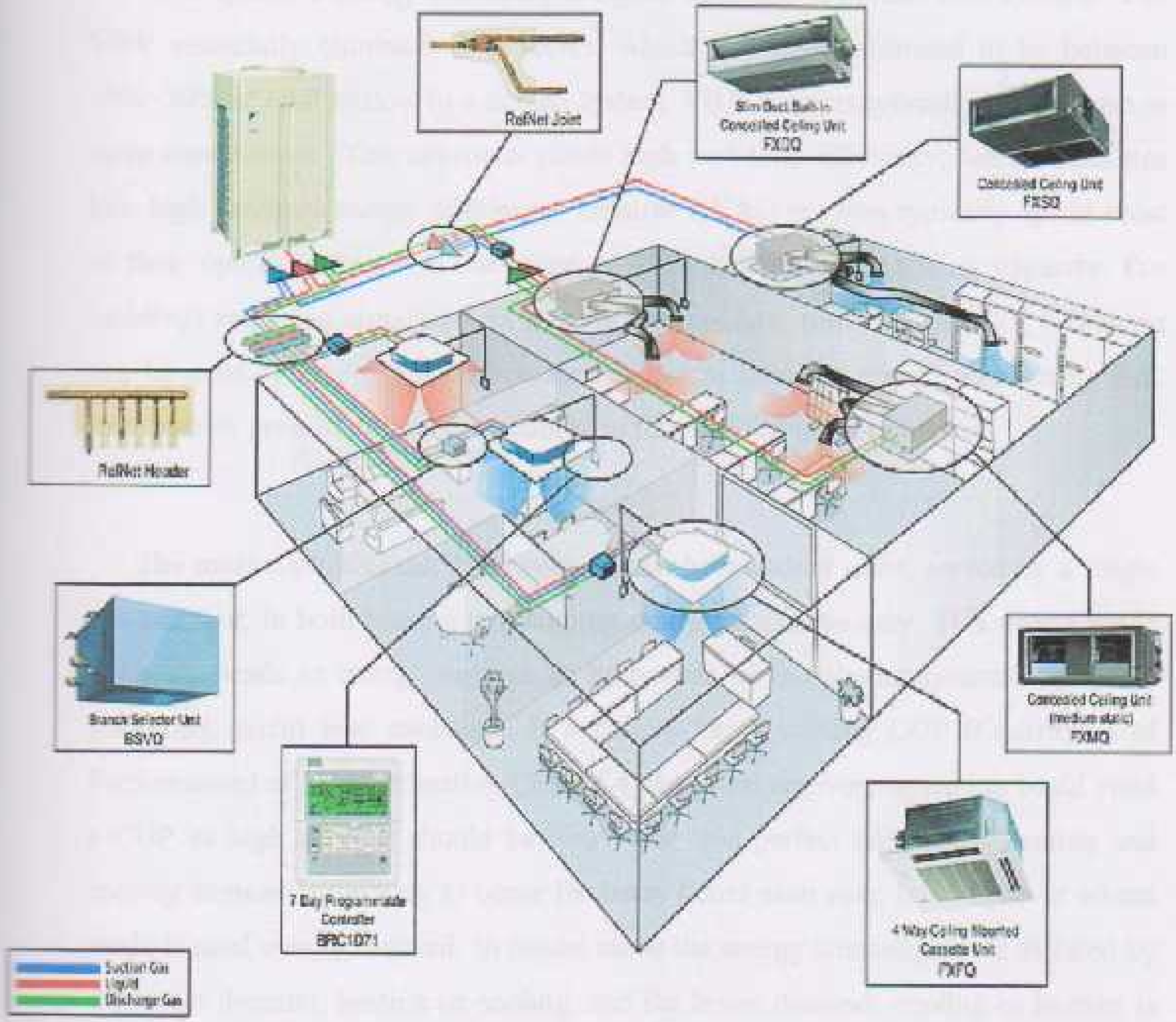


Figure 2.7 pipe heat recovery system

2.5 Features

2.5.1 High-Performance VRV System For Cold Climate

VRV system's energy efficiency is higher than that of normal duct systems. The VRV essentially eliminate duct losses, which are often estimated to be between 10%-20% of total airflow in a ducted system. VRV systems typically include two or three compressors. This approach yields high part-load efficiency, which translates into high seasonal energy efficiency, because HVAC systems typically spend most of their operating hours in the range of 40% to 80% of maximum capacity. For buildings requiring simultaneous heating and cooling, heat recovery VRV systems can be used. These systems circulate refrigerant between zones, transferring heat from indoor units of zones being cooled to those of zones being heated.

The most sophisticated VRV systems can have indoor units, served by a single outdoor unit, in both heating and cooling modes simultaneously. This mixed mode operation leads to energy savings as both ends of the thermodynamic cycle are delivering useful heat exchange. If a system has a cooling COP (Coefficient of Performance) of 3, and a heating COP of 4, then heat recovery operation could yield a COP as high as 7. It should be noted that this perfect balance of heating and cooling demand is unlikely to occur for many hours each year, but whenever mixed mode is used energy is saved. In mixed mode the energy consumption is dictated by the larger demand, heating or cooling, and the lesser demand, cooling or heating is delivered free. Units are now available to deliver the heat removed from space cooling into hot water for space heating, domestic hot water or leisure applications, so that mixed mode is utilized for more of the year.

2.5.2 Operation Range

VRV can be Installation in extreme temperature conditions is possible due to an increase in operational range cooling mode can be operated from : -15°C to 46°C and heating mode can be operated from -20°C to 21°C see figure 2.8 below.

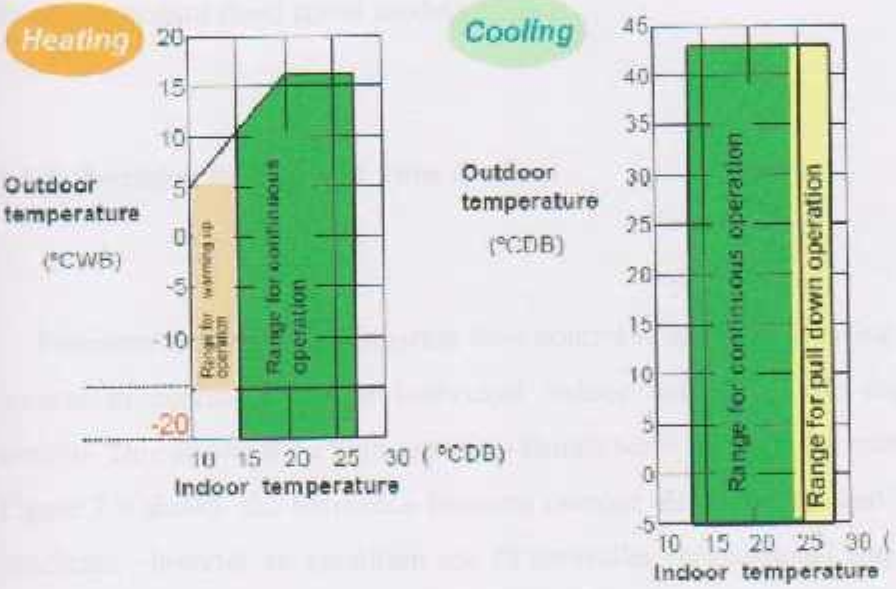


Figure 2.8 Operation Range

2.5.3 Low Noise Mode

Two low noise modes can be selected automatically by quiet priority setting and capacity priority setting depending on the usage environment and outside temperature load. Reductions in outdoor unit sound pressure levels have been achieved via:

- redesigned fan blades and inlet bell mouth
- a new high efficiency aero spiral fan with backward curved blade tips that reduce air turbulence and pressure loss.

- the redesigned bell mouth air inlet fitted with guide vanes at the intake that also reduces air turbulence around the blades

Using the latest technology, sound pressure levels down to 47 dB(A) in cooling (3 HP) are achieved. The sound pressure levels therefore up to 3 dB(A) lower than those of standard fixed speed models.

2.5.4 Precision Refrigerant Flow Control

Precision and Smooth refrigerant flow control is achieved by using a DC Inverter control in conjunction with individual indoor unit electronic expansion valve control. This allows for a high precision comfortable temperature control of ±0.5°C. Figure 2.9 shows the deference between inverter air condition and Traditional air condition , inverter air condition use PI controller (proportional and integral) that response to the temperature of the air deviating from the setpoint, the proportional and integral control signals occur simultaneously. The proportional component provides a relatively fast response to the deviation from the setpoint. The integral component is used to drive the controlled variable back toward the setpoint, eliminating the offset characteristic of proportional control. The two signals are additive.

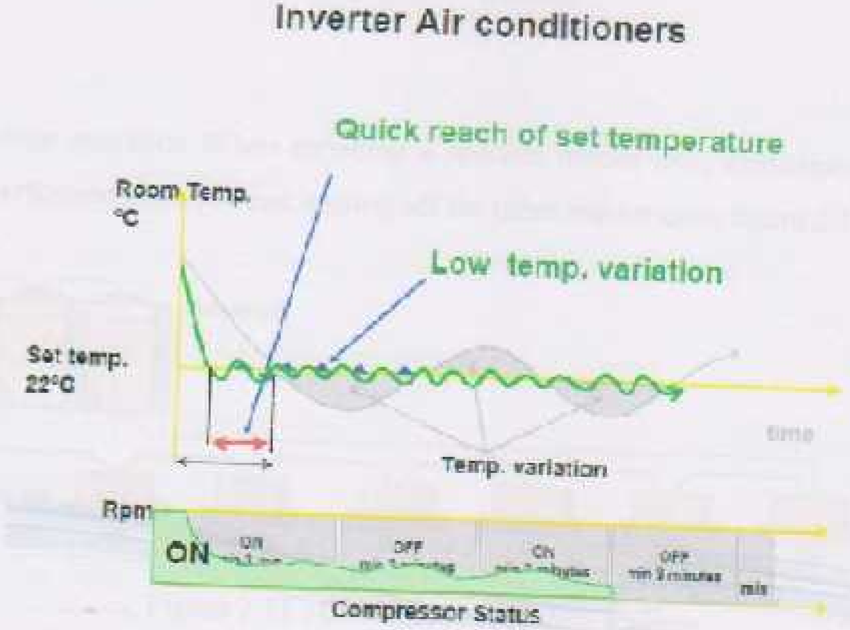


Figure 2.9: Inverter air conditioners

2.5.5 Individual Air Conditioning Control

The desired temperature conditions of each room are met due to the Individual thermostat control of each indoor unit see Figure 2.10

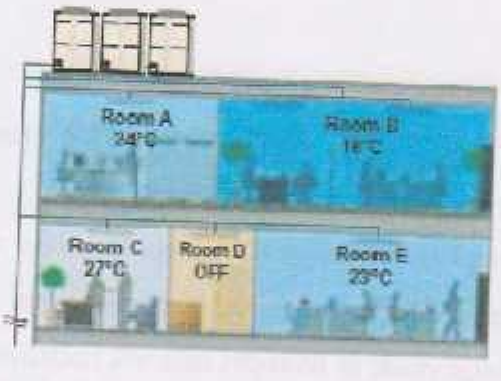


Figure 2.10: Individual Air Conditioning Control

2.5.6 Continuous Operation During Maintenance

Non-stop operation When servicing a specific indoor unit, maintenance can be performed even without turning off the other indoor units figure 2.11

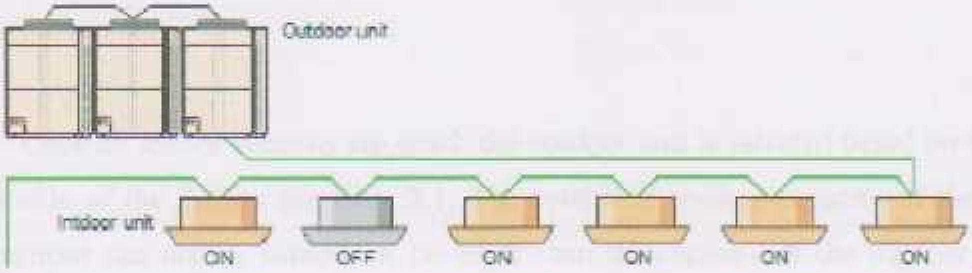


Figure 2.11 : Continuous operation scheme

2.5.7 Space saving

The VRV system allows you to use the available space more efficiently. Instead of having to incorporate a machine room in to your building plans, you can use this space for other purposes, such as a garage.

2.6 General Design Consideration

2.6.1 User Requirements

The user primarily needs space conditioning for occupant comfort. Cooling, dehumidification, and air circulation often meet those needs, although heating, humidification, and ventilation are also required in many applications. Components other than the base outdoor and indoor units may need to be installed for VRVsystems to satisfy all requirements.

2.6.2 Diversity And Zoning

The complete specification of a VRV system requires careful planning. Each indoor section is selected based on the greater of the heating or cooling loads in the area it serves. In cold climates where the VRV system is used as the primary source for heating, some of the indoor sections will need to be sized based on heating requirements.

Once all indoor sections are sized, the outdoor unit is selected based on the load profile of the facility see table 2.1. The combined cooling capacity of the indoor sections can match, exceed, or be lower than the capacity of the outdoor section connected to them. An engineer can specify an outdoor unit with a capacity that constitutes anywhere between 70% and 130% of the combined indoor units capacities. The design engineer must review the load profile for the building so that each outdoor section is sized based on the peak load of all the indoor sections at any given time. Adding up the peak load for each indoor unit and using that total number to size the outdoor unit likely will result in an unnecessarily oversized outdoor section.

Although an oversized outdoor unit in a VRV system is capable of operating at lower capacity, avoid over sizing unless it is required for a particular project due to an anticipated future expansion or other criteria. Also, when indoor sections are greatly oversized, the modulation function of the expansion valve is reduced or entirely lost. Most manufacturers offer selection software to help simplify the optimization process for the system's components.

Table 2.1 Sizing Example

Peak cooling load for Zone 1	3 ton
Peak cooling load for Zone 2	2.5 ton
Peak cooling load for Zone 3	4 ton
Zones peak load = 3 + 2.5 + 4	9.5 ton
Available sizes for outdoor unit	7.5 ton and 10 ton
Selection: Unless additional indoor units are planned for the future, select a 10 ton outdoor section.	

2.6.3 Installation

In deciding if a VRV system is feasible for a particular project, the designer should consider building characteristics, cooling and heating load requirements; peak occurrence; simultaneous heating and cooling requirements; fresh air needs; electrical and accessibility requirements for all system components; minimum and maximum outdoor temperatures; sustainability; and acoustic characteristics. The physical size of the outdoor section of a typical VRV is somewhat larger than that of a conventional condensing unit, with a height up to 6 ft. (1.8 m) excluding supports. The chosen location should have enough space to accommodate the condensing unit(s) and any clearance requirements necessary for proper operation

2.6.4 Refrigerant Piping Design

Building geometry must be studied carefully so that refrigerant piping lines are properly designed. The system should not be considered if the expected pipe lengths or height difference exceed those listed in the manufacturer's catalog. In buildings where several outdoor locations are available for the installation of the outdoor units, such as roof, and ground floor, each condensing section should be placed as close as possible to the indoor units it serves. Although manufacturers routinely increase the maximum allowable refrigerant pipe run, the longer the lengths of refrigerant pipes, the more expensive the initial and operating costs. For most VRV units, see figure 2.12

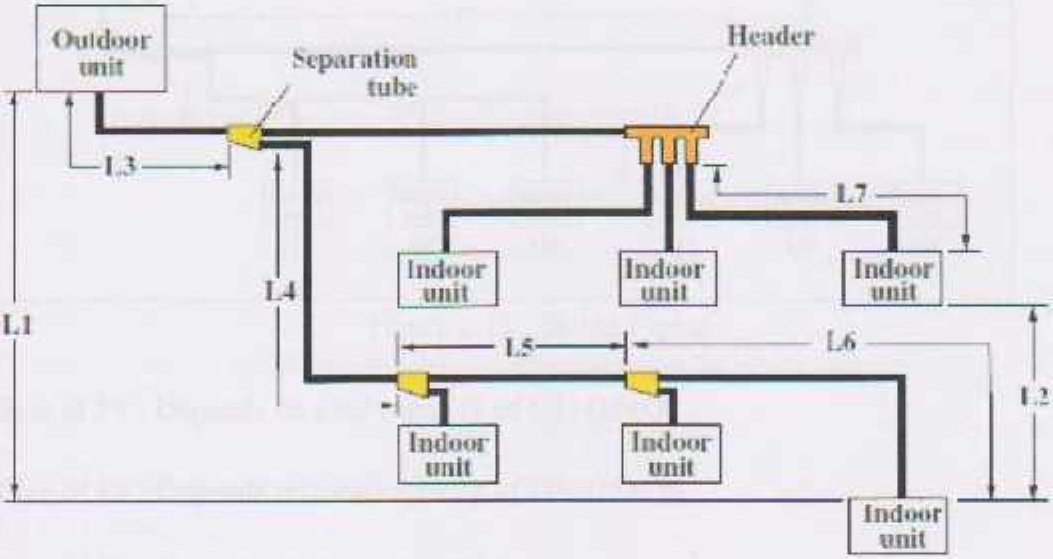


Figure 2.12 : Maximum Allowable Distances and Piping Lengths

Maximum height difference between:

1-Outdoor unit and indoor unit : 50m [L1]

2. Indoor unit and indoor unit : 15m [L2]

3. R.B. unit and R.B. unit : 15m [L8]

4. R.B. unit and indoor unit : 3m [L9]

Maximum piping length

1. From outdoor unit to first separation tube : 70m [L3]

2. From outdoor unit to last indoor unit : 100m [L3+L4+L5+L6]

3. From header or separation tube to indoor unit : 40m [L6], [L7]

Total piping length : 200m (Liquid pipe length), Figure 2.13 show sizing of pipe that depende on load in indoor unite.

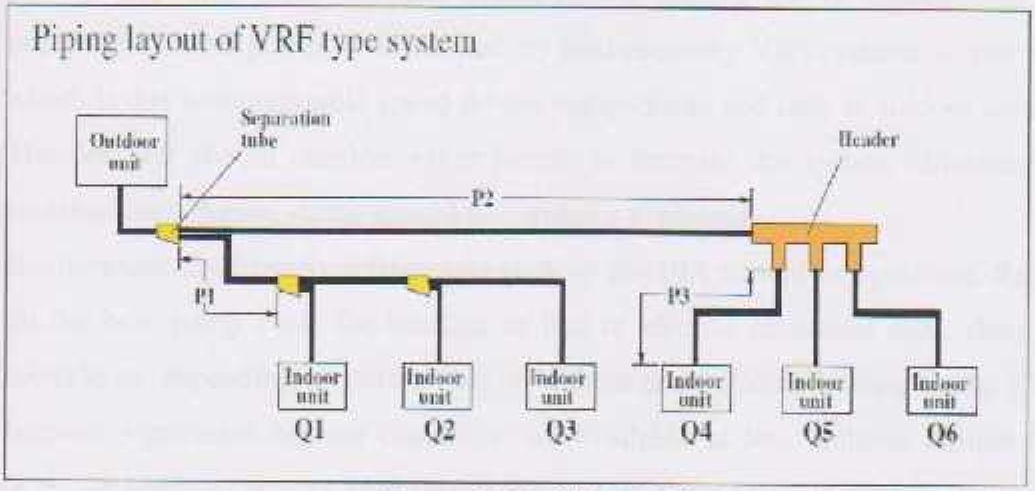


Figure 2.13 : Sizing Piping

Size of P1 : Depends on total capacity of Q1+Q2+Q3

Size of P2 : Depends on total capacity of Q4+Q5+Q6

Size of P3 : Depends on capacity of Q4

2.6.5 Maintenance Considerations

Ductless VRV indoor units have some considerations in reference to maintenance:

- Draining condensate water from the indoor and outdoor units
- Changing air filters
- Repairs
- Cleaning

Ease of maintenance depends on the relative position of the indoor and outdoor units and the room to ensure access for changing filters, repairing, and cleaning. The installer must make sure there is enough slope to drain condensate water generated by both the indoor and outdoor units. Depending on the location where the indoor unit is installed, it may be necessary to install a pump so that water drains properly.

2.6.7 Sustainability

VRV systems feature higher efficiencies in comparison to conventional heat pump units. Less power is consumed by heat-recovery VRVsystems at part load, which is due to the variable speed driven compressors and fans at outdoor sections. The designer should consider other factors to increase the system efficiency and sustainability. Again, sizing should be carefully evaluated.

Environmentally friendly refrigerants such as R-410A should be specified. Relying on the heat pump cycle for heating, in lieu of electric resistance heat, should be considered, depending on outdoor air conditions and building heating loads. This is because significant heating capacities are available at low ambient temperatures (e.g., the heating capacity available at 5°F (– 15°C) can be up to 70% of the heating capacity available at 60°F (16°C), depending on the particular design of the VRV system).

2.6.8 Advantages And Disadvantages Of VRV

VRV systems have many advantages over more traditional HVAC units. The advantages and disadvantages for a VRV system, when compared to a chilled system, are presented in Table 2.2.

Table 2.2 : advantages and disadvantages for a VRV system

Item	Description	Variable Refrigerant Flow AC System	Chilled Water AC System
1	Human Comfort	Partial – no humidity control, not so good air distribution	Good – true air conditioning
2	Process cooling, heating, humidification and dehumidification	Not applicable - no humidity control, not so good air distribution	Good - May by designed for any condition
3	Internal Air Quality	Partial – needs a auxiliary air make-up system and special filters No duct work is good	Good – may be designed for any condition. Ducts need to be cleanable

4	Initial Cost	Similar	Similar
5	Operational Cost	Little higher at full load 1.25 kW/ton	At full load 1.18 kW/ton
6	Cooling capacity	Good performance until 100 m equivalent length Poor performance above 100 m equivalent length	Distance is only a matter of pumps' selection and operational power consumption
7	Increasing cooling capacity	Not so easy, it may be necessary to change the refrigerant lines and the condensing unit	It could be done by changing the control valves and or the coils. Chiller plant doesn't change or chilled water pipes
8	Operation at partial load	Good performance and control	Good performance and control
9	Customer or tenant control on the operational cost	Good - full control Very important	No control on the operational and maintenance cost
10	Compatibility with standards, guides and regulations	Partial. It is necessary to solve the compatibilities issue during the design	Fully compatible
11	Long distance pipes	Up to 100 m is OK, more there is a cooling capacity reduction up to 75%	No problem.
12	Refrigerant management	Difficult it depends on the design of the system for monitoring, identification and repair	Concentrate in a single equipment easy and simple – Good
13	Customer operation	Easy and simple – Good Very important	Not so clear to customer - Acceptable
14	Malfunction Possibility	To many parts and components and long refrigerant lines – Acceptable	More reliable, just a few parts and equipment – Good
15	Operational life expectation	Up to 15 years - Acceptable	Up to 25 years – Good
16	Maintenance	Depends on the design, access may be a problem	No problem - Good
17	Sales strategy	It is necessary to verify, the say what the customer would like to listen, but not all is true	To much engineering stuff, difficult for the costumer to understand

Chapter Three

GENERAL DESIGN

PARAMETER

3.1 Introduction

In this chapter we will determine the general design parameters necessary for the design process, these parameters are divided into two categories:

1. the general outside condition. Such as the wind velocity, the average relative humidity, the outside temperature, the ground mean temperature.
2. Inside design parameters that satisfy human comfort .

3.2 The General Outside Design Parameters

Building located in Hebron, at longitude 35.1 degree east, latitude of 31.5 degree north. And the elevation is 1010 m from the sea level.
(see table 3.1)

Table 3-1 Climate information to the station

Station Name	Hebron
Longitude	35.1 East
Latitude	31.5 North
Height above sea level	1010 m

3.2.1 Dry Bulb Temperature:

The first outside design parameter is the temperature, for the Hebron its located in climate area of the West Bank. And the temperature records for the region at which the refrigeration station located is in the figure below.

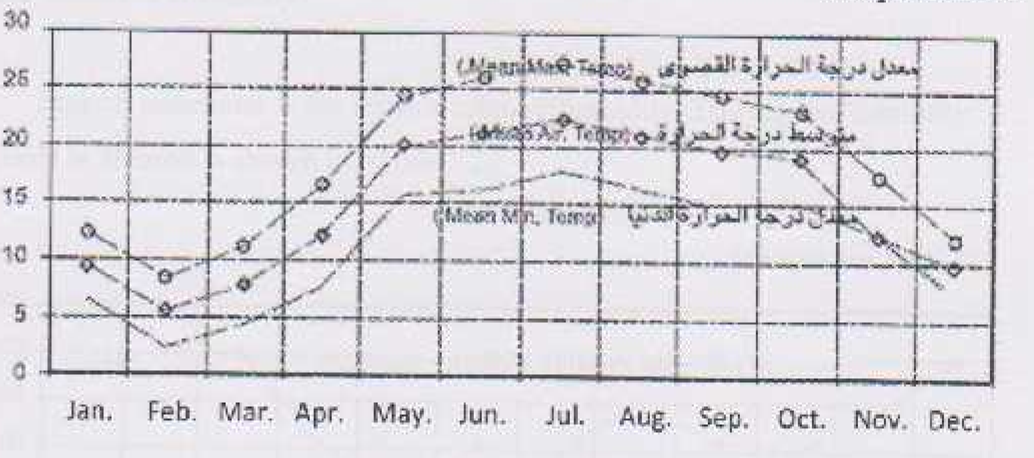


Figure 3-1: Temperature chart

As shown in this record. The maximum temperature is about 28 C. and the minimum temperature 3 C. the maximum temperature that is gained last for about 4 months. By taking in consideration the high temperature waves that happens in this region which may reach 34 C, we will consider 35 C as an outside design temperature.

And the maximum and minimum values for the temperature in the operating months which are shown in Table 3-2, illustrating the design temperature above.

Table 3.2 max. and min. DB. temperature in Hebron

mounth	Jan.	Feb.	Mar.	Apr.	May.	Jun.	Jul.	Aug	Sep.	Oct.	Nov	Dec.
Mean Max. Temp.(c°)	10.2	11.5	14.6	19.6	23.6	25.9	27.2	27.2	26.0	23.2	17.5	12.1
Mean Min. Temp.(c°)	4.0	4.7	6.5	9.9	13.2	15.8	17.0	17.0	15.9	14.0	9.9	5.6
Absolute Max. Temp.(c°)	21.4	21.0	23.6	32.6	34.0	33.5	38.0	33.4	34.6	31.6	31.6	22.0
Absolute Min.Temp.(c°)	-0.1	-3.0	-0.5	1.0	6.5	10.0	13.0	12.0	12.0	9.0	2.0	-0.4

The temperature of the ground is over the valued design temperature by about 8 degree centigrade. And so the ground design temperature will be considered to be 43 °C.

3.2.2 Relative Humidity:

Second parameter is the outside relative humidity. The relative humidity records in Hebron is shown in figure 3-2:

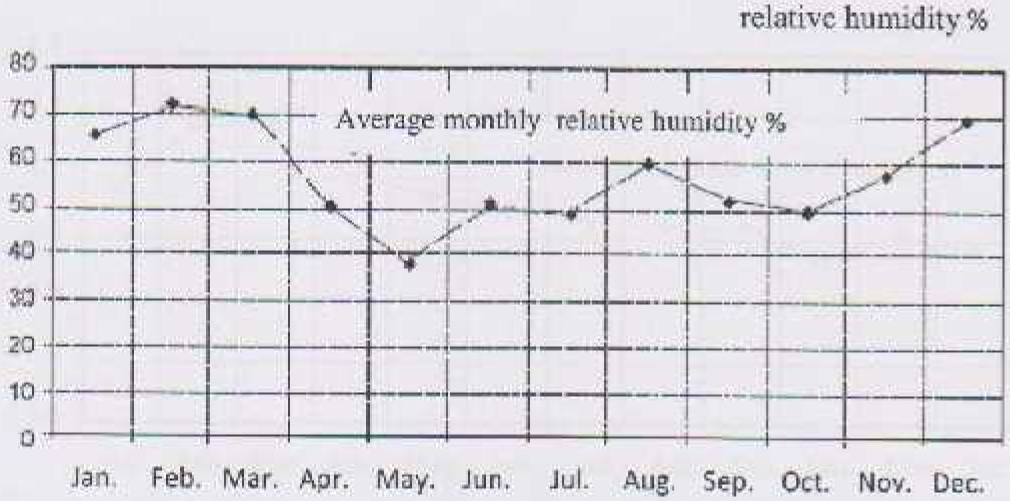


Figure 3-2: Outside relative humidity chart

The maximum relative humidity is in February, and the minimum is about May. The relative humidity affects on two components; the ventilation system, and the condensation unit. So the design relative humidity is 50%.

For the ventilation system, the outside relative humidity, when the relative humidity of the entering air increases, this will cause an increase in the relative humidity in the building , that will affect on human comfort . In addition, the heat rejection from the ventilation air increases, this causes an increase in the cooling load. This occurs in February. Which has the maximum relative humidity.

And for the condenser, the most critical relative humidity is the minimum relative humidity. Which occurs in May, i.e. specific volume increase, and so the enthalpy will increase, because the heat capacity of an air stream at these circumstances will have less heat capacity. Recording to the psychometric chart of air at the atmospheric pressure at that height.

3.2-3 Wind Velocity

The third outside parameter is the outside wind velocity. That as shown in the figure below. Shows the wind velocity all around the year for Hebron area.

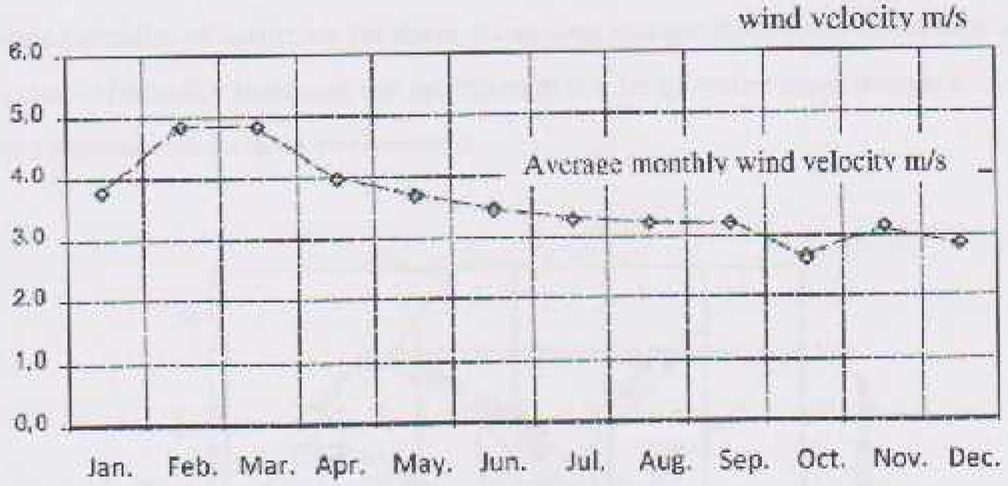


Figure 3-3: Wind velocity chart

The wind velocity ranges from 2.5 m/s to 4.8 m/sec. the wind velocity effects by two ways. Increasing the value of the outsider convection heat transfer coefficient. Which in turn increases the heat gain to the building . And also affecting on the heat rejection from the condensation unit.

3.3 The General Inside Design Parameters

3.3.1 Comfort Chart

The ASHRE comfort chart of figure (3.4) indicate the acceptable zones of selecting the inside operative temperature and the inside relative humidity for winter heating summer cooling and year-round air conditioning application.

The inside operative temperature and the ordinate is the inside humidity ratio or the dew temperature . the acceptable operative temperature values and the inside humidity ratio are considered as standard comfort zones for summer operation , winter operation or year – round application , as indicated on fig (3.4) it can be observed that the minimum operating temperature for comfort winter is 20 C (68 F) db temperature when the inside relative humidity is 50 % thus , the comfort zones fig 3.4 set the limits of both the operative in side temperature and the inside relative humidity of inside air for these zones .one can see from these zones that as the relative humidity increases the operative inside temperature must decrease to keep a desired comfortable environment.

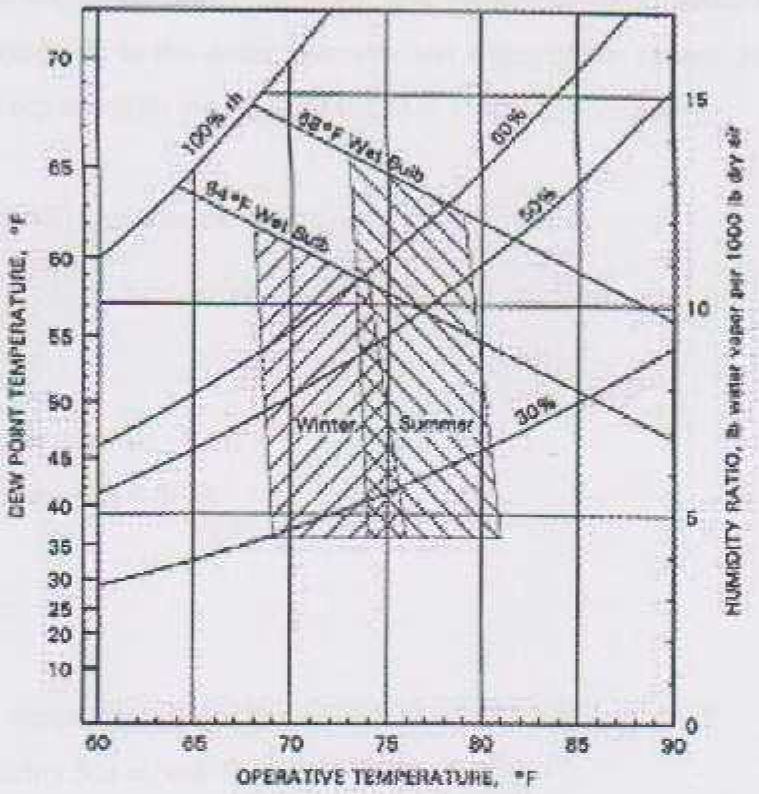


Figure 3.4 comfort zones of operative temperature and relative humidity for winter and summer

The inside design conditions refer to temperature, humidity, air speed and cleanliness of inside air that will induce comfort to occupants of the space at minimum energy consumption. There are several factors that control of selection of the inside design conditions and expenditure of energy to maintain those conditions

The outside design condition, The period of occupancy of the conditional space, the level of activity of the occupants in conditional space. And The type of building and its use.

Usually the range of temperature difference between inside and outside is 12 C . the relative humidity range from the conditioned space varies from 30% - 60% . a dry environment will be felt when the relative humidity falls below 30%, and sickness will be felt at relative humidity above 60%.

The Indore air speed is not designed as a parameter for comfort as long as it moves the treated air to the desired corners and edges of the spaces. However, it is desirable to keep it within the range of 0.1 to 0.35 m/s for comfort.

In this project will consider the inside design conditions as:

For winter:

Dry bulb temperature Td =22 C
Relative humidity RH = 50 %

For summer:

The dry bulb temperature Td = 22 C
Relative humidity RH = 50%

Chapter Four

COOLING AND HEATING CALCULATIONS

4.1 Introduction

The cooling load of a building consists of the following components:

1- heat gain transmitted through building stricter such as walls, floor and ceiling that are adjacent to unconditioned spaces. That heat transmitted is by temperature difference that exists on both sides of structures.

2-heat gain due to solar effect which includes:

a) solar radiation transmitted through the glass and absorbed by inside surface and furniture.

b) solar radiation absorbed by walls, glass windows, glass doors, and roofs that are exposed to solar radiation.

3) sensible and latent heat gain brought into the space as a result of 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 cooking, hot baths, or any other moisture producing equipment

6) sensible and latent heat produced by occupants.

The quantity of heat transmitted through the walls, doors, floor, and roof of space per unit of time is the function of three factors whose relationship is expressed in the following equation :

$$Q = U\,A\,\Delta T \qquad (4.1)$$

Where Q: the rate of heat transferred in watt (W).

U: the overall coefficient of heat transmission in W/m^2. C°.

A: the outsides surface area of the wall (m^2).

ΔT : total equivalent temperature difference (C°) .

But ΔT which takes into consideration the increase of wall temperature due to absorption of solar radiation, so ΔT is called corrected cooling load temperature differences $CLTD_{corr}$.

4.2 Overall heat transfer coefficients of 'C' building

A Wall could be made of several number of layers as shown in Figure 4.1

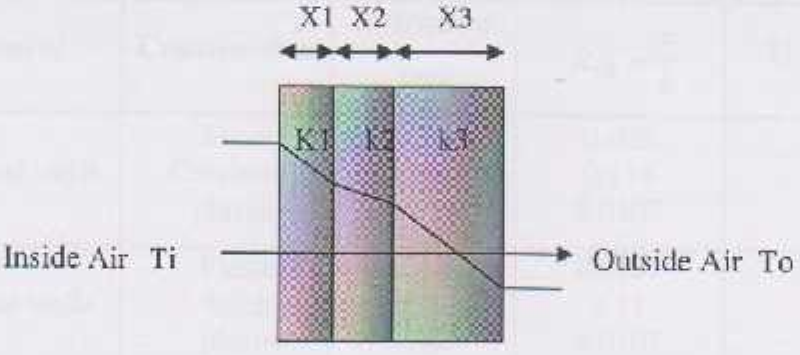


Figure 4.1 :heat transfer through composite material

Overall heat transfer coefficients (U) is identified in the following equation:

$$U = \frac{1}{Rth} = \frac{1}{\frac{1}{hin} + \frac{X1}{K1} + \frac{X2}{K2} + \frac{X3}{K3} + \frac{1}{hout}} \quad (4.2)$$

Where:

→ h_{in}: Convection coefficient (surface Conductance) of inside wall, floor, or ceiling.

→ h_{out} : Convection coefficient (surface conductance) of outside wall, floor, or roof.

→ X : Thickness of constructing material.

→ K : Thermal conductivity of material (see Table A-17 in appendix A)

Overall heat transfer coefficient for walls, roofs, glass, and floors calculated and tabulated as follows:

note: all construction of walls are shown in section plan in appendix (B-1)

Table 4.1 The construction of building

Element	Construction	Thickness (m)	$R_{th} = \frac{x}{k}$	U $W/m^2.c^{\circ}$
External walls	Stone Concrete plaster	0.05 0.20 0.015	0.022 0.114 0.0107	3.35
Internal walls	Plaster brick plaster	0.0107 0.1 0.005	0.0107 0.11 0.0107	2.9
Internal roof	ceramic tiles plaster concrete sand plaster	0.05 0.02 0.31 0.15 0.005	0.045 0.0142 0.177 0.5 0.0035	1.05
External roof	Plaster Concrete Slop concrete water memb. screed	0.005 0.31 0.10 0.01 0.02	0.0035 0.177 0.057 0.0071 0.026	2.37

glass	glass air glass	0.002 0.006 0.002	0.0018 0.24 0.0018	2.35
floor	ceramic tiles concrete sand plaster compact soil damp course concrete	0.05 0.10 0.15 0.02 0.2 0.10 0.10	0.045 0.057 0.5 0.014 0.133 0.07 0.057	0.97

4.3 Corrected cooling load temperature differences $CLTD_{corr}$

The CLTD values vary with hour of the day and it is function of environmental conditions and building parameters. The CLTD tables are derived from computer solutions using the transfer function method.

The value of CLTD extracted from (Table A-8 in appendix A) needs to be corrected so that the actual value is found for different cases, and hence it will be called corrected CLTD and can be calculated from the following equation:

$$CLTD_{corr} = (CLTD + LM)k + (25.5 - T_i) + (T_{O,M} - 29.4)f \tag{4.3}$$

Where LM is latitude correction factor, which is obtained from (Table A-1 in appendix A) for horizontal and vertical surfaces. The factor k is colour adjustment factor such that k = 1.0 for dark colored roof, and k = 0.5 for permanently light coloured roofs.

If the roof construction details are not specified or the construction is different from that specified in (Table A-8 in appendix A), approximate cooling load temperature differences can be obtained from (Table A-18 in appendix A) . This simplified table gives the CLTD for sunlit roofs of light, medium and heavy constructions.

The temperature difference (25.2-T_i) is a correction value for indoor design temperature where T_i is the room or inside design temperature, C . On the other hand, the temperature difference ($T_{o,m}$-29.4) is a correction factor for outdoor mean temperature $T_{o,m}$. It is related to the outdoor design temperature T_o, according the relation:

$$T_{om} = \left(\frac{T_{max} - T_{min}}{2}\right)$$

4.3.1 CLTD correct for all walls and roof in Building

CLTD corrected for walls and roof calculated and tabulated as follows:

TBLE 4-2 CLTD. LM value and $\mathbf{CLTD_{corr}}$

wall	CLTD	LM	$CLTD_{corr}$
N	8	0.5	13.35
NE	11	0.5	14.85
E	14	0.0	16.1
SE	13	-0.5	15.35
S	11	-1.6	13.8
SW	14	-0.5	15.85
W	15	0.0	16.6
NW	12	0.5	15.35
roof	16	0.5	25.6

4.3-2 CLTD Correct For Glass in Building.

The maximum cooling load temperature differences (CLTD) for convection heat gain for glass windows occur at 15:00 hours is 8 °C. From(Table A-13 in appendix A).

When substitute value of LM and CLTD in equation of $CLTD_{corr}$ and record in Table 4-3.

Table 4-3 $CLTD_{corr}$ and LM for glass

glass	LM	CLTD	$CLTD_{corr}$
N	0.5	8	16.15
NE	0.5	8	16.15
E	0.0	8	15.7
SE	-0.5	8	15.3
S	-1.6	8	14.4
SW	-0.5	8	15.3
W	0.0	8	15.7
NW	0.5	8	16.15

4.4 Sample load Calculations

Figure below show classroom in second floor.

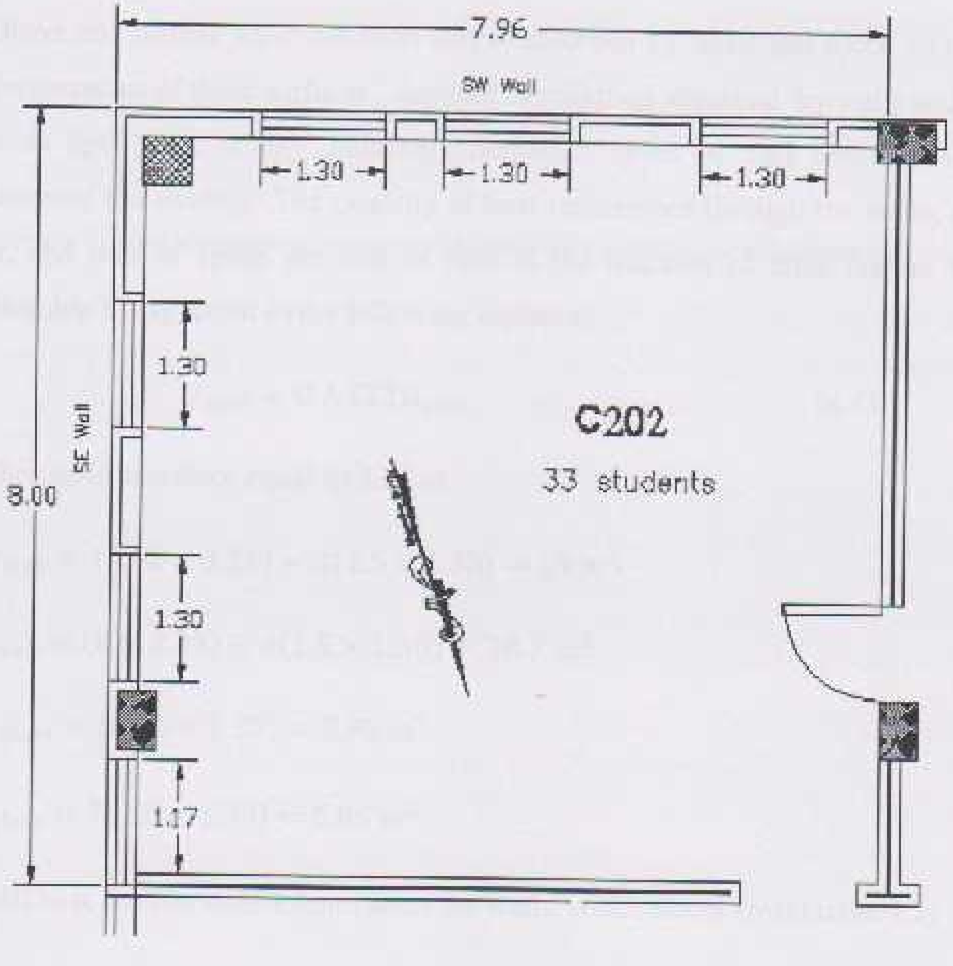


Figure 4.2 C202 Room

4.4.1 Cooling load Calculations For C202 Room.

4.4.1.1 Cooling load From Sunlit Walls And Roof

Direct and diffuse solar radiation that is absorbed by walls and roofs in raising the temperature of these surfaces . Amount of radiation absorbed by walls and roofs depends upon time of day ,building orientation ,types of wall construction and presence of the shading .The quantity of heat transmitted through the walls, doors, floor, and roof of space per unit of time is the function of three factors whose relationship is expressed in the following equation:

$$Q_{wall} = \mathrm{U\ A\ CLTD_{corr}} \qquad (4.4)$$

The height of one floor equal to 3.24 m

$A_{SW\ wall} = (8.02 \times 3.24) - 3(1.5 \times 1.30) = 19\ m^2$

$A_{SE\ wall} = (8 \times 3.24) - 3(1.5 \times 1.30) = 18.7\ m^2$

$A_{SW\ glass} = 3(1.5 \times 1.30) = 5.85\ m^2$

$A_{SE\ glass} = 3(1.5 \times 1.30) = 5.85\ m^2$

Overall heat transfer coefficient values for walls, roofs, floors (from table 4.1)

CLTD values from table (4.2)

$Q_{sw\ wall} = 19 \times 3.35 \times 15.85 = 1008.8\ W$

$Q_{SE\ wall} = 18.7 \times 3.35 \times 15.35 = 961.6\ W$

$Q_{roof} = 58 \times 2.37 \times 25.6 = 3519\ W$

$Q_{sw\ glass} = 5.85 \times 2.35 \times 15.3 = 210\ W$

$Q_{sE\ glass} = 5.85 \times 2.35 \times 15.3 = 210\ W$

Total load of room C202 from walls and roof = 5909.4 W

4.4.1.2 Transmitted Heat Gain Through The Glass

Solar radiation which falls on glass has three components which are

(1) Transmitted component: It represents the largest component, which is transmitted directly into the interior of the building or the space. This component represents about 42 to 87% of incident solar radiation, depending on the glass transmissibility value.

(2) Absorbed component: This component is absorbed by the glass itself and raises its temperature. About 5 to 50% of solar radiation it absorbed by the glass, depending on the absorptive value of the glass.

(3) Reflected component: This component is reflected by glass to the outside of the building. About 8% of the solar energy is reflected back by the glass .

If a certain building has a large area of exposed clear glass then solar radiation is considered a large part of the cooling load. The total cooling load due to exposed glass area is the sum of transmission load due to inside-outside glass surface temperature difference (heat conduction) and heat gain due to solar energy (heat radiation and convection). The amount of solar radiation that can be transmitted through glass depends upon the following factors:

1- Type of glass (Single, double or insulating glass) and availability of inside shading. (Such as venetian blinds, construction overhangs, wing walls, etc.).

2- Hour of the day, day of the month, and month of the year.

3- Orientation of glass area.(North, Northeast, East orientation, etc.)

4- Solar radiation intensity and solar incident angle.

5- Latitude angle of the location.

4.4.1.2.1 Transmission Heat Gain

Heat gain due to solar transmission through glass windows and glass doors is estimated by using Appendix A-10 to A-11 where the following factors are selected:

a) **Solar Heat Gain Factor (SHG):**

This factor represents the amount of solar energy that would be received by floor furniture and the inside walls of the room and can be extracted from (Table A-9 in appendix A)

b) **Shading Coefficient (SC):**

It accounts for different shading effects of the glass wall or window and can be extracted from (Table A-10 in appendix A) for a single and double glass without inside shading or from (Table A-11 in appendix A) for single and double glass , as well as, for insulating glass with internal shading (venetian blinds, curtains, drapes, roller shades, etc.) The shading coefficient, SC is defined as the ratio of solar heat gain of glass window of the pace to the solar heat gain of double strength glass.

c) **Cooling load Factor (CLF):**

This represent the effect of the internal walls, floor, and furniture on the instantaneous cooling load, and can be extracted from (Table A-19 in appendix A) for glass without interior shading or from (Table A-20 in appendix A) for glass with interior shading. It accounts for the variation of SHG factor with time, mass capacity of the structure and the internal shading. The transmitted cooling load is calculated as follows:

$$Q_{tr} = A(SHG)(SC)CLF) \qquad (4.5)$$

The values of the factors SHG(Table A-9 in appendix A), SC(Table A-11 in appendix A), CLF(Table A-20 in appendix A)

$Q_{tr\ SW\ glass} = 5.85(473)(0.51)(0.56) = 790\ W$

$Q_{tr\ SE\ glass} = 5.85(473)(0.51)(0.53) = 748\ W$

4.4.1.3 Cooling load Due To Infiltration

Infiltration is the leakage of outside air through cracks or clearances around the windows and doors. It provides fresh outside air needed for living comfort and health. The amount of this infiltration air depend mainly on the tightness of the windows and doors and on the outside wind speed and its direction or the pressure difference between the outside and inside of the room.

The methods are used to estimate the volumetric flow rate of infiltration into air conditioning spaces are the air change method (ACH) ,and the crack method .

The air change method assumes that the air volume in a space is replaced by outside air at certain number of time per hour .can be used the following equation:-

$$Q_{t.f} = \dot{m}(h_o - h_i) \quad (4.6)$$

Where $\dot{m}$: mass flow rate of air kg/s

h_o: Enthalpy of infiltration air at out temperature and outside relative humidity .

h_i: Enthalpy of infiltration air at inside temperature and inside relative humidity.

Where h_o , v_o, h_i are obtained from psychrometric chart at inside and outside design conditions as follows:

h_o: 91 Kj/Kg dry air

$v_o = 0.903\ m^3/Kg$ dry air

$h_i = 45.5$ Kj/Kg dry air

$v_i = 0.846\ m^3/Kg$ dry air

$h_o - h_i = 91 - 45.5 = 45.5$ kJ/kg dry air

$\dot{m} = \rho\dot{V}$ (4.7)

ρ : density of infiltration air.

$\dot{V}$: volumetric flow rate of infiltration air .

$$\rho = \frac{1}{v_o} = \frac{1}{0.903} = 1.1\ kg/m^3$$

$\dot{V}$ = (number of air change /hour) × volum of space (4.8)

The height of one floor equal to 3.24 m

Volume of room C202 = 7.66×7.57×3.24 =187.5 m^3

Number of air change per hour equal 2 (from Appendix A-7)

$\dot{V} = (2/3600) \times 187.5 = 0.104\ m^3/s$

$\dot{m} = 0.104 * 1.1 = 0.1144\ kg/s$

$Q_{t.f} = 0.1144 * 45.5 = 5.2$ kW

4.4.1.4 Cooling load Due To Ventilation

A ventilation air that mixes with room return air before flowing over the cooling cool. Can be calculation be the following equation :-

$Q_{ven} = \dot{m}(h_o - h_i)$ (4.9)

$\dot{m}$: mass flow rate of ventilation air kg/s .

h_o: Enthalpy of infiltration air at out temperature and out relative humidity .

h_i: Enthalpy of infiltration air at inside temperature and inside relative humidity.

Where h_o , v_o, h_i are obtained from psychrometric chart at inside and outside design conditions as follows:

h_o: 91 kJ/kg dry air

$v_o = 0.903\ m^3/kg$ dry air

$h_i = 45.5$ kj/kg dry air

$v_i = 0.846\ m^3/kg$ dry air

$h_o - h_i = 91 - 45.5 = 45.5$ kj/kg dry air

$$\dot{m} = \frac{\dot{V}}{v_o}$$

$\dot{V}_f$:volumetric flow rate of ventilation air .

v_o: specific volume at T_o and $\emptyset_{out}$.

$\dot{V}$ = number of person ×outside air requirements for mechanical ventilation

outside air requirements for mechanical ventilation is $3 \times 10^{-3}\ m^3/s$ per person (Table A-6 in appendix A)

number of person in room C202 equal 33 student.

$\dot{V} = 33 \times 3 \times 10^{-3} = 0.099\ m^3/s$

v_o at out side temperature and out side relative humidity equal 0.9 (psycumetric chart)

$$\dot{m} = \frac{0.099}{0.9} = 0.11\ kg/s$$

$$Q_{ven} = 0.11 * 45.5 = 5\ kW$$

4.4.1.5 Cooling load Due To Equipment

Sensible and latent loads arising from various equipment(lights, people, computer…etc) and appliances that are installed in a conditioned space are given in Table 3.8. The indicated heat dissipation rates from such equipment and appliances should be included when the cooling load is estimated. Care must be taken when considering such dissipation rates as all sensible or latent or partly sensible and partly latent.

4.4.1.5.1 Cooling load Due To lights

Can be calculation by the following equation:-

$$Q_{lt} = \text{N} \times \text{P} \times \text{CLF} \times \text{Diversity Factor} \qquad (4.10)$$

Where:

N: number of lights.

P: rated power 40 W.

CLF: the light cooling load factor from (Table A-20 in appendix A)

Diversity factor :0.4 for light from (Table A-14 in appendix A)

number of lights in room C202 equal 24 light

CLF =0.36

$$Q_{lt} = 24 \times 40 \times 0.36 \times 0.4 = 138 \text{ W}$$

4.4.1.5.2 Cooling load Due To Computers

Each computer produces heat equal to 100 W

Number of computers in C202 room is one so the load from it is 100 W.

4.4.1.6 Total Cooling load Due To Equipment

$Q_{equipments} = Q_{lt} + Q\ computer.$

$Q_{equipments} = 138 + 100 = 238\ W$

4.4.1.7 Cooling load Due To Occupants

Can be calculation be the following equation:-

$$Q_{O\ total} = Q_{sensible} + Q_{latent} \quad (4.11)$$

$$Q_{latent} = LHG \times \text{No. of people} \times \text{Diversity factor} \quad (4.12)$$

$$Q_{sensible} = SHG \times \text{No. of people} \times CLF \times \text{Diversity factor} \quad (4.13)$$

Where:

LHG : Heat gain latent = 57 W (Table A-4 in appendix A).

SHG : Heat gain sensible =71.5 W (Table A-4 in appendix A).

Diversity factor from Appendix A-14

CLF: cooling load factor (Table A-20 in appendix A) .

$Q_{latent} = 57 \times 33 \times 0.5 = 907.5\ W$

$Q_{sensible} = 71.5 \times 33 \times 0.72 \times 0.5 = 772.2\ W$

$Q_{total} = 907.5 + 772.2 = 1680\ W$

4.4.1.8 Total Cooling load

Total cooling load can be calculation be the following equation:-

$Q_{total} = Q_{walls} + Q_{glass} + Q_{infiltration} + Q_{ventilation} + Q_{equipments} + Q_{occupants}$

$Q_{C202} = 5.91 + 1.54 + 5.2 + 5 + 1.680 + 0.238 = 19.5$ kW

4.4.2 Heating load calculations for C202 Room.

4.4.2.1 Heat Gain Through Sunlit Walls , Glass And Roof

$Q = \text{U A}\,\Delta\text{T}$ (4.13)

Where Q: the rate of heat transferred in watt (W).

U: the overall coefficient of heat transmission in $W/m^2.c^{\circ}$.

A: the outsides surface area of the wall (m^2).

ΔT : total equivalent temperature difference (c°) .

$Q_{sw\ wall} = 19 \times 3.35 \times 18 = 1145$ W

$Q_{sE\ wall} = 18.7 \times 3.35 \times 18 = 1127$ W

$Q_{roof} = 58 \times 2.37 \times 18 = 2474$ W

$Q_{sw\ glass} = 5.85 \times 2.35 \times 18 = 247$ W

$Q_{sE\ glass} = 5.85 \times 2.35 \times 13 = 247$ W

Total load of room C202 from walls and roof = 5240 W

4.4.2.2 Heat Gain Due To Infeltration

$Q_{t.f} = \dot{m}(\mathrm{h_o} - \mathrm{h_i})$

$\dot{m}$: mass flow rate of air kg/s

h_o: Enthalpy of infiltration air at out temperature and out relative humidity in wenter .

$\mathrm{h_i}$: Enthalpy of infiltration air at inside temperature and inside relative humidity in wenter.

Where $\mathrm{h_o}$, $\mathrm{v_o}$, $\mathrm{h_i}$ are obtained from psychrometric chart at inside and outside design conditions as follows:

$\mathrm{h_o}$: 13 Kj/Kg dry air

$\mathrm{v_o} = 0.789\ \mathrm{m^3/Kg}$ dry air

$\mathrm{h_i} = 48$ Kj/Kg dry air

$\mathrm{v_i} = 0.853\ \mathrm{m^3/Kg}$ dry air

$\mathrm{h_i} - \mathrm{h_o} = 48 - 13 = 35$ kJ/kg dry air

$\dot{m} = \rho\dot{V}$

ρ : density of infiltration air.

$\dot{V}$: volumetric flow rate of infiltration air .

$$\rho = \frac{1}{\mathrm{v_o}} = \frac{1}{0.789} = 1.26\ kg/\mathrm{m^3}$$

$\dot{\mathrm{V}}$ = (nomber of aire change /hour) × volum of space .

The height of one floor equal to 3.24 m

Volume For room C202 $= 7.66 \times 7.57 \times 3.24 = 187.5\ \mathrm{m^3}$

Number of air change per hour equal 2 (Table A-7 in appendix A)

$\dot{V} = (2/3600) \times 187.5 = 0.104\ \mathrm{m^3}/s$

$\dot{m} = 0.104 \times 1.26 = 0.131\ kg/s$

$Q_{t.f} = 0.131 \times 35 = 4.6$ kw

4.4.2.3 Heat Gain Due To Ventelation

$Q_{ven} = \dot{m}(\mathrm{h_o} - \mathrm{h_i})$

$\dot{m}$: mass flow rate of ventilation air kg/s ..

$\mathrm{h_o}$: Enthalpy of infiltration air at out temperature and out relative humidity .

$\mathrm{h_i}$: Enthalpy of infiltration air at inside temperature and inside relative humidity.

Where $\mathrm{h_o}$, $\mathrm{v_o}$, $\mathrm{h_i}$ are obtained from psychrometric chart at inside and outside design conditions as follows:

$\mathrm{h_o}$: 13 kJ/kg dry air

$\mathrm{v_o} = 0.789\ \mathrm{m^3/kg}$ dry air

$\mathrm{h_i} = 48$ kJ/kg dry air

$\mathrm{v_i} = 0.853\ \mathrm{m^3/kg}$ dry air

$\mathrm{h_i} - \mathrm{h_o} = 48 - 13 = 35$ kJ/kg dry air

$$\dot{m} = \frac{\dot{V}}{v_o}$$

$\dot{V}_f$:Volumetric flow rate of ventilation air .

v_o: Specific volume at T_o and $\emptyset_{\mathrm{out}}$.

$\dot{V}$ = number of person ×outside air requirements for mechanical ventilation

outside air requirements for mechanical ventilation is $3 \times 10^{-3}\ \mathrm{m^3}/s$ per person (Table A-6 in appendix A)

Average number of person in room C202 is 33 person.

$\dot{V} = (3 \times 10^{-3}) * 33 = 0.099\ \mathrm{m}^3/s$

$\dot{m} = \frac{0.099}{0.789} = 0.125\ \mathrm{kg/s}$

$Q_{ven} = 0.125 \times 35 = 4.4\ \mathrm{kW}$

4.4.2.4 Heating load Due To Equipment

4.4.2.4.1 Heat Gain Due To lights

$Q_{lt} = \mathrm{N} \times \mathrm{P} \times \mathrm{CLF} \times \mathrm{Diversity\ Factor}$

Where:

N: number of lights.

P: rated power 40 W.

CLF: the light cooling load factor (Table A-20 in appendix A)

Diversity factor :0.4 for light from (Table A-14 in appendix A)

For room C202

number of lights for room C202 equal 24 light

CLF = 0.36

$Q_{lt} = 24 \times 40 \times 0.36 \times 0.4 = 138\ \mathrm{W}$

4.4.2.4.2 Heating load Due To Computers

Each computer produces heat equal to 100 W.

Number of computers in C202 room is one so the load from it is 100 W.

4.4.2.5 Total Heating load Due To Equipment

$Q_{equipments} = Q_{it} + Q\ computer$

$Q_{equipments} = 138 + 100 = 238$ W

4.4.2.6 Heat Gain Due To Occupants

Can be calculation be the following equation:-

$Q_{O\ total} = Q_{sensible} + Q_{latent}$

$Q_{latent} = \text{LHG} \times \text{No. of people} \times \text{Diversity factor}$

$Q_{sensible} = \text{SHG} \times \text{No. of people} \times \text{CLF} \times \text{Diversity factor}$

Where:

LHG : Heat gain latent = 57 W (Table A-4 in appendix A).

SHG : Heat gain sensible =71.5 W (Table A-4 in appendix A).

Diversity factor from (Table A-14 in appendix A)

CLF: cooling load factor (Table A-20 in appendix A) .

$Q_{latent} = 57 \times 33 \times 0.5 = 907.5$ W

$Q_{sensible} = 71.5 \times 33 \times 0.72 \times 0.5 = 772.2$ W

$Q_{total} = 907.5 + 772.2 = 1680$ W

4.4.2.7 Total Heating load

Total heating load can be calculation be the following equation:-

$Q_{total} = Q_{walls} + Q_{infiltration} + Q_{ventilation} - Q_{equipments} - Q_{occupants}$

$Q_{C202} = 5.2 + 4.6 + 4.4 - 1.680 - 0.238 = 12.3$ kW

4.5 load Calculation For C202 Room By Using Software.

We used software program (LG software) to calculate the load of the building so we calculated the load of C202 by equations and by using software , by comparative of them we note that the Results are similar.

we consider the room C202 as sample room of the building and have apply the software ,and the following is a review of how to use the program.

In the figure bellow there are the information of the room such as the length,width,and the space name.

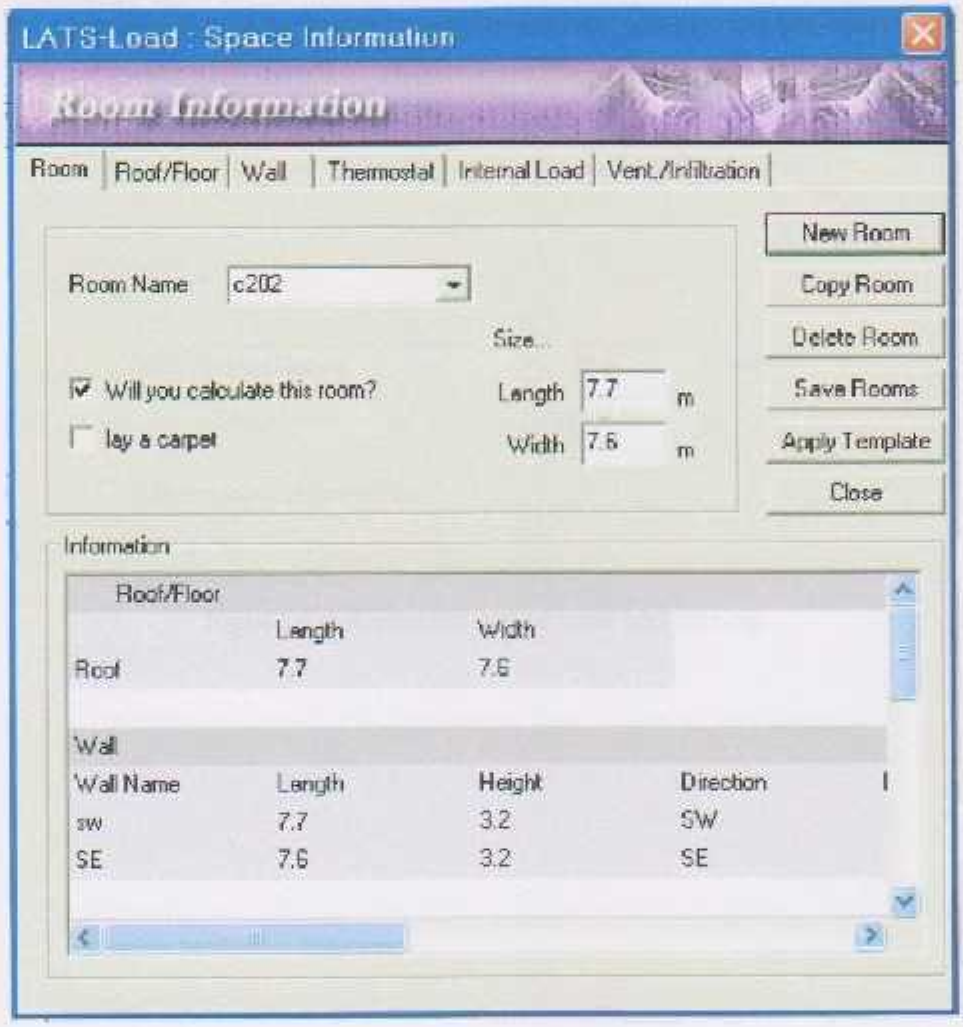


Figure 4.3: Room information

The figure (4.4) explain the construction of the floor and roof for room C202 and the Overall heat transfer coefficient of them and the length ,width of them.

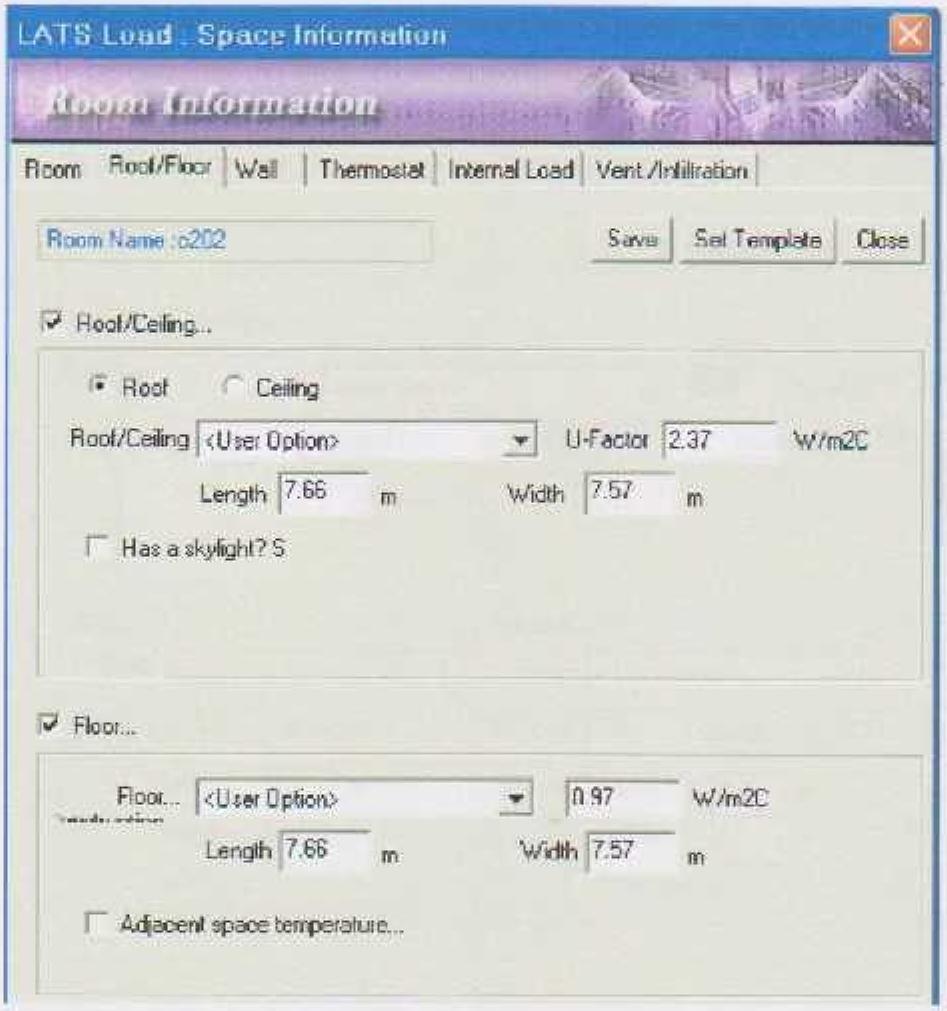


Figure 4.4 :Construction of roof and ceiling

The directions of the walls for room C202 and the constructions of them and the construction of the windows ,doors are shown in figure 4.5

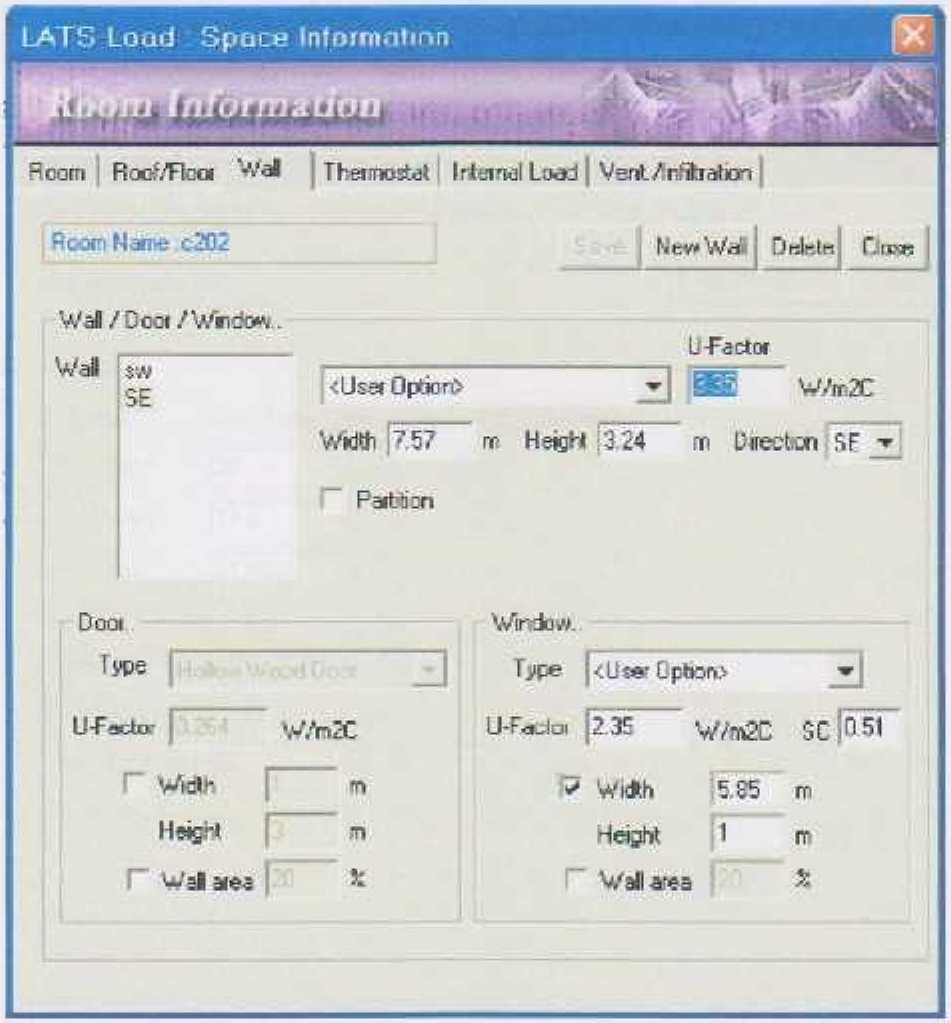


Figure 4.5 :Walls and windows direction

The figure (4.6) shows the indoor conditions in the cooling mode and heating where the indoor dew point temperature in the cooling or the heating is constant and the indoor relative humidity is 51%.

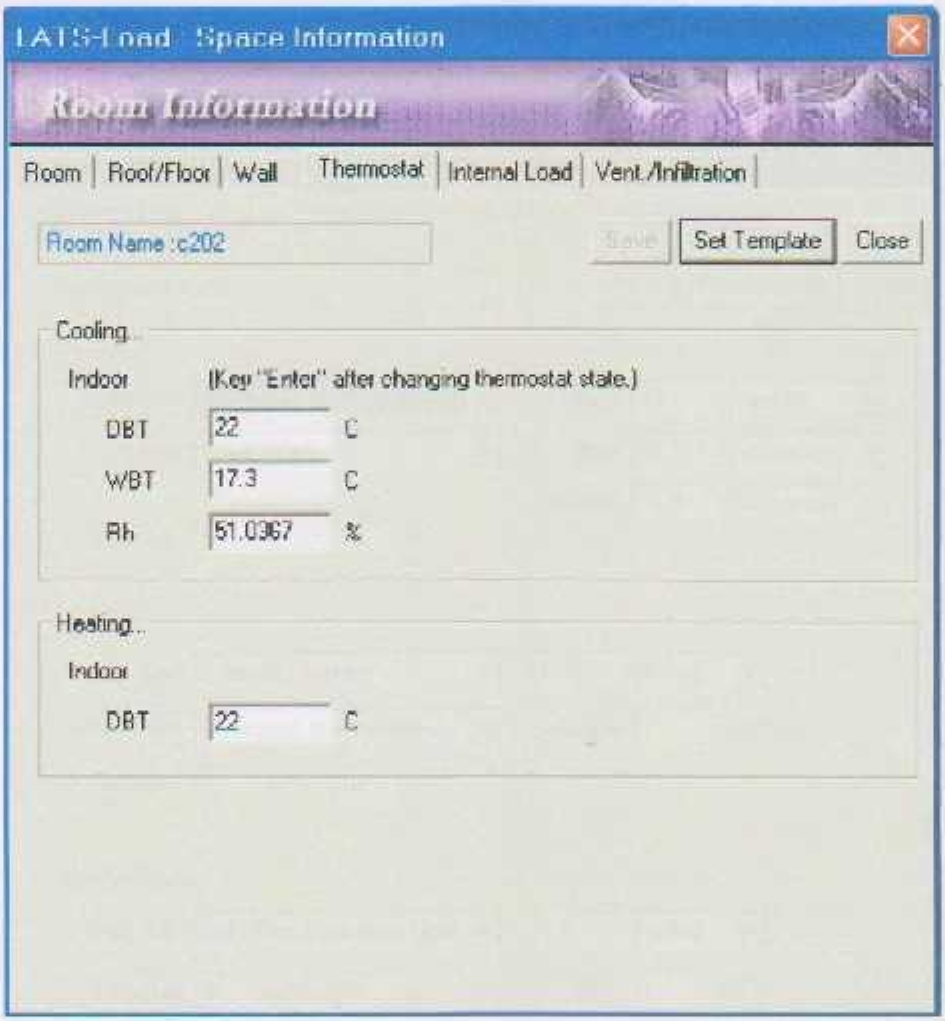


Figure 4.6 :Indoor conditions

The figure below shows the load resulting from existing equipment in the room addition, load resulting from people and lights inside the room and also explains the nature of the existing environment and the nature of the work done by people inside the apartment.

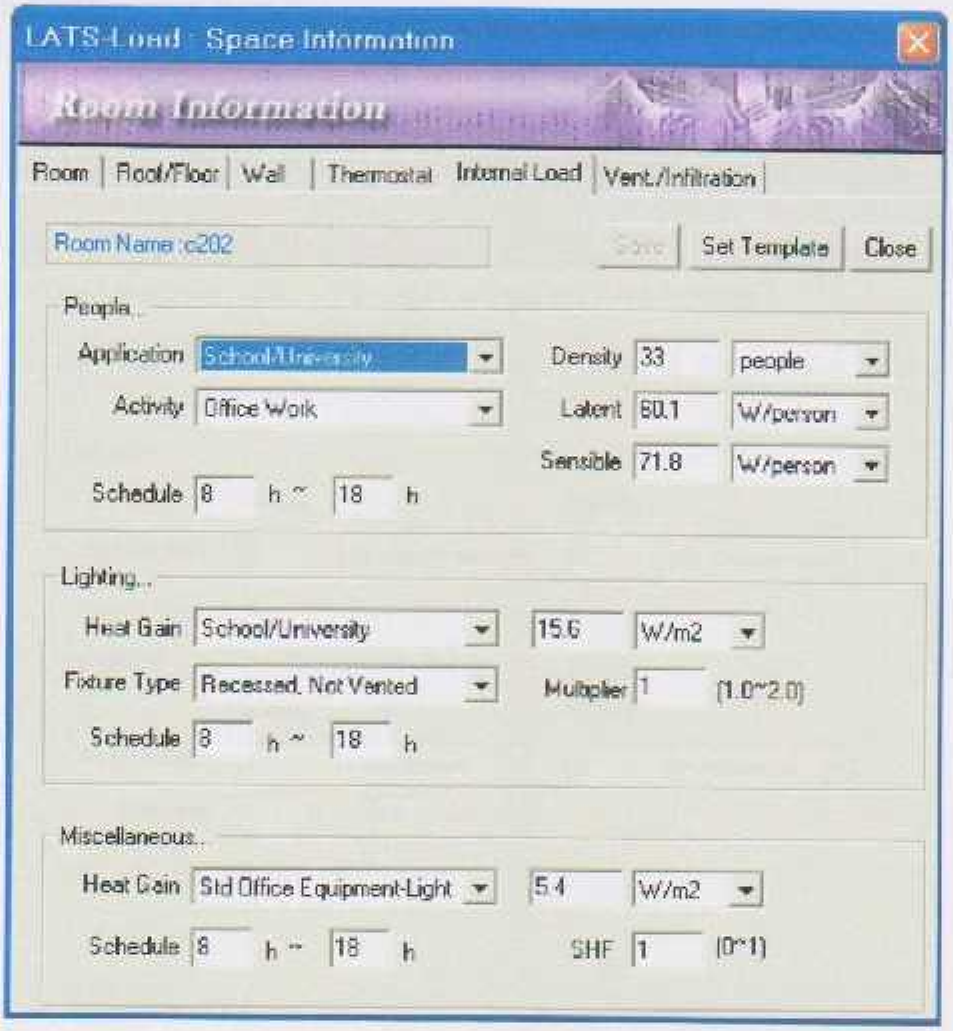


Figure 4.7 :load from people and equipment

In figure 4.8 explains the infiltration and ventilation of the building and the load which resulting from them.

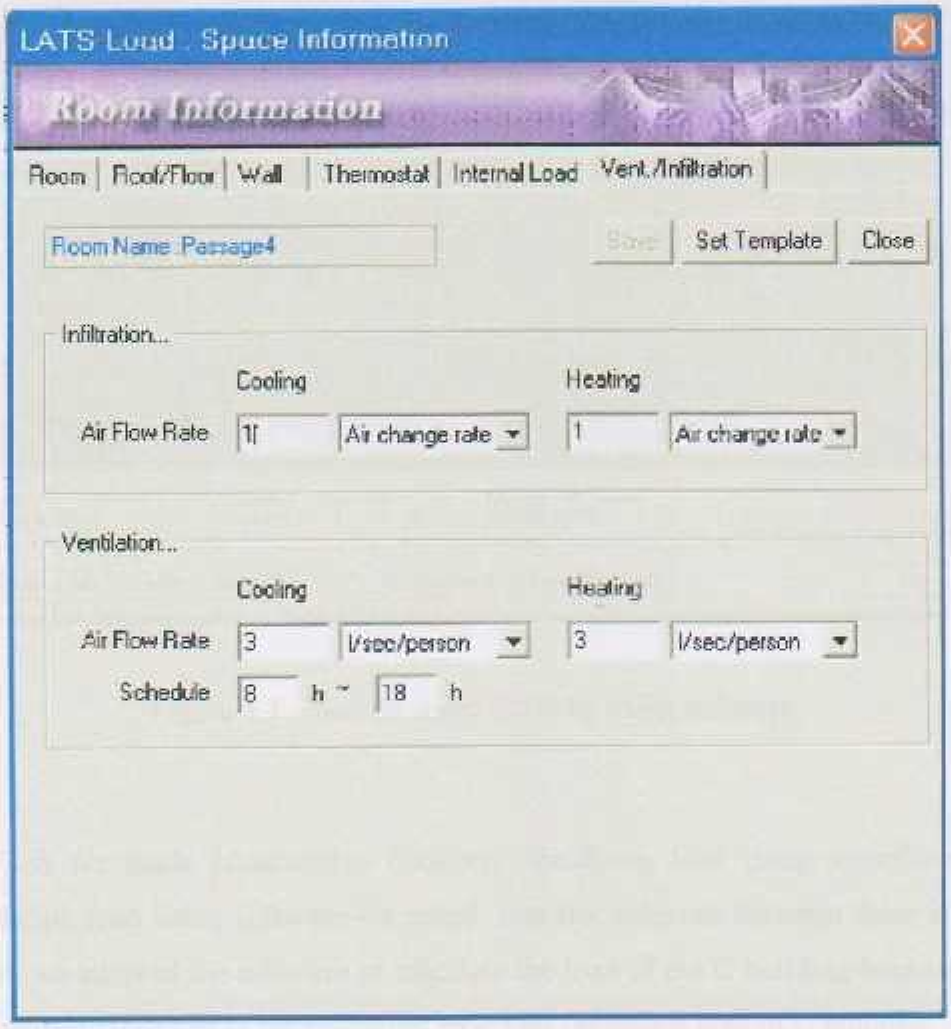


Figure 4.8 : Infiltration and ventilation

After insert the previous data to the program, the load of the room will calculated where the following figure shows the calculation of the room and load distribution in addition to the calculation of heating load.

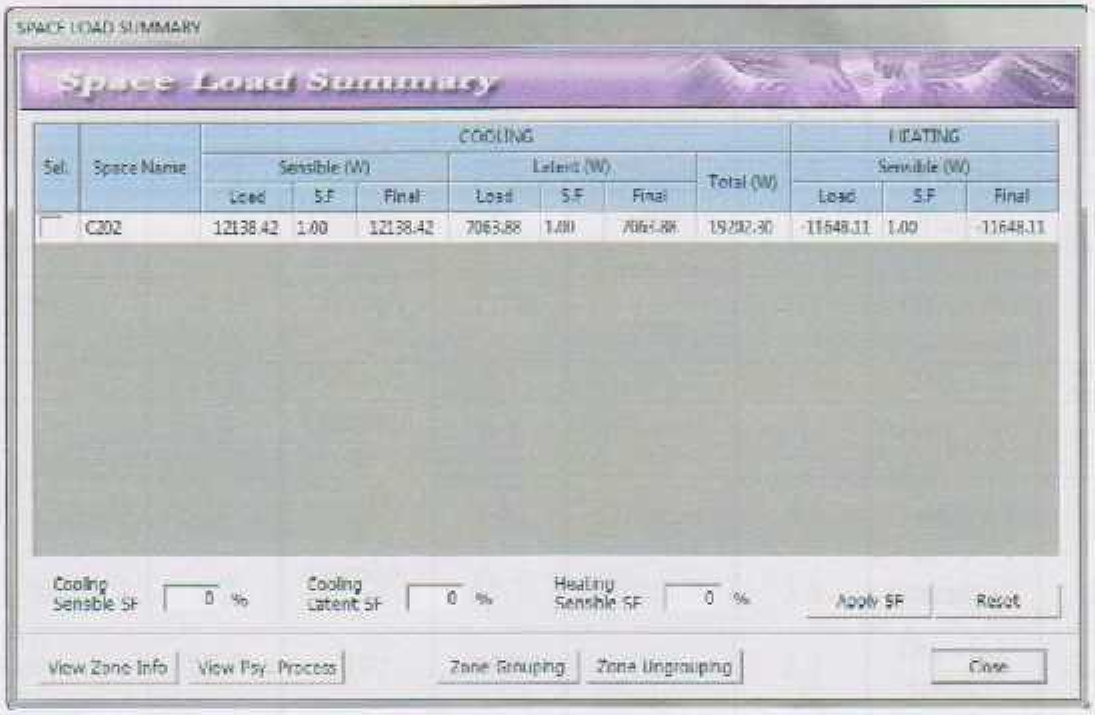


Figure 4.7 : load for room C202 by using software.

When we made comparative between calculation load using equations and calculation load using software we noted that the diffrence between them is very low so we adopted the software to calculate the load of the C building because it is high performance and in the following tables all the rooms load of C building.

4.6 Total Cooling and Heating Load

Table 4.4: Total cooling load

Second Floor		First Floor		Ground Floor		Basement Floor	
Room	**Q total (kW)**	**Room**	**Q total (kW)**	**Room**	**Q total (kW)**	**Room**	**Q total (kW)**
C202	19.2	Photo lab	19.3	office 1	3.5	Office 1	1.457
C201	17.4	Office1	6	Secretary1	1.5	Office 2	1.664
Office1	2.9	GIS lab	21.4	Drawing room	20	Office 3	2.255
C203	21.2	Office2	4.5	Office 2	4.3	Office 4	2.781
Office2	8.2	Corridor 1	8.7	Office 3	3	Office 5	1.791
Corridor 1	20	Corridor 2	6.8	Office 4	5.4	Corridor 1	18.792
Corridor 2	7.3	Corridor 3	15.1	Office 5	3.8	Corridor 2	21
Corridor 3	25.7	C102	15.5	Office 6	2.4	Drawing room 1	26.380
Meetings room	18.3	C101	5.3	Office 7	2.1	Drawing room 2	21.7
Office3	4.5	PC. lab	10.3	Office 8	2.8	Office 6	5.233
Secretary 1	2.3	CAD lab	10.3	counter	12		
Office4	11.5	Office3	6.8	Corridor1	8		
Office5	4.3	Office4	5.7	Corridor 2	12.7		
Office6	7.9	Office5	4.9	Corridor 3	8.3		
Office7	14	Corridor 4	6.7	Corridor 5	7		
Office8	3.5			Office 9	4		
Secretary 2	1.9			Office 10	5.2		
Office9	4.8			Corridor 4	9.6		
Office10	6.2			Corridor 6	14.3		
Secretary 3	3.7			Stairs	60		
Corridor 4	8.4						
TOTAL	**213**		**152**		**196**		**103**

Table 4.5: Total heating load

Second Floor		First Floor		Ground Floor		Basement Floor	
Room	Q total (kW)	Room	Q total (kW)	Room	Q total (kW)	Room	Q total (kW)
C202	12.8	Photo lab	12.8	office 1	2.3	Office 1	1
C201	11.6	Office1	4	Secretary1	1	Office 2	1.1
Office1	1.9	GIS lab	14.3	Drawing room	13	Office 3	1.5
C203	14	Office2	3	Office 2	2.8	Office 4	1.8
Office2	5.4	Corridor 1	5.8	Office 3	2	Office 5	1.2
Corridor 1	13	Corridor 2	4.5	Office 4	3.6	Corridor 1	12.5
Corridor 2	4.8	Corridor 3	10	Office 5	2.5	Corridor 2	14
Corridor 3	17	C102	10.3	Office 6	1.6	Drawing room 1	17.5
Meetings room	12.2	C101	3.5	Office 7	1.4	Drawing room 2	14.5
Office3	3	PC. lab	6.8	Office 8	1.8	Office 6	3.4
Secretary 1	1.5	CAD lab	6.8	counter	8		
Office4	7.6	Office3	4.5	Corridor1	5.3		
Office5	2.8	Office4	3.8	Corridor 2	8.5		
Office6	5.2	Office5	3.2	Corridor3	5.5		
Office7	9.3	Corridor 4	4.5	Corridor5	4.6		
Office8	2.3			Office 9	2.6		
Secretary 2	1.2			Office 10	3.5		
Office9	3.2			Corridor4	6.4		
Office10	4.1			Corridor6	9.5		
Secretary 3	2.5			Stairs	40		
Corridor 4	5.6						
TOTAL	**141**		**98**		**126**		**69**

Chapter Five

COMPONENTS SELECTION

5.1 Introduction

VRV system contains several important component ,one of these elements outdoor units ,which its capacity must be suit with the loads and it contains indoor units,and there is many models of outdoor and indoor units and the system consists of copper piping and control equpment.

The selection components of VRV System consist of :

a. Selection outdoor units of VRV system
b. Selection indoor units
c. Selection refrigerant copper pipes.
d. Selection the duct.

5.2 Indoor Units

VRV indoor units are modern, technologically advanced and come in ceiling mounted cassette, concealed ceiling, ceiling suspended, wall mounted and floor standing models. Recently, the range has been extended by the visually striking and much acclaimed round flow ceiling mounted cassette with its unique 360° air flow distribution pattern.

Designed to fit rooms of any size and shape, Daikin indoor units are also user friendly, quiet running, ultra reliable, easy to control and supply users with that relaxing 'extra something' to the indoor climate.

VRV air conditioning brings summer freshness and winter warmth to offices, hotels, department stores and many other commercial premises. It enhances the indoor environment and creates a basis for increased business prosperity and whatever the air conditioning requirement, a Daikin indoor unit will provide the answer. VRV air conditioning can be supplied via 26 different indoor unit models. And this is some models of indoor units which is used in our project.

Figure 5.1 wall mounted indoor unit, capacity rang (2.2-7 kw)

Figure 5.2 : Ceiling mounted cassette (round flow) indoor units

Figure 5.3 : ceiling mounted duct indoor units ,It is used for ducts

5.3 Outdoor Unit

Outdoor unit with a capacity that constitutes anywhere between 70% and 130% of the combined indoor units capacities. The design engineer must review the load profile for the building so that each outdoor section is sized based on the max load of all the indoor sections at any given time. Although an oversized outdoor unit in a VRV system is capable of operating at lower capacity, avoid over sizing unless it is required for a particular project due to an anticipated future expansion or other criteria. Also, when indoor sections are greatly oversized, the modulation function of the expansion valve is reduced or entirely lost. Most manufacturers offer selection software to help simplify the optimization process for the system's components, Figure 5.4 show outdoor units shape.

Figure 5.4 : outdoor unit

5.4 General Pipes Design

The pipes which used in VRV system is copper pipes. VRV offers an extended piping length of 165m (190m equivalent piping length) with a total system piping length of 1,000m.

In case the outdoor unit is located above the indoor unit the height difference is 50m standard. It can be extended to 90m .In case the outdoor unit is located below the indoor unit, the height difference is 40m standard. Height differences up to maximum 90marc possible .After the first branch, the difference between the longest piping length and the shortest piping length can be maximum 40m,provided that the longest piping length amounts to maximum 90m.

The figure below show the length of the pipes between outdoor units and indoor units.

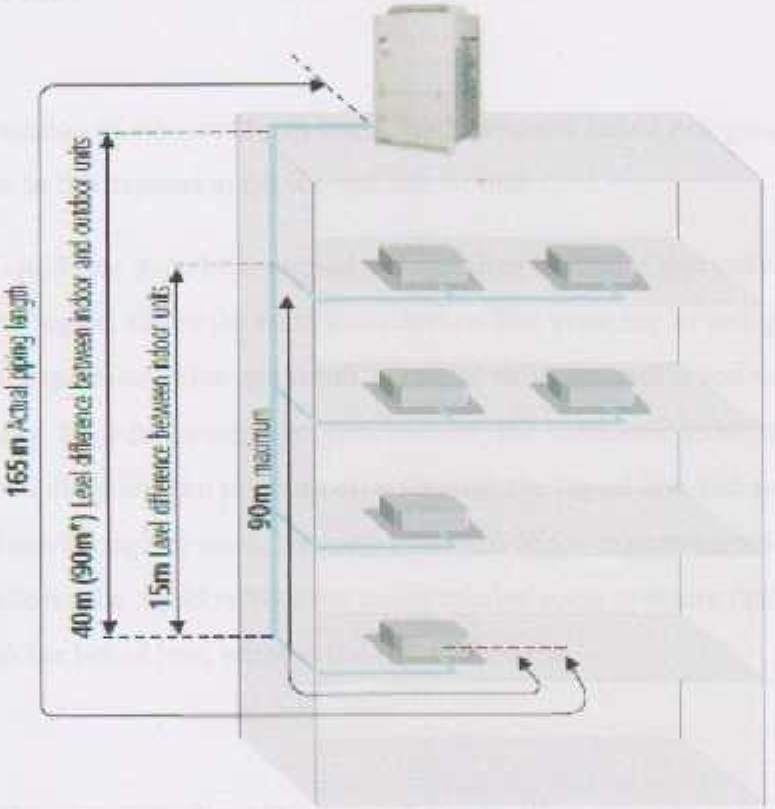


Figure 5.5 : Pipes length between outdoor &indoor units

The VRV piping of the refrigeration system is one of the most important designs of the VRV system, i.e. the VRV piping should assure many purposes such as:

1- Ensure adequate velocity to return oil to compressor at all steps of unloading.

2- Avoid excessive noise.

3- Minimize system capacity and efficiency loss and pipe erosion.

The VRV piping design of the system requires designing the liquid line that supplies the indoor cooling unit with liquid refrigerant, and the refrigerant suction line; which returns refrigerant from the indoor unit to the compressor.

5.4.1 Liquid Line

This section of pipe conducts warm, high-pressure liquid refrigerant from the condenser to the expansion device and indoor unit .

The liquid line must be designed and installed to ensure that only liquid refrigerant (no vapor) enters the expansion device. The presence of refrigerant vapor upstream of a expansion valve can result in erratic valve operation and reduced system capacity. In order to meet this requirement, the condenser must provide adequate subcooling, and the pressure drop through the liquid line and accessories must not be high enough to cause flashing upstream of the expansion device. Subcooling allows the liquid refrigerant to experience some pressure drop as it flows through the liquid line, without the risk of flashing.

The diameter of the liquid line must be as small as possible to minimize the refrigerant charge, therefore improving reliability and minimizing installed cost.

However, if the pipe is too small, the increased pressure drop may cause flashing upstream of the expansion device.

The refrigerant velocity in the liquid line shouldn't exceed 3 m/s to prevent noise and large pressure drops. And no under limit since the refrigerant will solute the oil and carry it by the way. As shown in Figure (5-6); the refrigerant velocity is about 2.8 m/s at the recommended liquid line diameter from the condensation unit manufacturer.

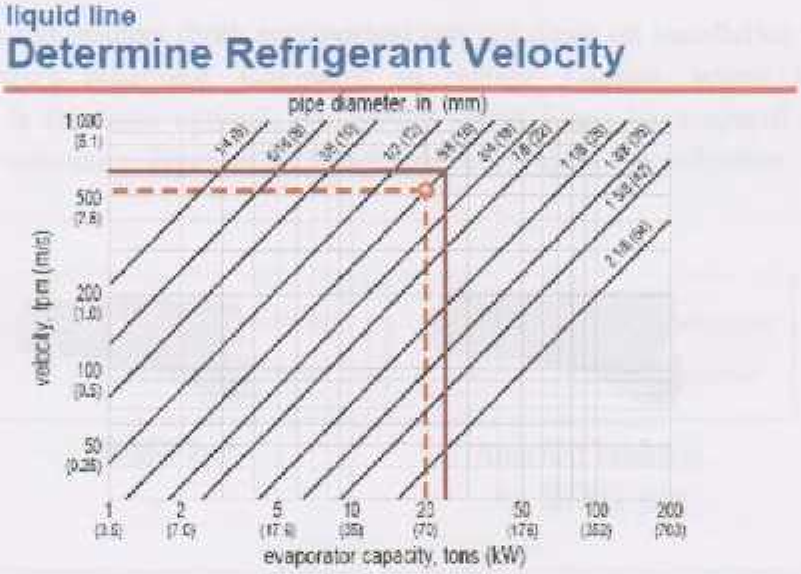


Figure (5-6) Evaporator Capacity and Velocity Diagram for Liquid Line

5.4.2 Suction Line

This pipe conducts low pressure refrigerant vapor from the evaporator to the compressor. The diameter of the suction line must be small enough that the resulting refrigerant velocity is sufficiently high to carry oil droplets, at all steps of compressor unloading. The refrigerant velocity inside a pipe depends on the mass flow rate and density of the refrigerant if the velocity in the pipe is too high, however, objectionable noise may result. Also, the pipe diameter should be as large as possible to minimize pressure drop and thereby maximize system capacity and efficiency.

5.4.3 Unified REFNET piping

The unified Daikin refnet piping system is especially designed for simple installation. The use of refnet piping in combination with electronic expansion valves, results in a dramatic reduction in imbalance in refrigerant flowing between indoor units, despite the small diameter of the piping.

refnet joints and headers (both accessories) can cut down on installation work and increase system reliability. Compared to regular T-joints, where refrigerant distribution is far from optimal, the Daikin refnet joints have specifically been designed to optimize refrigerant fl . The figure below show type of refnet.

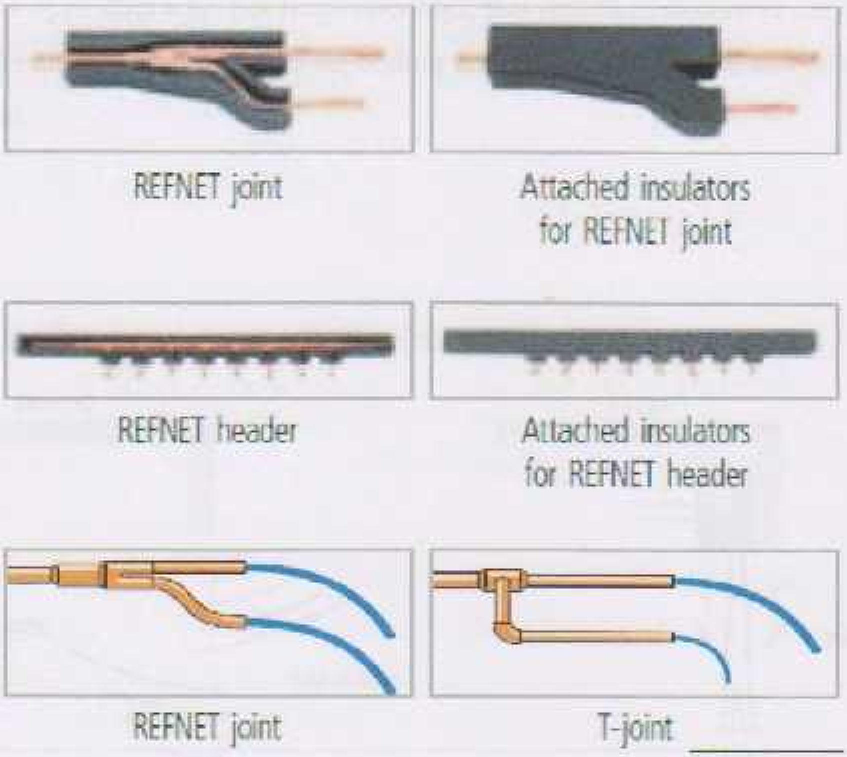


Figure 5.9 :Types of REFNET

5.5 Air Curtains

When dividing spaces with temperature differences, the air curtain prevents the natural flow of air caused by differing temperatures. The difference between indoor and outdoor temperature creates an imbalance in density, and therefore a pressure variation which causes the infiltration of outside air.

Figure (5-9) illustrates the loss of conditioned air before and after the installation of an air curtain at the doorway. As an example, during the winter heating season, the outside colder denser air flows through the bottom half of the doorway into building; The warmer, lighter air flows to the outside through the top half of the open doorway. For large openings, this results in excessive energy losses.The air curtain is selected depending on type of application, and door dimension.

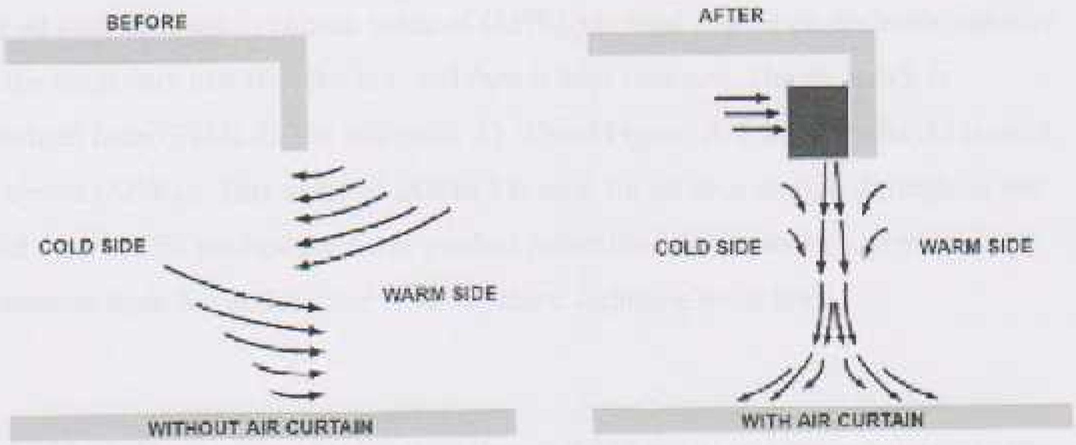


Figure (5-9) Air Curtain Working Principle

5.6 Sample Duct Sizing by Equal Pressure Drop Method

The duct in VRV system when large area to be conditioned ,where there are models of indoor unit (ducted heat pump VRV unit) selected to connect with duct, and the air passes through this duct to grills. Figure 5.10 show duct connect with ducted heat pump VRV unit in ground floor corridor 4 .

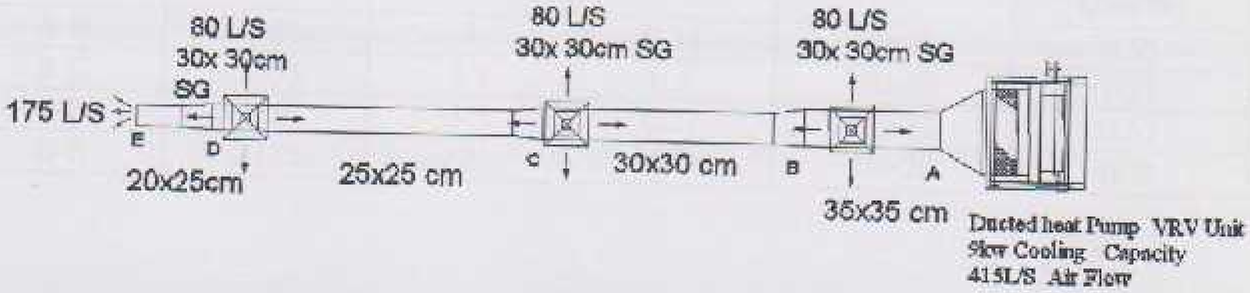


Figure 5.10 : Duct sample

Equal pressure drop method .this method is used in low velocity air conditioning systems where the pressure drop (ΔP/EL),through the duct network is kept constant for all duct sections. A chosen value of (ΔP/EL) is used , based on desirable velocity in the main duct just after the fan, and then is kept constant. The desirable is obtained from (Table A-3 in appendix A). Then (Figure A-1 in appendix A) is used to obtain (ΔP/EL). This value of (ΔP/EL) is used for all duct section throughout the duct system .this method produces gradual reduction of air velocity and duct diameters from fan outlet to air exit, therefore, reducing noise level.

For ducted heat pump VRV unit air flow is 0.415 m^3/s ,maximum velocity in the main duct on basis of the application environment. To avoid disturbing noise levels is 4 m/ s.

With this value of air velocity and the volumetric flow rate 0.415 for main duct A to B then figure A-1 (see appendix A), gives $(\Delta p/EL)_{A-B} = 0.55$ Pa/m and circular duct diameter 33 cm equivalent (35× 35 cm) rectangular duct(see Table A-12 in appendix A) .

TABLE 5-1: Volumetric flow rates and dimension for all duct sections.

Duct section	$\dot{V}$ $(m^3/_s)$	v $(^m/_s)$	d (cm)	h×w (cm)	$\Delta p/EL$ (Pa/m)
A-B	0.415	4	0.33	35×35	0.55
B-C	0.335	3.7	0.3	30×30	0.55
C-D	0.255	3.5	0.27	25×25	0.55
D-E	0.175	3	0.24	25×20	0.55

Pressure drop in duct (Δp):

$$\Delta p_{A-E} = \frac{\Delta P}{EL}\left(\sum L + \sum L_e\right) + \Delta P_{exit} \qquad 5.1$$

Where L : duct length.

L_e: equivalent length for reducer and elbow(see Table A-12 in appendix A) .

.

L= 1.16+1.38+2.47+2.54= 7.55 M

$L_e = 3L_e\ for\ reduser + 3L_e\ for\ 90\ rectangular\ elbow$

$L_e = 3(6) + 3(5) = 33\ m$

$\Delta P_{exit} = 10\ pa$

$\Delta p_{A-E} = 0.55(7.55 + 33) + 10 = 32.3\ Pa$

Pressure drop in duct less than external pressure drop for ducted heat pump VRV unit that equal 120 Pa

5.6.1 Duct Insulation

The most common insulation material for ducts is fiberglass. It is available in either a flexible or rigid form and comes in a variety of densities and thicknesses. The flexible blanket-type insulation is sold in rolls and is easy to apply to either round or rectangular ducts. Flexible insulation easily conforms to irregular surfaces. Rigid insulation comes in pre-formed boards bonded with a thermosetting resin, and works best on rectangular ducts (in some areas, ducts are constructed of a rigid insulation material, minimizing the need for additional insulation) . All duct insulation should have a foil or vinyl facing on the exterior side to prevent moisture from being absorbed into the fiberglass. Kraft paper- faced insulation should never be used on ducts because of its flammability and relatively poor moisture resistance. If any existing insulation has become wet, it should be replaced.

Chapter Six

COST EVALUATION AND CONCLUSION

6.1 Second Floor Cost

TABLE 6-1: Total Cost For Second Floor.

Unit	Capacity (kW)	Description	Number required	Cost for one piece (JD)
RXYQ-P(A)/P8(A)	105	Outdoor unit, small footprint combination	2	**21800**
FXFQ-P8	5.6	Indoor unit ,ceiling mounted cassette	2	**1050**
FXFQ-P8	7.1	Indoor unit ,ceiling mounted cassette	6	**1150**
FXFQ-P8	9.0	Indoor unit ,ceiling mounted cassette	4	**1250**
FXFQ-P8	11.2	Indoor unit ,ceiling mounted cassette	2	**1330**
FXAQ-P	2.2	Indoor unit ,wall mounted unit	1	**890**
FXAQ-P	2.8	Indoor unit ,wall mounted unit	1	**940**
FXAQ-P	3.6	Indoor unit ,wall mounted unit	2	**980**
FXAQ-P	4.5	Indoor unit ,wall mounted unit	1	**1040**
FXAQ-P	7.1	Indoor unit ,wall mounted unit	1	**1100**
FXMQ-MA	22.4	Large concealed ceiling unite	3	**3450**
RIFNET JOINT		For liquid and gas pipe	22+22	**150/PIECE**
Outdoor unit connections		For liquid and gas pipe	1+1	**250**
Duct			$65m^2$	**1450**
Axil Fan			12	**1080**
Installation				**4000**
TOTAL				**95000 JD**

6.2 First Floor Cost

TABLE 6-2: Total Cost For First Floor.

Unit	Capacity (kW)	Description	Number required	Cost for one piece (JD)
RXYHQ-P8/28	77	Outdoor unit, High COP combination	2	**15500**
FXFQ-P8	5.6	Indoor unit ,ceiling mounted cassette	4	**1050**
FXFQ-P8	7.1	Indoor unit ,ceiling mounted cassette	3	**1150**
FXFQ-P8	9.0	Indoor unit ,ceiling mounted cassette	3	**1250**
FXFQ-P8	11.2	Indoor unit ,ceiling mounted cassette	1	**1330**
FXAQ-P	2.2	Indoor unit ,wall mounted unit	1	**890**
FXAQ-P	2.8	Indoor unit ,wall mounted unit	2	**940**
FXAQ-P	3.6	Indoor unit ,wall mounted unit	1	**980**
FXAQ-P	4.5	Indoor unit ,wall mounted unit	1	**1040**
FXMQ-MA	22.4	Large concealed ceiling unite	1	**3450**
FXMQ-P7	14	concealed ceiling unite	1	**1990**
FXSQ-P	11	concealed ceiling unite	1	**1300**
RIFNET JOINT		For liquid and gas pipe	17+17	**150/PIECE**
Outdoor unit connections		For liquid and gas pipe	1+1	**250**
Duct			65 m^2	**1450**
Axial Fan			14	**1260**
Installation				**4400**
TOTAL				**72000 JD**

6.3 Ground Floor Cost

TABLE 6-3: Total Cost For Ground Floor.

Unit	Capacity (kW)	Description	Number required	Cost for one piece (JD)
RXYHQ-P8/36	98	Outdoor unit, High COP combination	2	**19950**
FXFQ-P8	7.1	Indoor unit ,ceiling mounted cassette	1	**1150**
FXFQ-P8	9.0	Indoor unit ,ceiling mounted cassette	1	**1250**
FXFQ-P8	11.2	Indoor unit ,ceiling mounted cassette	1	**1330**
FXAQ-P	2.2	Indoor unit ,wall mounted unit	1	**890**
FXAQ-P	3.6	Indoor unit ,wall mounted unit	1	**980**
FXMQ-MA	9	concealed ceiling unite	1	**1300**
FXMQ-MA	28.0	Large concealed ceiling unite	2	**3500**
FXMQ-P7	14	concealed ceiling unite	2	**1990**
FXSQ-P	11	concealed ceiling unite	3	**1300**
RIFNET JOINT		For liquid and gas pipe	10+10	**150/PIECE**
Outdoor unit connections		For liquid and gas pipe	1+1	**250**
DUCT			100 m^2	**2200**
Axial Fan			8	**720**
Installation				**5200**
TOTAL				**74000 JD**

6.4 Basement Floor Cost

TABLE 6-4: Total Cost for Ground Floor.

Unit	Capacity (kW)	Description	Number required	Cost for one piece (JD)
RXYHQ-P8/20	55	Outdoor unit, High COP combination	2	**11800**
FXFQ-P8	5.6	Indoor unit ,ceiling mounted cassette	1	**1050**
FXFQ-P8	11.2	Indoor unit ,ceiling mounted cassette	11	**1330**
FXFQ-P8	14.0	Indoor unit ,ceiling mounted cassette	1	**1600**
FXAQ-P	2.2	Indoor unit ,wall mounted unit	5	**890**
FXMQ-MA	9	concealed ceiling unite	1	**1650**
FXMQ-MA	22.0	Large concealed ceiling unite	1	**3450**
FXSQ-P	11	concealed ceiling unite	1	**1300**
RIFNET JOINT		For liquid and gas pipe	12+12	**150/PIECE**
Outdoor unit connections		For liquid and gas pipe	1+1	**250**
Duct			60 m^2	**1320**
Axial fan			4	**360**
Installation				**8400**
TOTAL				**67000JD**

Total cost for four floors of C building = 95000 +72000+74000+67000

= **310000 JD**.

Conclusion

Although the high installation cost of the VRV systems and limitation bellow:

- There is a limitation on the indoor coil maximum and minimum entering dry- and wet-bulb temperatures, which makes the units unsuitable for 100% outside air applications especially in hot and humid climates.
- The cooling capacity available to an indoor section is reduced at lower outdoor temperatures. This limits the use of the system in cold climates to serve rooms that require year-round cooling.
- The external static pressure available for ducted indoor sections is limited. For ducted indoor sections, the permissible ductwork lengths and fittings must be kept to a minimum. Ducted indoor sections should be placed near the zones they serve.

It has many features such as high efficiency operation design versatility high reliability easy installation comfort and convenience easy service and maintenance .

References

1- Heating and air conditioning , for residential Buildings., fourth edition .SI Version

2- Prenciple of refrigeration .Version SI.

3- J.P.Holman .Heat Transfer .Southern Methodist University. Ninth edition .Boston.1996.

4- www.DIKEN.com

5- DIKEN catalogs.

6- ASHRAE Handbook 1985

7- www.wikipedia.com

Software :

1-LG-LATS LOAD software.

2-DIKEN- XPRESS software.

Appendix

A

TABLE A-1 :Latitude –month correction factor LM, as applied to walls and horizontal roofs ,north latitudes

Lat.	Month	N	NNE NNW	NE NW	ENE WNW	E W	ESE WSW	SE SW	SSE SSW	S	Horizontal Roofs
16	December	-2.2	-3.3	-4.4	-4.4	-2.2	-0.5	2.2	5.0	7.2	-5.0
	Jan./Nov.	-2.2	-3.3	-3.8	-3.8	-2.2	-0.5	2.2	4.4	6.6	-3.8
	Feb./Oct.	-1.6	-2.7	-2.7	-2.2	-1.1	0.0	1.1	2.7	3.8	-2.2
	Mar/Sept.	-1.6	-1.6	-1.1	-1.1	-0.5	-0.5	0.0	0.0	0.0	-0.5
	Apr./Aug.	-0.5	0.0	-0.5	-0.5	-0.5	-1.6	-1.6	-2.7	-3.3	0.0
	May/July	2.2	1.6	1.6	0.0	-0.5	-2.2	-2.7	-3.8	-3.8	0.0
	June	3.3	2.2	2.2	0.5	-0.5	-2.2	-3.3	-4.4	-3.8	0.0
24	December	-2.7	-3.8	-5.5	-6.1	-4.4	-2.7	1.1	5.0	6.6	-9.4
	Jan./Nov.	-2.2	-3.3	-4.4	-5.0	-3.3	-1.6	-1.6	5.0	7.2	-6.1
	Feb./Oct.	-2.2	-2.7	-3.3	-3.3	-1.6	-0.5	1.6	3.8	5.5	-3.8
	Mar/Sept.	-1.6	-2.2	-1.6	-1.6	-0.5	-0.5	0.5	1.1	2.2	-1.6
	Apr./Aug.	-1.1	-0.5	0.0	-0.5	-0.5	-1.1	-0.5	-1.1	-1.6	0.0
	May/July	0.5	1.1	1.1	0.0	0.0	-1.6	-1.6	-2.7	-3.3	0.5
	June	1.6	1.6	1.6	0.5	0.0	-1.6	-2.2	-3.3	-3.3	0.5
32	December	-2.7	-3.8	-5.5	-6.1	-4.4	-2.7	1.1	5.0	6.6	-9.4
	Jan./Nov.	-2.7	-3.8	-5.0	-6.1	-4.4	-2.2	1.1	5.0	6.6	-8.3
	Feb./Oct.	-2.2	-3.3	-3.8	-4.4	-2.2	-1.1	2.2	4.4	6.1	-5.5
	Mar/Sept.	-1.6	-2.2	-2.2	-2.2	-1.1	-0.5	1.6	2.7	3.8	-2.7
	Apr./Aug.	-1.1	-1.1	-0.5	-1.1	0.0	-0.5	0.0	5.0	0.5	-0.5
	May/July	0.5	0.5	0.5	0.0	0.0	-0.5	-0.5	-1.6	-1.6	0.5
	June	0.5	1.1	1.1	0.5	0.0	-1.1	-1.1	-2.2	-2.2	1.1
40	December	-3.3	-4.4	-5.5	-7.2	-5.5	-3.8	0.0	3.8	5.5	-11.6

TABLE A-2 : Performance data for selecting return air diffusers

Face Velocity m/s	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
ΔP, Pa	2.0	4.0	7.5	12.5	17.4	22.4	29.9	39.8	49.8	69.7
Size, cm	Volumetric Flow Rate, L/s									
15 × 15	18	23	30	36	45	47	54	65	74	77
15 × 23	23	32	43	54	63	74	85	97	108	117
15 × 30	27	40	54	68	83	97	110	124	137	150
23 × 23	32	45	60	77	92	108	125	140	155	170
23 × 30	38	60	77	97	117	135	155	173	195	212
23 × 38	47	70	92	115	140	162	185	207	230	254
30 × 30	50	72	97	122	145	170	195	218	243	268
30 × 46	70	105	137	173	207	240	276	310	345	378
30 × 53	80	120	160	198	240	280	320	358	395	437
38 × 53	97	145	195	243	293	340	390	435	485	535
38 × 61	110	165	220	275	330	387	440	495	550	603
46 × 61	120	170	230	288	347	403	465	522	567	637
53 × 61	150	225	304	378	456	530	603	684	765	837
61 × 61	170	260	350	437	518	612	702	783	873	954

TABLE A-3 : recommended and maximum air velocities used in warm air heating systems

Description	Recommended Velocity, m/s			Maximum Velocity, m/s		
	Residence Buildings	Public Buildings	Industrial Buildings	Residence Buildings	Public Buildings	Industrial Buildings
Outside air intake	2.5	2.5	2.5	4.0	4.5	6.0
Heating coils	2.3	2.5	3.0	2.5	3.0	3.8
Cooling coils	2.3	2.5	3.0	2.5	3.0	3.5
Fan suction	3.5	4.0	5.0	4.5	5.0	7.0
Fan outlet	5.0-8.0	6.5-10.0	8.0-12.0	8.5	7.5-11.0	8.5-14.0
Main duct	4.0-4.5	5.0-6.5	6.0-9.0	4.0-6.0	5.5-8.0	6.5-11.0
Branch ducts	3.0	3.0-4.5	4.0-5.0	3.5-5.0	4.0-6.5	5.0-9.0
Branch risers	2.5	3.0-3.5	4.0	3.5-4.0	4.0-6.0	5.0-8.0

TABLE A-4 : instantaneous heat gain room occupants in units of watt

Type of Activity	Typical Application	Total Heat Dissipation Adult Male	Total Adjusted[a] Heat Dissipation	Sensible Heat, W	Latent Heat, W
Seated at rest	*Theater*:				
	Matinee	111.5	94.0	64.0	30.0
	Evening	111.5	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, supermarkets	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 halls	257.0	243.0	87.0	156.0
Walking at 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

TABLE A-5 : Design inside dry bulb temperature usually specified for sensible heating

Type of Space	T_{di} °C	Type of Space	T_{di} °C
Schools:		*Theaters:*	
Classrooms	22-24	Seating space	20-22
Assembly rooms	20-22	Lounge rooms	20-22
Gymnasiums	16-21	Auditorium	23-24
Toilets and bathes	21	*Hotels:*	
Locker rooms	18-20	Bedrooms and bathes	22-24
Kitchens	19	Dining rooms	22
Dining and lunch rooms	22-24	Kitchen and laundries	19
Play rooms	16-18	Ballrooms	18-20
Hospitals:		Toilets and service rooms	20
Private rooms	22-23	*Public rooms*	22-23
Patient rooms	23-24	*Steam bathes*	43
Operation rooms	21-25	*Foundries and boiler rooms*	10-16
Wards	22-23	*Paint shops*	27
Kitchens and laundries	19	*Factories:*	
Toilets	20	Light work	16-21
Bathrooms in general	23-27	Heavy work	14-20
Stores	21-23	*Swimming pools*	24

TABLE A-6 : Minimum outside air requirements for mechanical ventilation

Application	Maximum Occupancy Per 100 m²	Ventilation Air Requirements	
		L/s/Person	$L/s/m^2$
Game rooms	70	3.5-17.5	—
Ice arenas	—	—	2.50
Swimming pools	—	—	2.50
Gymnasium floors	30	10.0	
Ballrooms and discos	100	3.5-17.5	—
Bowling alleys	70	3.5-17.5	
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	2.5-12.5	—
Laboratories	30	10.0	—
Training shops	30	3.5-17.5	—
Music rooms	50	3.5-17.5	—
Libraries	20	2.5	—
Locker rooms	—		2.50
Corridors	—	—	0.50
Auditorium	150	8.0	—
Smoking areas	70	30.0	—
Hospitals:			

TABLE A-7 : Number of air change per hour in residences and our commercial application

Type of Room or Building	No. of Air Change per Hour
Rooms with no windows or exterior doors	0.5
Rooms with windows or exterior doors on one side only	1.0
Rooms with windows or exterior doors on two sides	1.5
Rooms with windows or exteriors doors on three sides	2.0
Entrance halls	2.0-3.0
Factories, machine shops	1.0-1.5
Recreation rooms, assembly rooms, gymnasium	1.5
Homes, apartments, offices	1.0-2.0
Classrooms, dining rooms, lounges, hospital rooms, kitchens, laundries, ballrooms, bathrooms	2.0
Stores, public buildings	2.0-3.0
Toilets, auditorium	3.0

TABLE A-8 : Cooling load temperature differences (CLTD) for sunlit roofs.

Roof No.	Description of Construction	U_m, W/m²·°C	Solar Time, h: 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	0.963	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	101.6 mm L.W. concrete	1.209	5	3	1	0	−1	−2	−2	1	5	11	18	25	31	36	39	40	40	37	32	25	19	14	10	7
4	50.8 mm H.W. concrete 25.4 mm (or 50.8 mm) insulation	1.170 (0.693)	7	5	3	2	0	−1	0	2	6	11	17	23	28	33	36	37	37	34	30	25	20	16	12	10
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	20	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	18	24	29	33	35	36	35	32	28	24	19	16
7	63.5 mm wood with 25.4 mm insulation	0.738	16	13	11	9	7	6	4	3	4	5	8	11	15	19	23	27	29	31	31	30	27	25	22	19
8	203.4 mm L.W. concrete	0.715	20	17	14	12	10	8	6	5	4	4	5	7	11	14	18	22	25	28	30	30	29	27	25	22
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	0.602	19	17	15	14	12	11	9	8	7	8	8	10	12	15	18	20	22	24	25	26	25	24	22	21

TABLE A-9 :Solar heat gain factor (SHG) for sunlit glass (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
Horizontal	555	685	795	855	874	871	861	836	770	672	552	498

TABLE A-10 : Shading coefficient (SC) for glass windows with interior shading

Type of Glass	Nominal Thickness, mm	Type of Interior Shading				
		Venetian Blinds		Roller Shade		
				Opaque		Translucent
		Medium	Light	Dark	White	Light
Single Glass						
Clear, regular	2.5-6.0	—	—	—	—	—
Clear, plate	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(gray sheet)	5.0-5.5	—	—	—	—	—
Heat Absorbing, plate	5.0-6.0	0.57	0.53	0.45	0.30	0.36
Pattern or Tinted, gray sheet	3.0-5.5	—	—	—	—	—
Heat Absorbing Plate 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-	—	—	—	—

TABLE A-11 : Shading coefficient (SC) for glass windows without interior shading

Type of Glass	Nominal Thickness, mm	Solar Trans.	Shading Coefficient, $W/m^2 \cdot K$ $h_o = 22.7$	$h_o = 17.0$
Single Glass				
Clear	3	0.84	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 A-13 : Cooling load temperature differences (CLTD) for convection heat gain for glass windows

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 A-12 : Circular equivalent diameters of rectangular ducts for equal pressure drop and flow rate

Lgth. Adj.	Length of One Side of Rectangular Duct , mm																			
	100	125	150	175	200	225	250	275	300	350	400	450	500	550	600	650	700	750	800	900
100	109																			
150	133	150	164																	
200	152	172	189	204	219															
250	169	190	210	228	244	259	273													
300	183	207	229	248	266	283	299	314	328											
400	207	235	260	283	305	325	343	361	378	409	437									
500	227	258	287	313	337	360	381	401	420	455	488	518	547							
600	245	279	310	339	365	390	414	436	457	496	533	567	598	628	656					
700	261	298	331	362	391	418	443	467	490	533	573	610	644	677	708	737	765			
800	275	314	350	383	414	442	470	496	520	567	609	649	682	722	753	787	818	847	875	
900	289	330	367	402	435	465	494	522	548	593	643	686	726	763	799	833	866	897	927	984
1000	301	344	384	420	454	486	517	546	574	626	674	719	762	802	840	875	911	944	976	1037
1200	324	370	413	453	490	525	558	590	620	677	731	780	827	872	914	954	993	1030	1066	1133
1400	344	394	439	482	522	559	595	629	662	724	781	835	886	934	980	1024	1066	1107	1146	1220
1600	362	415	463	508	551	591	629	665	700	766	827	885	939	991	1041	1088	1133	1177	1219	1298
1800	379	434	485	533	577	619	660	698	735	804	869	930	988	1043	1096	1146	1195	1241	1286	1371
2000	395	453	506	555	602	646	688	728	767	840	908	973	1034	1092	1147	1200	1252	1301	1348	1438
2200	410	470	525	577	625	671	715	757	797	874	945	1013	1076	1137	1195	1251	1305	1356	1406	1501
2400	424	486	543	597	647	695	740	784	826	905	980	1050	1116	1180	1241	1299	1355	1409	1461	1561
2600	437	501	560	616	668	717	764	810	853	935	1012	1085	1154	1220	1283	1344	1402	1459	1513	1617
2800	450	516	577	634	688	738	787	834	879	964	1043	1119	1190	1259	1324	1387	1447	1506	1562	1670

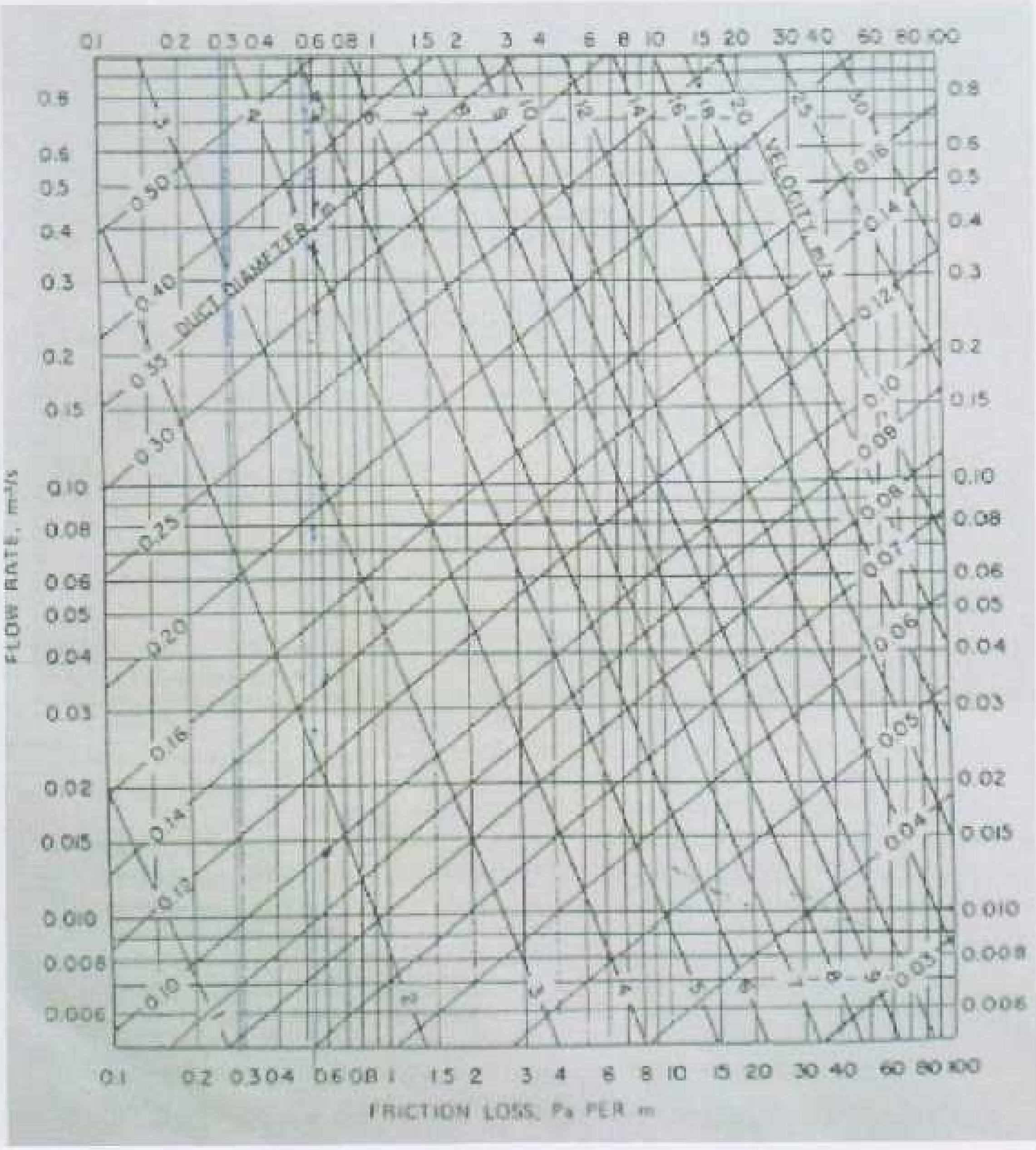


FIGURE A-1: Pressure drop (ΔP/EL) for low flow rates of air in galvanized steel ducts, based on round duct diameter.

TABLE A-14 : Diversity factor for selected application

Application	Diversity Factor	
	Lights	People
Peripheral aras 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 A-15 : Approximate values of equivalent length Le ,for commonly used fittings.

Fitting	L_e m
45° Round elbow	1.5
90° Four pieces elbow	3.0
Gradual reduction	6.0
45° Round Tee:	
Main run	1.5
Branch	11.0
90° Round Tee:	
Main run	1.5
Branch	15.0
90° Rectangular elbow	5.0
Abrupt round contraction or expansion	11.0

TABLE A-16 : Performance data for selecting return air diffusers

Face Velocity m/s	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
ΔP, Pa	2.0	4.0	7.5	12.5	17.4	22.4	29.9	39.8	49.8	69.7
Size, cm	**Volumetric Flow Rate, L/s**									
15 × 15	18	23	30	36	45	47	54	65	74	77
15 × 23	23	32	43	54	63	74	85	97	108	117
15 × 30	27	40	54	68	83	97	110	124	137	150
23 × 23	32	45	60	77	92	108	125	140	155	170
23 × 30	38	60	77	97	117	135	155	173	195	212
23 × 38	47	70	92	115	140	162	185	207	230	254
30 × 30	50	72	97	122	145	170	195	218	243	268
30 × 46	70	105	137	173	207	240	276	310	345	378
30 × 53	80	120	160	198	240	280	320	358	395	437
38 × 53	97	145	195	243	293	340	390	435	485	535
38 × 61	110	165	220	275	330	387	440	495	550	603
46 × 61	120	170	230	288	347	403	465	522	567	637
53 × 61	150	225	304	378	456	530	603	684	765	837
61 × 61	170	260	350	437	518	612	702	783	873	954

TABLE A-17 : thermal conductivity of some construction materials.

Material Number	Material	Density kg/m³	Thermal Conductivity W/m·°C
1	*Building Stone*		
	Marble	2,600	2.90
	Hard stone	2,500	2.20
	Firm stone	2,250	1.70
	Semi-firm stone	2,000	1.40
	Soft stone	1,750	1.05
	Granite	2,800	3.50 - 4.1
	Basalt and stones	2,800	3.50
	Lime stone	2500	1.3
2	*Sand*	1,800	0.70
	Soil	-	1.0 - 1.15
3	*Building brick*		
	Cement brick, solid	1,900	1.20
	Cement brick, with air gaps	1,600	1.00
	Common brick (low density)	1,400	0.72
	Face brick (high density)	1,200	1.27
	Glass brick	1,000	0.65
	Fire-clay brick	2,000	1.00
		1,800	0.80
		1,600	0.70
		1,400	0.60
		1,200	0.52
		1,000	0.47
	Clay	-	1.4
4	*Concrete, regular and reinforced*		
	Light concrete	2,300	1.75
		2,000	1.20
		1,800	1.00
		1,600	0.87
		1,400	0.72
		1,200	0.60
		1,000	0.47
	Foam concrete	1,600	0.68
		1,400	0.61
		1,200	0.52

Material Number	Material	Density kg/m^3	Thermal Conductivity W/m-°C
		1,000	0.43
		900	0.36
		800	0.30
		700	0.22
		600	0.19
		500	0.16
	Mortar	1800	0.72 – 1.05
5	*Tiles*		
	Terrazzo tiles	2,000	1.40
	Ceramic tiles	2,100	1.10
	PVC tiles	2,000	1.20
	Rubber tiles	1,500	0.23
	Rubber flooring	-	0.4
	Plastic tiles	-	0.2 – 0.5
6	*Cement Plaster*	2,000	1.20
		1,800	0.85
		1,400	0.70
		1,200	0.40
	Gypsum plaster	720	0.8
7	*Wood, natural*		
	Oak	800	0.21
	Pine	600	0.14
	Beech	800	0.17
	Mahogany	700	0.16
	Teak	700	0.17
	Red wood	-	0.11
8	*Wood boards*		
	Hard fiber boards	1,000	0.14
	Soft fiber boards	300	0.06
	Plywood boards	545	0.12
	Chip boards	800	0.15
	parquet	-	0.23
9	*Gypsum boards*	-	0.17
	Cork boards	800	0.95
10	*Window glass*		
	Regular	2,500	1.05
	Thermal sealing	2,250	1.10
11	*Metals*		

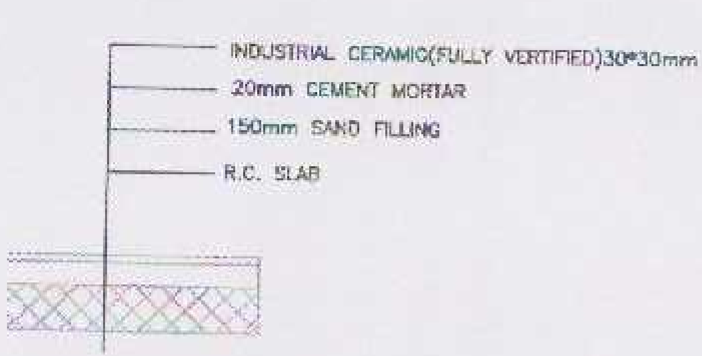
Material Number	Material	Density kg/m³	Thermal Conductivity W/m·°C
	Aluminum	2,300	200
	Copper	8,900	250
	Brass	8,400	130
	Cast iron	7,000	40
	Mild steel	7,800	45
	Stainless steel	7,800	15
12	*Insulating material*		
	Polystyrene boards	30	0.030
		25	0.034
		15	0.037
	Polyurethane boards	30	0.027 – 0.035
		37	0.030
	Rock wool	140	0.040
		130	0.036
		80	0.038
		50	0.039
	Glass wool	180	0.042
		130	0.045
		80	0.035
		65	0.032
		50	0.033
		25	0.040
	Cork boards	160	0.045
		145	0.042
		130	0.040
		110	0.039
	Cork particles	45 - 120	0.045
13	*Asphalt mix*	2,300	1.10
	Asphalt	2100	0.80
	Roll roofing	1,100	0.18

TABLE A-18 : Approximate CLTD values for sunlit roofs C.

Solar Time	Roof Construction		
	Light	Medium	Heavy
10:00	5	—	—
11:00	12	—	—
12:00	19	3	0
13:00	25	8	2
14:00	29	14	5
15:00	31	19	8
16:00	31	23	10
17:00	29	25	12
18:00	24	26	14
19:00	19	25	15
20:00	11	22	16

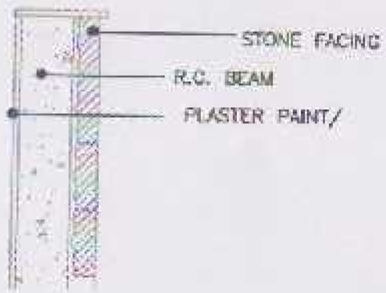
TABLE A-19 : cooling load factors(CLF) for glass windows without interior shading

Glass Facing	Building 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	0.38	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	0.34	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	0.33	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	0.24
	H	0.09	0.08	0.08	0.07	0.07	0.23	0.37	0.46	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	0.31	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	0.09	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	0.02	0.17	0.34	0.49	0.58	0.61	0.57	0.48	0.41	0.36	0.32	0.28	0.24
	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	0.29	0.26
	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	0.32	0.29	0.26	0.24
SE	L	0.05	0.04	0.04	0.03	0.03	0.13	0.28	0.43	0.55	0.62	0.63	0.57	0.48	0.42	0.37	0.33	0.28
	M	0.09	0.08	0.07	0.06	0.05	0.14	0.26	0.38	0.48	0.54	0.56	0.51	0.45	0.40	0.36	0.33	0.29
	H	0.11	0.10	0.10	0.09	0.08	0.17	0.28	0.40	0.49	0.53	0.53	0.48	0.41	0.36	0.33	0.30	0.27
SSE	L	0.07	0.05	0.04	0.04	0.03	0.06	0.15	0.29	0.43	0.55	0.63	0.64	0.60	0.25	0.45	0.40	0.35
	M	0.11	0.09	0.08	0.07	0.06	0.08	0.16	0.26	0.38	0.58	0.55	0.57	0.54	0.48	0.43	0.39	0.35
	H	0.12	0.11	0.11	0.10	0.09	0.12	0.19	0.29	0.40	0.49	0.54	0.55	0.51	0.46	0.39	0.35	0.31
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	0.45	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	0.58	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	0.55
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	0.59
	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



INTERNAL CIELING CONSTRUCTION

GROUND CONSTRUCTION



EXTERNAL WALL CONSTRUCTION

ROOF CONSRUCTION

Graduation Project
HEBRON/PALESTINE

Drawing Title:
SECTIONS

Ownership:
PALESTINE POLYTECHNIC UNIVERSITY

DRAWN BY:
Fadi Zamarah
Wisam Hraabat

PALESTINE POLYTECHNIC UNIVERSITY
Faculty of Engineering & Technology

GROUND FLOOR PLAN	SHEET #
DATE : 19/5/2011	A-21
SCALE : 1:100	