A PROJECT REPORT

ON

"DESIGN OF VARIABLE REFRIGERANT FLOW (VRF) SYSTEM"

Submitted to

UNIVERSITY OF MUMBAI

In Partial fulfillment of the Requirement for the Award of

Bachelor's degree in Mechanical Engineering

By

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Anjuman-I-Islam's Kalsekar Technical Campus, New Panvel

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CERTIFICATE

This is to certify that the project entitled

"DESIGN OF VRF SYSTEM"

Submitted by

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Is a record of bonafied work carried out by them, in the partial fulfilment of the requirement for the award of Degree of Bachelor of Engineering (Mechanical Engineering) at ***Anjuman-I-Islam's Kalsekar Technical Campus, Navi Mumbai*** under the University of Mumbai. This work is done during the year 2019-2020, under our guidance.

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Acknowledgement

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Project I Approval for Bachelor of Engineering

This Project entitled *"DESIGN OF VARIABLE FLOW (VRF) SYSTEM"* by ***Khan Shadab, Mandal Laxman, Satve Ganesh, Rakhange Eesa*** is approved for the degree of ***Bachelor of Engineering in Department of Mechanical Engineering.***

Examiners:

1) ____________________

2) ____________________

Supervisors:

1) ____________________

2) ____________________

Chairman:

____

Declaration

I declare that this written submission represents my ideas in my own words and where others ideas or words have been included. I have adequately cited and referenced the original sources. I also declare that I have adhered to all principles of academic honesty and integrity and have not misinterpreted or fabricated or falsified any idea/fact/source in my submission. I understand that any violation of the above will be cause for disciplinary action by the institute and can also evoke penal action from the sources which have thus not been properly cited or from whom proper permission has not been taken when needed.

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ABSTRACT

Air conditioner has become a very integral part of the modern society. The popularity of the air conditioner has only peaked ever since it's invention. However, it has been one of the noted reasons for the increase in Global Warming. This is due to the contribution of Refrigerants in the increase of greenhouse gases in the earth's atmosphere. It also uses a lot of energy. To save the environment from degradation as well as to save energy we need to keep upgrading the air conditioning appliance to provide more efficiency and use less refrigerant so as to cause less air pollution.

Normally the air-conditioner used is either split AC or centralized system however we are using Variable Refrigerant Flow (VRF) system which is a type of HVAC system combining both the types and providing the benefits of both.

The design is for commercial office situated in Maharashtra at longitude of 73°50' and latitude of 18° 31' with an elevation of 560m from sea level for a total air-conditioned space of 5113 sq. Ft and other office at longitude of 72° 52' and latitude 19° 5', 14 m above sea level situated in Andheri, Mumbai. The purpose is to achieve 24°C DBT and 17°C WBT from outside design conditions of 36°C DBT and 22°C WBT using standard data books.

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Chapter 1

INTRODUCTION

An **air conditioner** is a system or a machine nowadays commonly used as a household appliance as well as in offices, malls, restaurants, movie-theatres and vehicles. The primary goal of air-conditioning is human comfort which is why it qualifies more as a luxury than a need. HVAC is defined as the complete system of heating, ventilation and air conditioning.

VRF system is a combination of split AC and centralized system. It is having some advantages such as it is ductless system and it works on the principle of control flow of refrigerant with the help of electronic expansion valve which is connected with Indoor units. Outdoor and Indoor units are connected by means of communication cables. VRF system is mostly helpful in controlling of temperature individually, and zoning. Comparing with traditional HVAC systems COP of VRF system is high, noise produced is low and noise producing component are placed outside which does not affect human occupancy, VRF system require less space and does not require any mechanical rooms as in case of chiller based system as the compressors used in VRF system are of variable speed due to which power consumption is less where as other HVAC system uses fixed speed compressors.

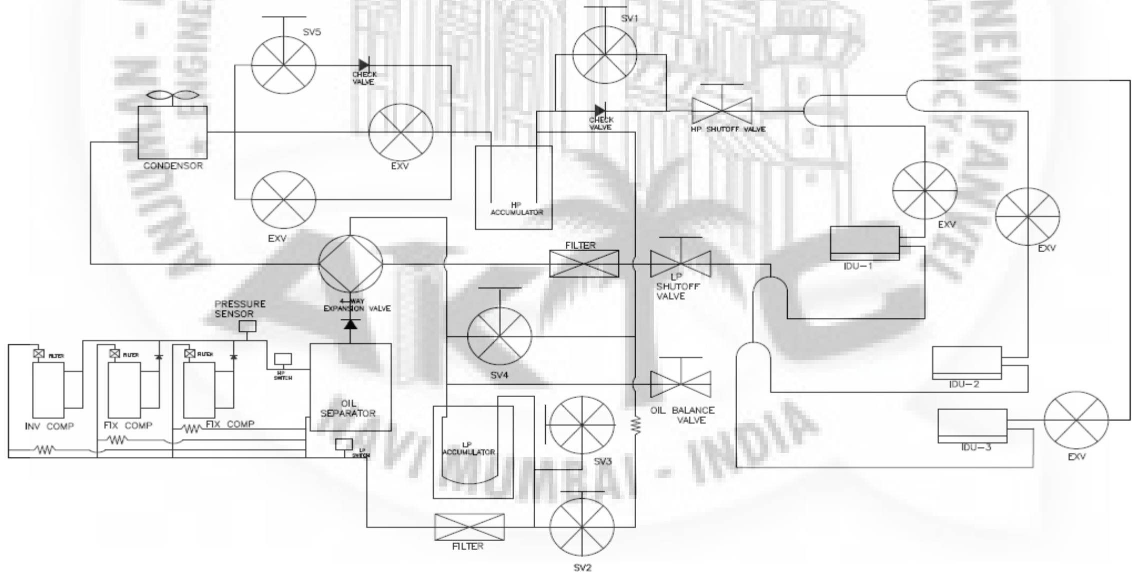

Fig.1.1. General VRF system layout

Key components:

1. Oil separator: Separates oil from gas refrigerant pumped out of the compressor and quickly returns it to the compressor. Separation efficiency is up to 99%.
2. Accumulator: Stores liquid refrigerant and oil to protect compressor from liquid hammering.

3. Electronic Expansion Valve (EXV): Controls refrigerant flow and reduces refrigerant pressure.
4. Four-way valve: Controls refrigerant flow direction. Closed in cooling mode and open in heating mode. When closed, the heat exchanger functions as a condenser; when open, the heat exchanger functions as an evaporator.
5. Solenoid valve SV2: Protects the compressor. If compressor discharge temperature rises above 100°C, SV2 opens and sprays a small amount of liquid refrigerant to cool the compressor. SV2 closes again once the discharge temperature has fallen below 90°C.
6. Solenoid valve SV4: Returns oil to the compressor. Opens once the compressor has run for 200 seconds and closes 600 seconds later and then opens for 3 minutes every 20 minutes.
7. Solenoid valve SV5: Enables fast defrosting. During defrosting operation, opens to shorten the refrigerant flow cycle and quicken the defrosting process. Closed in cooling mode.
8. Solenoid valve SV6: Allows refrigerant to bypass the expansion valves. Open in cooling mode when discharge temperature exceeds the limit. Closed in heating mode or standby.
9. High and low pressure switches: Regulate system pressure. When system pressure rises above the upper limit or falls below the lower limit, the high or low pressure switches turn off, stopping the compressor. After 10 minutes, the compressor restarts.

1.1 Purpose

The purpose of this project is to study the working, components and installation of a VRF system and to learn the advantages it has over the conventional methods of air conditioning as well as it's drawbacks and how those drawbacks can be eliminated for more efficient running and finally to design of a VRF system including the piping and wiring.

1.2 Goals

- To design a VRF system with the help of the study of different VRF components, heat load calculations and the piping and wiring diagrams.
- To achieve 24°C DBT and 17°C WBT from outside design conditions of 36°C DBT and 22°C WBT in the said VRF system using standard data books.

1.3 Objectives

- To study the different components, working and installation of the VRF system.
- To calculate the different parameters required for the design using the standard data books
- To cross check the manually calculated results with the software results.

Chapter 2

LITERATURE SURVEY

1. "A Course in Refrigeration and Air-Conditioning," by S. Domkundwar, S. C. Arora

   This is the standard book for Refrigeration and air conditioning. All the basic concepts about Air conditioning are provided in the book. The book describes about fundamental concepts of HVAC such as Comfort conditions, basic psychrometrics, Heat load and cooling load calculations, duct design etc. There are some standardize data also given in the book.

   Mainly the book is used in the research for understanding load calculations and duct designing which are described in subsequent sections.

2. ISHRAE HVAC Data book 2017

   ISHRAE HVAC data book is the standardize data book used by the engineers in India. The book includes all the information and the data for various aspects in HVAC. Book includes the climate conditions for all the regions in India likewise all the standards and constants used in HVAC calculations such as Load calculations, air distributions and duct designing. This book used in many calculations as needed in this research.

3. Voltas service manual VXV 6 2018

   Voltas service manual is needed to be referred at various stages of designing and installation such as piping, additional gas charge, indoor, outdoor installation and various other standard followed while installation of Voltas machine.

4. Handbook of Air conditioning system design, Carrier Air Conditioning Company

   This Handbook is very useful for designing the HVAC system at any location as needed. The book provides the total guide for design as well as installation of HVAC systems. Book describes the detailed procedure and also stepwise solutions for common day by day problems. The book provides practical data and techniques for developing economical HVAC system.

5. "Cooling Load Calculations", By H. M. Hashim, E. Sokolova, O Derevianko, , D. B. Solovev

   This is the Research paper published in IOP Conference of science and engineering in 2018. The article describes the thermal load calculation for buildings. The paper presents some consideration of cooling load calculation techniques. Numeral calculations are performed for a building floor which is added in the paper.

6. "Cooling Load Estimation for a Multi-story office building", By S. K. Sahu

   This is M. Tech. thesis submitted in NIT Rourkela in 2014. The thesis establishes the results of cooling load calculation of different climate conditions by using CLTD method for a multi-story building. Cooling load is calculated using MS-Excel program. End results are compared with the standard data given by ASHRAE and CARRIER Fundamental Hand Books

7. "Designing VRF Systems", By R. Afify

   In this paper, VRF system is explained thoroughly. The main advantage of a variable refrigerant flow (VRF) system is its ability to respond to fluctuations in space load conditions. By comparison, conventional direct expansion (DX) systems offer limited or no modulation in response to changes in the space load conditions. This article provides guidelines for determining the feasibility of a VRF system and discusses the factors that should be considered from initial planning through completion of a project.

8. Duct System Design Guide, McGill AirFlow Corporation

   Duct system design guide is the guide book for complete designing of duct systems. Various types of duct systems and their installation techniques are illustrated in the book. The book covers supply systems and exhaust/ return systems. There is a substantial appendix provided as reference for design work. The book gives the detailed description of Airflow fundamentals, designing and analysis of ducts.

9. Indian Standards of Air Ducts – Specification (IS 655:2006)

   IS 655:2006 has standards for Air distribution ducts. These standards are adopted by Refrigeration and air conditioning committee. The values are specified in metric system. This standard covers materials and constructional requirements of air ducts used for air conditioning and ventilation system. The standard is used for calculating PVC duct thickness discussed in subsequent section.

Chapter 3

METHODOLOGY

For Implementing the VRF System at any location many methods are carried out. The methodology used in this research work is based on manual calculations and predefined standards. These standards are specific according to the design which are taken from standard data books and manuals. All these manual calculations are verified by standard software applications.

General methodology to implement VRF system

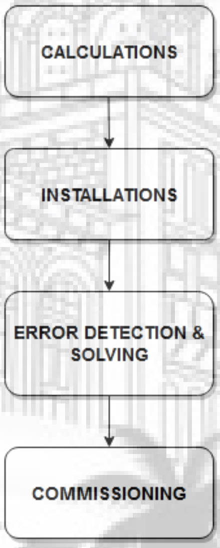

Fig.3.1. Methodology for implementing VRF system

STEP 1: Calculations

These are the manual calculations done by using standard data wherever applicable. In designing of VRF system the calculations are carried out to determine the requirements of the system. The calculations which are carried out are:

Cooling load estimations

Piping Design

Additional refrigerant calculations

Insulation

STEP 2: Installation

According to calculation results the Components are installed accordingly. These Components are

- Outdoor Unit
- Indoor Units
- Piping
- Insulation
- Wiring
- Testing
- Additional refrigerant

STEP 3: Error detection and solving

This is the checking of errors in communication system and wirings. These are identified and checked using standard manuals. The error solving is also done according to Standard manuals which are used to determine the errors.

STEP 4: Commissioning

After the installation some tests are made on the system to ensure the working of the system and to check for system failure etc. commissioning is simply validation of the system. This is created as a Commissioning report which is done by Testing & Commissioning engineer.

- Pressure test reports
- Commissioning reports

Chapter 4

PROJECTS

4.1. Project 1: Tricentis Office, Pune

This research article shows the designing of VRF system for commercial office situated in Maharashtra at longitude of 73° 50’ and latitude of 18° 31’ with an elevation of 560m from sea level for a total air-conditioned space of 5113 sq. Ft. The purpose is to achieve 24°C DBT and 17°C WBT from outside design conditions of 36°C DBT and 22°C WBT using standard data books. The article describes the design of VRF system consisting of 16 Indoor units in the conditioned space of 5113 sq.ft. Comprising of 1203 sq.ft of north facing wall, 260 sq.ft of west facing wall, 1252 sq.ft of south facing wall which are the main sources of Solar heat gains for the given office layout. The detailed procedure is given below.

Fig.4.1. Tricentis office floor layout

4.1.1. Cooling Load Calculation

It is a major part in any HVAC system. Cooling load refers to the amount of heat required to be removed from an enclosed area to achieve the required conditions. Therefore the aim is to calculate the total amount of heat generated or coming to the specified area.

It involves mainly two types of heat loads.

- Sensible heat load (mainly responsible to increase the temperature)
- Latent heat load (mainly responsible to increase the humidity)

Sensible heat sources involves direct heat gain from Sun through walls, window glasses, heat gain through partition, people, roof, ceiling, floor, lighting, equipment etc.
Whereas Latent heat gain mainly involves the latent heat coming from people; other sources are fresh air latent gain, food products or plantation (if any).

This section is mainly focused on Calculation of heat using the Load sheet (on Excel spreadsheet). Load sheet calculations is basically manual calculation of heat from the sources using the general heat equation

$$Q = UA\Delta T$$

Where
Q = the total heat gain (Btu/hr)
U = Overall heat transfer Coefficient [Btu/ (hr-ft^2 °F)]
A = Surface area (sq.ft)
ΔT = Temperature difference (°F)

A. Procedure using standard data book:

1) Outdoor and Indoor conditions:
   a) Outdoor conditions surrounding area at the time of Peak load.
   b) Indoor conditions required according to work-space.

2) Fresh Air:
   a) Fresh air CFM according to number of people.
   b) Fresh air CFM according to air changes according to applications.
   (The maximum CFM from above two is chosen for calculation purpose.)

3) Heat Loads (Heat producing sources):
   a) Sensible Heat load
      i) Exterior wall exposed to direct solar radiation
      ii) Exterior glass window
      iii) Glass partitioning
      iv) Revolving doors
      v) Ceiling
      vi) Floor
      vii) Outside fresh air sensible heat
      viii) Sensible heat produced by the people
      ix) Lighting load of the space
      x) Equipment load

   b) Latent heat load
      i) Latent heat of outside fresh air
      ii) Latent heat produced by the people

4) Estimation of Effective room sensible heat (ERSH) and Effective room latent heat (ERLH) and thereby calculating room total heat (RTH).

5) Estimation of Grand total heat (GTH) and selection of Tonnage of refrigeration for the given area.

TABLE 4.1
STANDARD VALUES OF COMMERCIAL OFFICE BUILDINGS

Sr.	Description	Standard Value
1.	Time of peak load for offices	4 PM for max. load
2.	Indoor conditions for offices	72°F to 75°F DBT, 50 to 55% RH
3.	Fresh air conditions CFM required for offices Air Changes	 5 CFM/person 4 to 10 per hour
4.	Person with light work Sensible heat load Latent heat load	 245 Btu/hr 202 Btu/hr
5.	Lighting load in offices	1.11 watt / sq.ft area
6.	Equipment load for medium offices	1 to 1.5 kW/ sq.ft

TABLE 4.2
HEAT TRANSFER COEFFICIENTS FOR BUILDING STRUCTURES

Sr.	Structure	Heat transfer coefficient U Btu/hrsq.ft °F	Assumption for current location
1.	Exterior wall exposed to direct solar radiation	0.33	Normal brick wall
2.	Glass window exposed to solar radiation	0.62	Exterior glass window
3.	Partition glass	1.13	Indoor glass
4.	Revolving doors	1.04	Slab doors
5.	Ceiling	0.27	Ceiling above ground floor
6.	Floor	0.40	Tile flooring

B. SAMPLE COOLING LOAD CALCULATION FOR 12 PEOPLE MEETING ROOM 2 (N-W CORNER ROOM)

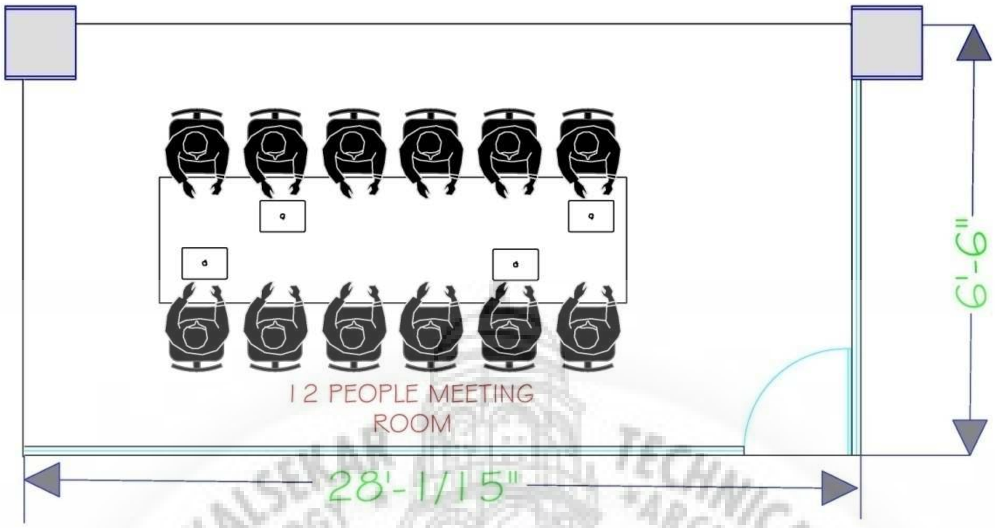

Fig.4.2. People meeting room layout (NW corner room)

Room specifications:
Area: 242 sq.ft
Height: 10 ft
No. of people: 12

Outside design conditions:
The office is situated at Pune, Maharashtra. The maximum temperature occurs in the month of May. The time of the peak load particularly for offices is 4 PM.
Therefore outside design conditions are:
DBT: 97 °F
WBT: 71 °F
% RH: 28 %
Absolute humidity: 80.08 gr/lb

Inside design conditions:
Inside air conditions are chosen for good productivity of the working staff which are:
DBT: 75 °F
WBT: 63 °F
% RH: 55 %
Absolute humidity: 76.25 gr/lbs

Fresh Air conditions: This is the fresh air required to prevent suffocation and to maintain the quality of the air.

F.A. CFM per person: 5 CFM/person
F.A. changes per hour: 5
Fresh Air CFM Calculation:
F.A. CFM on the basis of no. of people: 12 * 5 = 6
F.A. CFM on the basis of Air changes: Air changes/hr * Volume in cu.ft / 60 = 5 * 242 * 10 / 60 = 202 CFM
Fresh air required = Maximum of above two = 202 CFM

Sensible heat sources:

1. Heat gain through the exterior walls by direct solar radiation: It is directly calculated by relation Q = U*A*CLTD. For this particular room the exterior walls are at North and West side.
   i. North wall: The area of the wall is 228 sq.ft, Heat transfer coefficient for the wall is U = 0.33 Btu/hrsq.ft °F and the Equivalent temperature difference is = 12 °F
   Q = 228 * 0.33 * 12 = 903 Btu/hr
   ii. North wall: The area of the wall is 65 sq.ft, Heat transfer coefficient for the wall is U = 0.33 Btu/hrsq.ft °F and the Equivalent temperature difference is = 28 °F
   Q = 65 * 0.33 * 28 = 601 Btu/hr

2. Heat gain through the exterior glass by direct solar radiation: The heat load through glass is calculated by relation Q = U*A*CLTD. For this particular room the exterior glass is at the West side.
   i. West glass: Area for the glass is 40 sq.ft, Heat transfer coefficient U = 0.62 Btu/hr sq.ft °F and the equivalent temperature difference is 22 °F (Outside DBT - Inside DBT)
   Q = 40 * 0.62 * 22 = 546 Btu/hr

3. Heat gain through Glass partition: The heat load through glass partition is calculated by relation Q = U*A*CLTD. In commercial offices the rooms are partitioned by glass partitions having different properties than the exterior glass.
   Area of the glass partitioning for the room is 305 sq.ft, Heat transfer coefficient U = 1.13 Btu/hr sq.ft °F and the equivalent temperature difference is 10 °F
   Q = 305 * 1.13 * 10 = 3347 Btu/hr.

4. Heat gain through revolving doors: The heat load through revolving doors is calculated by relation Q = U*A*CLTD.
   Total area of the revolving doors is 30 sq.ft, Heat transfer coefficient U = 1.04 Btu/hr sq.ft °F and the equivalent temperature difference is 10 °F
   Q = 30 * 1.04 * 10 = 312 Btu/hr

5. Heat gain through the Ceiling: The heat load through Ceiling partition is calculated by relation Q = U*A*CLTD.
   Area of the Ceiling is 242 sq.ft, Heat transfer coefficient U = 0.27 Btu/hr sq.ft °F and the equivalent temperature difference is 10 °F
   Q = 242 * 0.24 * 10 = 653 Btu/hr

6. Heat gain through Floor: The heat load through the floor is calculated by relation Q = U*A*CLTD.
   Area of the floor is 242 sq.ft, Heat transfer coefficient U = 0.40 Btu/hr sq.ft °F and the equivalent temperature difference is 10 °F
   Q = 242 * 0.40 * 10 = 968 Btu/hr

7. Sensible heat gain through Fresh air: Sensible heat load from the fresh air is calculated by the relation Q = F.A. CFM * cp * density * temp. Diff, for the current conditions Q = 1.08 * CFM * temp. diff of outside and inside air
   Q = 202 * 1.08 * 22 = 4792 Btu/hr

8. Sensible heat from the people: This heat is calculated as Q = No. of people * Sensible heat of a person. People working in the offices have the sensible heat of 245 Btu/hr leaving from their body.
   Q = 12 * 245 = 2940 Btu/hr

9. Heat gain from Lighting: This is directly calculated by converting the watt load to Btu/hr. For the commercial offices the lighting watt required = 1.11 * sq.ft area of the room and the conversion factor for Watt to Btu/hr is 3.412
   Q = 242 * 1.11 * 3.412 = 916 Btu/hr

10. Heat produced by the equipment = the load from the equipment is estimated by the relation 1w/sq.ft for the offices. Therefore the Equipment load for this room is 242 * 1 / 1000 = 0.2 kW and the conversion factor of 3412.14
    Q = 0.2 * 3412.4 = 826 Btu/hr

11. Room sensible heat: This is the total Sensible heat load for the room is calculated by adding all the sensible heat loads listed above. For this room
    RSH = 16902 Btu/hr

12. Effective Room sensible heat load: ERSH = RSH + FOS, FOS is assumed as 10 % of the RSH = 1609 Btu/hr.
    ERSH = 18593 Btu/hr

Latent heat loads:

1. Latent heat load of Fresh air: Latent load coming from the outside fresh air is determined by Q = F.A. CFM * 0.68 * difference of absolute humidity of outdoor and indoor air.
   Q = 202 * 0.68 * 3.83 = 525 Btu/hr

2. Latent heat load of the people: Total latent heat produced by the people is estimated by Q = No. of people * Latent load of the person. The latent heat load the person working in the office is taken as 205 Btu/hr.
   Q = 12 * 205 = 2460 Btu/hr

3. Room latent heat load: Room latent heat load is the total Latent heat coming or produced in the room which is evaluated by adding all the latent heat loads.
   RLH = 2985 Btu/hr

4. Effective room latent heat load: ERLH = RLH + 5 % FOS = 3134 Btu/hr.

Effective Room Total heat:
Effective room total heat is the addition of Effective room sensible heat and Effective room latent heat

ERTH = ERSH + ERLH = 21727 Btu/hr

Grand total heat:
Grand total heat is calculated by adding 3 % factor of safety to the Effective room total heat. This FOS compensates the effects due to cracks, and other miscellaneous heat loads.

GTH = ERTH + 3 % FOS = 22379 Btu/hr

Air-conditioning tonnage required:
The Btu/hr heat load is converted in equivalent tonnage of refrigeration by diving 12000 from GTH.
Air conditioning tonnage required = 22379 / 12000 = 1.86 TR

Since the Tonnage calculated is 1.86 TR therefore 2 TR machine is selected for the Room

These Calculations are done using Excel sheets and similarly all cooling load calculations of each rooms/ zones are given below

HEAT LOAD DATA SHEET									
12 PEOPLE MEETING ROOM 1									
CUSTOMER	:	TRICENTS, PUNE		**DESIGN CONDITIONS**					
ESTIMATED BY	:	GANESH SATVE		**TIME OF PEAK LOAD:**			**4:00 PM**		
DATE	:	23/09/2019			**DB**	**% RH**	**WB**	**GR/LB**	
AREA	=	223	SQFT	**OUTSIDE**	97	28	71	80.08	
TOTAL HT	=	10.0	FEET	**INSIDE**	75	55	63	76.25	
				DIFF.	22	XXX	XXX	3.83	

PEOPLE :	12	NOS. X	5	CFM	=	60
5	AIR CHANGES PER HOUR				=	186
FRESH AIR					=	186

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR	EFFECTIVE ROOM LATENT HEAT			
EFFECTIVE ROOM SENSIBLE HEAT					F. A.	186	2.60	484
WALL N		0.33	12	0	PEOPLE	12	205	2460
WALL NE		0.33	14	0	RLH SUBTOTAL			2944
WALL E		0.33	14	0	FACTOR		5%	147
WALL SE		0.33	20	0	**ERLH TOTAL**			3091
WALL S		0.33	28	0	**EFFECTIVE ROOM TOTAL HEAT**			21472
WALL SW		0.33	34	0	**GRAND TOTAL HEAT SUBTOTAL**			21472
WALL W	68	0.33	28	628	FACTOR		3%	644
WALL NW		0.33	14	0	**GRAND TOTAL HEAT**			22116
GLASS N		0.62	22	0	**AIRCONDITIONING TONNAGE**			**1.84**
GLASS NE		0.62	22	0				
GLASS E		0.62	22	0	**ERSHF**			**0.86**
GLASS SE		0.62	22	0	INDICATED ADP			0.0
GLASS S		0.62	22	0	SELECTED ADP			0.0
GLASS SW		0.62	22	0	**DEHUMIDIFIER CFM**			**0**
GLASS W	102	0.62	22	1391				
GLASS NW		0.62	22	0	**RETURN AIR GAIN**			
ALL GLASS		1.13	10	0	STRUCT.	SQFT	FACTOR	BTU/HR
PRT. WL.	60	0.40	10	240	WALL N	0	0.33	0
PRT. GLS.	326	1.13	10	3684	WALL NE	0	0.33	0
DOOR	30	1.04	10	312	WALL E	0	0.33	0
ROOF		0.15	22	0	WALL SE	0	0.33	0
CEILING	223	0.27	10	602	WALL S	0	0.33	0
FLOOR	223	0.40	10	892	WALL SW	0	0.33	0
F. A.	186	1.08	22	4415	WALL W	0	0.33	0
PEOPLE	12	Nos.	245	2940	WALL NW	0	0.33	0
LIGHT (w)	223	1.11	3.41	844	PRT. WL.	0	0.40	0
EQPT. (kw)	0.2	1	3412.14	761	**3) TOTAL R.A. GAIN**			0
OTHER (kw)		1	3412.14	0				
RSH SUBTOTAL				16710				
FACTOR			10%	1671				
ERSH TOTAL				18381				

Fig.4.3. Cooling load calculation of 12 people meeting room 1

HEAT LOAD DATA SHEET								
12 PEOPLE MEETING ROOM 2								
CUSTOMER	:	TRICENTS, PUNE		**DESIGN CONDITIONS**				
ESTIMATED BY	:	GANESH SATVE		**TIME OF PEAK LOAD:**		**4:00 PM**		
DATE	:	23/09/2019			**DB**	**% RH**	**WB**	**GR/LB**
AREA	=	242	SQFT	**OUTSIDE**	97	28	71	80.08
TOTAL HT	=	10.0	FEET	**INSIDE**	75	55	63	76.25
				DIFF.	22	XXX	XXX	3.83

PEOPLE :	12	NOS. X	5	CFM =	60
5	AIR CHANGES PER HOUR			=	202
FRESH AIR				=	202

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N	228	0.33	12	903
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S		0.33	28	0
WALL SW		0.33	34	0
WALL W	65	0.33	28	601
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S		0.62	22	0
GLASS SW		0.62	22	0
GLASS W	40	0.62	22	546
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.		0.40	10	0
PRT. GLS.	305	1.13	10	3447
DOOR	30	1.04	10	312
ROOF		0.15	22	0
CEILING	242	0.27	10	653
FLOOR	242	0.40	10	968
F. A.	202	1.08	22	4792
PEOPLE	12	Nos.	245	2940
LIGHT (w)	242	1.11	3.41	916
EQPT. (kw)	0.2	1	3412.14	826
OTHER (kw)		1	3412.14	0
RSH SUBTOTAL				16902
FACTOR			10%	1690
ERSH TOTAL				18593

EFFECTIVE ROOM LATENT HEAT			
F. A.	202	2.60	525
PEOPLE	12	205	2460
RLH SUBTOTAL			2985
FACTOR		5%	149
ERLH TOTAL			3134
EFFECTIVE ROOM TOTAL HEAT			21727
GRAND TOTAL HEAT SUBTOTAL			21727
FACTOR		3%	652
GRAND TOTAL HEAT			22379
AIRCONDITIONING TONNAGE			**1.86**
ERSHF			**0.86**
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			**0**
RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.4. Cooling load calculation of 12 people meeting room 2

HEAT LOAD DATA SHEET					
5 PEOPLE MEETING ROOM					
CUSTOMER :	TRICENTS, PUNE	DESIGN CONDITIONS			
ESTIMATED BY :	GANESH SATVE	TIME OF PEAK LOAD:	4:00 PM		
DATE :	23/09/2019		DB	% RH	WB	GR/LB
AREA =	160 SQFT	OUTSIDE	97	28	71	80.08
TOTAL HT =	10.0 FEET	INSIDE	75	55	63	76.25
		DIFF.	22	XXX	XXX	3.83

PEOPLE :	5 NOS. X	5	CFM =	25
5	AIR CHANGES PER HOUR		=	133
FRESH AIR			=	133

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N		0.33	12	0
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S	43	0.33	28	397
WALL SW		0.33	34	0
WALL W		0.33	28	0
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S	64	0.62	22	873
GLASS SW		0.62	22	0
GLASS W		0.62	22	0
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.		0.40	10	0
PRT. GLS.	380	1.13	10	4294
DOOR	29	1.04	10	302
ROOF		0.15	22	0
CEILING	160	0.27	10	432
FLOOR	160	0.40	10	640
F. A.	133	1.08	22	3168
PEOPLE	5	Nos.	245	1225
LIGHT (w)	160	1.11	3.41	606
EQPT. (kw)	0.2	1	3412.14	546
OTHER (kw)		1	3412.14	0
RSH SUBTOTAL				12482
FACTOR			10%	1248
ERSH TOTAL				13731

EFFECTIVE ROOM LATENT HEAT			
F. A.	133	2.60	347
PEOPLE	5	205	1025
RLH SUBTOTAL			1372
FACTOR		5%	69
ERLH TOTAL			1441
EFFECTIVE ROOM TOTAL HEAT			15172
GRAND TOTAL HEAT SUBTOTAL			15172
FACTOR		3%	455
GRAND TOTAL HEAT			15627
AIRCONDITIONING TONNAGE			1.30
ERSHF			0.91
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			0

RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.5. Cooling load calculation of 5 people meeting room

HEAT LOAD DATA SHEET								
DISCUSSION ROOM 1								
CUSTOMER :	TRICENTS, PUNE			DESIGN CONDITIONS				
ESTIMATED BY :	GANESH SATVE			TIME OF PEAK LOAD:		4:00 PM		
DATE :	23/09/2019				DB	% RH	WB	GR/LB
AREA =	80	SQFT		OUTSIDE	97	28	71	80.08
TOTAL HT =	10.0	FEET		INSIDE	75	55	63	76.25
				DIFF.	22	XXX	XXX	3.83

PEOPLE :	4	NOS. X	5	CFM =	20
5	AIR CHANGES PER HOUR			=	67
FRESH AIR				=	67

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N		0.33	12	0
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S	34	0.33	28	314
WALL SW		0.33	34	0
WALL W		0.33	28	0
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S	51	0.62	22	696
GLASS SW		0.62	22	0
GLASS W		0.62	22	0
GLASS NW		0.62	22	0
ALL GLASS	50	1.13	10	565
PRT. WL.	33	0.40	10	132
PRT. GLS.	170	1.13	10	1921
DOOR	20	1.04	10	208
ROOF		0.15	22	0
CEILING	80	0.27	10	216
FLOOR	80	0.40	10	320
F. A.	67	1.08	22	1584
PEOPLE	4	Nos.	245	980
LIGHT (w)	80	1.11	3.41	303
EQPT. (kw)	0.1	1	3412.14	273
OTHER (kw)		1	3412.14	0
RSH SUBTOTAL				7512
FACTOR			10%	751
ERSH TOTAL				8263

EFFECTIVE ROOM LATENT HEAT			
F. A.	67	2.60	174
PEOPLE	4	205	820
RLH SUBTOTAL			994
FACTOR		5%	50
ERLH TOTAL			1043
EFFECTIVE ROOM TOTAL HEAT			9306
GRAND TOTAL HEAT SUBTOTAL			9306
FACTOR		3%	279
GRAND TOTAL HEAT			9585
AIRCONDITIONING TONNAGE			0.80
ERSHF			0.89
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			0

RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.6. Cooling load calculation of Discussion room 1

HEAT LOAD DATA SHEET					
DISCUSSION ROOM 2					

CUSTOMER :	TRICENTS, PUNE		DESIGN CONDITIONS					
ESTIMATED BY :	GANESH SATVE		TIME OF PEAK LOAD:		4:00 PM			
DATE :	23/09/2019			DB	% RH	WB	GR/LB	
AREA =	73	SQFT	OUTSIDE	97	28	71	80.08	
TOTAL HT =	10.0	FEET	INSIDE	75	55	63	76.25	
			DIFF.	22	XXX	XXX	3.83	

PEOPLE :	4	NOS. X	5	CFM =	20
5	AIR CHANGES PER HOUR			=	61
FRESH AIR				=	61

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR	EFFECTIVE ROOM LATENT HEAT			
EFFECTIVE ROOM SENSIBLE HEAT					F. A.	61	2.60	158
WALL N		0.33	12	0	PEOPLE	4	205	820
WALL NE		0.33	14	0	RLH SUBTOTAL			978
WALL E		0.33	14	0	FACTOR		5%	49
WALL SE		0.33	20	0	ERLH TOTAL			1027
WALL S		0.33	28	0	EFFECTIVE ROOM TOTAL HEAT			8771
WALL SW		0.33	34	0	GRAND TOTAL HEAT SUBTOTAL			8771
WALL W		0.33	28	0	FACTOR		3%	263
WALL NW		0.33	14	0	GRAND TOTAL HEAT			9034
GLASS N		0.62	22	0	AIRCONDITIONING TONNAGE			0.75
GLASS NE		0.62	22	0				
GLASS E		0.62	22	0	ERSHF			0.88
GLASS SE		0.62	22	0	INDICATED ADP			0.0
GLASS S		0.62	22	0	SELECTED ADP			0.0
GLASS SW		0.62	22	0	DEHUMIDIFIER CFM			0
GLASS W		0.62	22	0				
GLASS NW		0.62	22	0	RETURN AIR GAIN			
ALL GLASS	60	1.13	10	678	STRUCT.	SQFT	FACTOR	BTU/HR
PRT. WL.	40	0.40	10	160	WALL N	0	0.33	0
PRT. GLS.	226	1.13	10	2554	WALL NE	0	0.33	0
DOOR	20	1.04	10	208	WALL E	0	0.33	0
ROOF		0.15	22	0	WALL SE	0	0.33	0
CEILING	73	0.27	10	197	WALL S	0	0.33	0
FLOOR	73	0.40	10	292	WALL SW	0	0.33	0
F. A.	61	1.08	22	1445	WALL W	0	0.33	0
PEOPLE	4	Nos.	245	980	WALL NW	0	0.33	0
LIGHT (w)	73	1.11	3.41	276	PRT. WL.	0	0.40	0
EQPT. (kw)	0.1	1	3412.14	249	3) TOTAL R.A. GAIN			0
OTHER (kw)		1	3412.14	0				
RSH SUBTOTAL				7040				
FACTOR			10%	704				
ERSH TOTAL				7744				

Fig.4.7. Cooling load calculation of Discussion room 2

HEAT LOAD DATA SHEET

BREAK OUT AREA 1 WITH WORK TABLE 5

CUSTOMER	:	TRICENTS, PUNE		**DESIGN CONDITIONS**				
ESTIMATED BY	:	LAXMAN MANDAL		**TIME OF PEAK LOAD:**			**4:00 PM**	
DATE	:	23/09/2019			**DB**	**% RH**	**WB**	**GR/LB**
AREA	=	437	SQFT	**OUTSIDE**	97	28	71	80.08
TOTAL HT	=	10.0	FEET	**INSIDE**	75	55	63	76.25
				DIFF.	22	XXX	XXX	3.83

PEOPLE :	18	NOS. X	5	CFM =	90
5	AIR CHANGES PER HOUR			=	364
FRESH AIR				=	364

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR	**EFFECTIVE ROOM LATENT HEAT**			
EFFECTIVE ROOM SENSIBLE HEAT					F. A.	364	2.60	948
WALL N	110	0.33	12	436	PEOPLE	18	205	3690
WALL NE		0.33	14	0	RLH SUBTOTAL			4638
WALL E		0.33	14	0	FACTOR		5%	232
WALL SE		0.33	20	0	**ERLH TOTAL**			4870
WALL S		0.33	28	0	**EFFECTIVE ROOM TOTAL HEAT**			28874
WALL SW		0.33	34	0	**GRAND TOTAL HEAT SUBTOTAL**			28874
WALL W		0.33	28	0	FACTOR		3%	866
WALL NW		0.33	14	0	**GRAND TOTAL HEAT**			29741
GLASS N	165	0.62	22	2251	**AIRCONDITIONING TONNAGE**			**2.48**
GLASS NE		0.62	22	0				
GLASS E		0.62	22	0	**ERSHF**			**0.83**
GLASS SE		0.62	22	0	INDICATED ADP			0.0
GLASS S		0.62	22	0	SELECTED ADP			0.0
GLASS SW		0.62	22	0	**DEHUMIDIFIER CFM**			**0**
GLASS W		0.62	22	0				
GLASS NW		0.62	22	0	**RETURN AIR GAIN**			
ALL GLASS		1.13	10	0	STRUCT.	SQFT	FACTOR	BTU/HR
PRT. WL.		0.40	10	0	WALL N	0	0.33	0
PRT. GLS.		1.13	10	0	WALL NE	0	0.33	0
DOOR		1.04	10	0	WALL E	0	0.33	0
ROOF		0.15	22	0	WALL SE	0	0.33	0
CEILING	437	0.27	10	1180	WALL S	0	0.33	0
FLOOR	437	0.40	10	1748	WALL SW	0	0.33	0
F. A.	364	1.08	22	8653	WALL W	0	0.33	0
PEOPLE	18	Nos.	245	4410	WALL NW	0	0.33	0
LIGHT (w)	437	1.11	3.41	1654	PRT. WL.	0	0.40	0
EQPT. (kw)	0.4	1	3412.14	1491	**3) TOTAL R.A. GAIN**			0
OTHER (kw)		1	3412.14	0				
RSH SUBTOTAL				21822				
FACTOR			10%	2182				
ERSH TOTAL				24004				

Fig.4.8. Cooling load calculation of Breakout 1 with table 5

HEAT LOAD DATA SHEET			
BREAK OUT AREA 2			
CUSTOMER :	TRICENTS, PUNE		
ESTIMATED BY :	LAXMAN MANDAL		
DATE :	23/09/2019		
AREA =	450	SQFT	
TOTAL HT =	10.0	FEET	

DESIGN CONDITIONS				
TIME OF PEAK LOAD:	**4:00 PM**			
	DB	**% RH**	**WB**	**GR/LB**
OUTSIDE	97	28	71	80.08
INSIDE	75	55	63	76.25
DIFF.	22	XXX	XXX	3.83

PEOPLE :	18	NOS. X	5	CFM =	90
5	AIR CHANGES PER HOUR			=	375
FRESH AIR				=	375

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR	**EFFECTIVE ROOM LATENT HEAT**			
EFFECTIVE ROOM SENSIBLE HEAT					F. A.	375	2.60	977
WALL N		0.33	12	0	PEOPLE	18	205	3690
WALL NE		0.33	14	0	RLH SUBTOTAL			4667
WALL E		0.33	14	0	FACTOR		5%	233
WALL SE		0.33	20	0	**ERLH TOTAL**			4900
WALL S		0.33	28	0	**EFFECTIVE ROOM TOTAL HEAT**			27386
WALL SW		0.33	34	0	**GRAND TOTAL HEAT SUBTOTAL**			27386
WALL W		0.33	28	0	FACTOR		3%	822
WALL NW		0.33	14	0	**GRAND TOTAL HEAT**			28207
GLASS N		0.62	22	0	**AIRCONDITIONING TONNAGE**			**2.35**
GLASS NE		0.62	22	0				
GLASS E		0.62	22	0	**ERSHF**			**0.82**
GLASS SE		0.62	22	0	INDICATED ADP			0.0
GLASS S		0.62	22	0	SELECTED ADP			0.0
GLASS SW		0.62	22	0	**DEHUMIDIFIER CFM**			**0**
GLASS W		0.62	22	0				
GLASS NW		0.62	22	0	**RETURN AIR GAIN**			
ALL GLASS		1.13	10	0	STRUCT.	SQFT	FACTOR	BTU/HR
PRT. WL.	217	0.40	10	868	WALL N	0	0.33	0
PRT. GLS.		1.13	10	0	WALL NE	0	0.33	0
DOOR		1.04	10	0	WALL E	0	0.33	0
ROOF		0.15	22	0	WALL SE	0	0.33	0
CEILING	450	0.27	10	1215	WALL S	0	0.33	0
FLOOR	450	0.40	10	1800	WALL SW	0	0.33	0
F. A.	375	1.08	22	8910	WALL W	0	0.33	0
PEOPLE	18	Nos.	245	4410	WALL NW	0	0.33	0
LIGHT (w)	450	1.11	3.41	1703	PRT. WL.	0	0.40	0
EQPT. (kw)	0.5	1	3412.14	1535	**3) TOTAL R.A. GAIN**			0
OTHER (kw)		1	3412.14	0				
RSH SUBTOTAL				20442				
FACTOR			10%	2044				
ERSH TOTAL				22486				

Fig.4.9. Cooling load calculation of Breakout area 2

HEAT LOAD DATA SHEET

HALL

CUSTOMER	:	TRICENTS, PUNE		**DESIGN CONDITIONS**					
ESTIMATED BY	:	LAXMAN MANDAL		**TIME OF PEAK LOAD:**		**4:00 PM**			
DATE	:	23/09/2019			**DB**	**% RH**	**WB**	**GR/LB**	
AREA	=	741	SQFT	**OUTSIDE**	97	28	71	80.08	
TOTAL HT	=	10.0	FEET	**INSIDE**	75	55	63	76.25	
				DIFF.	22	XXX	XXX	3.83	

PEOPLE :	40	NOS. X	5	CFM =	200
5	AIR CHANGES PER HOUR			=	618
FRESH AIR				=	618

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N		0.33	12	0
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S	150	0.33	28	1386
WALL SW		0.33	34	0
WALL W		0.33	28	0
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S	223	0.62	22	3042
GLASS SW		0.62	22	0
GLASS W		0.62	22	0
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.	195	0.40	10	780
PRT. GLS.	506	1.13	10	5718
DOOR	60	1.04	10	624
ROOF		0.15	22	0
CEILING	741	0.27	10	2001
FLOOR	741	0.40	10	2964
F. A.	618	1.08	22	14672
PEOPLE	40	Nos.	245	9800
LIGHT (w)	741	1.11	3.41	2805
EQPT. (kw)	0.7	1	3412.14	2528
OTHER (kw)	1.5	1	3412.14	5118
RSH SUBTOTAL				51437
FACTOR			10%	5144
ERSH TOTAL				56581

EFFECTIVE ROOM LATENT HEAT			
F. A.	618	2.60	1608
PEOPLE	40	205	8200
RLH SUBTOTAL			9808
FACTOR		5%	490
ERLH TOTAL			10299
EFFECTIVE ROOM TOTAL HEAT			66880
GRAND TOTAL HEAT SUBTOTAL			66880
FACTOR		3%	2006
GRAND TOTAL HEAT			68886
AIRCONDITIONING TONNAGE			**5.74**
ERSHF			**0.85**
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			**0**
RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.10. Cooling load calculation of Hall

HEAT LOAD DATA SHEET

WORKTABLE 1 & 2

CUSTOMER	:	TRICENTS, PUNE	
ESTIMATED BY	:	LAXMAN MANDAL	
DATE	:	23/09/2019	
AREA	=	310	SQFT
TOTAL HT	=	10.0	FEET

DESIGN CONDITIONS				
TIME OF PEAK LOAD:		4:00 PM		
	DB	% RH	WB	GR/LB
OUTSIDE	97	28	71	80.08
INSIDE	75	55	63	76.25
DIFF.	22	XXX	XXX	3.83

PEOPLE :	15	NOS. X	5	CFM =	75
5	AIR CHANGES PER HOUR			=	258
FRESH AIR				=	258

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR	EFFECTIVE ROOM LATENT HEAT			
EFFECTIVE ROOM SENSIBLE HEAT					F. A.	258	2.60	673
WALL N	88	0.33	12	348	PEOPLE	15	205	3075
WALL NE		0.33	14	0	RLH SUBTOTAL			3748
WALL E		0.33	14	0	FACTOR		5%	187
WALL SE		0.33	20	0	ERLH TOTAL			3935
WALL S		0.33	28	0	EFFECTIVE ROOM TOTAL HEAT			24999
WALL SW		0.33	34	0	GRAND TOTAL HEAT SUBTOTAL			24999
WALL W		0.33	28	0	FACTOR		3%	750
WALL NW		0.33	14	0	GRAND TOTAL HEAT			25749
GLASS N		0.62	22	0	AIRCONDITIONING TONNAGE			2.15
GLASS NE	131	0.62	22	1787				
GLASS E		0.62	22	0	ERSHF			0.84
GLASS SE		0.62	22	0	INDICATED ADP			0.0
GLASS S		0.62	22	0	SELECTED ADP			0.0
GLASS SW		0.62	22	0	DEHUMIDIFIER CFM			0
GLASS W		0.62	22	0				
GLASS NW		0.62	22	0	RETURN AIR GAIN			
ALL GLASS		1.13	10	0	STRUCT.	SQFT	FACTOR	BTU/HR
PRT. WL.		0.40	10	0	WALL N	0	0.33	0
PRT. GLS.	105	1.13	10	1187	WALL NE	0	0.33	0
DOOR		1.04	10	0	WALL E	0	0.33	0
ROOF		0.15	22	0	WALL SE	0	0.33	0
CEILING	310	0.27	10	837	WALL S	0	0.33	0
FLOOR	310	0.40	10	1240	WALL SW	0	0.33	0
F. A.	258	1.08	22	6138	WALL W	0	0.33	0
PEOPLE	15	Nos.	245	3675	WALL NW	0	0.33	0
LIGHT (w)	310	1.11	3.41	1173	PRT. WL.	0	0.40	0
EQPT. (kw)	0.3	1	3412.14	1058	3) TOTAL R.A. GAIN			0
OTHER (kw)	0.5	1	3412.14	1706				
RSH SUBTOTAL				19149				
FACTOR			10%	1915				
ERSH TOTAL				21064				

Fig.4.11. Cooling load calculation of Work table 1 & 2

HEAT LOAD DATA SHEET

WORKTABLE 3 & 4

CUSTOMER	:	TRICENTS, PUNE	
ESTIMATED BY	:	SHADAB KHAN	
DATE	:	23/09/2019	
AREA	=	330	SQFT
TOTAL HT	=	10.0	FEET

DESIGN CONDITIONS				
TIME OF PEAK LOAD:		4:00 PM		
	DB	% RH	WB	GR/LB
OUTSIDE	97	28	71	80.08
INSIDE	75	55	63	76.25
DIFF.	22	XXX	XXX	3.83

PEOPLE :	18	NOS. X	5	CFM =	90
5	AIR CHANGES PER HOUR			=	275
FRESH AIR				=	275

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N	90	0.33	12	356
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S		0.33	28	0
WALL SW		0.33	34	0
WALL W		0.33	28	0
WALL NW		0.33	14	0
GLASS N	135	0.62	22	1841
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S		0.62	22	0
GLASS SW		0.62	22	0
GLASS W		0.62	22	0
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.		0.40	10	0
PRT. GLS.		1.13	10	0
DOOR		1.04	10	0
ROOF		0.15	22	0
CEILING	330	0.27	10	891
FLOOR	330	0.40	10	1320
F. A.	275	1.08	22	6534
PEOPLE	18	Nos.	245	4410
LIGHT (w)	330	1.11	3.41	1249
EQPT. (kw)	0.3	1	3412.14	1126
OTHER (kw)	0.5	1	3412.14	1706
RSH SUBTOTAL				19434
FACTOR			10%	1943
ERSH TOTAL				21377

EFFECTIVE ROOM LATENT HEAT			
F. A.	275	2.60	716
PEOPLE	18	205	3690
RLH SUBTOTAL			4406
FACTOR		5%	220
ERLH TOTAL			4627
EFFECTIVE ROOM TOTAL HEAT			26004
GRAND TOTAL HEAT SUBTOTAL			26004
FACTOR		3%	780
GRAND TOTAL HEAT			26784
AIRCONDITIONING TONNAGE			2.23
ERSHF			0.82
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			0

RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.12. Cooling load calculation of Work table 3 & 4

HEAT LOAD DATA SHEET								
WORKTABLE 6 & 7								
CUSTOMER	:	TRICENTS, PUNE		DESIGN CONDITIONS				
ESTIMATED BY	:	SHADAB KHAN		TIME OF PEAK LOAD:		4:00 PM		
DATE	:	23/09/2019			DB	% RH	WB	GR/LB
AREA	=	340	SQFT	OUTSIDE	97	28	71	80.08
TOTAL HT	=	10.0	FEET	INSIDE	75	55	63	76.25
				DIFF.	22	XXX	XXX	3.83

PEOPLE :	18	NOS. X	5	CFM =	90
5	AIR CHANGES PER HOUR			=	283
FRESH AIR				=	283

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N	92	0.33	12	364
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S		0.33	28	0
WALL SW		0.33	34	0
WALL W		0.33	28	0
WALL NW		0.33	14	0
GLASS N	138	0.62	22	1882
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S		0.62	22	0
GLASS SW		0.62	22	0
GLASS W		0.62	22	0
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.		0.40	10	0
PRT. GLS.	110	1.13	10	1243
DOOR		1.04	10	0
ROOF		0.15	22	0
CEILING	340	0.27	10	918
FLOOR	340	0.40	10	1360
F. A.	283	1.08	22	6732
PEOPLE	18	Nos.	245	4410
LIGHT (w)	340	1.11	3.41	1287
EQPT. (kw)	0.3	1	3412.14	1160
OTHER (kw)	0.5	1	3412.14	1706
RSH SUBTOTAL				21063
FACTOR			10%	2106
ERSH TOTAL				23169

EFFECTIVE ROOM LATENT HEAT			
F. A.	283	2.60	738
PEOPLE	18	205	3690
RLH SUBTOTAL			4428
FACTOR		5%	221
ERLH TOTAL			4649
EFFECTIVE ROOM TOTAL HEAT			27818
GRAND TOTAL HEAT SUBTOTAL			27818
FACTOR		3%	835
GRAND TOTAL HEAT			28653
AIRCONDITIONING TONNAGE			2.39
ERSHF			0.83
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			0

RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.13. Cooling load calculation of Work table 6 & 7

HEAT LOAD DATA SHEET				
WORKTABLE 8 & 9				
CUSTOMER	:	TRICENTS, PUNE		
ESTIMATED BY	:	SHADAB KHAN		
DATE	:	23/09/2019		
AREA	=	480	SQFT	
TOTAL HT	=	10.0	FEET	

DESIGN CONDITIONS				
TIME OF PEAK LOAD:		4:00 PM		
	DB	% RH	WB	GR/LB
OUTSIDE	97	28	71	80.08
INSIDE	75	55	63	76.25
DIFF.	22	XXX	XXX	3.83

PEOPLE :	30	NOS. X	5	CFM =	150
5	AIR CHANGES PER HOUR			=	400
FRESH AIR				=	400

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N		0.33	12	0
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S		0.33	28	0
WALL SW		0.33	34	0
WALL W		0.33	28	0
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S		0.62	22	0
GLASS SW		0.62	22	0
GLASS W		0.62	22	0
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.	64	0.40	10	256
PRT. GLS.		1.13	10	0
DOOR		1.04	10	0
ROOF		0.15	22	0
CEILING	480	0.27	10	1296
FLOOR	480	0.40	10	1920
F. A.	400	1.08	22	9504
PEOPLE	30	Nos.	245	7350
LIGHT (w)	480	1.11	3.41	1817
EQPT. (kw)	0.7	1	3412.14	2457
OTHER (kw)	0.7	1	3412.14	2388
RSH SUBTOTAL				26988
FACTOR			10%	2699
ERSH TOTAL				29687

EFFECTIVE ROOM LATENT HEAT			
F. A.	400	2.60	1042
PEOPLE	30	205	6150
RLH SUBTOTAL			7192
FACTOR		5%	360
ERLH TOTAL			7551
EFFECTIVE ROOM TOTAL HEAT			37238
GRAND TOTAL HEAT SUBTOTAL			37238
FACTOR		3%	1117
GRAND TOTAL HEAT			38355
AIRCONDITIONING TONNAGE			3.20
ERSHF			0.80
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			0

RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.14. Cooling load calculation of Work table 8 & 9

HEAT LOAD DATA SHEET

WORKTABLE 10 & 11

CUSTOMER :		TRICENTS, PUNE	
ESTIMATED BY :		SHADAB KHAN	
DATE :		23/09/2019	
AREA	=	511	SQFT
TOTAL HT	=	10.0	FEET

DESIGN CONDITIONS				
TIME OF PEAK LOAD:		4:00 PM		
	DB	% RH	WB	GR/LB
OUTSIDE	97	28	71	80.08
INSIDE	75	55	63	76.25
DIFF.	22	XXX	XXX	3.83

PEOPLE :	25	NOS. X	5	CFM =	125
5	AIR CHANGES PER HOUR			=	426
FRESH AIR				=	426

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N		0.33	12	0
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S		0.33	28	0
WALL SW		0.33	34	0
WALL W		0.33	28	0
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S		0.62	22	0
GLASS SW		0.62	22	0
GLASS W		0.62	22	0
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.		0.40	10	0
PRT. GLS.	105	1.13	10	1187
DOOR	20	1.04	10	208
ROOF		0.15	22	0
CEILING	511	0.27	10	1380
FLOOR	511	0.40	10	2044
F. A.	426	1.08	22	10118
PEOPLE	25	Nos.	245	6125
LIGHT (w)	511	1.11	3.41	1934
EQPT. (kw)	0.8	1	3412.14	2615
OTHER (kw)	0.5	1	3412.14	1706
RSH SUBTOTAL				27317
FACTOR			10%	2732
ERSH TOTAL				30048

EFFECTIVE ROOM LATENT HEAT			
F. A.	426	2.60	1109
PEOPLE	25	205	5125
RLH SUBTOTAL			6234
FACTOR		5%	312
ERLH TOTAL			6546
EFFECTIVE ROOM TOTAL HEAT			36594
GRAND TOTAL HEAT SUBTOTAL			36594
FACTOR		3%	1098
GRAND TOTAL HEAT			37692
AIRCONDITIONING TONNAGE			3.14
ERSHF			0.82
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			0

RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.15. Cooling load calculation of Work table 10 & 11

HEAT LOAD DATA SHEET								
WORKTABLE 12 & 13								
CUSTOMER	:	TRICENTS, PUNE		**DESIGN CONDITIONS**				
ESTIMATED BY	:	EESA RAKHANGE		**TIME OF PEAK LOAD:**		**4:00 PM**		
DATE	:	23/09/2019			**DB**	**% RH**	**WB**	**GR/LB**
AREA	=	400	SQFT	**OUTSIDE**	97	28	71	80.08
TOTAL HT	=	10.0	FEET	**INSIDE**	75	55	63	76.25
				DIFF.	22	XXX	XXX	3.83

PEOPLE :	18	NOS. X	5	CFM =	90
5	AIR CHANGES PER HOUR			=	333
FRESH AIR				=	333

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR	**EFFECTIVE ROOM LATENT HEAT**			
EFFECTIVE ROOM SENSIBLE HEAT					F. A.	333	2.60	868
WALL N		0.33	12	0	PEOPLE	18	205	3690
WALL NE		0.33	14	0	RLH SUBTOTAL			4558
WALL E		0.33	14	0	FACTOR		5%	228
WALL SE		0.33	20	0	**ERLH TOTAL**			4786
WALL S		0.33	28	0	**EFFECTIVE ROOM TOTAL HEAT**			32023
WALL SW		0.33	34	0	**GRAND TOTAL HEAT SUBTOTAL**			32023
WALL W		0.33	28	0	FACTOR		3%	961
WALL NW		0.33	14	0	**GRAND TOTAL HEAT**			32984
GLASS N		0.62	22	0	**AIRCONDITIONING TONNAGE**			**2.75**
GLASS NE		0.62	22	0				
GLASS E		0.62	22	0	**ERSHF**			**0.85**
GLASS SE		0.62	22	0	INDICATED ADP			0.0
GLASS S		0.62	22	0	SELECTED ADP			0.0
GLASS SW		0.62	22	0	**DEHUMIDIFIER CFM**			**0**
GLASS W		0.62	22	0				
GLASS NW		0.62	22	0	**RETURN AIR GAIN**			
ALL GLASS		1.13	10	0	STRUCT.	SQFT	FACTOR	BTU/HR
PRT. WL.	140	0.40	10	560	WALL N	0	0.33	0
PRT. GLS.	380	1.13	10	4294	WALL NE	0	0.33	0
DOOR	30	1.04	10	312	WALL E	0	0.33	0
ROOF		0.15	22	0	WALL SE	0	0.33	0
CEILING	400	0.27	10	1080	WALL S	0	0.33	0
FLOOR	400	0.40	10	1600	WALL SW	0	0.33	0
F. A.	333	1.08	22	7920	WALL W	0	0.33	0
PEOPLE	18	Nos.	245	4410	WALL NW	0	0.33	0
LIGHT (w)	400	1.11	3.41	1514	PRT. WL.	0	0.40	0
EQPT. (kw)	0.4	1	3412.14	1365	**3) TOTAL R.A. GAIN**			0
OTHER (kw)	0.5	1	3412.14	1706				
RSH SUBTOTAL				24761				
FACTOR			10%	2476				
ERSH TOTAL				27237				

Fig.4.16. Cooling load calculation of Work table 12 & 13

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HEAT LOAD DATA SHEET									
WORKTABLE 14 & 15									
CUSTOMER	:	TRICENTS, PUNE		**DESIGN CONDITIONS**					
ESTIMATED BY	:	EESA RAKHANGE		**TIME OF PEAK LOAD:**		**4:00 PM**			
DATE	:	23/09/2019			**DB**	**% RH**	**WB**	**GR/LB**	
AREA	=	355	SQFT	**OUTSIDE**	97	28	71	80.08	
TOTAL HT	=	10.0	FEET	**INSIDE**	75	55	63	76.25	
				DIFF.	22	XXX	XXX	3.83	

PEOPLE :	18	NOS. X	5	CFM	=	90
5	AIR CHANGES PER HOUR				=	296
FRESH AIR					=	296

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N		0.33	12	0
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S	107	0.33	28	989
WALL SW		0.33	34	0
WALL W		0.33	28	0
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S	160	0.62	22	2182
GLASS SW		0.62	22	0
GLASS W		0.62	22	0
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.	133	0.40	10	532
PRT. GLS.		1.13	10	0
DOOR		1.04	10	0
ROOF		0.15	22	0
CEILING	355	0.27	10	959
FLOOR	355	0.40	10	1420
F. A.	296	1.08	22	7029
PEOPLE	18	Nos.	245	4410
LIGHT (w)	355	1.11	3.41	1344
EQPT. (kw)	0.5	1	3412.14	1817
OTHER (kw)	0.5	1	3412.14	1706
RSH SUBTOTAL				22387
FACTOR			10%	2239
ERSH TOTAL				24626

EFFECTIVE ROOM LATENT HEAT			
F. A.	296	2.60	770
PEOPLE	18	205	3690
RLH SUBTOTAL			4460
FACTOR		5%	223
ERLH TOTAL			4683
EFFECTIVE ROOM TOTAL HEAT			29310
GRAND TOTAL HEAT SUBTOTAL			29310
FACTOR		3%	879
GRAND TOTAL HEAT			30189
AIRCONDITIONING TONNAGE			**2.52**
ERSHF			**0.84**
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			**0**
RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.17. Cooling load calculation of Work table 14 & 15

Similarly the heat load is calculated and Tonnage of refrigeration is selected for all the rooms and the final layout is shown below:

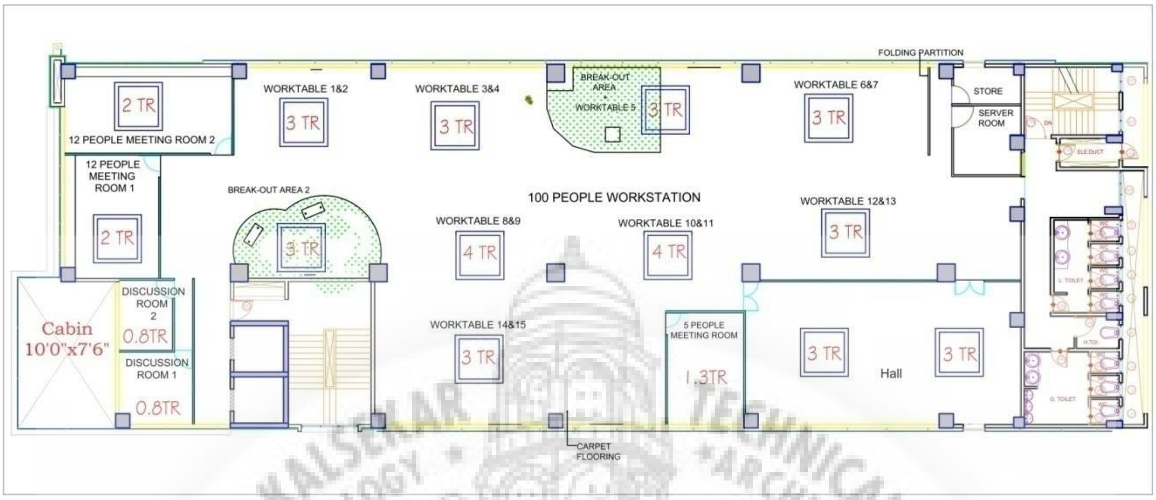


Fig.4.18. TR selection (Tricentis office)

All the cooling load calculation results are verified using the software HeatCad 2019

Cooling Rooms

Room	Area	Room Temp	AED	Cooling Load
5 people meeting room	173	75	NO	15,498
12 People meeting room 2	302	75	NO	22,669
12 People Meeting room 1	244	75	NO	22,382
BreakOut 1 + Table 5	452	75	YES	30,206
BreakOut Area 2	310	75	YES	28,627
Discussion room 1	118	75	NO	9,635
Discussion room 2	75	75	YES	9,112
Hall part1	396	75	NO	34,606
Hall part2	392	75	NO	32,816
Table 1 & 2	342	75	YES	26,254
Table 3 & 4	313	75	YES	26,829
Table 6 & 7	379	75	YES	30,637
Table 8 & 9	480	75	YES	40,392
Table 10 & 11	511	75	YES	37,746
Table 12 & 13	356	75	YES	33,538
Table 14 & 15	386	75	NO	31,872

Fig.4.19. Cooling load results by HeatCAD 2019 (Tricentis office)

4.1.2. Piping Design

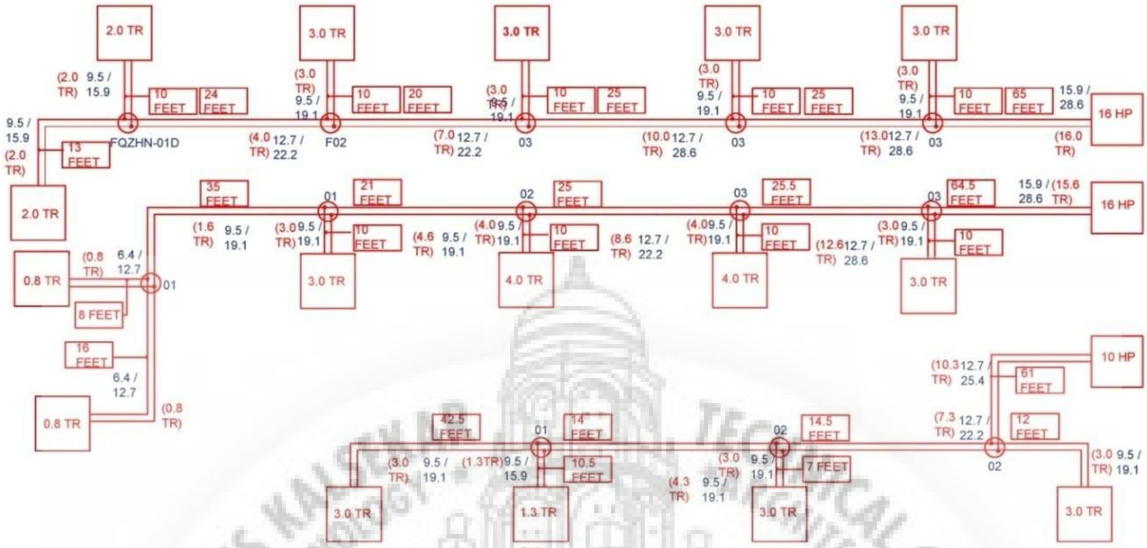


Fig.4.20. Schematic piping diagram

Major challenges faced while designing of VRF system is piping design. The following factors should be taken into consideration while designing.

The maximum piping length should not exceed 150m ASHRAE standard 15-2001. Distance between first bend and outdoor unit should be minimum 1m. Maximum distance between outdoor unit (master unit) and indoor unit (last slave) should be 90m. Minimum distance between first bend and u joint should be 500mm.

Mainly there are three types of pipe liquid pipe (discharge), gaseous pipe (suction) and drain pipe. Piping design is based on selection of liquid side line and gaseous side line for indoor and outdoor using predefined value set by the manufacturer. Diameter of pipe is selected form Table 4.3, Table 4.4

TABLE 4.3
PIPE SIZES FOR INDOOR UNITS

Capacity of indoor unit (TR)	Size of main pipe (mm)		
	Gas side	Liquid side	Available Branching pipe
<4.7	Φ19.1	Φ9.5	FQZHN-01C
4.7 to 6.5	Φ22.2	Φ9.5	FQZHN-02C
6.5 to9.4	Φ22.2	Φ12.7	FQZHN-02C
9.4 to13.1	Φ25.4	Φ12.7	FQZHN-03C
13.1to18.8	Φ34.9	Φ15.9	FQZHN-03C
18.8 to26.2	Φ41.3	Φ19.1	FQZHN-04C

TABLE 4.4
SIZE OF JOINT PIPE FOR 410A OUTDOOR UNIT

Model	When the equivalent length of all liquid pipe <90m The size of main pipe (mm)		
	Gas side	Liquid side	Available Branching pipe
8HP	Φ22.2	Φ12.7	FQZHN-02C
10HP	Φ25.4	Φ12.7	FQZHN-02C
12HP	Φ28.6	Φ12.7	FQZHN-03C
14HP-16HP	Φ28.6	Φ15.9	FQZHN-03C
24HP	Φ31.8	Φ15.9	FQZHN-03C
26HP-32HP	Φ34.9	Φ15.9	FQZHN-04C

.

A. Sample calculation of 10 HP system:

Tonnage of refrigeration from piping diagram for 10 HP system

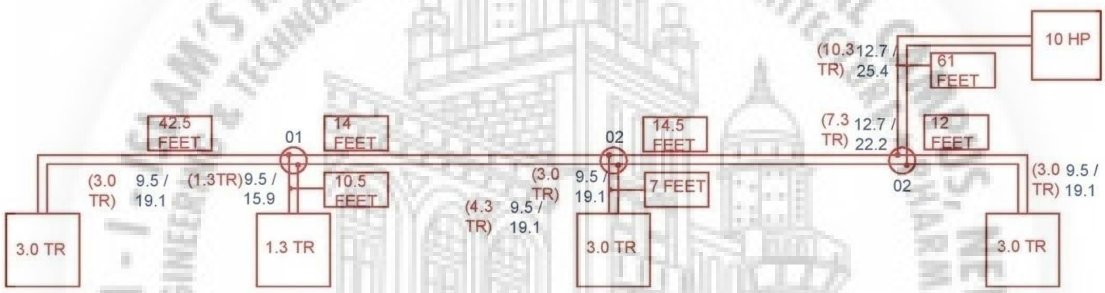


Fig.4.21. 10 HP System Piping diagram

Diameters are taken from Table 4.5

TABLE 4.5
SAMPLE CALCULATION OF PIPING FOR 10 HP SYSTEMS

Slave unit	Tonnage of refrigeration in (TR)	Diameters (mm)		Branching joint
		Gas side	Liquid side	
Last slave unit consider as first slave unit	3.0	19.1	9.5	FQZHN-01C
Combination of last and second last slave unit	3.0 + 1.3 = 4.3	19.1	9.5	FQZHN-01C
Combination of last three slave units	3.0 + 1.3 + 3.0 = 7.3	22.2	12.7	FQZHN-02C
Combination of all four slave units	3.0 + 1.3 + 3.0 + 3.0 = 10.3	25.4	12.7	-

Pipe selection for outdoor system is as follows:

TABLE 4.6
Piping Sizes for Outdoor Units

Outdoor units	DIAMETER		Branching joint
	Gas side	Liquid side	
10HP	Φ25.4	Φ12.7	FQZHN-02C
16HP	Φ28.6	Φ15.9	FQZHN-03C
16HP	Φ28.6	Φ15.9	FQZHN-03C

Results generated by software VXV selection software 2.7.7:

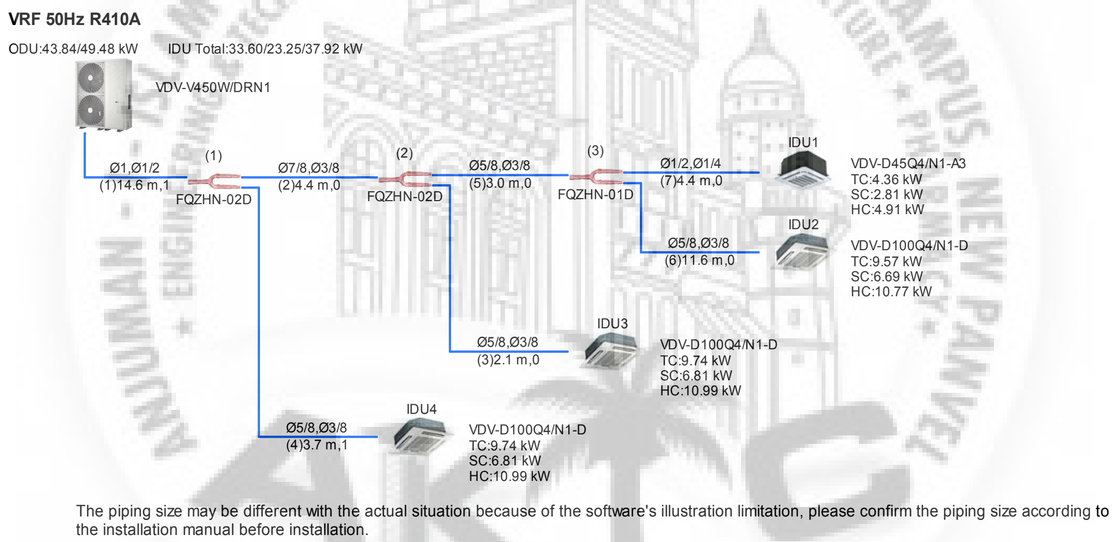


Fig.4.22. 10 HP system piping diagram by VXV selection software 2.7.7

B. 16HP system (middle) Piping design: (using VXV selection software)

TABLE 4.7
16HP system (middle) piping specifications

Pipe

No.	Length	Gas Pipe	Liquid Pipe
(1)	13.0 m	Ø1	Ø1/2
(2)	7.8 m	Ø1	Ø1/2
(3)	7.6 m	Ø7/8	Ø3/8
(4)	6.4 m	Ø5/8	Ø3/8
(5)	10.0 m	Ø5/8	Ø3/8
(6)	3.0 m	Ø5/8	Ø3/8
(7)	3.0 m	Ø5/8	Ø3/8
(8)	3.0 m	Ø5/8	Ø3/8
(9)	3.0 m	Ø5/8	Ø3/8
(10)	4.8 m	Ø1/2	Ø1/4
(11)	2.4 m	Ø1/2	Ø1/4

Branch Joint

No.	Load kW	Model
(1)	53.60	FQZHN-02D
(2)	43.60	FQZHN-02D
(3)	29.60	FQZHN-02D
(4)	15.60	FQZHN-01D
(5)	5.60	FQZHN-01D

Piping Diagram (16 HP middle system)

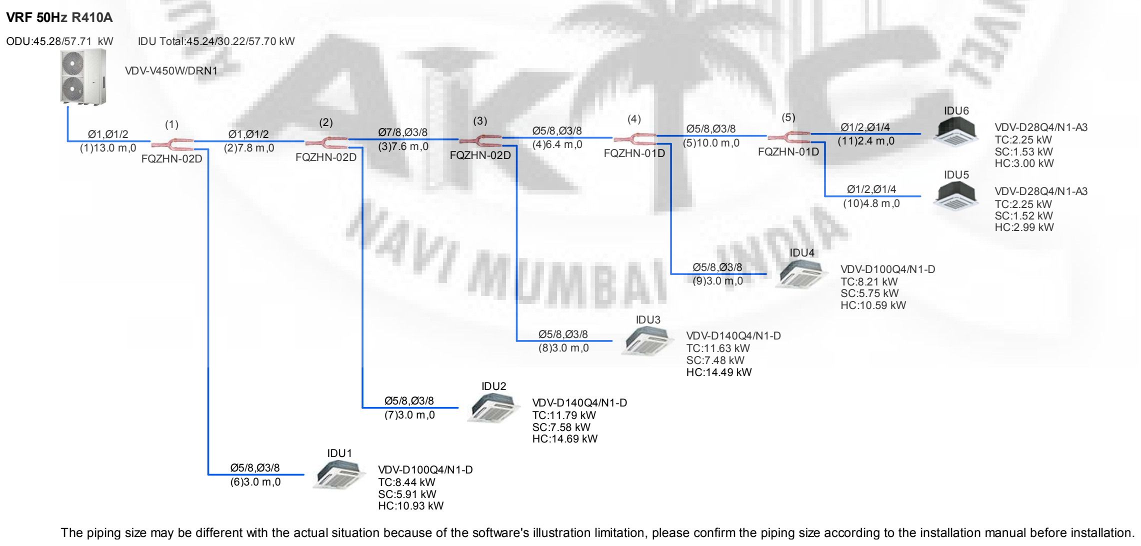


Fig.4.23. 16 HP system (middle) piping diagram by VXV selection software 2.7.7

C. Piping design for 16 HP system (Right) (using VXV selection software):

TABLE 4.8
16HP system (right) piping specifications

Pipe

No.	Length	Gas Pipe	Liquid Pipe
(1)	12.0 m	Ø1	Ø1/2
(2)	7.6 m	Ø1	Ø1/2
(3)	7.6 m	Ø1	Ø1/2
(4)	6.0 m	Ø7/8	Ø3/8
(5)	7.3 m	Ø5/8	Ø3/8
(6)	3.0 m	Ø5/8	Ø3/8
(7)	3.0 m	Ø5/8	Ø3/8
(8)	3.0 m	Ø5/8	Ø3/8
(9)	3.0 m	Ø5/8	Ø3/8
(10)	4.0 m	Ø5/8	Ø3/8
(11)	3.0 m	Ø5/8	Ø3/8

Branch Joint

No.	Load kW	Model
(1)	54.20	FQZHN-02D
(2)	44.20	FQZHN-02D
(3)	34.20	FQZHN-02D
(4)	24.20	FQZHN-02D
(5)	14.20	FQZHN-01D

Piping Diagram (16 HP Right system)

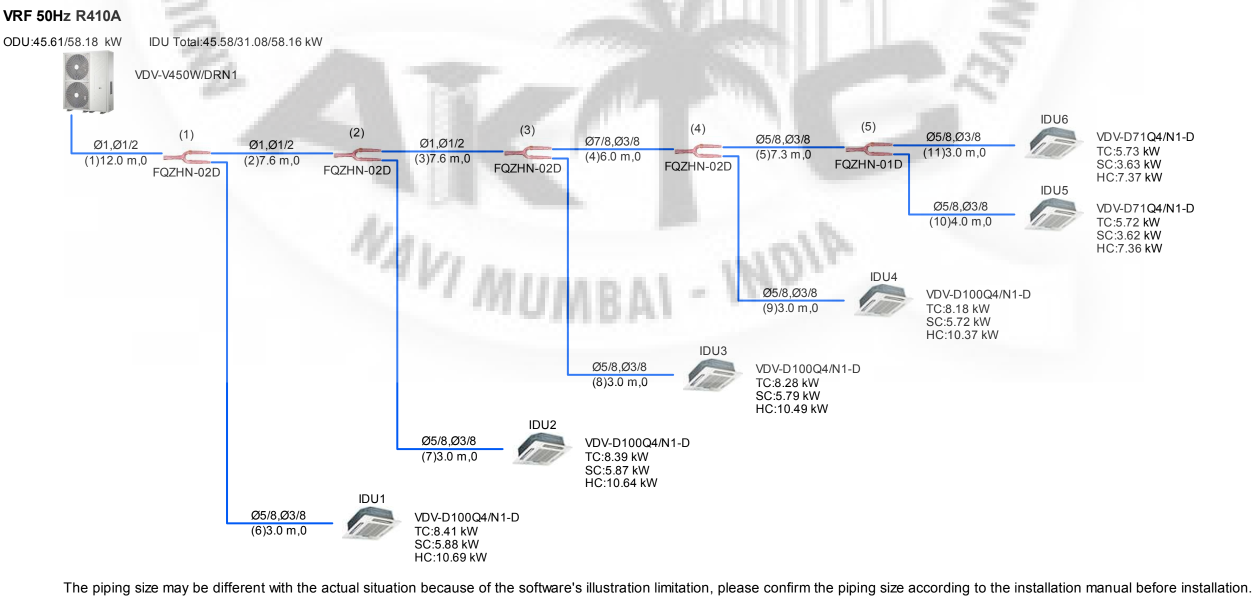


Fig.4.24. 16 HP system (right) piping diagram by VXV selection software 2.7.7

4.1.3. Additional refrigerant calculations

The actually installed VRF system which was been viewed while preparation of this paper .For three system installed, the factory charge of each system is as follows.

10 HP (6.2kg)
16HP middle (12kg)
16 HP right (12kg)

Factory charge refrigerant is not enough for evaporation. Pipe length increases refrigerant required per unit length also increases. Additional refrigerant required per meter unit of pipe is provided by the manufacturer in the Table 4.9

TABLE 4.9
ADDITIONAL REFRIGERANT REQUIRED PER METER OF THE PIPE

Sr. no	Pipe size of liquid side (in mm)	Additional refrigerant charge per meter (kg)
(L1)	ϕ6.35	0.022
(L2)	ϕ9.53	0.057
(L3)	ϕ12.7	0.110
(L4)	ϕ15.9	0.170
(L5)	ϕ19.1	0.260
(L6)	ϕ22.2	0.360
(L7)	ϕ25.4	0.520
(L8)	ϕ28.6	0.068

Formula for calculating extra refrigerant required in the system.

*Recharged volume: R(kg)=((L1*0.022) + (L2*0.057) + (L3*0.110) + (L4*0.170) + (L5*0.260) + (L6*0.360) + (L7*0.520) + (L8*0.068))*

A. Sample calculation of 10HP system:

Length of ϕ9.53 pipe in meter: 26.213(L2)
Length of ϕ12.7 pipe in meter: 13.321(L3)
R (kg) =26.213*0.057+13.321*0.110=2.96 kg

Similarly additional refrigerant required for the systems is calculated and shown below

TABLE 4.10
ADDITIONAL REFRIGERANT REQUIRED IN THE SYSTEMS

16 HP (right system)	5.50 kg
16 HP (middle system)	4.78 kg
10 HP (Left system)	2.96 kg

B. Results generated by software VXV selection software 2.7.7:

TABLE 4.11
ADDITIONAL REFRIGERANT REQUIREMENT REPORT (10 HP left system)

IDU Quantity	4/15
Combination Ratio	76.67%
Additional refrigerant charge	3.0 kg = 4.42(1/4) * 0.022 + 26.29(3/8) * 0.057 + 15.63(1/2) * 0.110
Factory refrigerant charge	6.2 kg
Total refrigerant charge	9.2 kg
Total Pipe Length	46.3428 m / 250 m

TABLE 4.12
ADDITIONAL REFRIGERANT REQUIREMENT REPORT (16 HP middle system)

IDU Quantity	6/15
Combination Ratio	119.11%
Additional refrigerant charge	4.69 kg = 7.23(1/4) * 0.022 + 37.52(3/8) * 0.057 + 21.77(1/2) * 0.110
Factory refrigerant charge	12.00 kg
Total refrigerant charge	16.69 kg
Total Pipe Length	66.52 m / 250 m

TABLE 4.13
ADDITIONAL REFRIGERANT REQUIREMENT REPORT (16 HP right system)

IDU Quantity	6/15
Combination Ratio	120.44%
Additional refrigerant charge	5.06 kg = 33.27(3/8) * 0.057 + 28.74(1/2) * 0.110
Factory refrigerant charge	12.00 kg
Total refrigerant charge	17.06 kg
Total Pipe Length	62.01 m / 250 m

4.1.4. Optimum insulation of pipe

Previous methods used for optimum insulation are by depending upon heating pipe and cooling pipe.
Heating pipe consist of HPGP (high pressure gas pipe) & LPLP (low pressure liquid pipe) and Cooling pipe consists of LPGP (low pressure gas pipe) & LPLP (lo pressure liquid pipe).

TABLE 4.14
OPTIMUM THICKNESS OF INSULATION

Operating mode	Pipe line	Optimum insulation thickness (mm)
Heating pipe	HPGP	16 to 20
	LPLP	11 to 13
Cooling pipe	LPGP	7 to 8
	LPLP	

Recent methods deals with optimal insulation of different pipes used in VRF system which work under different condition, minimum insulation thickness of piping in hot humid environment varies in given table:

TABLE 4.15
MINIMUM INSULATION THICKNESS FOR VARIOUS PIPE DIAMETERS

Pipe Outer Diameter (mm)	**Minimum Insulation Thickness (mm)**
Φ6.35	15
Φ9.53	
Φ12.7	
Φ15.9	20
Φ19.1	
Φ22.2	
Φ25.4	
Φ28.6	
Φ31.8	
Φ38.1	
Φ41.3	25
Φ44.5	
Φ54.0	

Critical insulation is by the formula

$$Rc = k / h$$

Rc = Critical insulation in mm

Material used for insulation - Elastomeric Foam

Thermal conductivity - 0.038915 W/m K

Outside convective heat transfer coefficient - 6, 10, 14, 18, 22 W/ m^2 K

TABLE 4.16

CRITICAL INSULATION CALCULATION BY FORMULA

Thermal conductivity (W/m K)	Heat transfer coefficient (W/m^2 K)	Critical insulation (mm) Rc = k/ h
0.038915	6	15.41
	10	25.69
	14	35.97
	18	46.25
	22	56.53

4.1.5. Outdoor Unit installation

The purpose of the outdoor unit is to exchange heat with the outdoor environment, either by expelling heat (when cooling) or absorbing heat (while heating). Heat is exchanged with the outdoor environment using a heat exchanger filled with R-410a refrigerant, which is then pumped throughout the building to one of many indoor units. Variable speed compressors are used so that lower compressor speeds can be used during part-load circumstances.

Fig.4.25. Outdoor unit installation

TABLE 4.17
OUTDOOR UNIT SPECIFICATIONS

Name	Model	Weight (kg)	Base gas(kg)	Add gas(kg)	Power Supply	CR%	Heating load(kW)
10hp	VDV-V450W/DRN1	185.00	6.2	3.32	380-415-3-50	76.6	9.35
16hp (middle)	VDV-V450W/DRN2	275.00	12.00	4.69	380-415-3-50	119.11	15.34
16hp (right)	VDV-V450W/DRN2	275.00	12.00	5.06	380-415-3-50	120.44	15.43

4.1.6. Indoor Units installation

Refrigerant from an outdoor unit is pumped to one of many indoor evaporator units, each of which is responsible for heating and cooling an individual zone in a building. Indoor evaporator units control the amount of heat being dumped to (or collected from) a space using linear or electronic expansion valves (EEVs). Indoor evaporator units exchange heat between the refrigerant and ambient air by blowing air over the unit's evaporator coil. During the heat exchange, the refrigerant either condenses (when in heating mode) or evaporates (when in cooling mode).

Fig.4.26. Indoor units installation

TABLE 4.18
INDOOR UNIT SPECIFICATIONS

IDU Name	Model	Weight(kg)	Rated Power(W)	HC (kW)
5 People meeting room	VDV-D45Q4/N1-A3	56.00	160	4.91
Machine 1 Machine 2 Work table – 1,2,3,4,5,6,11,12,13,14	VDV-D100Q4/N1-D	32.00	56	10.99
12 people meeting room 1,2	VDV-D71Q4/N1-D	26.00	82	7.36
Discussion room 1,2	VDV-D28Q4/N1-A3	16.00	50	3.00
Work table – 8,9,10,11	VDV-D140Q4/N1-D	32.00	170	14.69

4.1.7 Piping Installation

One of the toughest parts in VRF installation is piping. The steps in piping are straightening, cutting, bending and flaring which are shown in the following figure.

Fig.4.27. Piping installation

1 When counting the total piping length, the actual length of the distribution pipes which between the first branch joint and MS should be double.Max. Piping length from the first branch Joint to the farthest indoor unit: 295ft (90m) Max .Equivalent single piping length: 656ft (200m) Max. Pipinglength from MS to its downstream indoor unit: 131ft (40m) the first line branch pipe Max. Level difference between Indoor unit and outdoor unit: 361ft (110m) Max. Level difference between indoor units: 98ft (30m).

2 The equivalent length of each branch pipe is 1.64ft (0.5m).

3 The allowable piping length from the first branch joint to the farthest indoor unit should be equal to or less than 131ft (40m), but when the following conditions are all met, the allowable length can be extended to 295ft (90m).)

4 When the outdoor unit is higher than indoor units and the level difference is over 65.6ft(20m), it is recommended to set an oil return bend every 32.8ft(10m) in the gas pipe of the main pipe, the specification of the oil return bend refers to below figure. Unit: in.(mm)

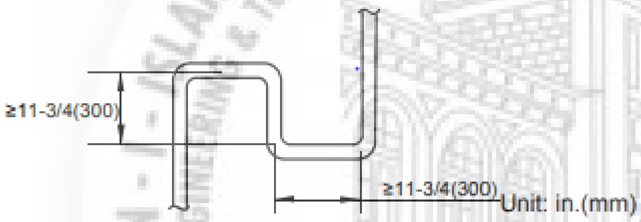


Fig.4.28. Oil return bend

5 When the outdoor unit is lower than indoor units and the level difference is more than 131ft (40m), the liquid pipe of the main pipe need to increase one size

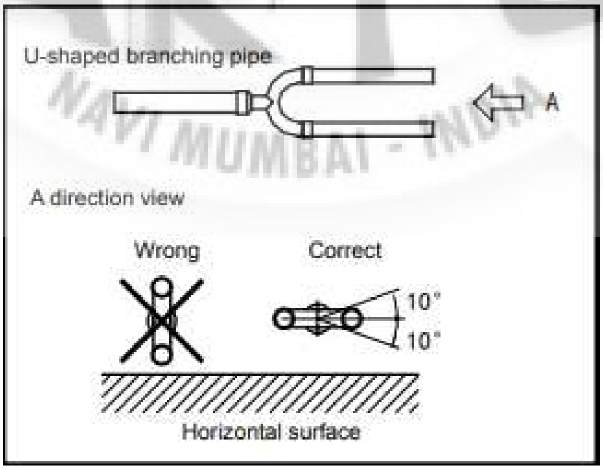


Fig.4.29. U Joint

6 Caution for brazing

Brazing is a metal-joining process in which two or more metal items are joined together by melting and flowing a filler metal into the joint, the filler metal having a lower melting point than the adjoining metal.

♣ Make sure to blow through with nitrogen when brazing. Blowing through with nitrogen prevents the creation of large quantities of oxidized film on the side of the pipe. An oxidized film adversely affects valves and compressors in the refrigerating system and prevents proper operation.

♣ The nitrogen pressure should be set to 0.02MPa (just enough so it can be felt on the skin) with a pressure-reducing valve.

♣ Do not use anti-oxidants when brazing the pipe joints. Residue can clog pipes and break equipment.

♣ Do not use flux when brazing copper-to-copper refrigerant piping. Use phosphor copper brazing filler alloy (BCuP) which does not require flux.

♣ Flux has an extremely harmfully influence on refrigerant pipe systems. For instance, if chlorine based flux is used, it will cause pipe corrosion in particular, if the flux contains fluorine, it will deteriorate the refrigerant oil.

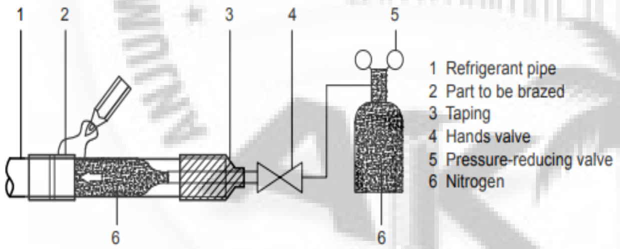


Fig.4.30. Brazing

4.1.8. Insulation Installation

Purpose of Insulation

- Prevent condensate water adhere to the gas pipe
- Protect people from hurt of high temp.
- Avoid energy loss

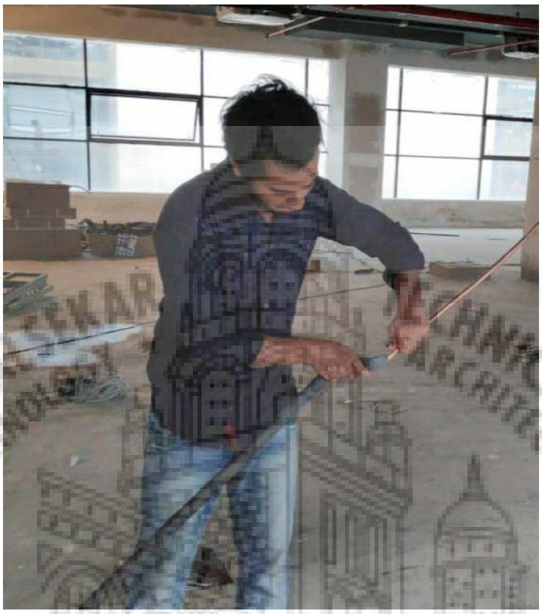


Fig.4.31. Installing insulation over gas pipe

4.1.9. Wiring

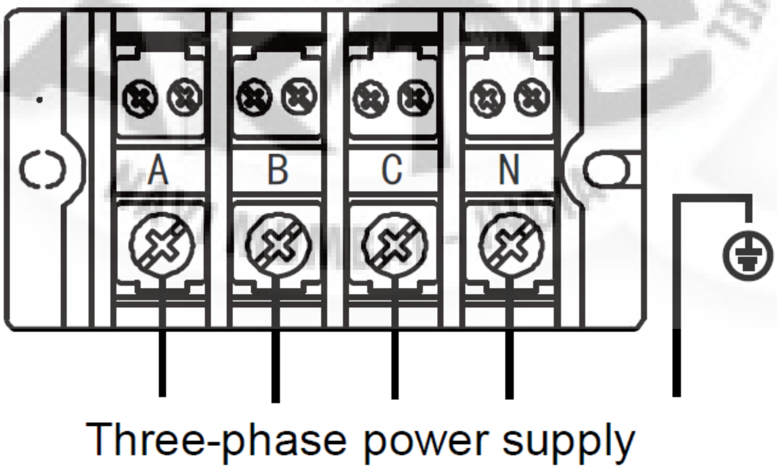


Fig.4.32. Three Phase power supply to the outdoor unit

TABLE 4.19
IMPORTANT PARAMETERS FOR OUTDOOR UNITS

Model	Units				Power supply			compressor
	HZ	Voltage (V)	Min (V)	max (V)	MCA (A)	TOCA (A)	MFA (A)	RLA (V)
10HP	50/60	380~415	342	440	20.6	22.1	25	17.4
16HP	50/60	380~415	342	440	33.4	32.8	35	27.9

A. Notes:

1 RLA (Rated Load Amp) is based on the conditions, Indoor temp. 75°F DB/ 63°F WB, Outdoor temp.97°F DB/ 71°F WB.

2 TOCA (Total Over-current Amps) means the total value of each OC set.

3 MSC (Max Starting Amps) means the Max. Current during the starting of compressor.

4 Voltage range. Units are suitable for use on electrical systems where voltage supplied to unit terminals is not below or above listed range limits.

5 Maximum allowable voltage variation between phases is 2%

6 Selection wire size based on the larger value of MCA or TOCA

7 MFA (Max. Fuse Amps) is used to select the circuit breaker and the ground fault circuit interrupter (earth circuit breaker).

MCA: Min. Circuit Amps. (A)

MSC: Max. Starting Amps. (A)

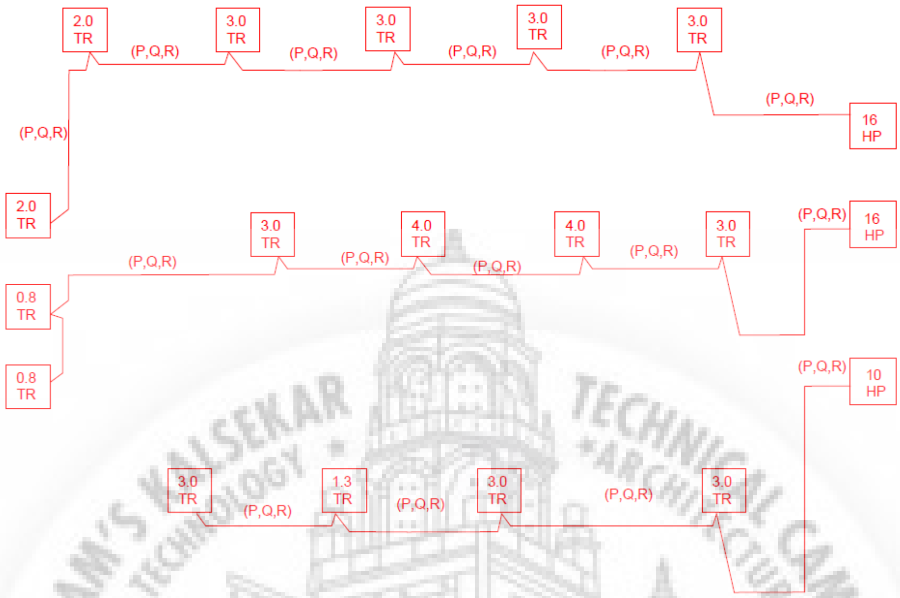
OFM: Outdoor Fan Motor.

FLA: Full Load Amps. (A)

KW: Rated Motor Output (KW)

B. Communication wire between outdoor unit and indoor unit

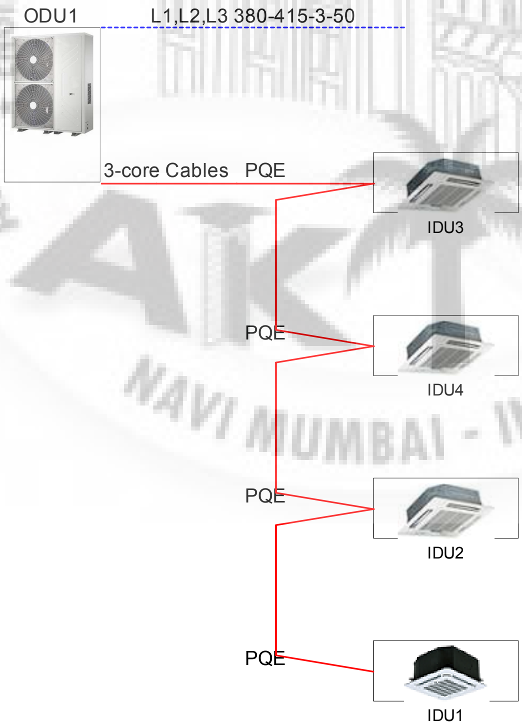
Communication wire of indoor/outdoor unit adopts 3-core shielded wire (≥0.0012in.2 (0.75mm2) which has polarity



,

Fig.4.33. Schematic Wiring diagram

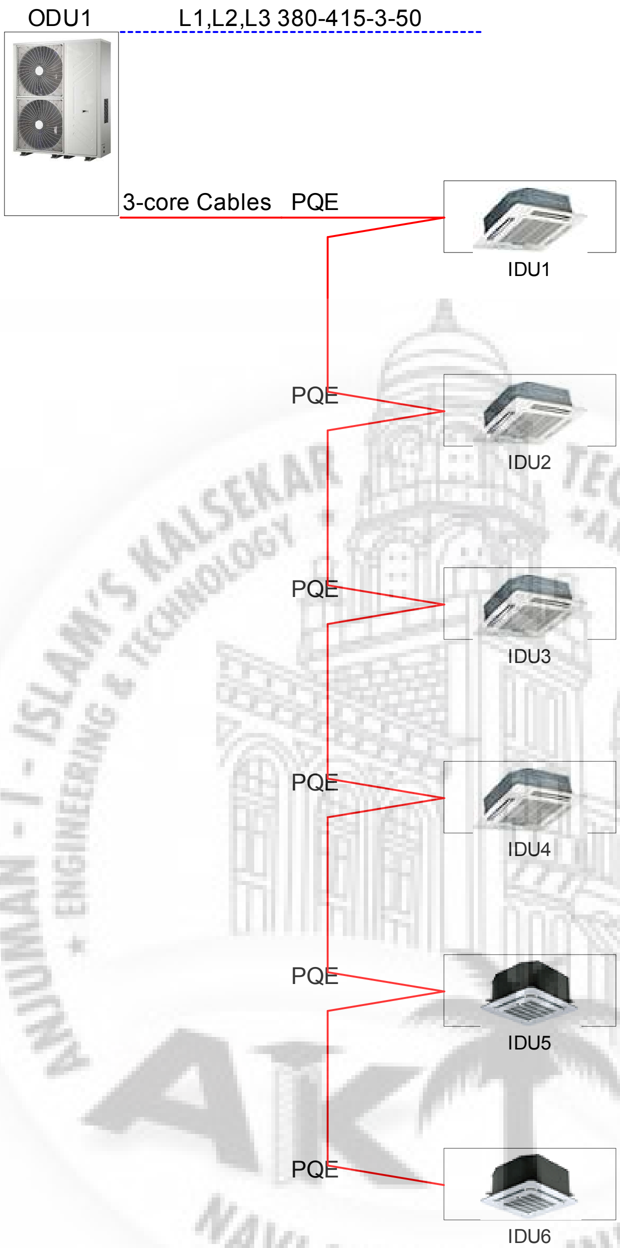
C. Results generated by software VXV selection software 2.7.7 (10 HP system) :



The wiring diagram may be different with the actual situation because of software's illustration limitation, please confirm the wiring diagram according to the installation manual before installation.

Fig.4.34. 10 HP system wiring diagram by VXV selection software 2.7.7

D. Results generated by software VXV selection software 2.7.7 (16 HP system) :



The wiring diagram may be different with the actual situation because of software's illustration limitation, please confirm the wiring diagram according to the installation manual before installation.

Fig.4.35. 16 HP system wiring diagram by VXV selection software 2.7.7

Wiring Diagram of Modular Outdoor Unit

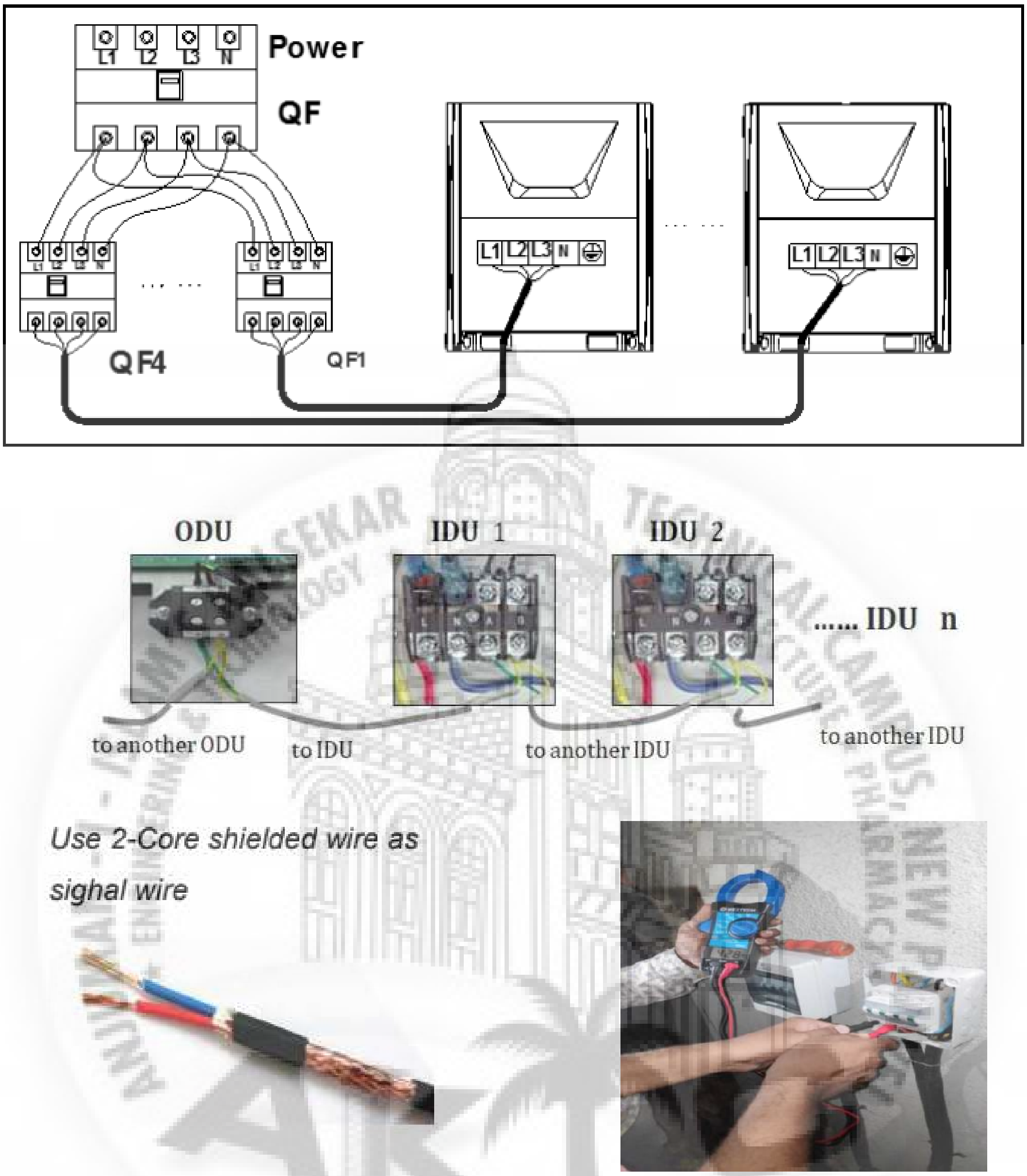


Fig.4.36. Wiring diagram of Modular outdoor unit

Fig.4.37. Wiring in Indoor and Outdoor unit

4.1.10. Testing

1. Air tight Test (Leakage test)

With this **test**, the HVAC technician first removes any remaining freon or Puron and pumps compressed **nitrogen** into the refrigeration system. The **leaks** become audible because the **nitrogen** is held at a higher pressure than the refrigerant can operate under normal conditions

Charge pressured nitrogen after connecting indoor/outdoor unit piping to do airtight test. Cautions:

1. Pressured nitrogen [4.3MPa (44kg/cm) for R410A] should be used in the airtight test.
2. Tighten high pressure/low pressure valves before applying pressured nitrogen.
3. Apply pressure from air vent mouth on the high pressure/low pressure valves.
4. The high pressure/low pressure valves are closed when applying pressured nitrogen.
5. The airtight test should never be use any oxygen, flammable gas or poisonous gas.

Fig.4.38. Air tight test (Nitrogen leakage test)

2. Vacuum Test

Air conditioning **vacuum** pump is used to remove unwanted air and water vapor from the air conditioning system when it is under service. The evacuation process is the next step in which the **vacuum** pump is used to remove the air and moisture from the refrigerant system.

Vacuum should be done from both liquid side and gas side simultaneously

Fig.4.39. Vacuum test

4.1.11. Additional Refrigerant

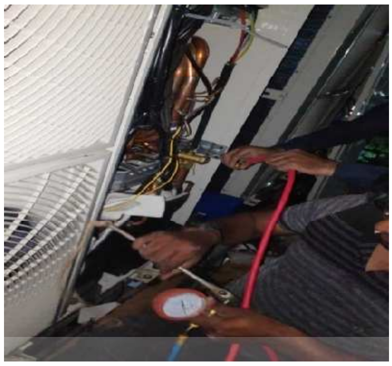
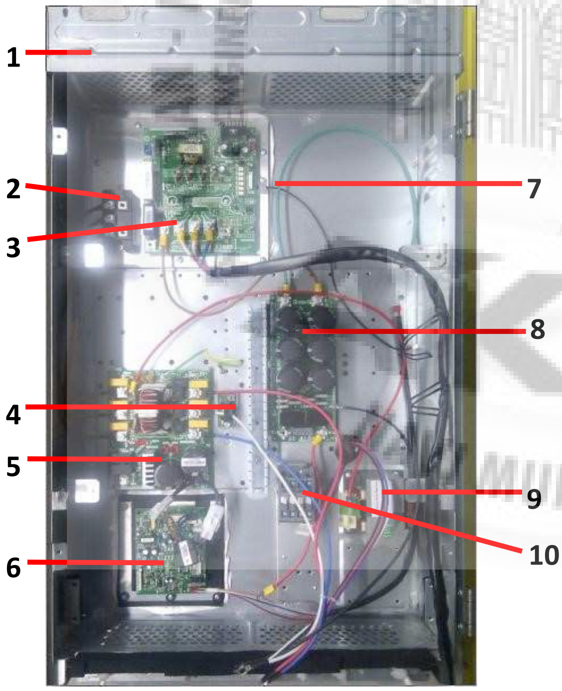


Fig.4.40. Additional refrigerant

4.1.12. Error Detection & solving

Outdoor electric control box assembly

A. **For 8/10/12HP**



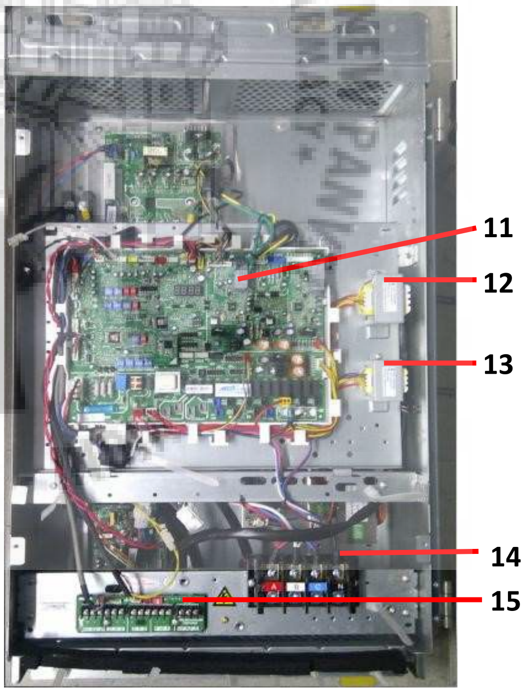


Fig.4.41. Bottom and top layer of Outdoor electric control box 8/10/12HP

No.	Content
1	Electric control box assembly
2	Three phase bridge rectifiers
3	Inverter module
4	Outdoor current detection board
5	Outdoor power supply board
6	DC fan module
7	Temperature sensor
8	Filter board
9	Reactor
10	Contactor
11	Main PCB
12	Power transformer
13	Power transformer
14	Terminal block, 4P
15	Intermediate adapter board

TABLE 4.20
ERROR CODES TABLE

Error code	Content	Note
E0	Communication error between outdoor units	Only display on the faulty slave unit
E1	Phase sequence error	Display on the faulty unit
E2	Communication error between indoor units and the master unit.	Only display on the master unit
E3	Reserved	/
E4	Malfunction of outdoor ambient temperature sensor (T4)	Display on the faulty unit
E5	Malfunction of power supply voltage	Display on the faulty unit
E6	Reserved	/
E7	Malfunction of discharge temperature sensor	Display on the faulty unit
E8	Faulty outdoor unit address	Display on the faulty unit
xE9	Driver model is mismatched	When x is 1, it means A system; 2 means B system
xH0	Malfunction of communication between main control chip and inverter driver chip	Display on the faulty unit
H1	Malfunction of communication between main control chip and communication chip	Display on the faulty unit
H2	Faulty outdoor unit quantity (decreased)	Only display on the master unit
H3	Faulty outdoor unit quantity (increased)	Only display on the master unit
xH4	P6 protection appears three times in 60 minutes	Display on the faulty unit Cannot be recovered until re-power on
H5	P2 protection appears three times in 60 minutes	Display on the faulty unit Cannot be recovered until re-power on
H6	P4 protection appears three times in 100 minutes	Display on the faulty unit Cannot be recovered until re-power on

H7	Faulty indoor units quantity	Display on the master unit Cannot be recovered until unit quantity recover
H8	Malfunction of pressure sensor for discharge pipe	The discharge pressure Pc≤0.3MPa(43.5PSI)
H9	P9 protection appears three times in 60 minutes	Display on the faulty unit Cannot be recovered until re-power on
Hc	Reserved	/
F0	PP protection appears three times in 150 minutes	Display on the faulty unit Cannot be recovered until re-power on
C7	PL protection appears three times in 100 minutes	Display on the faulty unit Cannot be recovered until re-power on
yHd	Slave units malfunction	Y stands for corresponding slave unit, y=1, 2, 3.
P0	Temperature protection of inverter compressor	Display on the faulty unit
P1	High pressure protection	Display on the faulty unit
P2	Low pressure protection	Display on the faulty unit
xP3	Over current protection of compressor	Display on the faulty unit
P4	Discharge temperature protection	Display on the faulty unit
P5	Condenser high temperature protection	Display on the faulty unit
xP6	Inverter module protection	When X is 1, it means A inverter module; 2 means B inverter module
P9	Fan module protection	Display on the faulty unit
PL	Temperature protection of inverter module	Display on the faulty unit
PP	Insufficient overheat degree protection of compressor discharge	Display on the faulty unit
xL0	Inverter module error	Display after P6 displaying for one minute
xL1	DC generatrix low voltage error	Display after P6 displaying for one minute
xL2	DC generatrix high voltage error	Display after P6 displaying for one minute
xL3	Reserved	/
xL4	MCE error/ synchronization/ closed loop	Display after P6 displaying for one minute
xL5	Zero speed protection	Display after P6 displaying for one minute
xL6	Reserved	/
xL7	Phase sequence error	Display after P6 displaying for one minute
xL8	Frequency difference in one second more than 15Hz protection	Display after P6 displaying for one minute
xL9	Frequency difference between the real and the setting frequency more than 15Hz protection	Display after P6 displaying for one minute
r1	Refrigerant amount is a little insufficient	/
r2	Refrigerant amount is obviously insufficient	/
r3	Refrigerant amount is seriously insufficient	/
R1	Refrigerant amount is a little excessive	/
`	Refrigerant amount is seriously excessive	/

- **E0: Communication error between outdoor units**

  The error only displays on faulty slave unit, all the ODUs will be standby.

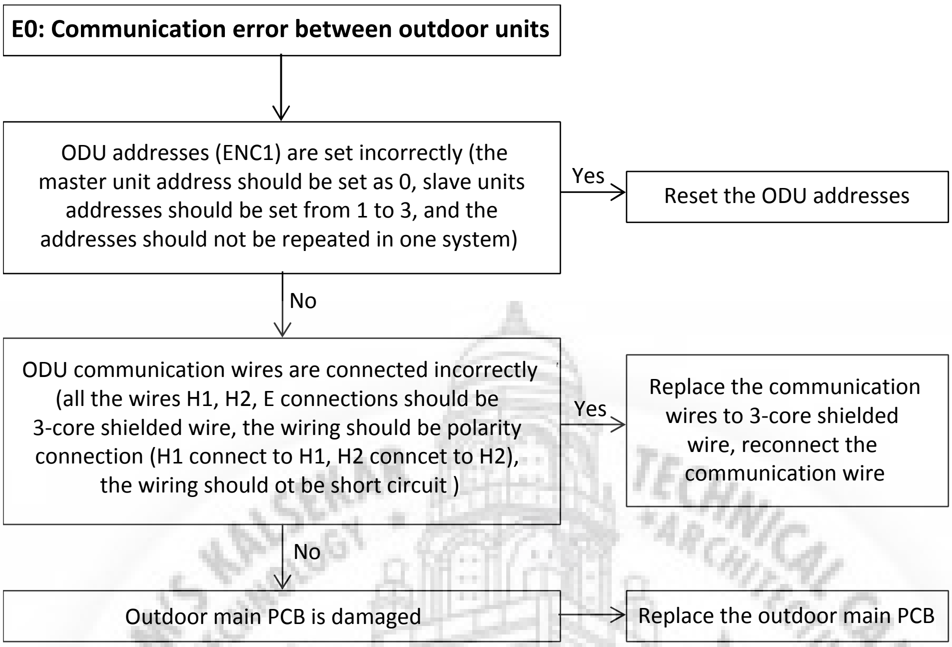


Fig.4.42. E0 error troubleshooting

Fig.4.43. 2-core shielded wire (x) and 3-core shielded wire (/)

- **E1: Phase sequence error**

  The error only displays on faulty unit, all the ODU will be standby.

Fig.4.44. E1 error showing in outdoor unit

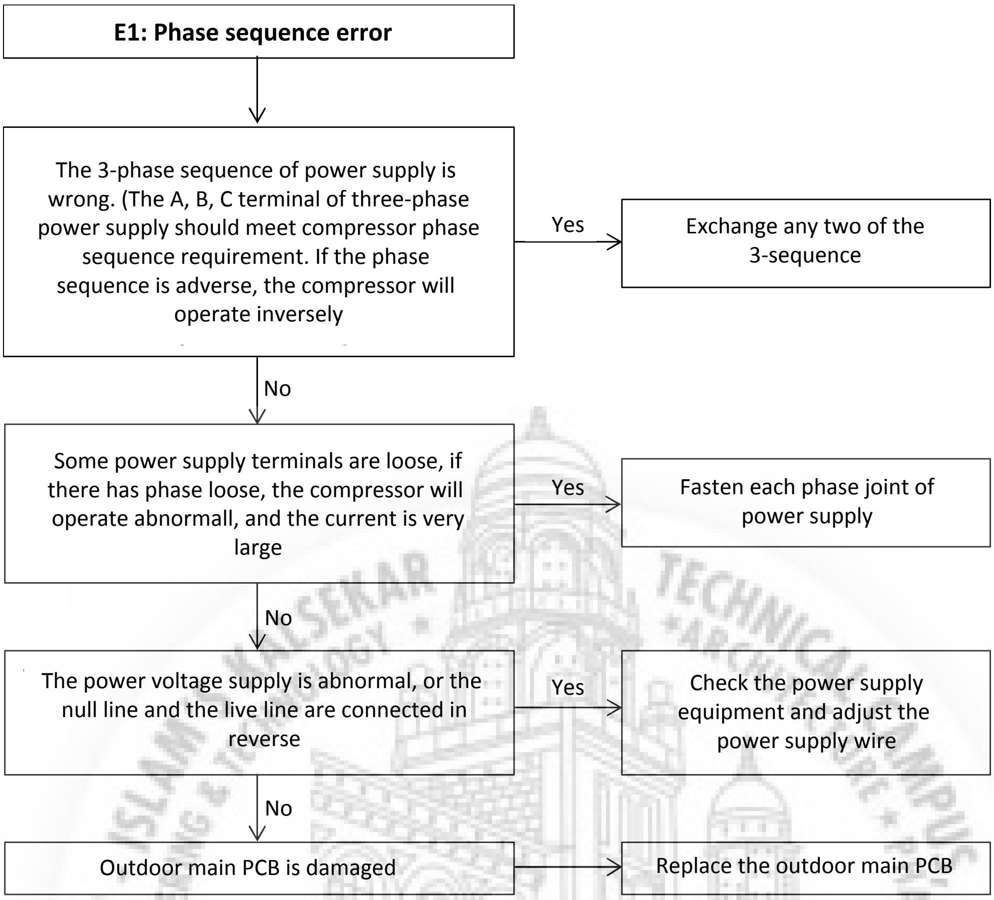


Fig.4.45. E1 error troubleshooting

If the wiring connection of each outdoor unit is according to A, B, C phase sequence, when the quantity of outdoor units is large, the current difference between C phase and A, B phase will be very large for the power supply load of each outdoor unit is on C phase, it is very easy to lead to air switch break and wiring terminal burnout. So, when the quantity of outdoor units is large, the phase sequence should be staggered, then the current can be distributed to the three phases equally

Step 1: Compressor check

Measure the resistance between each two of U, V, W terminals of the compressor, all the resistance should be the same and equal to 0.9~5 Ohms. (Fig. A and Fig. B)

Measure the resistance between each of U, V, W terminals of the compressor to ground (Fig. C), all the resistance should trend to infinity (Fig. D), otherwise the compressor has been malfunction, needs to be replaced.

Fig A

Fig B


Fig C

Fig D

Fig.4.46. Compressor check

If the resistance values are normal, then go to step 2.

Step 2: Module check

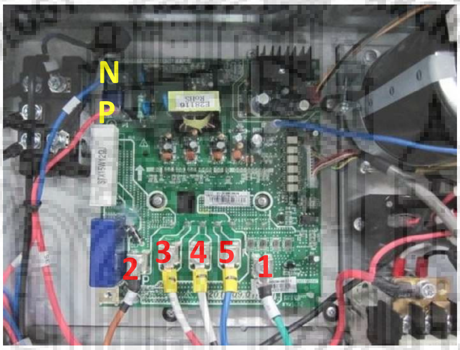


Fig.4.47. Module Check

1. DC voltage between terminal P and terminal N should be 1.41 times of the local power supply voltage.
2. DC voltage between terminal 1 and 2 should be 510V～580V.
3. Disconnect the terminal 3, 4, and 5 from inverter compressor. Measure the resistance between any two terminals among terminal 1, 2, 3, 4, 5. All the values should be infinity. If any of the value approximates to 0, the inverter module is damaged and should be replaced.

After replaced the inverter module, if the system is still abnormal, then go to step 3.


Fig.4.48. Checking for working of system

4.1.13. Commissioning

16HP
center side

VOLTAS

PRESSURE TEST REPORT

NAME OF THE PROJECT:	Tricentis
ADDRESS:	Hinjewadi
CIRCUIT NO:	1

INITIAL READINGS	DATE	TIME	AMB. TEMP. (in °C)	PRESSURE (in PSI)	REMARKS
GAS	17/8/19	7:30 PM	32°	460 PSI	
LIQUID	—	—	—	450 PSI	

FINAL READINGS	DATE	TIME	AMB. TEMP. (in °C)	PRESSURE (in PSI)	REMARKS
GAS	20/8/19	11:15 AM	36°	455 PSI	
LIQUID	—	—	—	445 PSI	

* Pressure test should be conduct by holding pressure at 580-620 PSI for copper pipe lines & 380-450 PSI for indoor units for minimum duration of 24 hrs.

Amaan Cooling Solutions
Patel
Proprietor

Customer Name:

Dealer Name:

Contact No:

Contact No: 9049817186

TRICENTIS INDIA PRIVATE

AMAAN COOLING SOLUTIONS PUNE

Customer Sign & Stamp

Customer Sign & Stamp

Fig.4.49. Pressure test of 16HP system (right)

(2)
16HP

VOLTAS

PRESSURE TEST REPORT

NAME OF THE PROJECT:	Tricon+18
ADDRESS:	Hinjewadi
CIRCUIT NO:	2

INITIAL READINGS	DATE	TIME	AMB. TEMP. (in °C)	PRESSURE (in PSI)	REMARKS
GAS	18/8/19	9:30 PM	30°	485 PSI	
LIQUID	—	—	—	480 PSI	

FINAL READINGS	DATE	TIME	AMB. TEMP. (in °C)	PRESSURE (in PSI)	REMARKS
GAS	21/8/19	10:AM	26°	480 PSI	
LIQUID	—	—	—	475 PSI	

* Pressure test should be conduct by holding pressure at 580-620 PSI for copper pipe lines & 380-450 PSI for indoor units for minimum duration of 24 hrs.

Customer Name:

Contact No:

Customer Sign & Stamp

Amaan Cooling Solutions
Proprietor

Dealer Name:

Contact No: 9049817786

AMAAN COOLING SOLUTIONS PUNE

Customer Sign & Stamp

Fig.4.50. Pressure test of 16HP system (middle)

(3)
10HP

VOLTAS

PRESSURE TEST REPORT

NAME OF THE PROJECT:	Tricentis
ADDRESS:	Hinjewadi
CIRCUIT NO:	3

INITIAL READINGS	DATE	TIME	AMB. TEMP. (in °C)	PRESSURE (in PSI)	REMARKS
GAS	17/08/19	7 PM	32°	450 PSI	
LIQUID	—	—	—	470 PSI	

FINAL READINGS	DATE	TIME	AMB. TEMP. (in °C)	PRESSURE (in PSI)	REMARKS
GAS	20/08/19	11:AM	36°	445 PSI	
LIQUID	—	—	—	463 PSI	

* Pressure test should be conduct by holding pressure at 580-620 PSI for copper pipe lines & 380-450 PSI for indoor units for minimum duration of 24 hrs.

Amaan Cooling Solutions
Proprietor

Customer Name:

Contact No:

Customer Sign & Stamp

Dealer Name:

Contact No: 9049817786

Customer Sign & Stamp

Fig.4.51. Pressure test of 10HP system (left)

COMMISSIONING REPORT

PROJECT--	M/s Tricentis India Pvt Ltd
SSP Name--	Amaan Cooling
GO-	
System No-	42 HP

No	Sequence	SYSTEM OPERATION PARAMETERS	10 HP(LEFT)
1	0	Local Capacity of ODU	8
2	1	Total capacity of Outdoor unit	22
3	2	Total Requirement of Outdoor unit capacity	1
4	3	Operating mode	4
5	4	Operting Fan Spped	5
6	5	T2 / T2B AVERAGE TEMP. OF EVP	8
7	6	T3 Pipe Temp	3
8	7	T4 environment temperature	32
9	8	Inverter exhaust temperature	50
10	9	Non-Inverter exhaust temperature	11
11	10	Heat Dissipitated surface temp	0
12	11	Electronic expansion valve temp	60
13	12	Inverter Input current	8
14	13	Non - Inverter Input current	0
15	14	Exhaust press	0
16	15	Proirity mode	4
17	16	Indoor Unit Quantity	4
18	17	Working Indoor Quantity	4
19	18	Last fault or Protection	

Remark -	Note All readings are variable as per Load & Conditions (For Internal purpose only)
Customer Name -- Stamp & Sign	Voltas Commissioning Engg. Name -- Stamp & Sign

TRICENTIS INDIA PRIVATE LIMITED

Amaan Cooling Solutions

Proprietor

AMAAN COOLING SOLUTIONS PUNE

Fig.4.52. Commissioning report 1

COMMISSIONING REPORT

PROJECT--	M/s Tricentis India Pvt Ltd			
SSP Name--	Amaan Cooling		V5p Series	
GO-				
System No-	42 HP			
No	Sequence	SYSTEM OPERATION PARAMETERS	16HP(CENTER)	16HP(RIGHT)
1	0	ODU UNIT ADDRESS	0	0
2	1	ODU UNIT ITSELF CAPACITY	18	18
3	2	Moduler outdoor unit qty.	1	1
4	3	Setting of Indoor unit qty.	6	6
5	4	Total capacity of Outdoor unit	0	0
6	5	Total Requirement of Indoor unit capacity	58	63
7	6	Tolal requirement of main unit cor. capacity	58	63
8	7	Operation Mode	2	2
9	8	THIS ODU unit actual operation CAPACITY	55	60
10	9	FAN SPEED A	7	7
11	10	FAN SPEED B	7	7
12	11	T2 / T2B AVERAGE TEMP. OF EVP	7	5
13	12	T3 Pipe Temp	31	33
14	13	T4 Ambient Temp.	30	32
15	14	DISCHARGE TEMP. OF INVERTER A	75	86
16	15	DISCHARGE TEMP. OF INVERTER B	70	87
17	16	Reserve	45	48
18	17	CURRENT OF Inverter comp A	11	11
19	18	CURRENT OF Inverter comp B	10	11
20	19	Opening Degree of EXV A	43	60
21	20	Opening Degree of EXV B	43	60
22	21	HIGH pressure (Display valueX 0.1 MPA)	20	20
23	22	T3B	41	41
24	23	Quantity of INDOOR UNITS	6	6
25	24	Qty of working indoor unit	6	6
26	25	Priority mode	4	4
27	26	Night Noise control Mode	0	0
28	27	STATIC PRESSURE MODE	0	0
29	28	DC Voltage A	55	55
30	29	DC Voltage B	53	53
31	30	Reserve	57	57
32	31	Reserve	88	88
33	32			

Remark -

Note
All readings are variable as per Load & Conditions
(For Internal purpose only)

Customer
Name --
Stamp & Sign

TRICENTIS INDIA PRIVATE LIMITED

Voltas Commissioning Engg.
Name -- Sharad Mandal
Stamp & Sign

Amaan Cooling Solutions
Patel
Proprietor
AMAAN COOLING SOLUTIONS PUNE

Fig.4.53. Commissioning report 2

VRF Commissioning Report	Operation data of Unit	
	Cust Name-	M/s Tricentis India Pvt Ltd
	SSP Name-	Amaan Cooling
	GO No-	
V5 P	Project Eng.	Mr.Bharat Mandal
ODU Model No	ODU Serial No.	
1 VDV-280W/DRN1	SR.3407717820893260100006	
2 VDV-V450W/DRN1	SR.3407342451191190100014	
3 VDV-V450W/DRN1	SR.3407342451191190100011	

Operation data of outdoor unit (Cooling)	Readings are veriable as Load & condition		
Unit-42 HP	10 HP(LEFT)	16HP(CENTER)	16HP(RIGHT)
Run Voltage V	RY-410 / YB-413 / RB-408, E/N - 0 Volt	RY-410 / YB-413 / RB-408, E/N - 0 Volt	RY-410 / YB-413 / RB-408, E/N - 0 Volt
Total current of run A	R-12.5 / Y-10.5 / B-13.9	R-25 / Y-17.9 / B-23.7	R-20.6 / Y-21.8/ B-26.6
High-pressure pressure PSI	400 PSI		
Low-pressure pressure PSI	120 PSI		
Inlet air temperature °C	31 °C		
Outlet air temperature °C	39 °C		

Operation data of outdoor unit(Cooling) — All readings are variable as per Load & Conditions (For Internal purpose only)

IDU No.	Position	Model	Sr No of Indoor unit	Set Temp	Room Temp
1	10 HP(LEFT)	VDV-D112Q4/N1-D	C7032044231116A14400066	22°C	22°C
2		VDV-D112Q4/N1-D	C7032044231116A14400061	22°C	22°C
3		VDV-D71Q4/N1-D	C7032044231116A14400085	22°C	22°C
4		VDV-D71Q4/N1-D	C7032044231116A14400009	22°C	22°C
5		VDV-D112Q4/N1-D	C7032044231116A14400022	22°C	22°C
6		VDV-D112Q4/N1-D	C7032044231116A14400048	22°C	22°C
7					
8	16HP(CENTER)	VDV-D28Q4/N1-A3	SR 3407342631291120100141	22°C	22°C
9		VDV-D28Q4/N1-A3	SR 3407342631291120100062	22°C	22°C
10		VDV-D112Q4/N1-D	SR 3407342640591140100055	22°C	22°C
11		VDV-D140Q4/N1-D	SR 3407342640591140100063	22°C	22°C
12		VDV-D112Q4/N1-D	SR 340734260491150100056	22°C	22°C
13		VDV-D140Q4/N1-D	SR 3407342840691140100065	22°C	22°C
14					
15	16HP(RIGHT)	VDV-D112Q4/N1-D	SR 3407342640491180100038	22°C	22°C
16		VDV-D45Q4/N1-A3	SR 3407342631491130100114	22°C	22°C
17		VDV-D112Q4/N1-D	SR 3407342640491180100032	22°C	22°C
18		VDV-D112Q4/N1-D	SR 3407342640491150100052	22°C	22°C

Customer
Name --
Stamp & Sign

Voltas Commissioning Engg.
Name -- Bharat Mandal
Stamp & Sign

Tricentis India Private Limited
Amaan Cooling Solutions Pune
Amaan Cooling Solutions
Proprietor

Fig.4.54. Commissioning report 3

4.2. Project 2: MSD office, Andheri

In this project we have designed the VRF system for the MSD office located in Andheri, Mumbai. The total air-conditioned space is 772 sq. Ft. The purpose is to achieve 24^{o}C DBT and 17^{o}C WBT from outside design conditions of 36^{o}C DBT and 22^{o}C WBT using standard data books. The article describes the design of VRF system consisting of 5 Indoor units

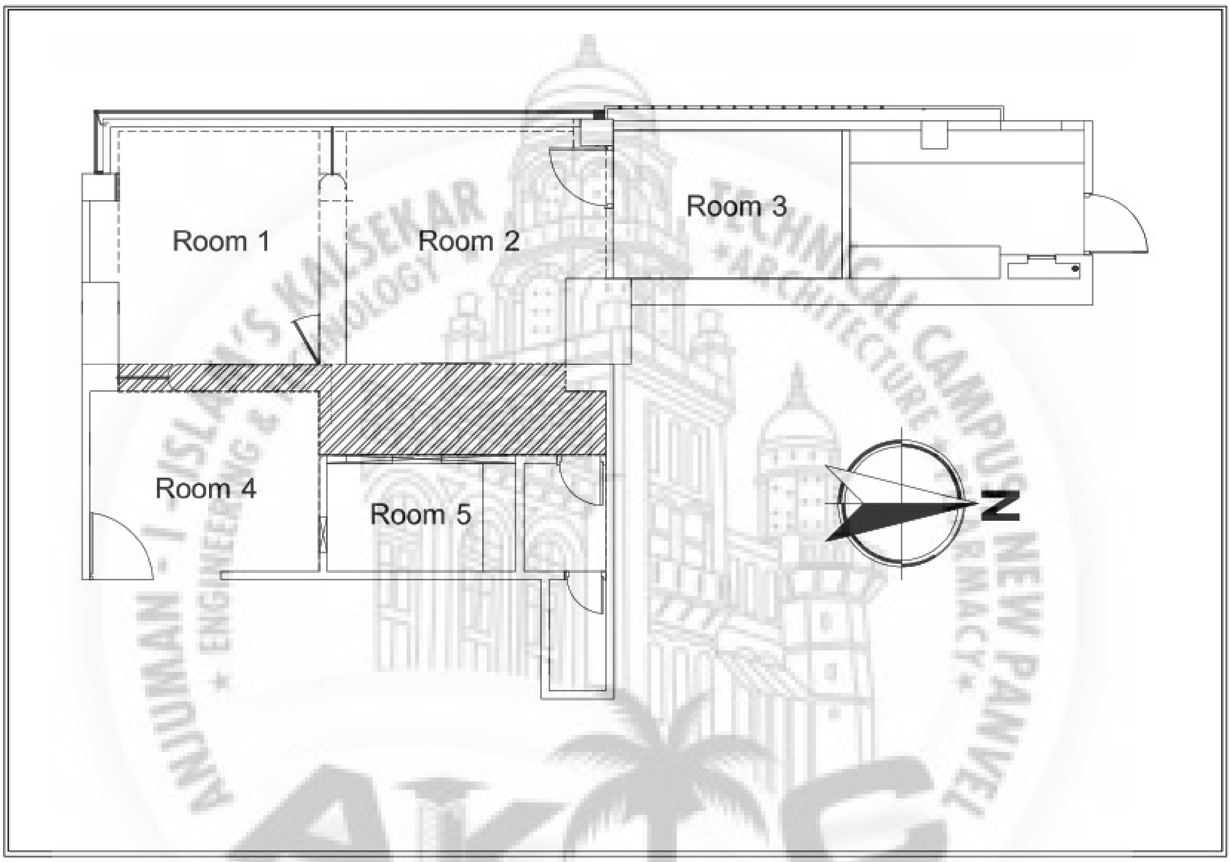


Fig.4.55. MSD office floor layout

4.2.1. Calculations

All the calculation which are required in designing VRF system are described in the previous section. Same method is used for the current project also. Cooling load calculation sheets are given below

HEAT LOAD DATA SHEET

ROOM 1

CUSTOMER	:	MSD OFFICE	
ESTIMATED BY	:	Ganesh Satve	
DATE	:	16/01/2020	
AREA	=	211	SQFT
TOTAL HT	=	10.0	FEET

DESIGN CONDITIONS				
TIME OF PEAK LOAD:		4:00 PM		
	DB	% RH	WB	GR/LB
OUTSIDE	90	39	71	83.3
INSIDE	75	55	64	72.1
DIFF.	15	XXX	XXX	11.2

PEOPLE :	10	NOS. X	5	CFM =	50
8	AIR CHANGES PER HOUR			=	281
FRESH AIR				=	281

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR	EFFECTIVE ROOM LATENT HEAT			
EFFECTIVE ROOM SENSIBLE HEAT					F. A.	281	7.62	2143
WALL N		0.33	12	0	PEOPLE	10	205	2050
WALL NE		0.33	14	0	RLH SUBTOTAL			4193
WALL E		0.33	14	0	FACTOR		5%	210
WALL SE		0.33	20	0	ERLH TOTAL			4402
WALL S		0.33	28	0	EFFECTIVE ROOM TOTAL HEAT			22165
WALL SW		0.33	34	0	GRAND TOTAL HEAT SUBTOTAL			22165
WALL W	43	0.33	28	397	FACTOR		3%	665
WALL NW		0.33	14	0	GRAND TOTAL HEAT			22830
GLASS N		0.62	22	0	AIRCONDITIONING TONNAGE			1.90
GLASS NE		0.62	22	0				
GLASS E		0.62	22	0	ERSHF			0.80
GLASS SE		0.62	22	0	INDICATED ADP			0.0
GLASS S		0.62	22	0	SELECTED ADP			0.0
GLASS SW		0.62	22	0	DEHUMIDIFIER CFM			0
GLASS W	101	0.62	22	1378				
GLASS NW		0.62	22	0	RETURN AIR GAIN			
ALL GLASS		1.13	10	0	STRUCT.	SQFT	FACTOR	BTU/HR
PRT. WL.	406	0.40	10	1624	WALL N	0	0.33	0
PRT. GLS.		1.13	10	0	WALL NE	0	0.33	0
DOOR	31	1.04	10	322	WALL E	0	0.33	0
ROOF		0.15	22	0	WALL SE	0	0.33	0
CEILING	211	0.27	10	570	WALL S	0	0.33	0
FLOOR	211	0.40	10	844	WALL SW	0	0.33	0
F. A.	281	1.08	22	6684	WALL W	0	0.33	0
PEOPLE	10	Nos.	245	2450	WALL NW	0	0.33	0
LIGHT (w)	211	1.11	3.41	799	PRT. WL.	0	0.40	0
EQPT. (kw)	0.3	1	3412.14	1080	3) TOTAL R.A. GAIN			0
OTHER (kw)		1	3412.14	0				
RSH SUBTOTAL				16148				
FACTOR			10%	1615				
ERSH TOTAL				17763				

Fig.4.56. Cooling load calculation of Room 1

HEAT LOAD DATA SHEET								
ROOM 2								
CUSTOMER :	MSD OFFICE			DESIGN CONDITIONS				
ESTIMATED BY :	Ganesh Satve			TIME OF PEAK LOAD:		4:00 PM		
DATE :	16/01/2020				DB	% RH	WB	GR/LB
AREA	=	175	SQFT	OUTSIDE	90	39	71	83.3
TOTAL HT	=	10.0	FEET	INSIDE	75	55	64	72.1
				DIFF.	15	XXX	XXX	11.2

PEOPLE :	10	NOS. X	5	CFM	=	50
8	AIR CHANGES PER HOUR				=	233
FRESH AIR					=	233

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N		0.33	12	0
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S		0.33	28	0
WALL SW		0.33	34	0
WALL W	50	0.33	28	462
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S		0.62	22	0
GLASS SW		0.62	22	0
GLASS W	117	0.62	22	1596
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.	307	0.40	10	1228
PRT. GLS.		1.13	10	0
DOOR	69	1.04	10	718
ROOF		0.15	22	0
CEILING	175	0.27	10	473
FLOOR	175	0.40	10	700
F. A.	233	1.08	22	5544
PEOPLE	10	Nos.	245	2450
LIGHT (w)	175	1.11	3.41	662
EQPT. (kw)	0.3	1	3412.14	896
OTHER (kw)		1	3412.14	0
RSH SUBTOTAL				14728
FACTOR			10%	1473
ERSH TOTAL				16201

EFFECTIVE ROOM LATENT HEAT			
F. A.	233	7.62	1777
PEOPLE	10	205	2050
RLH SUBTOTAL			3827
FACTOR		5%	191
ERLH TOTAL			4018
EFFECTIVE ROOM TOTAL HEAT			20219
GRAND TOTAL HEAT SUBTOTAL			20219
FACTOR		3%	607
GRAND TOTAL HEAT			20826
AIRCONDITIONING TONNAGE			1.74
ERSHF			0.80
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			0

RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.57. Cooling load calculation of Room 2

HEAT LOAD DATA SHEET				
ROOM 3				

CUSTOMER	:	MSD OFFICE		DESIGN CONDITIONS					
ESTIMATED BY	:	Laxman Mandal		TIME OF PEAK LOAD:			4:00 PM		
DATE	:	16/01/2020			DB	% RH	WB	GR/LB	
AREA	=	151	SQFT	OUTSIDE	90	39	71	83.3	
TOTAL HT	=	10.0	FEET	INSIDE	75	55	64	72.1	
				DIFF.	15	XXX	XXX	11.2	

PEOPLE :	10	NOS. X	5	CFM =	50
8	AIR CHANGES PER HOUR			=	201
FRESH AIR				=	201

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N		0.33	12	0
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S		0.33	28	0
WALL SW		0.33	34	0
WALL W	44	0.33	28	407
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S		0.62	22	0
GLASS SW		0.62	22	0
GLASS W	102	0.62	22	1391
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.	315	0.40	10	1260
PRT. GLS.		1.13	10	0
DOOR	38	1.04	10	395
ROOF		0.15	22	0
CEILING	151	0.27	10	408
FLOOR	151	0.40	10	604
F. A.	201	1.08	22	4784
PEOPLE	10	Nos.	245	2450
LIGHT (w)	151	1.11	3.41	572
EQPT. (kw)	0.2	1	3412.14	773
OTHER (kw)		3.4	3412.14	0
RSH SUBTOTAL				13043
FACTOR			10%	1304
ERSH TOTAL				14347

EFFECTIVE ROOM LATENT HEAT			
F. A.	201	7.62	1533
PEOPLE	10	205	2050
RLH SUBTOTAL			3583
FACTOR		5%	179
ERLH TOTAL			3763
EFFECTIVE ROOM TOTAL HEAT			18110
GRAND TOTAL HEAT SUBTOTAL			18110
FACTOR		3%	543
GRAND TOTAL HEAT			18653
AIRCONDITIONING TONNAGE			1.55
ERSHF			0.79
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			0

RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.58. Cooling load calculation of Room 3

HEAT LOAD DATA SHEET								
ROOM 4								
CUSTOMER	:	MSD OFFICE		DESIGN CONDITIONS				
ESTIMATED BY	:	Shadab Khan		TIME OF PEAK LOAD:		4:00 PM		
DATE	:	16/01/2020			DB	% RH	WB	GR/LB
AREA	=	157	SQFT	OUTSIDE	90	39	71	83.3
TOTAL HT	=	10.0	FEET	INSIDE	75	55	64	72.1
				DIFF.	15	XXX	XXX	11.2

PEOPLE :	10	NOS. X	5	CFM =	50
8	AIR CHANGES PER HOUR			=	209
FRESH AIR				=	209

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR	EFFECTIVE ROOM LATENT HEAT			
EFFECTIVE ROOM SENSIBLE HEAT					F. A.	209	7.62	1594
WALL N		0.33	12	0	PEOPLE	10	205	2050
WALL NE		0.33	14	0	RLH SUBTOTAL			3644
WALL E		0.33	14	0	FACTOR		5%	182
WALL SE		0.33	20	0	ERLH TOTAL			3826
WALL S		0.33	28	0	EFFECTIVE ROOM TOTAL HEAT			17467
WALL SW		0.33	34	0	GRAND TOTAL HEAT SUBTOTAL			17467
WALL W		0.33	28	0	FACTOR		3%	524
WALL NW		0.33	14	0	GRAND TOTAL HEAT			17991
GLASS N		0.62	22	0	AIRCONDITIONING TONNAGE			1.50
GLASS NE		0.62	22	0				
GLASS E		0.62	22	0	ERSHF			0.78
GLASS SE		0.62	22	0	INDICATED ADP			0.0
GLASS S		0.62	22	0	SELECTED ADP			0.0
GLASS SW		0.62	22	0	DEHUMIDIFIER CFM			0
GLASS W		0.62	22	0				
GLASS NW		0.62	22	0	RETURN AIR GAIN			
ALL GLASS		1.13	10	0	STRUCT.	SQFT	FACTOR	BTU/HR
PRT. WL.	466	0.40	10	1864	WALL N	0	0.33	0
PRT. GLS.		1.13	10	0	WALL NE	0	0.33	0
DOOR	38	1.04	10	395	WALL E	0	0.33	0
ROOF		0.15	22	0	WALL SE	0	0.33	0
CEILING	157	0.27	10	424	WALL S	0	0.33	0
FLOOR	157	0.40	10	628	WALL SW	0	0.33	0
F. A.	209	1.08	22	4974	WALL W	0	0.33	0
PEOPLE	10	Nos.	245	2450	WALL NW	0	0.33	0
LIGHT (w)	157	1.11	3.41	594	PRT. WL.	0	0.40	0
EQPT. (kw)	0.3	1	3412.14	1071	3) TOTAL R.A. GAIN			0
OTHER (kw)		5	3412.14	0				
RSH SUBTOTAL				12401				
FACTOR			10%	1240				
ERSH TOTAL				13641				

Fig.4.59. Cooling load calculation of Room 4

HEAT LOAD DATA SHEET

ROOM 5

CUSTOMER :	MSD OFFICE		
ESTIMATED BY :	Eesa Rakhange		
DATE :	16/01/2020		
AREA =	78	SQFT	
TOTAL HT =	10.0	FEET	

DESIGN CONDITIONS				
TIME OF PEAK LOAD:	4:00 PM			
	DB	% RH	WB	GR/LB
OUTSIDE	90	39	71	83.3
INSIDE	75	55	64	72.1
DIFF.	15	XXX	XXX	11.2

PEOPLE :	10	NOS. X	5	CFM =	50
8	AIR CHANGES PER HOUR			=	104
FRESH AIR				=	104

STRUCT.	SQFT	FACTOR	TEMP.	BTU/HR
EFFECTIVE ROOM SENSIBLE HEAT				
WALL N		0.33	12	0
WALL NE		0.33	14	0
WALL E		0.33	14	0
WALL SE		0.33	20	0
WALL S		0.33	28	0
WALL SW		0.33	34	0
WALL W		0.33	28	0
WALL NW		0.33	14	0
GLASS N		0.62	22	0
GLASS NE		0.62	22	0
GLASS E		0.62	22	0
GLASS SE		0.62	22	0
GLASS S		0.62	22	0
GLASS SW		0.62	22	0
GLASS W		0.62	22	0
GLASS NW		0.62	22	0
ALL GLASS		1.13	10	0
PRT. WL.	335	0.40	10	1340
PRT. GLS.		1.13	10	0
DOOR	31	1.04	10	322
ROOF		0.15	22	0
CEILING	78	0.27	10	211
FLOOR	78	0.40	10	312
F. A.	104	1.08	22	2471
PEOPLE	10	Nos.	245	2450
LIGHT (w)	78	1.11	3.41	295
EQPT. (kw)	0.2	1	3412.14	532
OTHER (kw)	2	1	3412.14	6824
RSH SUBTOTAL				14758
FACTOR			10%	1476
ERSH TOTAL				16234

EFFECTIVE ROOM LATENT HEAT			
F. A.	104	7.62	792
PEOPLE	10	205	2050
RLH SUBTOTAL			2842
FACTOR		5%	142
ERLH TOTAL			2984
EFFECTIVE ROOM TOTAL HEAT			19218
GRAND TOTAL HEAT SUBTOTAL			19218
FACTOR		3%	577
GRAND TOTAL HEAT			19794
AIRCONDITIONING TONNAGE			1.65
ERSHF			0.84
INDICATED ADP			0.0
SELECTED ADP			0.0
DEHUMIDIFIER CFM			0

RETURN AIR GAIN			
STRUCT.	SQFT	FACTOR	BTU/HR
WALL N	0	0.33	0
WALL NE	0	0.33	0
WALL E	0	0.33	0
WALL SE	0	0.33	0
WALL S	0	0.33	0
WALL SW	0	0.33	0
WALL W	0	0.33	0
WALL NW	0	0.33	0
PRT. WL.	0	0.40	0
3) TOTAL R.A. GAIN			0

Fig.4.60. Cooling load calculation of Room 5

Report generated by HeatCAD 2019 Software

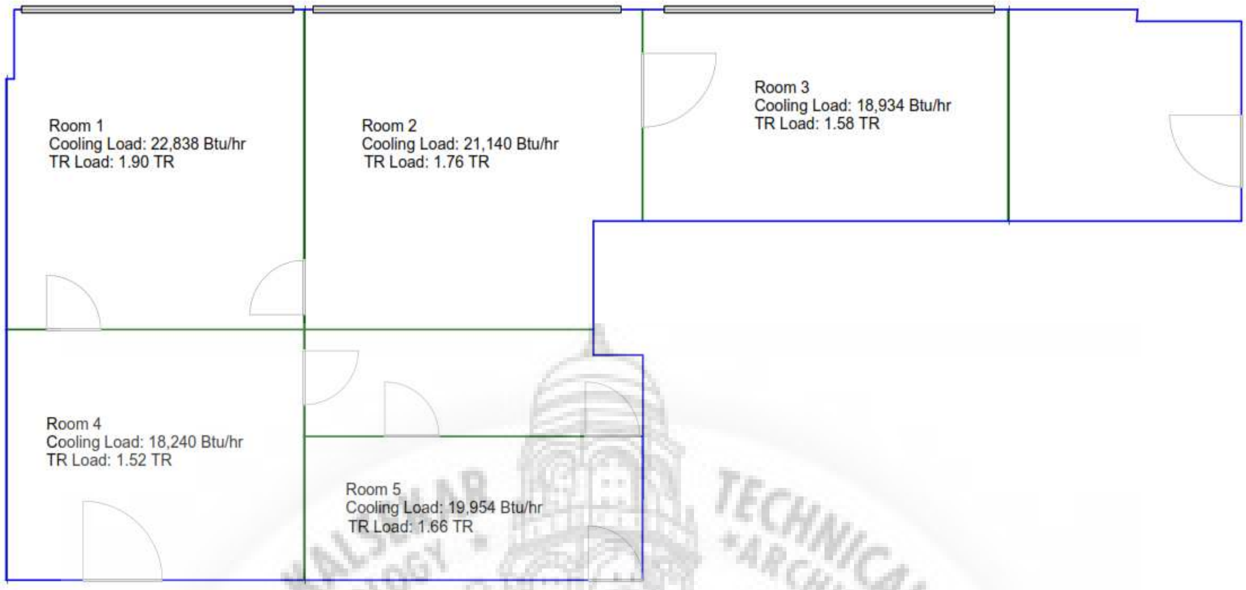


Fig.4.61. Cooling load report by HeatCAD 2019 (MSD office)

TR Selection for the Office Layout:

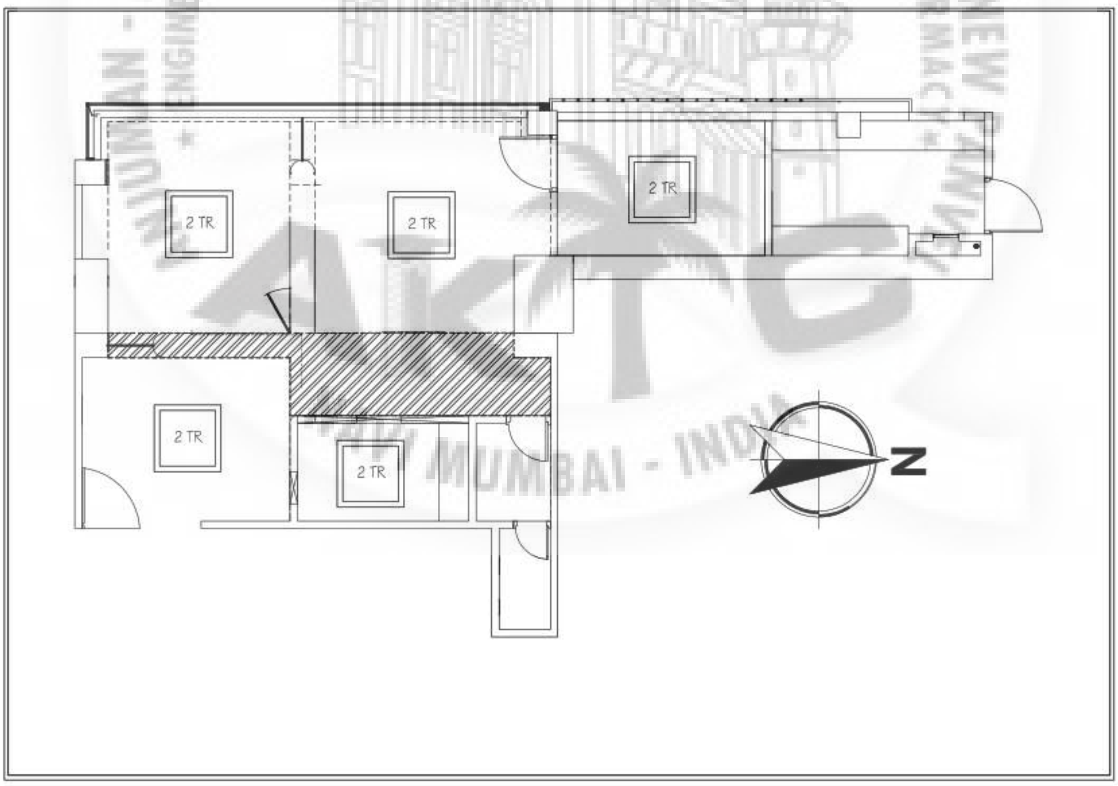


Fig.4.62. TR selection (MSD office)

4.2.2. Installations

Installations of the Components required are same as discussed in above sections which include

- Outdoor unit (Compressor)
- Indoor units (Evaporators)
- Piping
- Wirings

4.2.3. Error solving

Error detection and solving is done as per the standards shown in previous sections. By referring these standards the errors are detected and solved accordingly in this project

4.2.4. Testing & Commissioning

Testing and commissioning is done for the validation of the system. The detailed process is described in the previous section.

4.3. Project 3: Suggestion for Implementing PVC material for ventilation ducts over conventional GI ducts

This project is the suggestion for the Tricentis offie in Pune. In VRF systems Ventilation ducting is used for fresh air supply. Generally these ducts are made of Galvanized Iron. This article describes the implementation of PVC as a material for ventilation ducts instead of common GI ducts. Duct designing and calculations are done with the help of Computer based software applications and standard data. PVC ducting is efficient, low power and Cost effective alternative for the conventional ducting systems.

A Variable Refrigerant Flow System is silent, precise and highly energy efficient HVAC technology, best for individualized comfort control. But the system has no provision for outside air supply to the air conditioned space. The ventilation air is delivered to the occupied space through a duct. G.I ducts are most commonly used for the ventilation purpose and this paper suggests PVC for duct material as it has many advantages over G.I ducts. PVC is light in weight and smooth than G.I. Its absolute roughness is only 10% of roughness of galvanized iron. Thus it has less pressure drop. A PVC duct has around 33% less pressure drop per 100ft of duct length. Therefore the power requirement for the fan unit is also low. Therefore the PVC ducts become very cost effective and energy efficient solution over conventional ducting.

4.3.1. Designing of ducts

Designing of ducts includes different shapes of ducts as rectangular ducts, circular ducts, oval ducts. In this paper design of ducts is obtained by circular ducts by using PVC. For designing of ducts different methods are used as flows:

1. Equal friction method.
2. Velocity reduction method.
3. Static regain method.

In these methods the calculations are either done manually using standard friction charts or by using Duct Sizer application as discussed below:

A. Using Friction chart:

Horizontal lines describes the flow rate or air quantity in (CFM) cubic feet per minute

Vertical lines depicts the head loss in (in WC/100 ft) inches of water column per 100 feet.

Inclined lines show the velocity in (fpm) feet per minute and equivalent diameter in inches.

It can be seen easily in figure below.

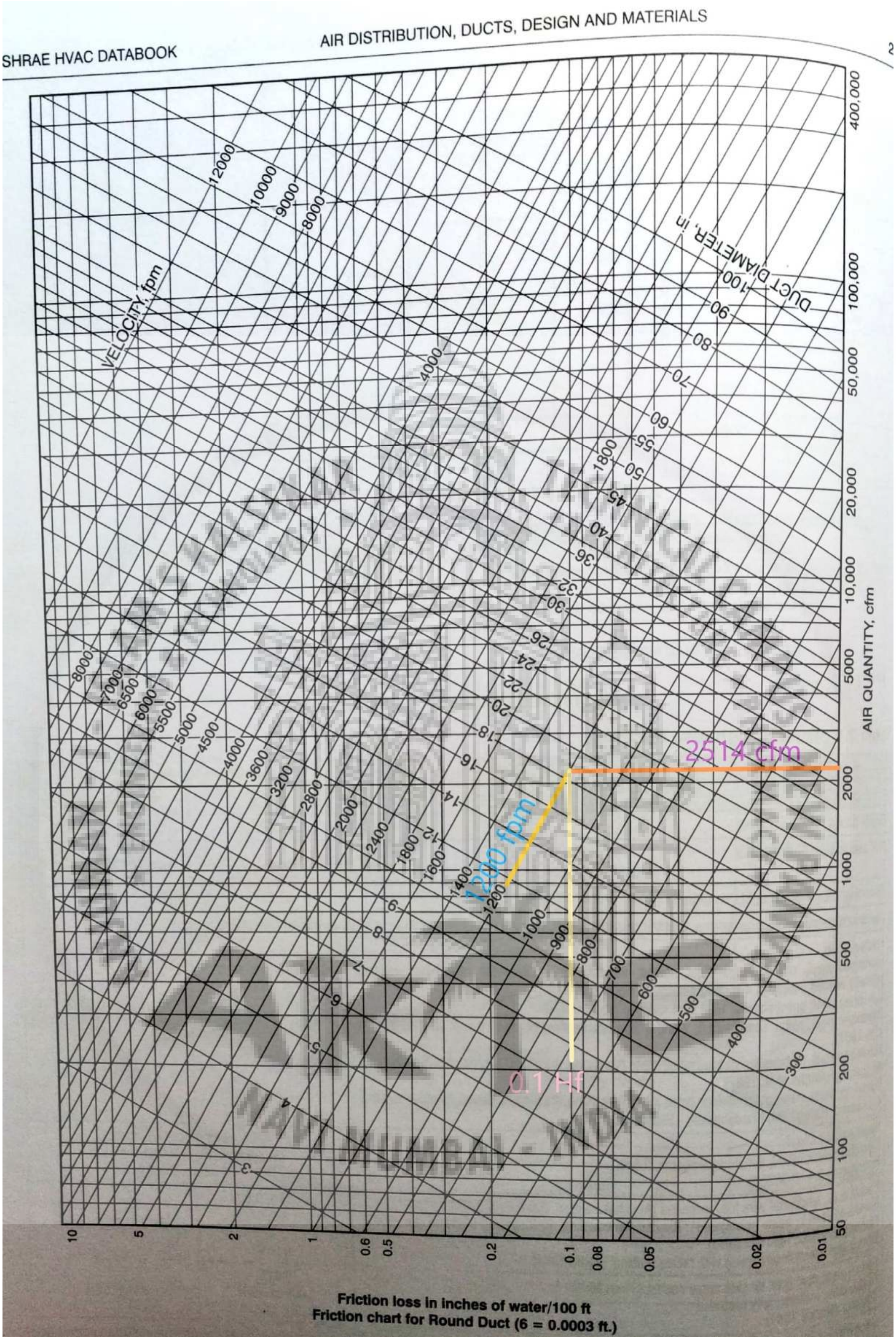


Fig.4.63. Friction Chart

Figure 4.63 shows the friction chart which evaluates the value of head loss as 0.1 in WC/100 ft by plotting the value of air quantity is equal to 2514 CFM as total flow rate and velocity equal to 1200fpm for office building.

B. Using Duct Sizer application:

To overcome some difficulties while plotting friction chart and for time saving, for removing error and for increasing accuracy, Duct Sizer application is used. It is simple to use as compared to friction chart

Duct Sizer application work on algorithm which gives the approximate value as we can get from the friction chart

Different parameters are required for designing by using duct sizer are as follows:

- Flow rate in cubic feet per minute (CFM): Total flow rate is equal to 2514 CFM
- For private offices duct velocity is equal 1200 feet per minute (fpm)
- Equivalent Diameter of ducts in inches

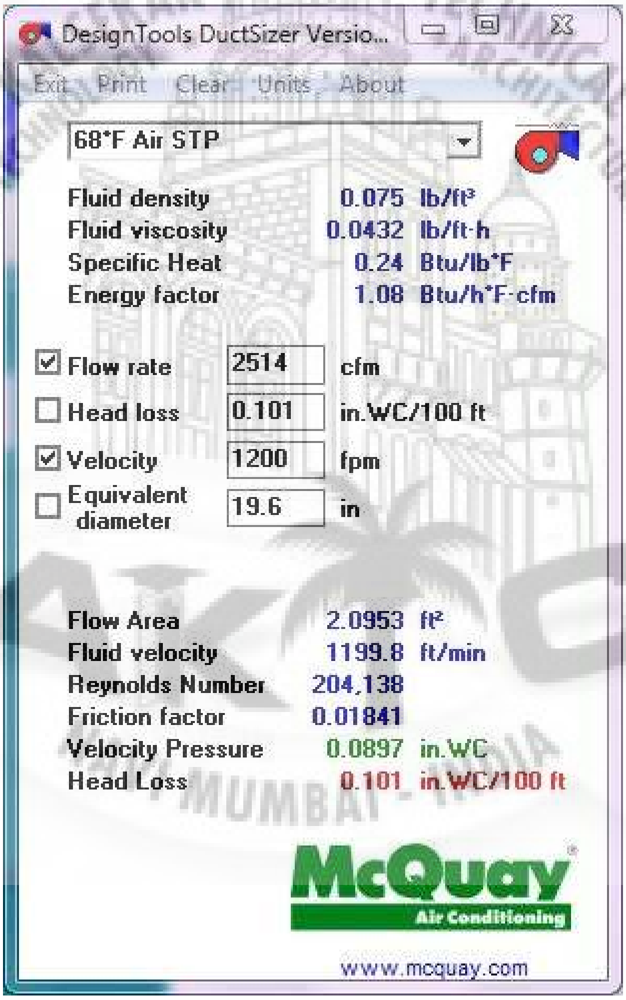


Fig.4.64. Calculation of Head loss and eq. Dia by using McQuay DuctSizer 6.4

From figure 4.64, By using total flow rate equal to 2514 CFM and velocity equal to 1200 fpm and we got head loss as 0.101 inch of water per 100 feet (in WC/100 ft)

It can be determined by using head loss as 0.101 in WC/100ft as constant value got from previous evaluation and 15% of CFM corresponding to each indoor unit.

For example,

- 2TR requires 800 CFM of air supply
- Therefore, 15% of 800cfm is equal to 120cfm which is required for ducting respectively.

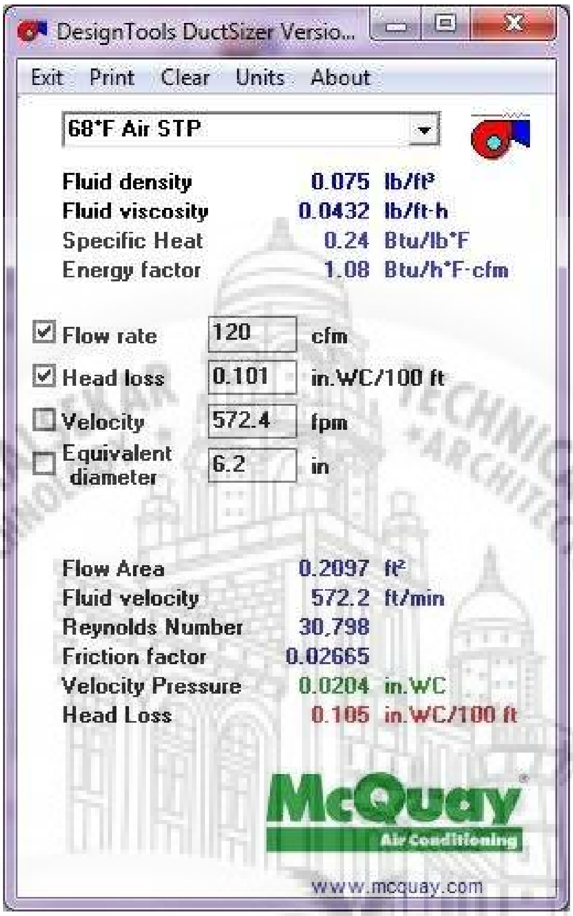


Fig.4.65. Calculation of eq. Dia with help of previous head loss value

From figure 4.65, we got equivalent diameter of specific duct equal to 6.2 inch and the method used as equal friction method and same is used to find further ducts diameters.

Similarly remaining ducts diameters are shown in the below diagram.

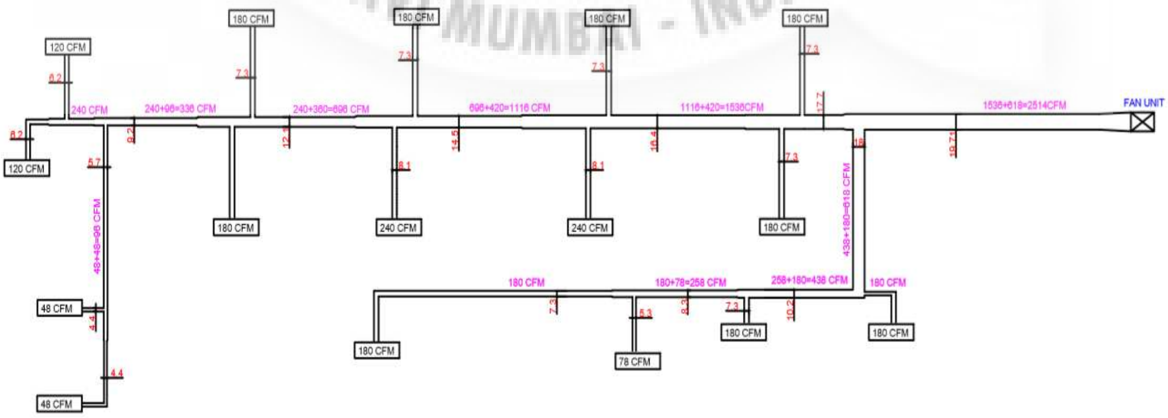


Fig.4.66. Duct layout (Diameters)

4.3.2. Pressure calculations and thickness selection

In duct designing Pressure calculation refers to calculation of duct internal resistance to the flow of air. That is the pressure required for efficient flow of air through the duct.

There are mainly three types of pressure losses calculated while designing the ducts

1. Static pressure: It is resistance provided by the duct walls for the flow of air, which needs to overcome to make the flow possible.

2. Velocity pressure: It is the pressure developed by the flowing fluid. For calculation purpose it can be referred as pressure required to flow the air with required velocity.

3. Total pressure: It is the Sum of static and velocity pressure. This is the total pressure generated by the air when it flows through the duct.

For a particular path the pressure is estimated by adding all the pressures at every sections of the path.

For the calculation of total pressure loss and fan selection the path with highest pressure loss is taken into the consideration and that path is called as 'Critical Path'. It is either the longest path for fluid flow or a path with maximum duct fittings or bends to yield maximum resistance (pressure).

Types of pressure losses (Calculated using ASHRAE Duct fitting database 6.0)

1. Duct fitting losses;

   i. Straight duct friction loss

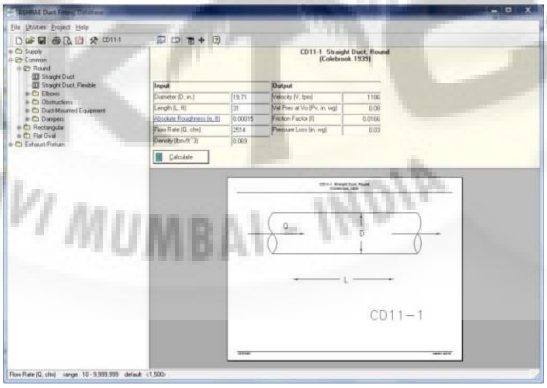


Fig.4.67. Pressure loss for straight duct

ii. Branching friction loss

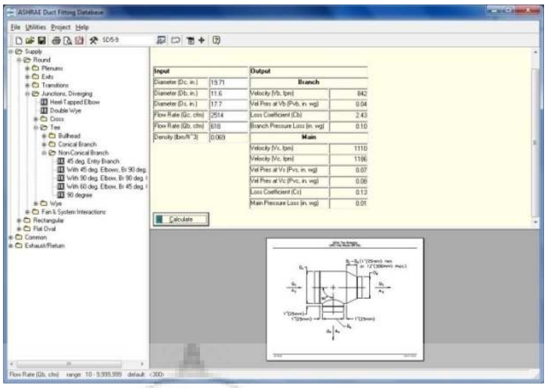


Fig.4.68. Pressure loss for Branching

iii. Elbow

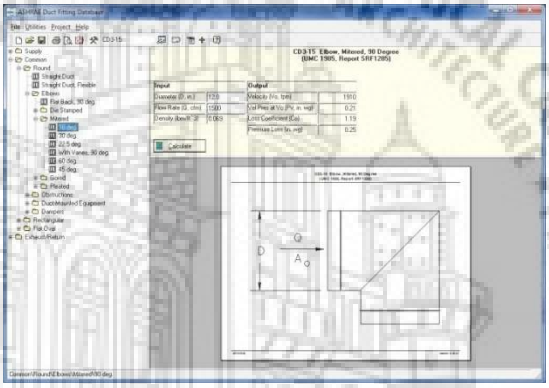


Fig.4.69. Pressure loss in elbow joint

2. Equipment losses (Determined directly from manufactures catalogues)
   i. Air Filters (AAFL catalogue)
   ii. Volume control dampers (VFCD Catalogue)

A. Duct System layout

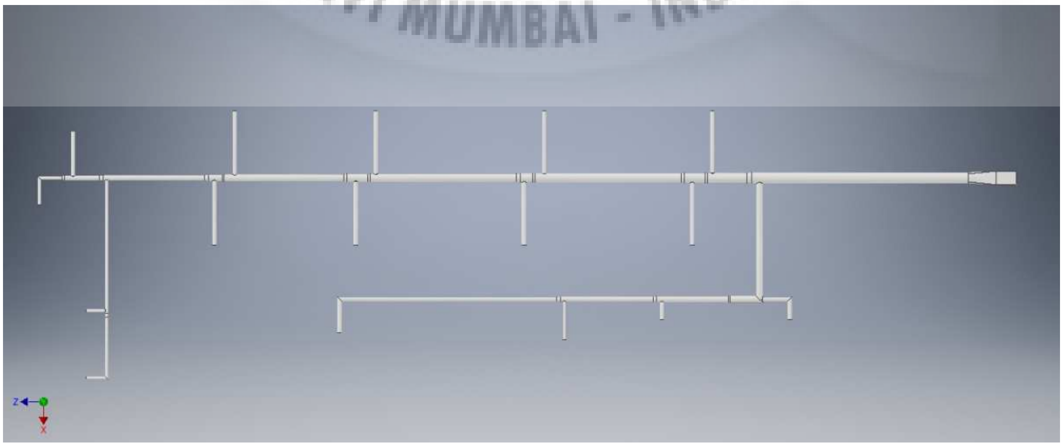


Fig.4.70. Ducting layout for given office

B. Critical Path for Pressure calculation

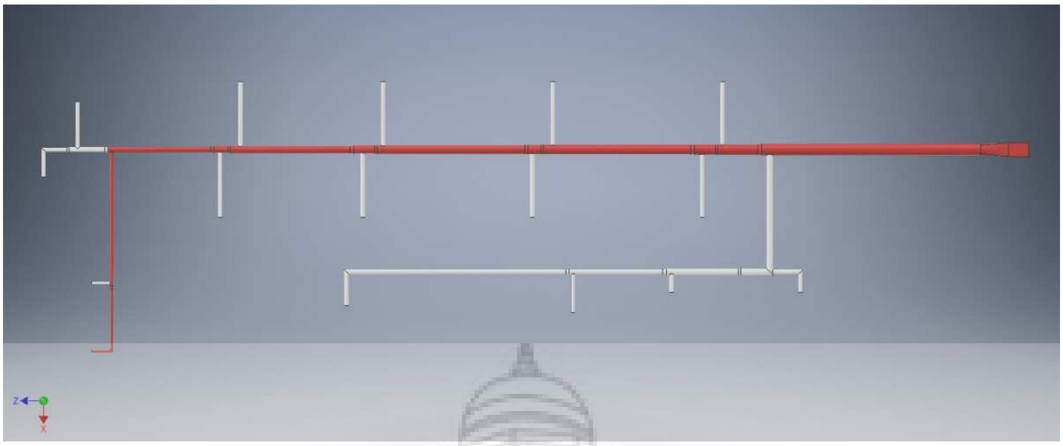


Fig.4.71. Critical path

C. Total pressure:

The pressure for this path is calculated with help of ASHRAE Duct fitting Database 6.0. The result is:

For the Critical path,

Total pressure = 1.89 inches of WC. = 472 Pa

Estimation of Duct Thickness

Thickness of the duct is directly estimated from IS - 655 / 2006

Table 6 Thickness of Sheet for Rigid Polyvinyl Chloride Duct

(*Clause* 6.2.1)

Classification of Duct by Pressure	Low Pressure Duct and Medium Pressure Duct	High Pressure Duct Pa		Thickness of Sheet, *Min* mm
	$p \leq 1\,000$	$1\,000 < p \leq 1\,500$	$1\,500 < p \leq 2\,000$	
(1)	(2)	(3)	(4)	(5)
Long side of duct	$l \leq 500$	$l \leq 500$	—	3
	$500 < l \leq 1\,000$	—	$l \leq 500$	4
	$1\,000 < l \leq 2\,000$	$500 < l \leq 2\,000$	$500 < l \leq 2\,000$	5
	$2\,000 < l$	$2\,000 < l$	$2\,000 < l$	6

Fig.4.72. IS standard thickness chart for PVC

For 472 Pa internal pressure, Thickness = 3mm

4.3.3. Fan Selection

Fan is selected according to pressure required in the duct and the total flow rate. Fans are selected according to Fan curves. Fan curves are plotted as Pressure Vs Flow rate.

Properties which are determined from fan curve.

i. Fan speed

ii. Pitch angle

iii. Motor Power

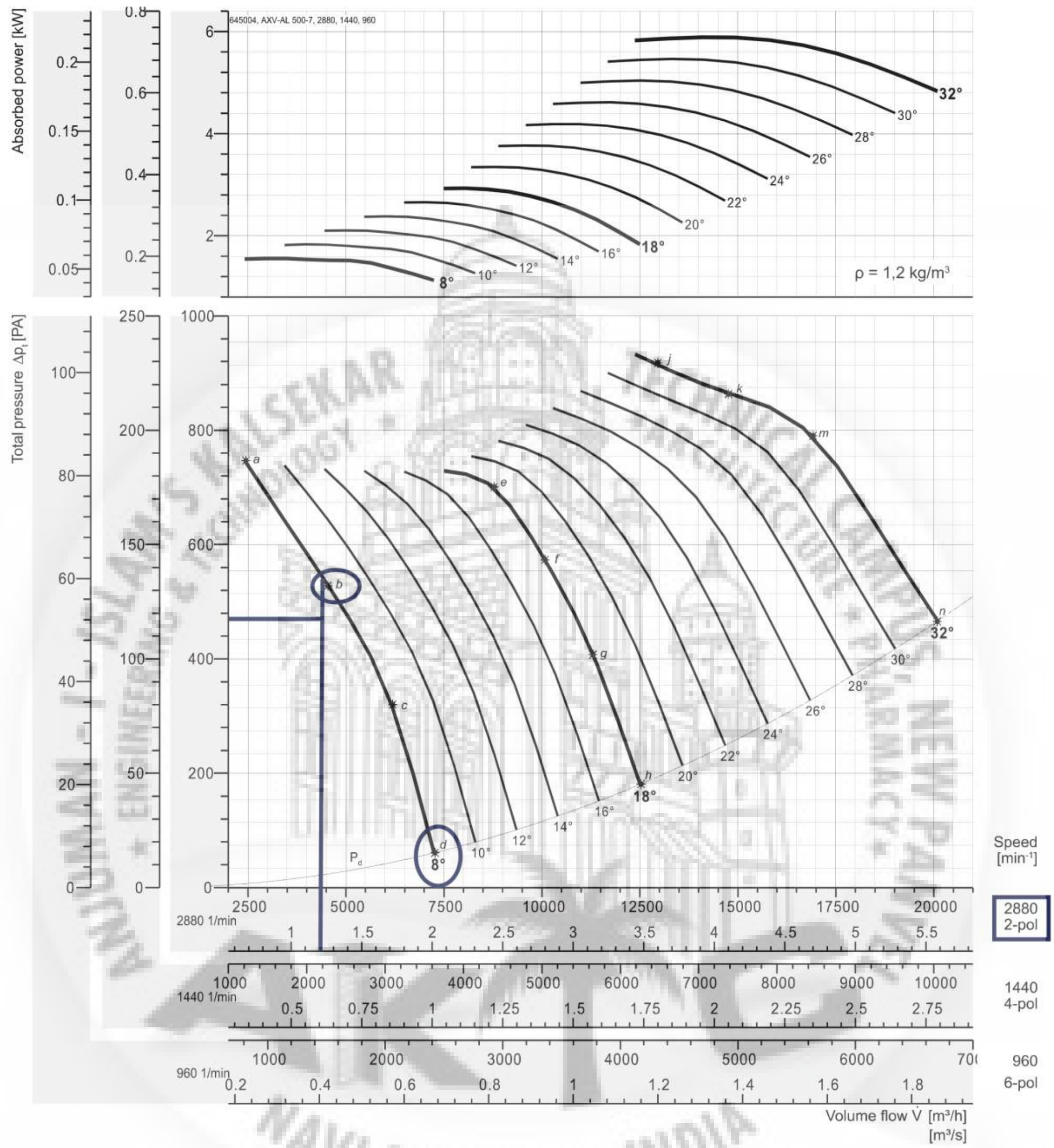
Selection of Fan for the Current duct layout:

- Pressure = 472 Pa
- Air flow rate = 2514 CFM

  = 4271 m^3/hr

  = 1.19 m^3/s
- On the fan curve vertical axis shows the pressure and horizontal axis is for flow rate. Above values are plotted on Fan curve to determine the required fan properties.

Performance Curve — AXV 500-7, 50 Hz



Peak absorbed power [kW]

n [min⁻¹]	Pitch angle [°] 8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40
960	0,058	0,068	0,078	0,088	0,098	0,108	0,124	0,141	0,155	0,171	0,187	0,202	0,218	-	-	-	-
motor	0,37																
1440	0,194	0,228	0,262	0,297	0,331	0,366	0,419	0,471	0,524	0,577	0,630	0,683	0,736	-	-	-	-
motor	0,37						0,55			0,75							
2880	1,55	1,83	2,10	2,37	2,65	2,93	3,35	3,77	4,19	4,61	5,04	5,46	5,89	-	-	-	-
motor	2,2			3			4		5,5				7,5				

Fig.4.73. Fan selection using fan curve

Fan properties selected from Fan Curve

i. Fan Speed: 2880 rpm
ii. Pitch angle: 8°
iii. Motor Power: 1.55 kW

4.3.4. Cost Comparison

Initial cost and running cost is to be considered while cost comparison. Material cost, cost of fan-motor and damper is considered to be initial cost. Energy consumption, maintenance and many other are to be considered in running cost.

Figure 4.74 represent the mass of PVC material required for ducting which is 885.843 lb.
As, 1 lb = 0.453592 kg So, 885.83 lb = 401.81 kg.
Considering standard rate 1kg PVC material to be 72 Rs.\kg (finished product)
The total cost of material accounts for: 401.81*72 = 28,930.32 Rs.

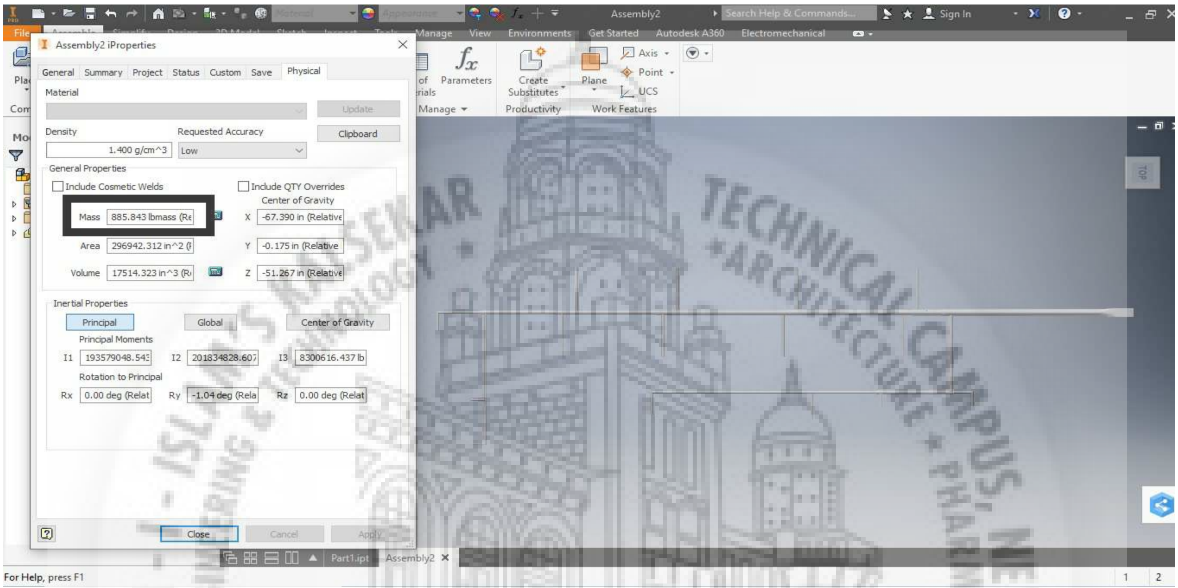


Fig.4.74. Mass estimation of PVC ducting using Inventor Software

Figure 4.75 represent the mass of GI material required for ducting which is 1728.94 lb, which is equivalent to 784.23 kg. When considered standard rate of 1 kg of GI to be 70 Rs\kg.
Therefore, total cost is 784.23*70 = 54,896.1 Rs.

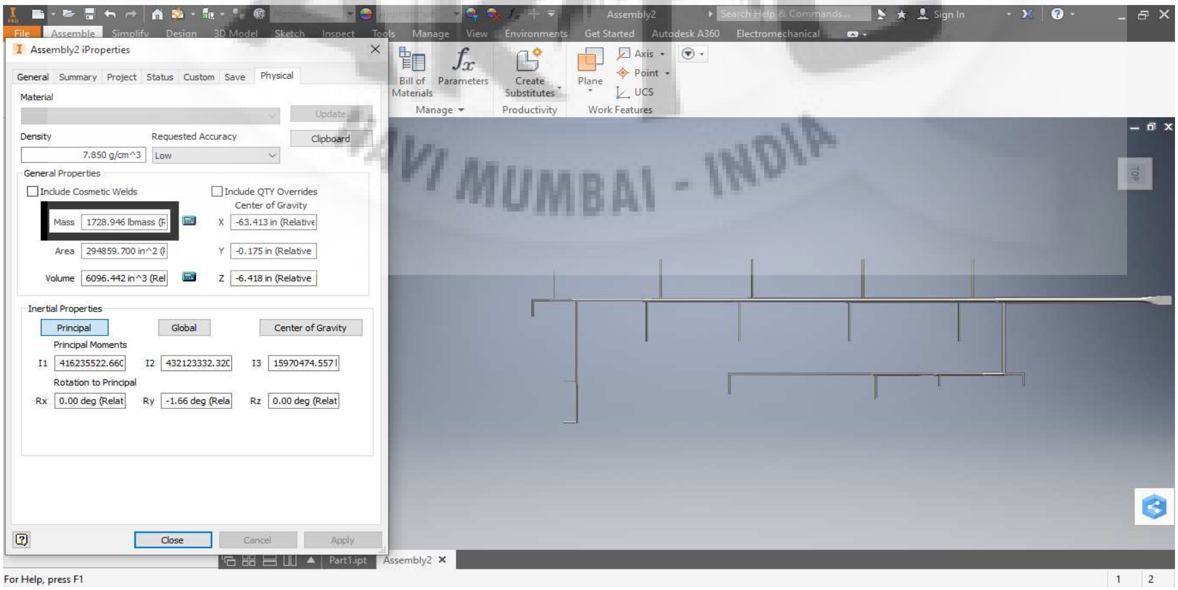


Fig.4.75. Mass estimation of G.I ducting using Inventor Software

Savings in rupees (for material) = 54,896.1 - 28,930.32 = 25,965.78 Rs.

While considering fan-motor cost there is no substantial difference in fan size so the cost also don't differ a lot as mentioned in Table 4.23 – motor specification for PCV and GI. Where the cost 1.55 kW motor, required in PVC ducting is 13,037 Rs., conversely cost of 2.2 kW motor, required in GI ducting is 13,958.2 Rs.

Saving in rupees (fan-motor) = 13958.2 – 13037 = 921.2 Rs.

Other costs which are present are shown in Table 4.21

TABLE 4.21
COST ELEMENTS PRESENT IN PVC OR GI DUCT SYSTEMS

Material	PVC	GI
Fasteners (nut & bolt)	Absent	Present
Adhesive	Present	Absent
Thermal insulation	Absent	Present

As we can see form the above table that ducting of PVC requires less equipment as compared to GI ducting.
For a rough estimation consider 3 kg of adhesive will be required (advised by normal contractor), cost per kg of adhesive is 140 Rs\kg so the total cost accumulates to 3*140 = 420 Rs.

Now let's assume that total 10 kg of nuts and bolts are required in ducting per kg cost of nuts and bolts are 55 Rs.\kg. The cost of nuts and bolts would be 55*5 = 275 Rs.

Thermal insulation is available per sq. ft and the price is 20 Rs.\ sq. ft. From figure 15 total surface area for GI ducting is 294859.7 sq. inch i.e. 2047.63 sq. ft, so the cost for insulation comes to 2047.63*20 = 40,940 Rs.

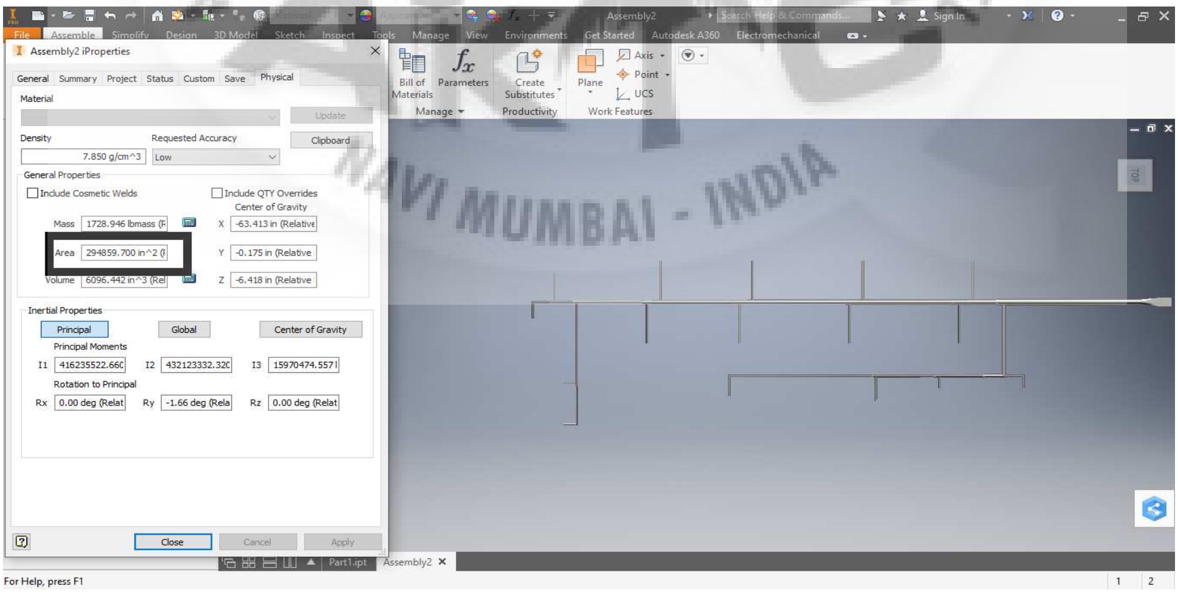


Fig.4.76. Surface area of G.I ducting using Inventor Software

TABLE 4.22
COST COMPARISONS OF PVC DUCTING & G.I DUCTING

Items	GI Ducting (Rs)	PVC Ducting (Rs)	Difference (Rs)
Material	54,896.1	28,930.32	25,965.78
Fan-motor	13,958.2	13,037	921.2
Adhesive	-	420	- 420
Fasteners (nut & bolt)	275	-	275
Thermal insulation	40,940	-	40,940
		` Total:	67681.98

From Table 4.21 we get total saving in initial cost, there many other factor such as labour and time i.e. not much skilled labour is required for ducting of PVC compared to GI ducting, also the time required for PVC ducting is less.

Consider running cost, the cost associated with electricity consumption and maintenance of ducts.

Motor installed in both systems has power capacity as follows:

Motor installed in PVC ducting consumes a power of 1.5 kW compared to 2.2 kW required in GI ducting. Average per unit watt rate in different regions of India has been taken from PHD chamber of commerce and industry, which is 9.85 Rs. /kW for Maharashtra.

Cost in rupees (formula):

*Cost per day = per unit rate * Power consumed (kW) * no. of hours in a day*

Cost in PVC ducting = 9.85*1.5 *12 = 177.3 Rs.\day

Cost GI ducting = 9.85*2.2*12 = 252.16 Rs.\day

Cost comparison between PVC & GI duct is shown below graphically:

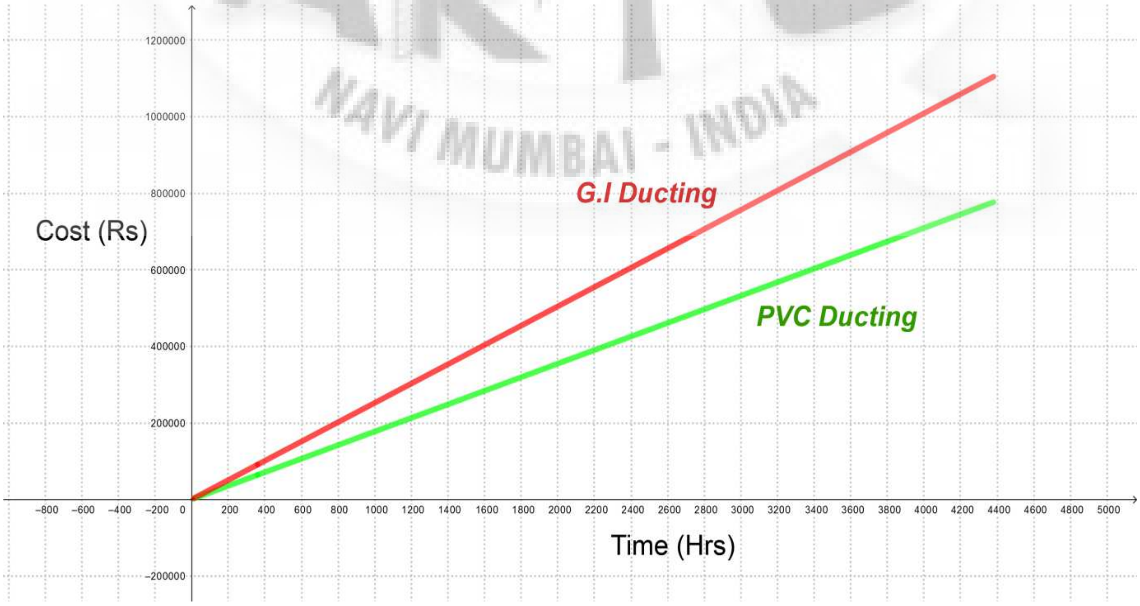


Fig.4.77. Power consumption cost vs. Time graph

4.3.5. Physical properties and life comparisons

- Availability: PVC pipes are easily available as compared to GI pipes for ducting.
- Weight: PVC pipes are lighter than GI pipes and are easy for installation.
- Cost: Cost of PVC Pipes is comparatively lower than GI pipes as already proved above.

G.I pipes are not always suitable for duct work due to its corrosive, heavy weight, inflexible properties as well as high cost. Because of these reasons use of PVC pipes may solve the above problems.

TABLE 4.23

TOTAL COMPARISON BETWEEN PVC DUCTING AND G.I DUCTING

Sr. No.	Particulars	GI ducting	PVC ducting
1	Roughness	0.003 ft	0.00015 ft
2	Pressure loss	0.12 in WC/ 100ft	0.08 in WC/ 100ft
3	System Pressure	710 Pa	472 Pa
4	% Pressure loss	-	33% less
5	Density	7.850 g/cm^3	1.400 g/cm^3
6	Mass	1729 lbs (784 kg)	886 lbs (402 kg)
7	Duct thickness	0.6 mm	3 mm
8	Fan Motor Power Pitch angle	2.1 kW 12°	1.5 kW 8°

`

Life of PVC ducts

The average lifespan of PVC pipes is 50 to 70 years. While according to many of the plastic manufacturer it may last up to 100 years or more. The lifespan of the PVC pipes is also depending upon the environmental conditions and where it has been fitted.

Whereas galvanized steel pipes have an average life expectancy of 30 to 40 years. This is comparably lower than PVC pipe

Chapter 5

CONCLUSION

Project 1 and 2:

The cooling loads estimated in above sections are based on manual load sheet calculation method which provides accurate and reliable result. The results verified by Heat cad 2019 software and both the results are almost same with only 5% of error. Pipe size and joint selection by manual method and from VXV selection software 2.7.7 are tallied with no errors. Additional refrigerant required in the system calculated by manual method when compared with software report gave an error of 1.13 %. Optimum insulation thickness and wiring diagram are selected from company service manual.

Project 3:

From the costing table difference in cost can be easily compared. With 45.47% saving in material, 6.6% saving in fan motor and 98.98% saving in other accessories. Average percentage saving while using PVC ducting over GI ducting accumulates to 50.35%, further long term saving can be achieved in terms of running cost and maintenance cost. From figure 16 one can easily depict the substantial increase energy consumption and its cost for GI ducting with respect to PVC ducting and long term gain can easily be achieved with PVC ducting.

Chapter 6

FUTURE SCOPE

Project 1 and 2:

According to this research, the India HVAC market is expected to attain a size of $5.9 billion by 2024, progressing at a CAGR of 7.0% during the forecast period (2019-2024). The major factors driving the growth of the market are increasing number of high-rise buildings, hypermarkets in Tier-II cities, and shopping complexes and malls, where HVAC systems are an essential requirement. Furthermore, the market is predicted to progress due to the on-going smart city projects in the nation. Variable refrigerant flow (VRF) technology has gradually expanded its market presence and steadily gaining market share throughout the world. According to recently published report of Bonafide Research "India Air Conditioning Market Overview", variable refrigerant flow air conditioning market is expected to grow with a CAGR of 12.62% over next six years. In India, VRF systems are gaining its market shares both on medium sized commercial buildings and premium segmented dwellings.

Project 3:

Few disadvantages of PVC material include material transportation is difficult, custom GI duct shapes are easily available in the field than PVC duct shapes which often must be specially ordered from a manufacturer, damaged parts can be easily replaced in GI ducting and PVC material is more prone to cracking and failure in term of strength as it is weaker. Fitting for PVC cost more but it can be overcome by custom manufacturing. Although having few disadvantages PVC ducting can be used in application specific places. With upcoming advancement in technology these disadvantages can be overcome with time.

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Project Completion Certificates

A. Certificate from Amaan Cooling Solution (Project 1)

AMAAN COOLING SOLUTION

Voltas authorized channel partner

To Whomsoever It May Concern

This is to certify that candidates

1. **Mr Laxman Mandal**
2. **Mr Shadab Khan**
3. **Mr Ganesh Satve**
4. **Mr Eesa Rakhange**

Who are students of **ANJUMAN-I- ISLAM 'S KALSEKAR TECHNICAL CAMPUS**, have successfully completed the Project on "**DESIGN OF VRF SYSTEM**". At our company "**AMAAN COOLING SOLUTIONS**" during the academic year 2019-2020.

They have assisted us in the following work:

- Heat/ Cooling Load estimation.
- Pipe size Calculations
- Thermal Insulation Calculations.
- Refrigerant Quantity Calculations.
- Communication cable wiring.
- Trouble shooting an error post installation.

They have also helped us through various processes of installation of indoor/outdoor units and piping of the VRF system.

They have done this project by themselves as a team. All the necessary details were provided from our side for the establishment of the project .They also suggested us for fresh air supply in the system for better human comfort. We are satisfied by their work and appreciate their effort to accomplish the project at most sincerity.

Patel

PATEL SARFARAJ

(PROPRIETOR)

AMAAN COOLING SOLUTIONS

AMAAN COOLING SOLUTIONS PUNE

Address: Shop no-2,Opposite Lifeline Hospital , Kausarbaugh Co-op society ,Kondhwa,Pune -411048
Email ID:amaancs7@gmail.com Mobile no: 9049817786
GSTNO: 27AZLPP7329P1ZD

B. Certificate from VOLTAS (Project 1)

VOLTAS

To Whomsoever It May Concern

This letter is provided with the reference to the undertaking submitted by the students of **ANJUMAN-I-ISLAM 'S KALSEKAR TECHNICAL CAMPUS** dated **14/10/2019**

This is to acknowledge that the candidates

1. **Mr. Laxman Mandal**
2. **Mr. Shadab Khan**
3. **Mr. Ganesh Satve**
4. **Mr. Eesa Rakhange**

Have successfully completed the project on "**DESIGN OF VRF SYSTEM**" with our system solution provider "**AMAAN COOLING SOLUTIONS**" at **TRICENTIS HINJAWADI, PUNE** during the academic year 2019-2020.

They were involved with us from the starting to fully accomplishment of the project. They used to us to follow-up us from time to time. All the calculations which were been done by them have totally matched with our standard calculations with minimal errors. They were present almost each and every time during the crucial period of project.

We have seen them taking all the technical knowledge involved in the installation of the project. They have also helped our solution provider through various processes of installation of indoor/outdoor units and piping of the VRF system.

All the necessary details were provided from our side for the establishment of the project. Their suggestion for fresh air supply in the system using PVC made ducts were of great importance. We found them extremely inquisitive and hard working. They were also willing to put their efforts and get in to depth of the subject to understand it better.

Their association with the company was fruitful and we wish all the best for the future endeavours.

Bharat

VOLTAS LIMITED ★ DPG ★ PUNE - 411003 ★

Bharat M
(Senior Engineer)
VRF Design, Execution and Commissioning

VOLTAS LIMITED
Domestic Projects Group
14, Mumbai - Pune Road, Wakadewadi, Pune - 411003.
Tel : 020-66297444/408/433, 66011776 Fax : 020-66046601 Website : www.voltas.com
Registered Office : Voltas House 'A', 5th Floor, Dr. Babasaheb Ambedkar Road, Chinchpokli, Mumbai - 400033.
Corporate Identity Number L29308MH1954PLC009371
A TATA Enterprise

C. Certificate from Aircon Solutions (Project 2)

Aircon Solutions

(HVAC & ELECTRO MECHANICAL ENGINEERS)

Office : H.No. 0329, Shop No. 3, Sec- 12 E, Khairne, Bonkode Gaon, Navi Mumbai - 400 709.
Mob : 9867244878, 9029796215 E-mail : airconhvac2015@ gmail.com.

To Whomsoever It May Concern

This is to certify that project work entitled "**DESIGN OF VRF SYSTEM**" is a bonafide work carried by the candidates.

1. **Mr Laxman Mandal**
2. **Mr Shadab Khan**
3. **Mr Ganesh Satve**
4. **Mr Eesa Rakhange**

Who are students of **ANJUMAN-I- ISLAM 'S KALSEKAR TECHNICAL CAMPUS**. Have completed the project during the academic year 2019-2020.

As the part of project they have designed the vrf system by taking following factors into consideration:

- Heat/ Cooling Load estimation.
- Piping of the indoor and outdoor system.
- Thermal Insulation thickness Calculations.
- Additional Refrigerant required in the system.

They have helped our technical team in the following ways:

- Installation of indoor and outdoor unit.
- Communication cable wiring.
- Trouble shooting any error post installation.

They have completed their project in timely manner with punctuality and sincerity. Their association with us was fruitful and we wish them luck for their future endeavours.

For AIRCON SOLUTIONS

AUTHORISED SIGNATORY

Achievement

Certificates from JETIR for research paper publication (Project 3)

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Eesa Rakhange

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