ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Thermal Comfort for Passenger Train from Dire-Dawa to Djibouti

By

Mekonnen Tadesse

A Thesis Submitted to the School of Graduate Studies, Addis Abeba University, in Partial Fulfilment of the Requirements for the Degree of Masters of Science in Railway Mechanical Engineering

Advisor

Demiss Alemu (Dr. Ing.)

March, 2015

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DECLARATION

I, the undersigned, declare that the thesis entitled "Thermal Comfort for Passenger Train from Dire-Dawa to Djibouti" comprises my own work. In compliance with internationally accepted practices, I have duly acknowledged and referenced all materials used in this work. I understand that non-adherence to the principle of academic honesty and integrity, misrepresentation/ fabrication of any idea/data/fact/source/ will constitute sufficient ground for disciplinary action by the university.

__________________ ______________

Mekonnen Tadesse Date

This is to certify that the above declaration made by the candidate is correct to the best of my knowledge

______________________ _____________

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ACKNOWLEDGMENT

Alhamdulillah praise is to Allah, the cherisher, sustainer of the universe. My at most gratitude is to lord Almighty, Allah, who gave me the courage and strength to go through.

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ABSTRACT

People spend a considerable time in vehicles; cars, trains, planes, buses or subway. This is the reason why the thermal comfort has more and more paid attention. In one hand people try to make comfort whatever they are at home, office or in car. In the other hand the thermal conditions in the cabin of vehicles and passenger compartments directly influences on the drivers and passengers safety,

Now a day the railway transport infrastructure in Ethiopia is the major issues in relation to the development of the country. To make the sector more competitive with other transportation modes it is better to design effective air conditioning system for a rail vehicle locally based on the specific working environment, so as to make passengers more comfortable. After all, passengers want to have a comfortable journey and arrive at their destination rested, so keeping train passenger compartment at a pleasant temperature is crucial.

The main objective of this thesis is designing of air conditioning system for a rail vehicle passenger compartment locally, while travelling through the route DireDawa-Djibouti.

In this paper the thermal load analysis is done in two ways; by using Analytical methods based on ASHRAE for the Design and Application of Heating, Ventilation and Air Conditioning Equipment for Rail Passenger Vehicles and by using Hourly analysis program (HAP) software.

During thermal load analysis car body conductive heat gain, solar radiation heat gain, glazing conductive heat gain, glazing solar radiation heat gain, passenger and operator cooling loads, ventilation cooling load, interior lighting heat gains, fan and fan motor heat gains, passenger door cycling air-infiltration cooling load and other miscellaneous internal heat gains have been considered. And the total cooling load using both methods HAP and Analytical methods is calculated as 74.2kw and 78.22kw respectively based on passenger compartment comfort parameters to be kept at a temperature of 23.5^0C *and a relative humidity of 50%. And finally the suitable vehicle AC system has been designed.*

Keyword: Cooling load, Thermal comfort, Air conditioning

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NOMENCLATURES

A	Car-body thermal envelope area
A/C	Air conditioning
ASHRAE	American Society Heating, Refrigeration and Air conditioning Engineers
CATIA	Computer aided three dimensional interpolate aided
CLTD	The cooling load temperature differential
CWB	Coincident Wet Bulb Temperature
DB	Dry-bulb
DP	Dew Point Temperature
FRP	Fiber re-inforced plastic
HAP	Hourly analysis program
HBM	Heat Balance Method and
HVAC	Heating ventilation and air conditioning
IAQ	Indoor air quality
LP	Low pressure
LST	Local standard time
MDR	Mean Daily Range
OP	Low Oil pressure
PM	Particular matter
PVB	Polyvinyl butyral
PVC	Polyvinyl chloride
Q	Thermal energy transfer rate
Q_{Ref}	Reflected radiation loads
RH	Relative humidity
RMPU	Roof mounted packing unit
SC	Solar coefficient
SHGC	Solar heat gain coefficient
TPU	Thermoplastic polyurethane
WB	Wet-bulb
WFM	Weighting Factor Method
3D	Three dimensional

d	Declination
F_{sa}	Lighting special allowance factor
F_{ul}	Lighting use factor
h	Hour angle
h_{fg}	Latent heat of fussion
hi	Interior convective air film coefficient
ho	Exterior convective air film coefficient
I_{DN}	Direct radiation
S	surface area of the roof exposed on direct sun ray (m2)
ΔT	Exterior to interior temperature differential
S_R	Surface area of the roof exposed on direct sun ray
U	Car-body thermal transmittance
UA	Factor car body heat loss per unit temperature difference
W	Total light wattage,
α	Surface azimuth angle
β	Altitudes angle
τ	Transmissivity of the surfaces
$\sum$	Tilt angle
ξ	Wall azimuth angle
θ	Incident angle
Υ	Solar azimuth angle
ωo	Outdoor humidity ratio
ωi	Indoor humidity ratio

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CHAPTER ONE

INTRODUCTION

1.1. Background

Air conditioning which deals with the comforts of human beings in an enclosed conditioned space is known as Comfort Air Conditioning. There are a number of factors that influence the comfort conditions. Comfort air conditioning has been defined by Dr. D.W. Carrier as under: “Artificial simultaneous control within enclosures of variable humidity, temperature, air motion and air cleanliness.”

The rail passenger vehicle HVAC system is designed to provide a comfortable and pleasant interior thermal environment to the passengers during all expected external ambient environmental conditions. For that purpose, appropriately sized and controlled heating, cooling, and air supply components are integrated into the passenger car design. Based on the thermal load analysis and the specification the HVAC equipment configuration which may consist of single or multiple unitized HVAC units or split system components mounted beneath, within, or on top of the vehicle but in this thesis we use roof mounted HVAC units. In addition to the HVAC units and depending on the application, other HVAC components like ducting, dampers, diffusers, exhaust fans, booster fans, etc., may need to be integrated into the passenger car design.

Thermal comfort in rail vehicles represents a subjective sensation of heat balance that occurs in the human body when environmental parameters - air temperature, air humidity, radiant temperature, air velocity, human level activity and clothing insulation - are in a range of well-defined values.[3]

The research on thermal comfort has been in progress for many years. The study of thermal comfort in vehicles was developed from basic thermal comfort research and applied work related to factories and buildings. Initial research dealt mainly with agricultural vehicles and public transport systems such as subways, trains and buses [4]. Passenger thermal comfort in vehicles has been a concern throughout the development of the automobile and is known to affect the automobile driver’s arousal level and ability to concentrate. The current importance of this issue in vehicle design is evident by the high cost associated with providing this feature in automobiles. In the United States, approximately 26 billion litters of fuel are consumed annually for cooling vehicle passenger compartments [5]. This figure can be reduced considerably by improving climatic control systems, lowering the solar heating load and delivering cooling air more efficiently while keeping the required thermal comfort levels inside the vehicle.

Improving the comfort conditions of rail vehicles is an important factor in increasing the attractiveness of rail transportation system. In particular, providing air conditioning in the rolling stock can play a significant role in making public transport a viable alternative to the private car.

This research is mainly to address a detailed analysis of the comfort parameters and present the possible ways of improving thermal comfort by designing air conditioning system for passenger rail vehicle from Diredawa to Djibouti port locally.

Temperature, humidity and air velocity can be called independent factors. There is also second group of factors affecting thermal comfort individual human feelings which are much harder to define, because each person has their own preferences for thermal comfort. The symptoms of thermal discomfort are intensive sweat production and increment of heart beat frequency.[6]

When assessing thermal comfort in a confined space, we must examine both the general temperature in the room, as well as the uneven distribution of heat in the room. Typically when people refer to temperature, they mean the temperature of the air; however, thermal comfort depends on more than simply air temperature. The mean radiant temperature entails averaging the temperatures of each surface in the room. Combined with the air temperature this produces an overall measure, the mean operative temperature. However, even this measure has its limitations, as how close a person will be to a particular surface is usually variable, and in these cases different surfaces will dominate at different times. Differences in temperature within a room or across a body can create a sense of discomfort.

While temperature is the most important factors in generating a phenomenological sense of thermal comfort, Relative humidity plays a large part in conjunction with temperature to provide a sense of discomfort. High levels of relative humidity can work against the evaporative cooling effects of sweating and leave the body prone to over-heating. Air velocity plays a role in the perception of thermal comfort. In hot weather, as the body tries to cool itself, the flow of air across the body will assist evaporative cooling from sweating. When air has a high relative humidity, the air next to the sweating body may become saturated with moisture, but by moving the air next to the body away and bringing in fresh, lower-humidity air, the evaporation of sweat can continue. Mechanisms of convection can further move the heat generated by metabolic processes from the skin and into the surrounding air. All this leads to continued cooling, and the higher the velocity of air, the more effective is the process.

Thermal environment in passenger cars/trains differs from those in buildings and is often highly non-uniform and asymmetric, from the following reasons:

- Interior volume is small, compared with the number of persons;
- Changing of microclimate parameters could be rapid (vehicle is changing the orientation to the sun, etc);
- The shape of the cab interior is complex;
- Glazing area is large in comparison to cab surface;
- Passengers seat near surfaces with temperatures which could be significantly higher or lower than interior air temperatures;
- Passengers are not able to change position within the cab, and changes of body posture are limited

Thermal comfort is achieved when passengers perceive the air temperature, humidity, air movement, and heat radiation of their surroundings as ideal and would not prefer warmer or colder air or a different humidity level.

1.2. Air Conditioning Equipments in Railway Coaches

This consists of the following: -

- Evaporator Unit and Compressor,
- Condenser Unit.
- Gauge panel.
- A/C control panel.
- Air Duct.
- Refrigerant piping & joints.
- Wiring.

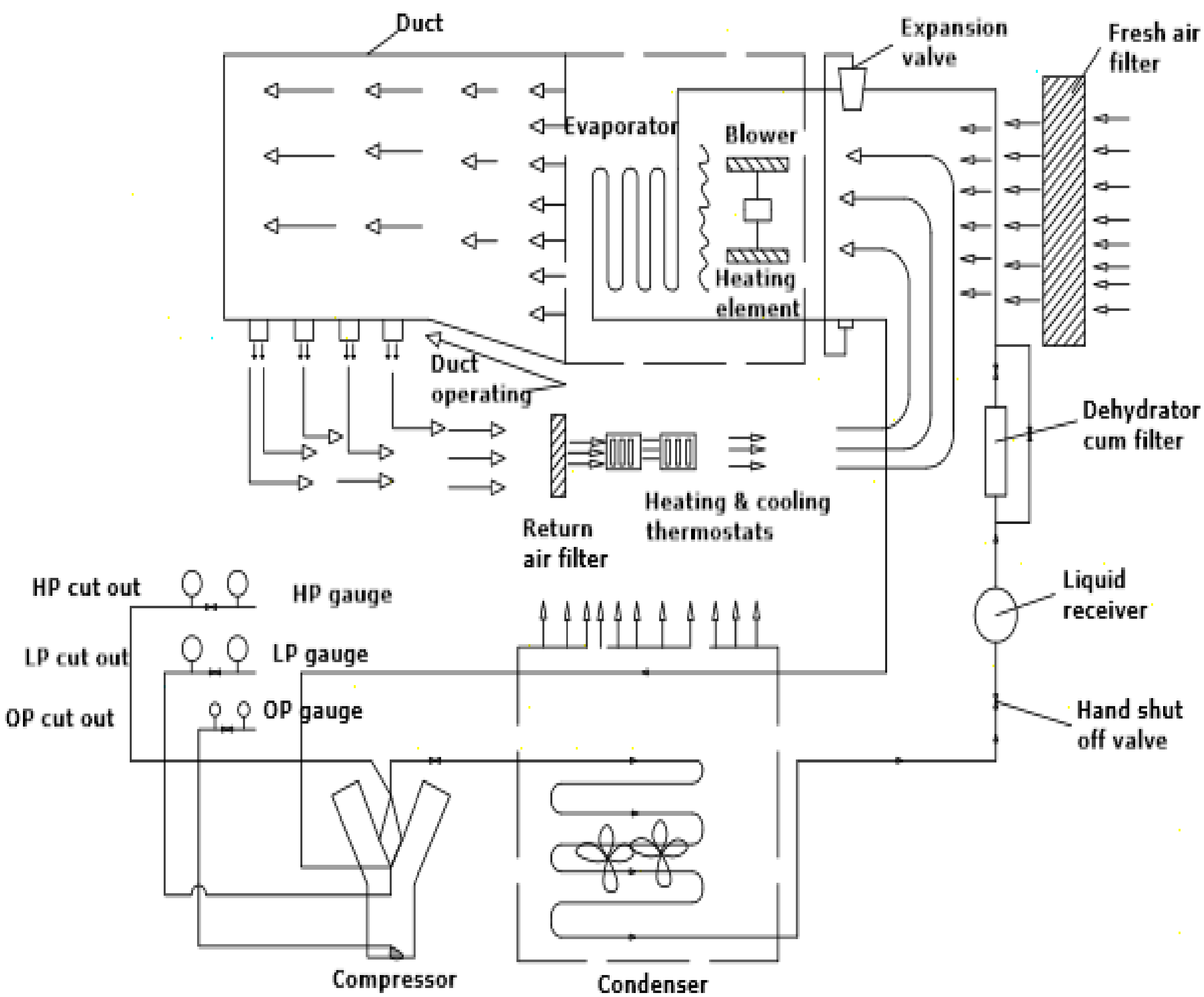


Fig 1: Schematic diagram of air conditioning system for A/C coaches [24]

1.2.1. Evaporator Unit

The evaporator unit consists of a thermostatic expansion valve, a heat exchanger, a resistance heating unit and centrifugal blower driven by a motor The thermostatic expansion valve controls quantity of high pressure liquid refrigerant and allow to expand to a lower pressure corresponding to the load demand. The expanded refrigerant passes through the distributor into the heat exchanger consisting of finned copper tubes. The return air from the air conditioned compartment is mixed with fresh air and this mixture is drawn / blown through the heat exchanger, where heat in the air is transferred to the cool refrigerant causing cooling of the air and the evaporation of the refrigerant inside the tubes. The cooled air is led through the ducting to the various compartments and diffused by means of air diffusers. Filters are provided in the fresh air and return air path to eliminate dust. When the outside ambient temperature is very low, heater is switched on according to the setting of the thermostats.

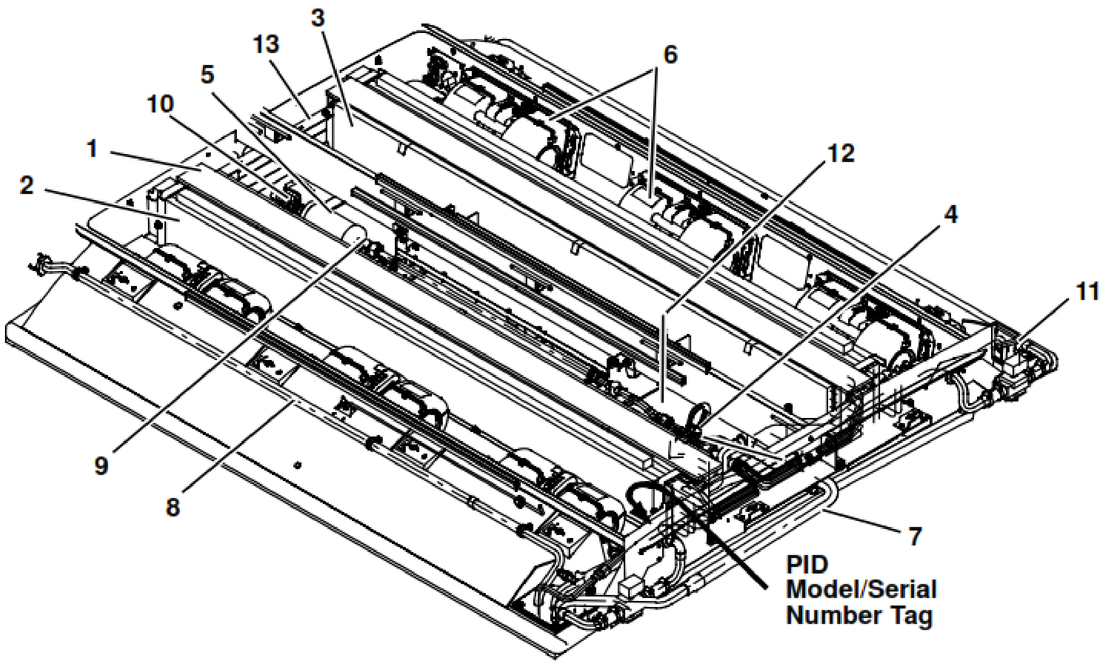


Fig 2: Evaporator Section Components [25]

1. Evaporator Coil Assembly
2. Heat Coil
3. Evaporator Return Air Filter
4. Expansion Valve
5. Filter Drier
6. Blower & Motor Assembly
7. Suction Line
8. Discharge Line
9. Service Valve
10. Liquid Line Solenoid
11. Heat Line Connection
12. Control Panel
13. Fresh Air Damper

1.2.2. Compressor

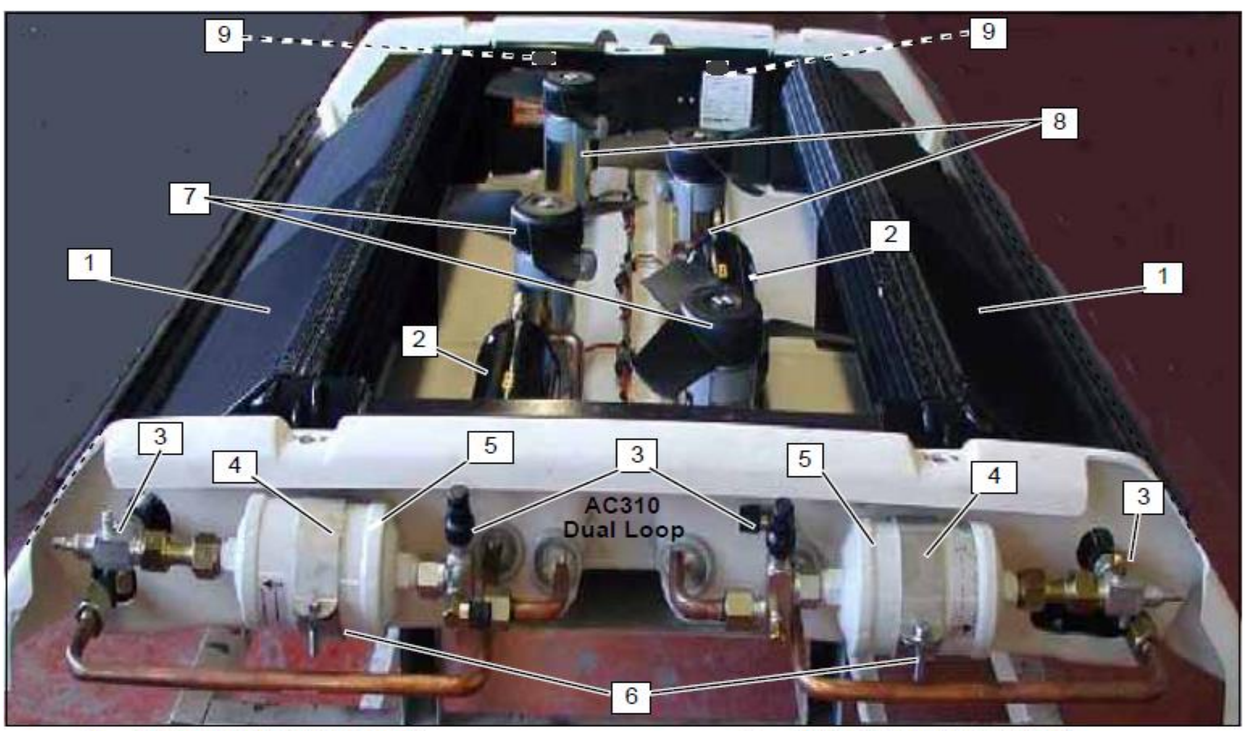
The refrigerant vapour drawn from the evaporator is compressed by means of a multi cylinder reciprocating compressor and compressed according to the load demand. The work done due to compressor, raises the temperature of the refrigerant vapour.

The compressor assembly includes the refrigerant compressor, clutch assembly, suction and discharge service valves, high pressure switch, low pressure switch, suction and discharge servicing (charging) ports and electric solenoid unloaders.

1.2.3. Condenser

The condenser serves the function of extracting the heat absorbed by the refrigerant vapour in the evaporator and the heat absorbed during the compression process. The condenser consists of a heat exchanger, which is forced-air-cooled by means of two or three axial flow impeller fans. The refrigerant vapour is liquefied when ambient cool air is passed through the heat exchanger. The refrigerant liquid leaving the condenser is led into the liquid receiver from where it proceeds to the expansion valve on the evaporator. The liquid receiver is a cylindrical container which contains a reserve of the refrigerant liquid. A dehydrator and filter are also provided to ensure that the refrigerant is free from moisture and dust particles.

The condensing section includes the cover, left and right condenser coils, fan and motor assemblies, receiver and service valves as shown in the figure below.



1. Condenser Coil Assembly
2. Receiver
3. Service Valve (High Side)
4. Filter Drier Upper Support
5. Filter Drier
6. Filter Drier Lower Support
7. Condenser Fan and Motor Assembly
8. Condenser Motor Support
9. Discharge Line Check Valve (Location)

Fig 3: Condensing Section Components [24]

1.2.4. Gauge panel

Gauge panel consists of pressure gauges (HP, LP, and OP) and pressure cut-outs to protect the compressor against, (i) High pressure, (ii) Low pressure and (iii) low oil pressure.

1.2.4.1. High pressure *cut-out*

It is a safety device against build-up of excessive delivery pressures and protects the compressor and piping system from damage. It is a pressure operated switch which switches off the compressor drive motor when the pressure exceeds a pre-set value. The plant cannot be restarted unless the cut-out is reset manually.

1.2.4.2. Low pressure cut-out

It is also a pressure operated switch similar to the H.P. cut-out switch, but it shuts down the compressor if the suction pressure drops down below the setting point. It protects the system against unduly low evaporator temperatures and formation of frost on the evaporator. No manual reset is provided on this and therefore the compressor starts automatically if the suction pressure rises above the pre-set value.

1.2.4.3. Low oil pressure cut-out

It ensures adequate lubrication of compressor to avoid piston seizure due to less lubricating oil or failure of oil pump.

1.2.5. A/C control panel

The control of the air conditioning system is achieved by means of air conditioning control panel. The design of the various elements in the control panel takes into account the system safety requirements.

1.2.6. Air duct

The air conditioning system includes three air ducts as follows:

a. Fresh (Inlet) air duct.

b. Main air duct.

c. Return air duct.

a. Fresh (Inlet) air duct

This is provided at the rate of two per A/C plant. It is mounted on the side wall just below the roof evaporator unit. There is an opening in the side wall with louver hinge door arrangement and with the provision to house a fresh air filter. The fresh (inlet) air duct has been designed with damper valve to control the quantity of fresh air to be drawn into the compartment. This arrangement has been standardized for all types of air conditioned coaches.

b. Main air duct

The cross section of the main air duct has been designed in such a way that air velocity inside the duct shall not be higher than 350 metre/min. in order to reduce turbulence and noise due to air motion in the duct. For the same reason the main air duct has been connected to evaporator outlet by means of an intermediate transition duct made of fire resistant canvas to prevent transmission of noise produced by the blower unit.

The aperture of air diffuser has been designed to deliver the required quantity of air into the compartment at a velocity not greater than 250m/min. This diffuser is provided with a knob to deflect the air to the required angle.

c. Return air duct.

Actually there is no separate return air duct provided in A/C coaches. The return air is drawn through the return air filters directly from the compartment of A/C coach, the corridor acts as return air duct and the return air is drawn through return air filters located on the ceiling.

1.2.7. Refrigerant piping and joints

The refrigerant piping consists of the suction line (from the evaporator out let to compressor inlet); discharge line (from compressor outlet to condenser inlet) and liquid line (from the liquid receiver to the inlet side of expansion valve), connections to the gauge panel from the compressor delivery side (high pressure side), low pressure side and from the compressor crank case. The lubricating oil connections are also part of the piping system.

Main pipelines are jointed with couplers or elbows by means of silver brazing whereas joints to various components like gauges pressure cut-outs, hand shut off valves, expansion valve, strainer etc. are connected by means of flare joints to facilitate easy removal of the above elements for replacement and inspection.

1.2.8. Wiring

All wiring has been done by means of multi-stranded PVC insulated copper cables to specification. All cables have been laid on steel trough/conduits for easy maintenance and prevent fire hazards. Crimped type of connections has been adopted throughout. All the terminal boards are of fire retardant FRP material, Reliability of wiring has been made very high.

1.2.9. Temperature setting

The temperature inside the air conditioned compartment is controlled by mercury in glass thermostats with different settings as mentioned below. Operation of cooling or heating takes place in accordance with ambient conditions. The temperature control thermostats are fitted in the return air passage. Two types of thermostats are used, one for controlling the cooling and the other for controlling the heating. Both these thermostats are alike, each consisting of a sealed glass tube containing a column of mercury.

1.3. Air conditioning refrigeration cycle

When air conditioning (cooling) is selected by the controller, the unit operates as a vapour compression system using R--134a as the refrigerant (See Figure 4 below). The main components of the system are the reciprocating compressor, air--cooled condenser coils, receiver, filter--drier, thermostatic expansion valve, liquid line solenoid valve and evaporator coils. The compressor raises the pressure and the temperature of the refrigerant and forces it into the condenser tubes. The condenser fan circulates surrounding air (which is at a temperature lower than the refrigerant) over the outside of the condenser tubes. Heat transfer is established from the refrigerant (inside the tubes) to the condenser air (flowing over the tubes). The condenser tubes have fins designed to improve the transfer of heat from the refrigerant gas to the air; this removal of heat causes the refrigerant to liquefy, thus liquid refrigerant leaves the condenser and flows to the receiver. The receiver serves as a liquid refrigerant reservoir so that a constant supply of liquid is available to the evaporators as needed and acts as a storage space when pumping down the system. The receiver is equipped with two sight glasses to observe refrigerant charge level. The refrigerant leaves the receiver and passes through the charge isolation valve to the liquid line solenoid valve. From the liquid line solenoid valve the refrigerant enters the filter--drier where an absorbent keeps the refrigerant clean and dry.

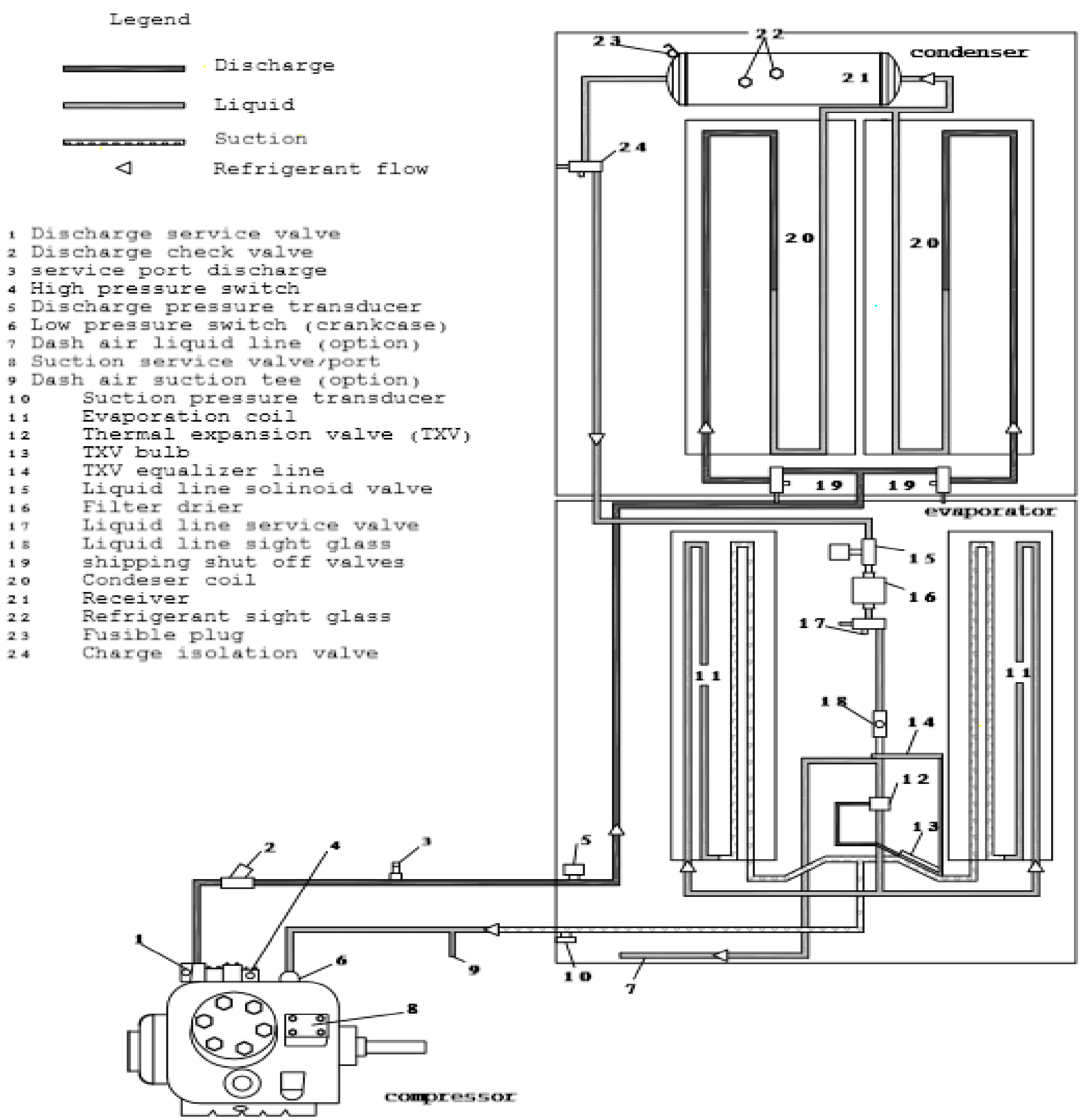


Fig 4: Refrigerant Flow Diagram [25]

From the filter--drier, the liquid refrigerant then flows through the liquid line service valve to the thermostatic expansion valve. The liquid line is equipped with a sight glass to observe the refrigerant for restricted flow. The thermostatic expansion valve reduces pressure and temperature of the liquid and meters the flow of liquid refrigerant to the evaporator to obtain maximum use of the evaporator heat transfer surface. The low pressure, low temperature liquid that flows into the evaporator tubes is colder than the air that is circulated over the evaporator tubes by the evaporator fans. Heat transfer is established from the evaporator air (flowing over the tubes) to the refrigerant (flowing inside the tubes). The evaporator tubes have aluminium fins to increase heat transfer from the air to the refrigerant; therefore the cooler air is circulated to the interior of the passenger coach. Liquid line solenoid valve closes during shutdown to prevent refrigerant flow. The transfer of heat from the air to the low temperature liquid refrigerant in the evaporator causes the liquid to vaporize. This low temperature, low pressure vapour passes through the suction line and returns to the compressor where the cycle repeats.

1.4. Thesis Objective

1.4.1. General Objective

The main objective of this study is to transfer rail vehicle air conditioning system design technology so that local rail car manufacturers can incorporate such system in the future.

1.4.2. Specific Objectives

The specific objectives of this thesis are:

- Cooling load estimation
- Duct design using duct sizer software,
- Develop 3D Duct work modelling with RMPU using CATIA software,
- Compare the calculated cooling Load in both methods,
- Based on the total cooling load and air flow rate select suitable HVAC equipment

1.5. Statement of the Problems

Railway transportation is new for Ethiopia, so all technologies related to this sector are now imported from abroad which causes the country to incur billions of Dollar. But in the future the country has a plan to improve in-house capacity so that we can handle all the design and manufacturing of railway transportation equipment locally; and rail vehicle air conditioning system is among them.

Extremely hot thermal condition in public transportation in particular railway transportation could affect the healthiness of passengers and drivers, which in turn make the sector unprofitable with other modes of transportation system.

Thermal comfort levels may vary in time and location, taking in to account our geographical location and weather condition, The function of the designed AC system is to compensate for the continuous changes of passenger compartment loads in order to maintain the passenger comfort within a thermal ‘comfort zone’. Slight temperature and air flow imbalance can easily cause respiratory malfunction in humans.

1.6. Significance of the research

Comfort and in particular thermal comfort in rail vehicles is of major importance for enhancing the attractiveness of public transport and make the rail transportation mode more enjoyable. This makes the sector (railway transportation system) more preferable and competitive over other transportation modes. In addition optimum thermal comfort in public transport keeps passengers more healthy and comfortable. And this paper is mainly focuses on bringing such importance on locally manufactured rail vehicles. By doing so, Ethiopian Railway Corporation, passengers and crew, all are benefited from such systems. By far financially Ethiopia will be benefited since we acquire technology transfer related to such design.

1.7. Methodology

The following systematic steps were guiding me in the process of completing this thesis so as to meet each of the objectives.

- Related literature review has been carried out for passenger rail vehicle thermal comfort
- Data collection
    - Previous years seasonal weather condition of that particular area (Diredawa-Djibouti) from national meteorology service agency. Only two years data is available.
    - Rail vehicle detailed dimension and vehicle capacity from ERC
- Cooling load estimation using analytical methods
- Cooling load estimation and system sizing using the aid of HAP software
- Supply duct design using Ductsizer software and 3D modeling

- Selection of HVAC system components from product catalogues

1.8. Limitation and Scopes of the Study

1.8.1. Limitation of the study

In this thesis the following limitations have been indicated:

- Only concepts of air conditioning design will be considered,
- Only using the route of Dire-dawa to Djibouti
- Due to inaccessibility of seasonal weather data, only the previous two years weather condition is considered in the analysis

1.8.2. Scope of the study

The scope of this thesis is estimating the cooling loads through the route of Dire-Ddawa to Djibouti and designing of the HVAC system components up on the result obtained in the analysis.

1.9. Thesis Organization

This thesis has a total of six chapters. The first chapter is the introductory part which is briefly introduces the concept of thermal comfort, Railway Coach Air Conditioning System and A/C Equipments, objective of the thesis, problem statement, and significance of the study, limitation and scopes.

The second Chapter presents the literature review and assesses theoretical considerations while we design Air Conditioning system for a rail vehicle and thermal comfort parameters. And the third Chapter deals with the analysis part, thermal load analysis using mathematical formulation with the aid of ASHRAE hand book for the Design and Application of Heating, Ventilation and Air Conditioning of Rail Passenger Vehicles and using Hourly Analysis Program (HAP) software.

The fourth chapter deals with the selection of RMPU, diffuser and grilles, duct work design, selection of suitable fan based on the analysis result above. The fifth chapter is result and discussion part of the thesis. And the sixth chapter is the last chapter which contains Conclusion, recommendation and future works in the study.

CHAPTER TWO

LITERATURE REVIEW

2.1. Literature Review

Gabriel Haller defines Thermal Comfort as in the ISO 7730 standard as being "That condition of mind which expresses satisfaction with the thermal environment". A definition most people can agree on, but also a definition, which is not easily converted into physical parameters.

The peoples' interest in the railroad has been particularly elevated with the opening of express railway in April 2004. Especially, as the new express railway is expected to solve the problem of the chronic traffic congestion on the roads and the resulted distribution expenses and thus the transportation time is shortened accordingly, the utility value of railroad is significantly elevated. Therefore, the transportation share of railroad is also rapidly rising. These benefits come from the advantage of high speed added to the existing merits of the railroad such as economical efficiency, punctuality, and mass-transportation and, therefore, the demand for railroad use is expected to be continuously risen.[26]

However, the railroad still has many environmental problems to be solved. Among them, the indoor air quality of passenger cabins is the most frequently indicated one. The indoor air quality (IAQ) of railroad became particularly important issue as it is rather hard to ventilate indoor air when the airtight space is cramped with many people that use the railroad at the same time. Therefore, large quantities of contaminants may exist in the indoor air of passenger cabins and such harmful substances will threat the health of persons; in particular, appropriate control of indoor air quality is essentially required in case of long-distance operation when passengers may be exposed to harmful substances for a long time[26].

The pollutants and the sources are analyzed during the time that passengers use the cabins of railroad vehicles. Also, particulate matter (PM) separator and CO_2 absorption system were developed to reduce PM and CO_2 in the passenger cabins according to the IAQ measurement data and source analysis. PM were removed from intake with installation of a cyclone type separator at the front end of air-conditioner; PM circulating indoor of passenger cabin were removed with use of a roll-type filter that is installed at the ceiling and may be replaced every six months. In case of CO_2 the absorption system was filled with the mixture of zeolite and special substances; when the CO2 level of passenger cabin exceeds a certain level, the air flows in the absorption system and the CO2 level is controlled by the absorption performance of zeolite.The study was the first challenge to develop a purification system for railroad vehicles and it was expected to improve the indoor air quality of mass transportations and thus to improve the health and comfort of passengers using railroads.[2].

Efficient design of mobile air conditioning (AC) has been the center of attention of automotive manufacturers and academic researchers during the last few decades. A clear understanding of the heating and cooling loads, encountered by the passenger cabin, is a key prerequisite for an efficient design of any mobile AC system. The function of AC system is to compensate for the continuous changes of cabin loads in order to maintain the passenger comfort within a thermal ‘comfort zone’. Fanger's model of thermal comfort has been extensively used in AC research and applications as the basis for comfort assessment [7]. Based on Fanger's model, Ingersoll et al. developed a human thermal comfort calculation model specific to automobile passenger cabins. While performing thermal load calculations for a vehicle cabin, their model can be used to assess the corresponding status of thermal comfort [8].

To understand and to address the thermal comfort issue, various papers have been studied and listed below.

Taeyoung Han et al. presented a paper on virtual thermal comfort engineering (VCTE). Paper describes basic methods of VTCE and demonstrates the capabilities including the effects of solar load for various solar incidence angles, glass properties, and also surrounding radiation heat, air velocity magnitude, and air temperatures on the thermal comfort of the occupants in a simplified passenger compartment [9].

Radu MUSAT1 et al. wrote details about parameters and models of the vehicle thermal comfort. Thermal comfort inside and a cabin and factors affecting it are detailed in this paper. Effect of air temperature, air velocity, human activity level, clothing insulation and combined effect of all these parameters was considered for this study. Specific models for thermal comfort are analyzed and used to estimate thermal comfort of the passenger. Humidity effect plays minor role in thermal comfort if value ranges from 30% to 70% and air velocity and temperatures play very important role in cabin thermal comfort is justified in this paper. This is useful for referring data of simulations and test with this paper. This gives better idea for unsteady state thermal comfort analysis [10].

K. David Huang et al. Contributed to thermal comfort work by introducing air conditioning system of an intelligent vehicle-cabin. This paper describes cabin temperature and velocities for passenger thermal comfort and experimental data of various temperatures over seat and human body parts for thermal comfort. This paper shows importance of temperature distribution over manikin for cabin thermal comfort [11].

Dusan Fiala et al. worked on fiala model which is mainly built for indoor conditions. This paper describes accuracy of thermal comfort prediction of fiala model. This paper also justifies that model can be used for indoor thermal comfort simulations, also for automotive industry to predict passengers' responses to the transient and asymmetric boundary conditions found inside car cabins [12].

Tzu-Ping Lin et al. worked on passenger thermal comfort predictions in short and long haul vehicles and discussed comfort zones according to vehicle size. Passengers in short-haul vehicles habitually adjust the air outlets to increase thermal comfort, while passengers in long haul vehicles prefer to draw the drapes to reduce discomfort from extended exposure to solar radiation. This paper refers that neutral temperatures for short and long haul vehicles are 26.20C and 27.40C, while comfort zones are 22.4-28.90C and 22.4-30.10C respectively. Also overall insulation of clothing worn by passenger and thermal adaptive behavior in vehicles differ from those in their living and working spaces. The result of this study provide a valuable reference for practitioners involved in determining the adequate control and management of in-cabin vehicle thermal environment, as well as facilitating design of bus and trains, ultimately contributing to efforts to achieve a balance between the thermal comfort satisfaction of passengers and energy conserving measures for air-conditioning in mass transportation vehicle [13].

ASHRAE Handbook of Fundamentals provides two major thermal load calculation methodologies: Heat Balance Method (HBM) and Weighting Factor Method (WFM) [14]. HBM is the most scientifically rigorous available method and can consider more details with less simplifying assumptions [15]. An advantage of HBM is that several fundamental models can be incorporated in the thermal calculations. Although HBM is more accurate than WFM, it is easier to implement WFM for load calculation in a passenger vehicle [16]. However, when more detailed information of the vehicle body and thermal loads is available, HBM is the preferred choice.

Zheng et al. devised a simple method to calculate vehicle's thermal loads. They calculated the different loads such as the radiation and ambient loads. A case study was performed and the results were validated using wind tunnel climate control tests. The different loads were separately calculated and summed up to give the total heat gain or loss from the cabin [17]. Arici et al. developed a computer code for simulating the dynamic operation of a climate control system for a typical vehicle. They considered the transition of both the cabin temperature and the relative humidity as predicted by the principles of heat and mass transfer; a lumped system model was developed. The components of the AC system were also simulated to work alongside the thermal load balance equations [18]. Ding and Zito also used a lumped model for the cabin and solved the corresponding transient heat transfer differential equation analytically [19].

Methods put forth by ASHRAE and other bodies often assume the cabin as a “lumped system” and do not take the three-dimensional distribution of temperature and flow parameters into account. Finding exact solutions for distribution of air flow, temperature, humidity, etc. is often complex, thus numerical methods have been employed for such simulations. Alexandrov et al. used two- and three dimensional CFD simulations to investigate the effect of various parameters such as car velocity and outside temperature on the performance of mobile AC systems. They simulated the flow in a typical vehicle and found a maximum temperature difference of about 7°C between two points in the car cabin. They concluded that issues such as low air circulation zones in the cabin can be resolved by designing better air inlet and outlet configurations [20].

The goal of cooling and heating load estimations is to provide thermal comfort for passengers efficiently. To do this, it is necessary to control the AC cycle to maintain the air quality within the desirable limits specified by a comfort model. Khayyam et al. collected a set of models to calculate the different types of thermal loads encountered in a vehicle [21].

Wei and Dage developed an intelligent cabin climate control system based on human sensory response to comfort factors. They used passive remote infrared sensors to measure passenger skin temperatures. An intelligent climate controller then controlled the parameters such as the blower speed to provide passenger thermal comfort [22].

The Heat Balance Method (HBM) allows continuous improvement of load calculation models through applying empirical models and correlations. The essence of this method is the conservation of mass and thermal energy. Because of this approach, much geometrical and thermal detail should be known before applying it to a specific enclosure. This makes HBM an intensive method which is avoided in some engineering applications. On the other hand, due to the fundamental approach of HBM, knowing more details can improve the accuracy of HBM models [23].

This paper aims to estimating thermal loads in a passenger compartment and design a proper air conditioning system for a rail vehicle. Although other load estimation models exist as mentioned above, this study mainly sticks on ASHRAE Guideline 23P (1) and HAP software. The results of the load estimation in turn used to design AC system equipments.

In this study, a simple method to calculate vehicle heat load is developed. The cooling load temperature differential (CLTD) method is used to calculate the heat gain of a sunlit roof and wall (door). This is done in one step by using ASHRAE data. The calculation presented here takes into account the geometrical configuration of the vehicle compartment including glazing surfaces (shading), windshield and roof angle, and vehicle orientation. Special attention is given to the calculation of direct and diffuse incidence solar radiation through the windshield and skylight glass. The vertical glass' solar heat gain is evaluated by using ASHRAE data. The U value method is used to calculate heat transfer between the outside and inside cabin. Heat gains from infiltration, occupant, and HVAC unit blower motors are considered in the cooling load calculation.

2.2. General Considerations in Designing Railway Air Conditioning system

Air conditioned coach has to work under widely varying conditions of ambient temperature, latitude, passenger load etc. In deciding the capacity of the plant, certain assumptions regarding number of adverse conditions of the working are to be made and based on these assumptions the plant capacity required can be determined.

2.2.1. General Comfort Considerations

ASHRAE Standard 55, Thermal Environmental Conditions for Human Occupancy, may be referenced to determine the railcar interior thermal environmental factors that will provide environmental conditions that will be acceptable to a majority of the passengers [1]

The following factors, typically encountered during rail travel, should also be considered:

- Changes in regional microclimates caused by local geography along the railcar's route.
- Air infiltration, drafts, and rapid temperature changes associated with door openings.
- Thermal Stratification (Vertical Air Temperature Variation).
- Velocity of air at passenger level (seated and standing passengers).
- Radiation effects of windows and structure.
- Solar loading variation due to dynamically changing solar orientation.
- Clothing insulation.
- Temperature recovery capability and pre-conditioning for passenger occupancy.
- Contribution of HVAC system to overall vehicle noise
- Length of journey.

Thermal comfort is influenced by:

- Personal factors (degree of activity, clothing, journey time)
- Spatial factors (radiant temperature, temperature of enclosing surfaces)
- Ventilation factors (air temperature, air speed, relative humidity).

These factors have complex effects on the heat balance of passengers. Thus all contributing factors must be considered in order to achieve conditions which will be perceived as comfortable by a majority of passengers.[26]

2.2.1.1. Air Motion

Controlling air motion within an acceptable range in a passenger railcar is an essential element of passenger comfort. Compared to a residential or commercial building HVAC system, the high occupant density and restricted space of a typical rail passenger vehicle make air distribution very challenging. Higher air velocities are normally desirable during high temperature and humidity operation when maximum cooling is required whereas high velocities constitute uncomfortable drafts when in heating or low level cooling operation.

2.2.1.2. Temperature Variation

Depending on the type of service conditions that influence the temperature balance in the passenger compartment can change rapidly. Because it is highly impractical to accurately simulate these dynamic conditions in a test environment, evaluation of these criteria are best made under stabilized conditions with all doors closed for the duration of the evaluation measurements.

2.2.1.3. Temperature and Humidity Design Range

Actual interior temperatures may be expected to exceed the design range when a combination of extreme environmental conditions, passenger overload, or operation occurs beyond the rating conditions. The limits to which interior temperature may exceed the design range are defined in some cases in the equipment section and may have a specific test or analysis requirement for conformance.

2.2.1.4. Thermal Recovery Rate

The need for a properly designed passenger rail car HVAC system is to maintain stable interior environmental comfort throughout its specified range of external ambient conditions and changing passenger loads is of primary importance. However, consideration must also be given to the ability of heating and cooling equipment to respond quickly to sudden changes in the thermal load which is a common occurrence in a rail passenger vehicle.

2.2.1.5. Surface Temperatures

a. Influence on Thermal Comfort

Passenger comfort can be adversely influenced by excessive non uniform thermal radiation from the surrounding surfaces. This parameter is referred to as radiant temperature asymmetry.

b. Safety Considerations

To minimize the potential for injury, the design of heating apparatus should not produce excessive temperatures on surfaces that may be exposed to passengers or crew members.

2.2.1.6. Solar Radiation Mitigation

Glazing specifically designed to reduce the effects of solar thermal radiation and to reduce interior car thermal loads is recommended for all vehicles that operate exposed to the exterior environment.

2.2.1.7. Spaces Separated From the Main Passenger Compartment

Some vehicles, mostly long distance inter-city type and some used in commuter service, have transient areas that are not intended for prolonged occupancy such as vestibules and toilet rooms and as such have less demanding thermal comfort requirements than the passenger occupied areas. Some are indirectly controlled, relying solely on branch conditioned air ducts from the passenger compartment, while other applications have thermostatically controlled local heaters. Few, if any, have local air conditioning devices or controls for unoccupied areas. The required comfort criteria in these areas can vary widely depending on the level of service and passenger expectations.

2.2.1.8. Supply Air Temperature

The majority of rail passenger vehicle HVAC systems distribute conditioned air from ceiling mounted grilles or diffusers. Floor and sidewall distribution systems are also used in some applications. Care must be exercised in the layout, design, and adjustment of the air distribution devices to assure uniform air distribution and mixing while avoiding uncomfortable drafts.

2.2.2. Ventilation

2.2.2.1. General consideration for Ventilation

Outside air is typically introduced into a rail passenger vehicle's HVAC system, return air stream by means of a separate ventilation fan or is drawn into the return air stream by the HVAC system's main supply air blower. The outside air passes through separate outside air filters or combined outside/re-circulated air filters prior to entering the HVAC system cooling, dehumidification, and heating apparatus. In addition, depending on the type of service, number of station stops, and car configuration, some unprocessed outside air may also enter the passenger area directly when doors are opened. To mitigate the introduction of unprocessed outside air due to door openings, drafts, and air infiltration due to door seal leakage when the vehicle is in motion, a positive pressurization of 12.46Pa to 37.37Pa is normally recommended (measured at static conditions). Pressurization is achieved naturally by maintaining a well-sealed car body structure and environmental seals (outside doors, windows, etc.), such that sufficient static pressure is created by the supply of outside air and its restricted natural exhaust leakage through small gaps and resilient seals. On particularly well sealed vehicles it may be necessary to add exhaust grilles (active or static as the application demands).

2.2.2.2. Variable Ventilation Control

Recent increased emphasis on reducing energy consumption plus the improvement in design and reliability of variable speed motor designs and variable damper controls have now led to these features being specified in some new vehicle contracts. Some control schemes being considered are indirect means of matching ventilation airflow to passenger load using load (weight) sensors or electronic passenger counting systems. Other more direct means such as CO_2 concentration sensors are also under consideration. More research on the effectiveness of variable airflow control schemes is needed before they are widely accepted by the industry.

The use of automatic powered dampers has, however, been adopted on many newer designs based on the ability to expeditiously shut off outside air from the vehicles by the operator if smoke or other hazardous exterior conditions are encountered. Another benefit of controlled powered dampers is to temporarily throttle-off outside air during car start-up to more quickly bring the car temperature to the proper interior temperature after an extended layover period where the HVAC system is shut down.

2.2.2.3. Air Exhaust

The oversupply of air in most rail passenger vehicles resulting from the introduction of outside ventilation air is normally exhausted naturally by the resulting positive pressurization and ex-filtration through the small gaps around structural members and outside door and window seals. Intercity coaches and similar vehicles that have few exterior doors often require static exhaust grilles to relieve excessive pressurization (Excessive pressurization can unnecessarily require higher fan power ratings and energy consumption to maintain the required outside airflow).

In addition, in localized areas where concentrated heat is generated (equipment lockers, food galleys, etc.), or objectionable odors may exist (toilets, food preparation areas, etc.), local forced exhaust fans are recommended to direct this undesirable air away and directly to the outside without adversely affecting the surrounding passenger environment. When forced exhaust is required, care must be taken to ensure that sufficient outside air is introduced to maintain positive pressurization in the passenger section.

2.2.3. Capacity Calculations

2.2.3.1. Introduction

Air conditioning loads arise from energy that flows in to a passenger coach through its envelop, solar gain through windows, infiltration and ventilation which brings in outside air that needs to be cooled and/or dried, plus heat and moisture that are generated within the compartment.

An accurate assessment of rail passenger vehicle heating and cooling loads is required for the selection of HVAC equipment that provides acceptable system performance and comfort.

Compared to buildings and other human habitats fixed in a given geographical position, rail passenger vehicles are often subject to significant environmental variation as a result of their motion. These variations include tunnels, changes in elevation, orientation with respect to solar heat gains, heat transfer effects due to wind, etc. These environmental influences affect not only the heating and cooling loads, but also the performance of the HVAC equipment itself.

With the possible exception of building passenger elevators, at design capacity no other form of public transportation exhibits the same degree of passenger loading per unit volume as subway and intra-city rail passenger vehicles. Determining the effect of passenger loading on system performance begins with capacity calculations.

2.2.3.2. Climatic and Environmental Design Criteria

The actual climatologically data conditions were taken from Ethiopian Metrology Service Agency throughout the track alignment but we take the critical hottest region.

a. Cooling Ambient Design Conditions

Cooling dry-bulb temperature/mean coincident wet-bulb temperature data are typically used to determine the required cooling capacity of rail HVAC equipment. The design cooling capacity may be influenced by interior temperature pull-down requirements.

b. Heating Ambient Design Conditions

Heating dry-bulb temperature data is used for determining the required heating capacity of HVAC equipment. The design heating capacity may be influenced by interior temperature pull-up requirements.

c. Street/Roadbed Temperatures

If applicable to the specific project, the designer should account for heat gain due to thermal radiation from the street or roadbed to the under-frame and lower sidewalls of the vehicle and resultant localized elevated DB temperatures.

Key factors to consider when this level of accuracy is deemed necessary include:

- Elevated street/roadbed temperature due to solar heating.
- Short distance from street/roadbed to under-frame.
- High surface thermal emissivity.

The net heat flux between the two surfaces depends on the thermal emissivity of the exposed exterior floor area and roadbed, surface areas of the roadbed and floor area, temperatures of the two surfaces, and "view factor" of the two surfaces. Typical thermal emissivity of both surfaces is approximately 0.9.There may be combinations of vehicle design, operation, and environmental conditions that reduce these additional cooling loads to insignificance; however, the designer should consider all of these factors and make an informed judgment as to how they should be addressed in the cooling load assumptions.

d. Roof-Mounted and Under frame-Mounted Equipment

Localized under-frame and roof temperatures may be higher than those calculated from climatological data due to heat rejection from adjacent equipment. These include propulsion motors, braking resistor banks, friction braking systems, HVAC condensing units, and other heat-generating equipment. The effect of these combined convective and radiative heat gains must be accounted for as required.

e. Elevation

Reduced air density (increased specific volume) at higher elevations can have a significant effect on the volumetric airflow rate necessary to achieve the desired interior conditions. These effects are usually disregarded at elevations less than 304.8 meters. The use of psychrometric software that includes elevation in the calculation algorithms is recommended.

2.2.4. Car body Heat Gain

2.2.4.1. Conductive Heat Gain

Where car body heat transfer data are available from vehicle climate room (CR) tests, these data generally do not include the effects of solar radiation. Typical testing involves maintaining the environmental chamber at a constant (low) temperature, with the vehicle interior heated by auxiliary electric heaters or the vehicle's sidewall/floor heating system. Air-moving devices may be used to simulate wind effects on exterior surfaces due to vehicle motion. At thermal equilibrium, the car body heat loss per unit temperature difference (UA factor) is calculated from the exterior/interior temperature differential and heater power input. The UA factor includes heat loss due to glazing as well as opaque surfaces.

2.2.4.2. Solar Radiation Heat Gain

The total heat gain through glazing is the sum of the solar radiant heat gain and the conductive heat gain due to exterior/interior temperature differentials.

2.2.4.3. Car body Conductive Heat Gain

a. Empirical Analysis

Empirical data from environmental testing may be available to the car builder for use in calculating car body heat transfer. Unknown or partially-known variables such as thermal bridging, thermal energy flow paths, workmanship effects, thermal energy storage effects, sealing effectiveness, material properties, glazing performance, etc. may lead to assumptions that make computational analysis less accurate. Typical values of UA factor per unit length are provided in Table 1 for various rail passenger vehicle constructions and applications.

Table 1: Car body Thermal Performance for Various Vehicle Types [27]

[Static Conditions (average exterior air velocity < 8kmph)]

Application or Vehicle Type	**UA Factors per meter of car length [$\frac{w}{mk}$]**
Inter-city and Commuter Car, Single Level, Stainless Steel Car body	18
Subway Car, Stainless Steel Carboy	23
Commuter Car, Single Level, Aluminium Carboy	23
Bi- level commuter car, stainless Steel Car body	24.5
Light Rail Vehicle, Steel Carboy	24.5
Light Rail Vehicle, Double-Articulated, Composite-Clad Car body	24.5
Light Rail Vehicle, Aluminium Car body	24.5
Bi-Level Commuter Car, Aluminium Carboy	33

Using data from Table 1 the UA factor for a specific vehicle is calculated as the product of the UA factor per meter of car length and the vehicle length. Car body conductive heat gain or loss at design temperatures is the product of the UA factor and the exterior/interior temperature differential.

b. Computational Analysis

If empirical data are not available, or if a computational analysis is desired, the car body conductive heat gain can be calculated using the heat transfer equation:

$$\dot{Q} = \mathrm{UA}(\Delta\mathrm{T}) \quad (1)$$

Where:

$\dot{Q}$ - Thermal energy transfer rate [w]

U – Car body thermal transmittance (includes exterior/interior convective air film coefficients) [$\frac{w}{m^2k}$]

A – Car body thermal envelope area [m^2]

ΔT -exterior to interior temperature differential [K]

This equation assumes an average thermal transmittance for the entire car body and uniform interior and exterior temperatures. The thermal envelope area (interior or exterior) may be determined by direct measurement or taken from drawings/3D models. In practice, vehicle thermal transmittance is a summation of individual transmittance values for specific opaque and glazed sections, weighted by their respective areas. The recommended use of the values in Table 1, can be adjusted as required for specific vehicle construction or car builder experience. Car body glazing and opaque surface solar heat gain should be accounted.

2.2.4.4. Car body Solar Radiation Heat Gain

To account for the heating effects of solar radiation on opaque surfaces, the sol-air temperature is calculated and used in the heat transfer equation. As stated in the Non-residential Cooling and Heating Load Calculations chapter of ASHRAE Handbook Fundamentals, "Sol-air-temperature is the outdoor air temperature that, in the absence of all radiation changes gives the same rate of heat entry into the surface as would the combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with outdoor air."

Once calculated for a specific date, time, surface orientation, and other pertinent variables, the sol-air temperature is used to determine the instantaneous heat gain of the surface.

2.2.4.5. Glazing Conductive Heat Gain

a. General Considerations

If the vehicle UA factor is known or is to be estimated using data from Table 1, the UA factor includes conductive heat gain through glazing.

The glazing conductive heat gain may be calculated from the thermal characteristics of the glazing and the assumed values of the exterior and interior convective air film coefficients. Information from the glazing supplier should be reviewed to determine appropriate thermal conductivity or conductance values.

Glazing conductive heat gain is calculated in the same manner as car body conductive heat gain. Dual-pane windows and other insulated glazing units are increasingly specified for use in environments subject to temperature extremes; and when energy conservation, condensation resistance, glazing acoustic performance, and enhanced thermal comfort are important considerations.

b. Exterior Convective Air Film Coefficient [h_o]

Glazing conductive heat gain calculations include exterior convective air film coefficients representative of those encountered during revenue service. One simple method is to assume the design exterior wind speed to be equal to the average vehicle speed attained in service.

The convection coefficients *ho* and *hi* depend on the orientation of the surface and the air /wind velocity. Here, the following estimation is used to estimate the convection heat transfer coefficients as a function of vehicle speed [27].

$$h_o = 0.6 + 6.64\sqrt{V} \quad \text{.......} \quad (2)$$

Where *h* is the convection heat transfer coefficient in [$\frac{w}{m^2\,k}$] *and V* is the vehicle speed in *m/s*.

c. Interior Convective Air Film Coefficient [hi]

Glazing thermal-performance empirical testing and standardized computational procedures typically include rating at one or more interior convective air film coefficient conditions.

A value of h_i = 8.3[$\frac{w}{m^2\,k}$] is recommended for both heating and cooling load calculations.

d. Glazing Thermal Conductivities

A typical rail passenger vehicle window is fabricated from soda-lime-silica float glass, with a thermal conductivity of approximately $1.0\frac{W}{mK}$. Polycarbonate may be included in the glazing system to provide additional impact resistance; this material has a thermal conductivity of approximately $0.2\frac{W}{mK}$. Glazing is typically of laminated construction, with a polyvinyl butyral (PVB) or thermoplastic polyurethane (TPU) interlayer. To meet penetration-resistance requirements, windshields in particular may have more than one interlayer. PVB and TPU interlayer thermal conductivities are approximately $0.2\frac{W}{mK}$.

2.2.4.6. Glazing Solar Radiation Heat Gain

General Considerations:

Glazing solar radiation heat gain consists of two main parts: (1) solar radiation transmitted directly through the glazing and (2) solar radiation absorbed by the glazing and this energy subsequently transferred to the interior by conduction, convection, and thermal radiation.

The metric typically used to quantify solar radiation heat gain is the solar heat gain coefficient (SHGC). The SHGC denotes the fraction of incident solar radiant energy that enters the vehicle interior by all heat-transfer mechanisms.

The total incident solar radiation is the sum of direct, sky diffuse, and diffuse ground-reflected solar radiation.

An older metric used to quantify solar radiation heat gain is the shading coefficient (SC). The SC is similar (though not identical) to the SHGC and may be used in vehicle glazing heat gain calculations.

Information from the glazing supplier should be reviewed to determine appropriate values for use in glazing solar radiation heat gain calculations.

2.2.5. Passenger and Operator Cooling Loads

Passenger/operator sensible and latent cooling loads are often provided in transit agency technical specifications. In this case the total cooling load is the product of the individual sensible or latent load for the various activities (seated, standing, operating personnel, etc.), and the number of persons in those activities.

Table 2 is based on the *Non-Residential Cooling and Heating Load Calculations in* ASHRAE hand book chapter and provides recommended heat gain values for passengers and operators in various rail applications.

Table 2: Recommended Heat-Gain Values for Passengers and Operators

Application, Activity	Total Heat[W/person]		% Sensible Heat
	Adult Male	Adjusted,M/F	
Inter-city and commuter Car, Seated	132	117	61
Inter-city and Commuter Car, Standing	139	132	56
Subway and Light Rail Vehicle, Seated	139	132	56
Subway and Light Rail Vehicle, Standing	161	147	50
Operator's Cab, Operators	132		56

2.2.6. Outside Air (Ventilation) Cooling Loads

Cooling loads due to outside air (ventilation) can be significant, and in some cases, represent the largest single vehicle load.

With designers having almost universal access to personal computers, outside-air load calculations are usually prepared using psychrometric software.

Simplified equations that calculate total, sensible, and latent heat gains in airflow-related air-conditioning processes have been used for many years. These include:

Total heat gain = 4.5 x volumetric airflow rate x Δh (change in enthalpy)................................ (3)

Sensible heat gain = 1.10 x volumetric airflow rate x Δt (change in DB temperature)............... (4)

Latent heat gain = 4840 x volumetric airflow rate x ΔW (change in humidity ratio)................. (5)

These equations are valid for air at standard conditions only. (at sea level); For elevations other than sea level the equations above are corrected by multiplying them by the ratio of air pressure at the specific elevation divided by air pressure at sea level.

2.2.7. Passenger Door Cycling Air-Infiltration Cooling Loads

Infiltration due to passenger door cycling can represent significant cooling loads. This is most evident with vehicles that make frequent station stops on exposed alignments, such as urban trams. These loads may be insignificant for vehicles with infrequent revenue service station stops, such as commuter rail. Although these loads are transient in nature, they should be considered in the load calculations, if considered significant in a particular application, to demonstrate that the HVAC equipment has sufficient capacity to recover to the design interior temperatures during steady state operating conditions when the doors are cycled.

Empirical and computational data are available to assist the designer in quantifying air exchange rates at door opening events. Input parameters for these calculations include door size, number of doors opened, number of door-opening events per hour, time required to fully open and fully close the doors, length of time the doors are fully open, as well as pertinent psychrometric properties of interior and exterior air.

Heat gain through doorways from air exchange is calculated as follows:

$$\dot{Q}_d = \dot{Q}_t \mathrm{D_t D_f}(1 - \mathrm{E}) \quad (6)$$

Where:

$\dot{Q}_d$ -the resulting passenger door cycling cooling load [W]

$\dot{Q}_t$ - Average total (sensible + latent) heat gain [W]

D_t - doorway open-time factor/ decimal portion of time doorway is open

D_f -doorway flow factor (ratio of actual air exchange to fully established flow)

E -effectiveness of doorway protective device (0 for typical rail passenger vehicle doorways without air Curtains or other devices to minimize infiltration)

The following equation is used to determine average heat gain ($\dot{Q}_t$):

$$\dot{Q}_t = \mathrm{A_d}(\mathrm{hi} - \mathrm{hr})\rho_r\left(1 - \frac{\rho_i}{\rho_r}\right)^{0.5}(\mathrm{gH_d})^{0.5}\ \mathrm{Fm} \quad (7)$$

Where:

$\dot{Q}_t$ - total sensible and latent load [w]

A_d -doorway area [m^2]

h_i - enthalpy of infiltration air [$\frac{kJ}{kg}$]

h_r - enthalpy of refrigerated air[$\frac{kJ}{kg}$]

ρ_i - density of infiltration air [$\frac{kg}{m^3}$]

ρ_r - density of refrigerated air [$\frac{kg}{m^3}$]

g - gravitational constant [$\frac{m}{s^2}$]

H_d - doorway height [m]

F_m - density factor

$$F_m=[\frac{2}{(1+(\rho_r/\rho_i)^{1/3})}]^{1.5} \quad (8)$$

To account for multiple door openings at specified intervals, the following factor is introduced.

$$D_t = \frac{(P\Theta_p + 60\Theta_o)}{3600\Theta_d} \quad (9)$$

where:

D_t - decimal portion of time doorway is open

P - number of doorway passages

Θ_p - door open-close time, seconds per passage

Θ_0 - Time door simply stands open,

Θ_d - daily (or other) time period, h

2.2.8. Interior Lighting Heat Gains

Essentially, all the energy supplied for lighting is eventually given off as heat to the interior of the car. From the *Non-Residential Cooling and Heating Load Calculations* chapter of ASHRAE *Handbook –Fundamentals*, in general, the instantaneous rate of sensible heat gain from electric lighting may be calculated from the following equation:

$$\dot{Q}_l = \mathrm{W F_{ul} F_{sa}} \quad (10)$$

Where: - $\dot{Q}_l$ - heat gain [W]

W - Total light wattage [W]

F_{ul} - lighting use factor and,

F_{sa} - lighting special allowance factor

2.2.9. Fan and Fan Motor Heat Gains

In typical rail passenger vehicle applications, the fans and fan motor are located in the supply airstream or inside the vehicle. In this case, the sensible heat gain is the total power provided to the fan motor. Fan and fan motor sensible heat gains may be determined using the methods outlined in the Electric Motor section of the *Non-Residential Cooling and Heating Load Calculations* chapter of ASHRAE *Handbook –Fundamentals.*

2.2.10. Miscellaneous Internal Heat Gains

These sensible and latent heat gains are produced by heat-rejecting equipment and components occupying the interior spaces of rail passenger vehicles. These include door-operator equipment, communication system components, power supplies, video recorders, food service equipment, and other equipment.

The designer should determine or obtain an estimate of the heat gains from this equipment to be included in the cooling load analysis.

CHAPTER THREE

THERMAL LOAD ANALYSIS AND SYSTEM DESIGN

3.1. Thermal Load Analysis using Analytical Method

A thorough heating and cooling load analysis is fundamental to the selection of air-conditioning equipment that will provide specification-compliant performance. An accurate analysis also minimizes system over-sizing and the associated negative effects on thermal comfort, humidity control, equipment reliability, equipment maintenance, energy efficiency, and first cost.

3.1.1. Cooling Load Analysis

To do the analysis we were taken the necessary data for extreme hot climate area called Aisha through the selected route DireDawa-Djibouti from Ethiopian Meteorology Service Agency (like location ,elevation ,latitude ,altitude ,daily Mean range, and hourly dry bulb and wet bulb) and vehicle dimensions from Ethiopian Railways Corporation Rolling Stocks Specifications.

Given data:

The geographical information of the selected area called Aisha is given as:

Elevation: 721m, Latitude: $10^{0}45”$ N, Longitude: $42^{0}34”$E

And from weather data the external environmental condition is given as shown below:

DBT = Dry Bulb Temperature = 40°C;

CWB = Coincident Wet Bulb Temperature = 26.4°C;

MDR = Mean Daily Range (K) = 16.8

The main target of the design is to maintain the passenger coach internal compartment within the recommended thermal comfort zone and in this case the temperature to be maintained at 23.5°C DB and relative humidity of 50%.

Upon the information gathered from Ethiopian Railway Corporation overall vehicle type is Inter-city and commuter, two cabs, YZ25G Hard-seat Passenger Car from Ethiopian Railway corporation Rolling stocks specification [Appendix VIII], Thermal envelope dimensions (exterior): 25500mm long x 3105mm wide x 3543mm height, Passenger-compartment window and cab side-window tilt from horizontal: 90° and maximum vehicle speed is 120km/h.

Car body UA- factor for this type of vehicle per metre of car length can be taken from ASHARE recommended value as:

$$\frac{\mathrm{UA}}{\text{vehicle length}} = 24.5\frac{\mathrm{w}}{\mathrm{mk}}$$

Car body UA factor × train length = 25.5m×24.5$\frac{\mathrm{w}}{\mathrm{mk}}$= **625**$\frac{\mathbf{w}}{\mathbf{k}}$

And from equation (1) above we can calculate Car body Conductive Load (including glazing) as:

$$\dot{Q} = \mathrm{UA}(\Delta \mathrm{T})$$

$$\dot{Q} = 625\frac{w}{k} \mathrm{x}\ (40°\mathrm{c} - 23.5°\mathrm{c}) = \mathbf{10.31kw}$$

- **Car body Glazing Solar-Radiation Cooling Load calculation**

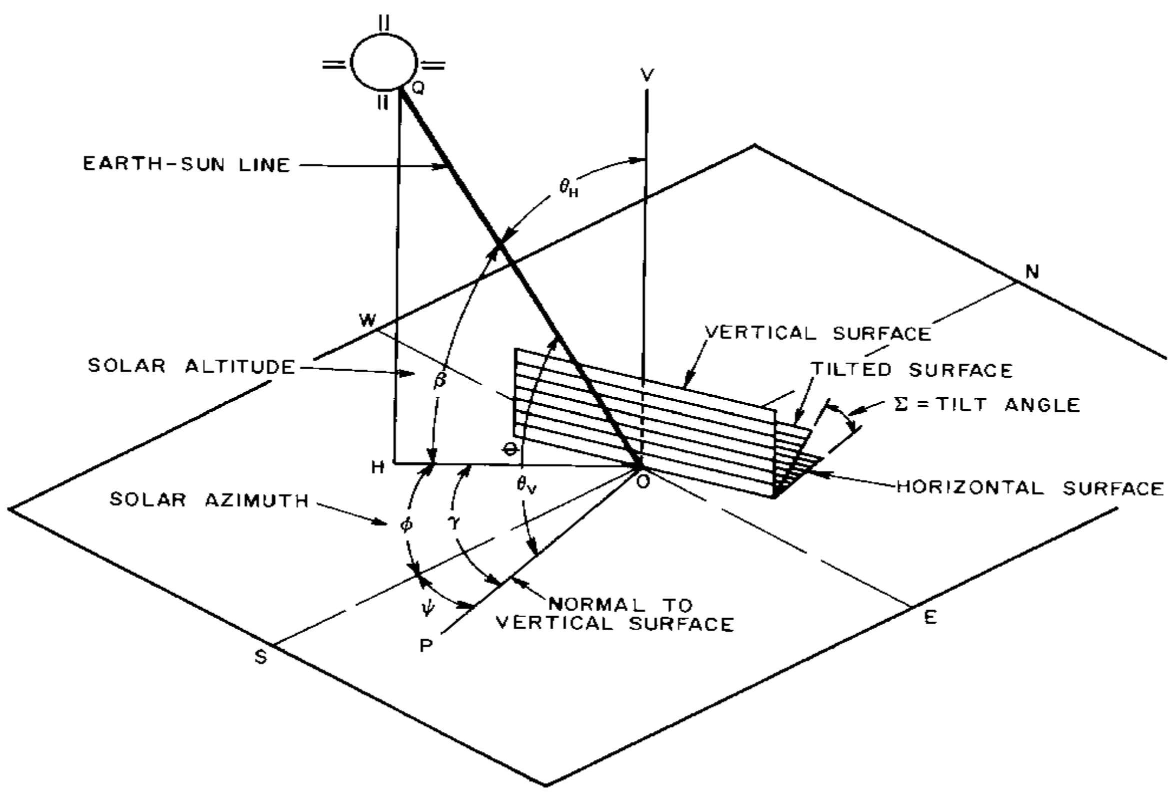


Fig 5: Solar Angles for Vertical and Horizontal Surfaces

From technical specification of the selected vehicle type above:

Side Window Area (one side only, including both cabs): = 14.22m^2

Windshield Area: = 2.9m^2

Cab side window area (per cab, one side): = 0.712m^2

June is the hottest month in Aisha according to the data available from Ethiopia Meteorology Agency.

Assume the sky to be cloudless and knowledge of clear-sky solar radiation at various times of year and day is required by several calculation methods for heat gains in HVAC loads and solar energy applications.

Direct radiation load ($\dot{Q}_{Dir}$) is given by the following formula:

$$\dot{Q}_{\text{Dir}}=\sum_{surfaces} S\tau \dot{I}_{\text{DN}}\text{Cos}\theta \quad (11)$$

Where: S – surface area of the roof exposed on direct sun ray (m^2)

τ – Transmissivity of the surfaces

I_{DN} - Direct radiation from sun on the Roof, Wall, Floor

θ - Incident angle

The commonly used models for air conditioning calculations are the one suggested by ASHRAE. According to this model, the direct radiation (I_{DN}):

$$I_{\text{DN}}= \frac{A}{\exp(\frac{B}{sin\beta})} \quad (12)$$

Where:

A: the apparent solar irradiation which is taken as 1230 W/m^2 for the months of December and January and 1080 W/m^2 for mid-summer

B: atmospheric extinction coefficient which takes a value of 0.14 for winter, 0.21 for summer.

β: The altitude angle

Reflectivity of the ground = 0.2

And declination, d is given by the following formula

$$d = 23.47 sin\frac{360(284+N)}{365} \text{.........} (13)$$

Where; N-day of the year start from January 1st

For June 29 at Aisha N=180, So

$$d = 23.47 sin\frac{360(284+180)}{365}$$

$d = \mathbf{23.26^0}$

And at 16:00 the Hour angle, h = 60^0

Tilt angel, $\sum = 0$ (horizontal surface)

Wall azimuth angle, $\xi = 0^0$ (South surface)

Wall azimuth angle, $\xi = 90^0$ (West surface)

Wall azimuth angle, $\xi = 180^0$ (North surface)

Knowing the above parameters we can calculate the altitude angle as shown below:

$$\beta = \sin^{-1}(\cos l \cos h \cos d + \sin l \sin d) \text{.........} (14)$$

$$\beta = \sin^{-1}(\cos 10.45 \cos 60 \cos 23.26 + \sin 10.45 \sin 23.26)$$

β= **31.56^0**

Therefore from equation (8) above direct radiation from sun can be calculated as:

$$I_{DN} = \frac{A}{\exp(\frac{B}{sin\beta})} = \frac{1080}{\exp(\frac{0.21}{sin 31.56})} = \mathbf{723.06}\frac{w}{m^2}$$

Direct radiation load on the roof is given by:

$(\dot{Q}_{Dir,R})$ = $S_R\ \tau_R I_{DN,R} \mathrm{Cos}\theta$ .... (15)

Where: S_R is surface area of the roof and given by; S_R =L×W= 25.5m×3.105m =79.2m^2

τ_R = is the roof surface element transmissivity = 0

Since surface element transmissivity of roof, wall and floor is zero the direct radiation through those elements becomes zero.

$(\dot{Q}_{Dir,R})= (\dot{Q}_{Dir,W}) = (\dot{Q}_{Dir,F}) =0$, b/c $\tau_R = \tau_W = \tau_F =0$

Direct, Diffuse, and Reflected radiation loads on the Glass surface:

The direct radiation load through the glass surface is given by the following formula:

$\dot{Q}_{Dir,G}$ = $S_G\ \tau_G I_{DN,R} \mathrm{Cos}\theta$ ....(16)

Where: τ_G = 0.5 given [28 & 29]

We can calculate the radiation load for all orientation and take the maximum one; in this case west orientation has the maximum load as shown below.

Wall azimuth angle, ξ = 90^0(West surface)

And solar azimuth angle is given by

$$\gamma = \sin^{-1}\left(\frac{\cos d * \sin h}{\cos\beta}\right) = \sin^{-1}\left(\frac{\cos 23.26 * \sin 60}{\cos 31.56}\right) \qquad (17)$$

$$\gamma = \mathbf{69}^{o}$$

Similarly Surface azimuth angle can be calculated by the following formula

$$\alpha = [\pi - (\gamma + \zeta] \times F; \text{ where F=1 for afternoon} \qquad (18)$$

$$\alpha = [\pi - (69 + 90)] \times 1 = \mathbf{21^0}$$

And Incident angle of sun's ray is given by the following formula

$$\theta = \cos^{-1}(\sin\beta\cos\Sigma + \cos\beta\cos\alpha\sin\Sigma) \qquad (19)$$

For vertical surfaces $\Sigma = 90^{0,}$ therefore the incident angle becomes

$$\theta_{ver} = \cos^{-1}(\cos 31.56 \cos 21) = \mathbf{37.3^0}$$

a. ***Direct Radiant Load ($\dot{Q}_{Dir}$)***

$$\dot{Q}_{Dir,G} = S_G\, \tau_G I_{DN} \cos\theta_{ver}$$

$$= 14.22 * 0.5 \times 723.06 \cos 37.3$$

$$\dot{Q}_{Dir,G} = \mathbf{4089.5W}$$

b. ***Diffuse Radiant Load ($\dot{Q}_{Dif}$)***

$$\dot{Q}_{Dif} = \sum_{surfaces} S\tau \dot{I}_D \,;$$

Where diffuse radiation (I_D) is given by

$I_D = C\, I_{DN}\, F_{WS}$ (w/m^2)

C-constant=0.135 for mid-summer

=0.058 for winter

F_{WS} = view factor $= \frac{1+\cos\Sigma}{2}$

Σ=tilt angle and its value will be 0 for horizontal surfaces and 90^0 for vertical surfaces this implies

F_{WS} = 1, for horizontal surfaces and

F_{WS} = 0.5, for vertical surfaces

Therefore;

$I_D = 0.135 \times 0.5 \times 723.06 = 48.81$ w/m^2

$$\dot{Q}_{Dif} = \sum_{surfaces} S\tau \dot{I}_D$$

$= 14.22 \times 0.5 \times 48.81 = \mathbf{347.04w}$

c. Reflected radiation loads

$$\dot{Q}_{\text{Ref}} = \sum_{surfaces} S\tau \, \dot{I}_{\text{Ref}}$$

$\dot{I}_{\text{Ref}} = (i_{DIR} + i_D) \times \rho_g \times F_{WG}$

ρ_{g} = ground reflectivity factor = 0.2

Angle factor; $F_{WG} = \frac{1-\cos\Sigma}{2}$

$F_{WG} = 0$; for horizontal surfaces and

$F_{WG} = 0.5$; for vertical surfaces

$\dot{I}_{\text{Ref}} = (723.06 + 48.81) \times 0.2 \times 0.5 = 77.19\text{w/m}^2$

$\dot{Q}_{\text{Ref}} = 14.22 \times 0.5 \times 77.19 = \mathbf{548.82W}$

In a similar manner we can calculate radiation load at the face of the train as shown below:

Direct Radiant Load ($\dot{Q}_{Dir}$) is given by:

$$\dot{Q}_{\text{Dir,G}} = S_G \, \tau_G I_{DN,R} \text{Cos}\theta_{\text{ver}}$$

$$= 2.9 \times 0.5 \times 723.06 \cos 37.3$$

$$\dot{Q}_{\text{Dir,G}} = \mathbf{834W}$$

Diffuse Radiant Load ($\dot{Q}_{Dif}$) also calculated by:

$$\dot{Q}_{\text{Dif}} = \sum_{surfaces} S\tau \dot{I}_{\text{Dif}}$$

$$= 2.9 \times 0.5 \times 48.81 = \mathbf{70.77W}$$

And finally the reflected radiation loads through the face of the train estimated as

$$\dot{Q}_{\text{Ref}} = \sum_{surfaces} S\tau \, \dot{I}_{\text{Ref}}$$

$=2.9\times0.5\times77.19=$ **111.93W**

To get the maximum radiant load on the surface take the largest value at which orientation of the sun having higher: In this case west orientation has the highest value.

$\dot{Q}_{\text{Dir}}=\sum_{surfaces} S\tau\dot{I}_{\text{DN}}\cos\theta_{\text{ver}} = 4089.5\text{W} + 834\text{W} = 4923.5\text{W}$

$\dot{Q}_{\text{Dif}}=\sum_{surfaces} S\tau\dot{I}_{\text{Dif}} = 347.04\text{W} + 70.77\text{W} = 417.81\text{W}$

$\dot{Q}_{\text{Ref}}=\sum_{surfaces} S\tau\dot{I}_{\text{ref}} = 548.82\text{W} + 111.93\text{W} = 660.75\text{W}$

$\dot{Q}_{\text{R}} = \dot{Q}_{\text{Di}} + \dot{Q}_{\text{Dif}} + \dot{Q}_{\text{Ref}} = 4923.5\text{W} + 417.81\text{W} + 660.75\text{W}$

$\dot{Q}_{\text{R}} = 6002.06\text{W}$ **= 6.00206kW**

Total Car body Glazing Solar-Radiation Instantaneous Cooling Gain (Load) = **6.002kw**,

Car body Opaque Surfaces Solar-Radiation Cooling Load:

In some rail passenger vehicle applications, the effects on the design cooling load of exterior-surface heating by solar radiation may need to be considered.

As calculated before Carbody UA-factor = $625\frac{\text{w}}{\text{k}}$

Thermal envelope area (exterior) = (2 x 25.5 x3.543) + (2 x25.5 x 3.105) + (2 x3.543 x3.105)

$= 361.05\text{m}^2$

UA = $625\frac{\text{w}}{\text{k}}$

Car body average thermal transmittance (U) $=\frac{625}{361.05} = 1.73\frac{\text{w}}{\text{m}^2\text{k}}$

Total glazing area = $(2 \times 14.22\text{m}^2) + (2 \times 2.9\text{m}^2) = 34.24\text{m}^2$

Glazing average U-factor (thermal transmittance) is assumed to be $5.394\frac{\text{w}}{\text{m}^2\text{k}}$ for the glazing of this vehicle. Therefore average opaque-surface transmittance can be calculated as shown below.

$(U_{\text{glazing}}$ x glazing area$) + (U_{\text{opaque surfaces}}$ x opaque surface area$) = (U_{\text{total}}$ x carbody area$)$

Solving for $U_{\text{opaque surfaces}}$:

$U_{\text{opaque surfaces}}$ = (U_{total} x carbody area) - (U_{glazing} x glazing area) / opaque surface area

$$U_{\text{opaque surfaces}} = \frac{[\,(1.73\frac{w}{m^2k}\times 361.05\text{m}^2) - (5.394\,\frac{w}{m^2k}\times\ 34.24\text{m}^2\,)]}{(361.05\text{m}^2 - 34.24\text{m}^2)} = 1.346\frac{\text{w}}{\text{m}^2\text{k}}$$

Sol-air temperature calculated as per the *Non-residential Heating and Cooling Load Calculations* chapter of the ASHRAE *Handbook – Fundamentals* take into account changes in ambient temperature throughout the design day. This requires the determination of hourly ambient temperatures that are used to calculate hourly sol-air temperatures. Rail vehicle cooling load calculations are typically based on a single design ambient temperature; therefore, any difference between the ASHRAE-method hourly design day temperature and design ambient temperature used for the rail vehicle must be taken into account. The sol-air temperature be calculated for June16:00 pm LST (local standard time).

Maximum opaque surface cooling gains typically occur when the vehicle sidewalls are approximately perpendicular to the sun; therefore, for these calculations, the vehicle center-line is oriented north-south.

June, 16:00 pm LST sol-air temperatures are calculated for vertical east/west surfaces (sidewalls). Sol-air temperature is an equivalent or an effective outdoor temperature that combines the effects of convection and radiation and given by:

$$T_{\text{sol-air}} = T_0 + (\alpha_D I_D + \alpha_d I_d - R)/h_0 \qquad (20)$$

Where: T_0 – outside air temperature

$\alpha_D I_D$ – direct radiation of opaque surface

$\alpha_d I_d$ – diffiuse radiation of opaque surface

R - reflected Radiation of opaque surface

h_0 – heat transfer coefficient of outside surface

A value of 0.20 is assumed for the absorptance/outer-surface heat transfer coefficient. June16:00 pm LST hourly design day temperature is 41^0c, 1^0c higher than the 40^0c cooling design temperature. The calculated sol-air temperatures must be reduced by 1^0c before determining the conductive gain. Sol-air temperature is based on the design cooling ambient temperature.

Instantaneous cooling gain [$\dot{Q} = \mathrm{UA}(\Delta\mathrm{T})$]

North: $1.346\frac{w}{m^2k}$x [(3.543 x3.105) – 2.9)] m^2 x (21.2) °C = 231.16w

South : $1.346\frac{w}{m^2k}$x [(3.543 x3.105) – 2.9)] m^2 x (21.2)°C = 231.16w

East: $1.346\frac{w}{m^2k}$x [(25.5 x3.543) – 14.22)] m^2 x (21.2) °C = 2172.18w

West: $1.346\frac{w}{m^2k}$ x [(25.5 x3.543) – 14.22)] m^2 x (21.2) °C = 2172.28w

*Total opaque-surface instantaneous cooling gain = **4.8kw***

Passenger and Operator Cooling Loads:

Inter-city and commuter seated passenger/operator sensible and latent cooling loads are taken from ASHRAE hand book.

Table 3: Inter-city and commuter Rail Vehicle Passenger/Operator Sensible and Latent Cooling Loads

	Seated	**Operator**
Number of occupants	118	1
Individual sensible Load [w/person]	71.37	73.92
Individual Latent Load [w/person]	45.63	58.08
Individual Total Load [w/person]	117	132

Therefore, from the above table we can get the total sensible and latent cooling load as shown below.

Table 4: Total occupant cooling load summary

	Sensible	**Latent**	**Total**
Total Occupant Cooling Load [kw]	8.5	5.44	13.94

- ***Ventilation (Outside-Air) Load***

Take ventilation rate as 8 l/s/person at cooling-capacity design conditions.

Total number of occupants: 119

The ventilation rate is 119persons x 8 l/s/person = 952 l/s = 0.952m^3/s

To go through the ventilation load calculation first we draw the psychometric chart at the given elevation (h=721m), as shown in the annex VII.

From psychometric chart:

For inside conditions: 23.5°C (DBT) and 50% RH:

$\omega_i = 0.01$ kg/kg, $h_i = 48.79$ kJ/kg

For outside conditions: 40°C (DBT) and 48% RH:

$\omega o = 0.02$ kg/kg, $h_o = 103.998$ kJ/kg, Va=1.005m^3/kg

Mass flow rate of ventilated air is;

$m_v = \frac{0.952}{1.005} = 0.95$kg/s

Sensible heat transfer rate due to ventilation is given by the following formula

$\dot{Q}$s$=mCp\Delta t$;

Where; m-mass flow rate

Cp-specific heat of water vapour

$\dot{Q}$s$=0.952 \times 1.0216 \times (40 - 23.5) = \mathbf{16.01\boldsymbol{kw}}$

And latent heat transfer rate due to ventilation also calculated by

$$\dot{Q}l = mh(\omega o - \omega i) = 0.952 \times 2501 \times (0.02 - 0.01) = \mathbf{23.81\boldsymbol{kw}}$$

Where; ωo & ωi are outdoor and indoor humidity ratios

h_{fg}-latent heat of vaporization

Total; $\dot{Q}t = 16.01 + 23.81 = \mathbf{39.82\boldsymbol{kw}}$

Interior Lighting Heat Gains:

From the Non-Residential Cooling and Heating Load Calculations chapter of ASHRAE Handbook –Fundamentals, in general, the instantaneous rate of sensible heat gain from electric lighting may be calculated from the following equation:

$\dot{Q}\mathrm{l} = \mathrm{WF_{ul}F_{sa}}$

For passenger railcar application, the lighting use factor, Ful, and special allowance factor, Fsa , can be taken as unity.

Therefore; let's take $F_{ul}= F_{sa}=1$

Fluorescent lights 50.8mm long = 30 Nos. ,

Incandescent lampas in the train = 16 Nos,

Fluorescent Light = 24 W (for a single Fluorescent Light)

Total Fluorescent Light = 24×30w = 720W = **0.72kW**

Incandescent lamp = 15 W (for a single Incandescent lamp)

Total Incandescent lamps = 15W×16 =240W = **0.24KW**

$\mathrm{W} = 0.96\mathrm{kW}$

$\dot{Q}l = \mathrm{WF_{ul}F_{sa}}$ = **0.96KW**

Passenger Door Cycling Air-Infiltration Cooling Loads:

From eqn (6) above heat gain through doorways from air exchange is calculated as follows:

$$\dot{Q}_d = \dot{Q}_t \mathrm{D_t D_f}(1 - \mathrm{E})$$

Where:

$\dot{Q}_d$ -the resulting passenger door cycling cooling load [W]

$\dot{Q}_t$ - Average total (sensible + latent) heat gain [W]

D_t - doorway open-time factor/ decimal portion of time doorway is open

D_f - doorway flow factor (ratio of actual air exchange to fully established flow)

E -effectiveness of doorway protective device (0 for typical rail passenger vehicle doorways without air Curtains or other devices to minimize infiltration)

And from eqn (7)

$$\dot{Q}_t = A_d(\mathrm{h_i} - \mathrm{h_r})\rho_r(1 - \frac{\rho_i}{\rho_r})^{0.5}(\mathrm{gH_d})^{0.5}\ \mathrm{F_m}$$

where:

$\dot{Q}_t$ - total sensible and latent load [w]

A_d -doorway area [m^2] = 2.9 m^2

h_i - enthalpy of infiltration air [$\frac{kJ}{kg}$] = 103.988$\frac{kJ}{kg}$

h_r - enthalpy of refrigerated air[$\frac{kJ}{kg}$] = 48.79$\frac{kJ}{kg}$

ρ_i - density of infiltration air [$\frac{kg}{m^3}$] = 1.088$\frac{kg}{m^3}$

ρ_r - density of refrigerated air [$\frac{kg}{m^3}$] = 1.173$\frac{kg}{m^3}$

g - gravitational constant [$\frac{m}{s^2}$] = 9.8$\frac{m}{s^2}$

H_d - doorway height [m] = 2m

F_m - density factor

$$\mathrm{F_m}=[\frac{2}{(1+(\rho_r/\rho_i)^{1/3})}]^{1.5} = 0.9808$$

Therefore;

$$\dot{Q}_t = A_d(\mathrm{h_i} - \mathrm{h_r})\rho_r(1 - \frac{\rho_i}{\rho_r})^{0.5}(\mathrm{gH_d})^{0.5}\ \mathrm{F_m}$$

$$= 30.62\text{kw}$$

The 30.62kw cooling load resulting from fully established flow through a single passenger door is equivalent to an air infiltration rate of approximately 732.05 l/s, which is large.

To compensate for the relatively small interior volume of rail passenger vehicles and resultant inability to provide fully established flow the entire time the passenger doors are fully opened, the door factor (D_f) is used. This factor is typically used to account for objects near the doorways that may impede airflow.

For two passenger doors opened on one side of vehicle: (D_f) = 0.5

To account for multiple door openings at specified intervals, the following factor is introduced.

From eqn (9) above; $D_t = \frac{(P\Theta_p + 60\Theta_o)}{3600\Theta_d}$

where:

D_t - decimal portion of time doorway is open

P - number of doorway passages = 2 = [60 minutes divided by 30 minutes per cycle]

Θ_p - door open-close time, seconds per passage = 6 = [3 seconds to open, 3 seconds to close]

Θ_0 - Time door simply stands open = 3.8 = [2 x (120 – (2 x 3)) s / 60]

Θ_d - daily (or other) time period, h = 1

Therefore using passenger door-opening data above:

$D_t = \frac{(2\times6 + 60\times3.8)}{3600\times1}$ = 0.067

And finally the resulting passenger door cycling cooling load is:

$\dot{Q}_d = 30.62 \times 0.067 \times 0.5 \times 1$

= **1.026kw** per door

Total cooling load for two doors; $\mathbf{\dot{Q}_d = 2 \times 1.026 = 2.052\mathit{kw}}$ Out of which 0.825kw is the sensible load and the remaining 1.227kw is the latent load.

Due to the long average distance between stations, the doors are cycled less frequently as compared with light rail vehicles.

Miscellaneous Internal Heat Gains:

These sensible and latent heat gains are produced by heat-rejecting equipment and components occupying the interior spaces of rail passenger vehicles. These include door-operator equipment, communication system components, power supplies, video recorders, food service equipment, and other equipment.

Table 5: summary of Miscellaneous Internal Heat Gains

Cooling Load	**Sensible**	**Latent**	**Total**
Miscellaneous Internal Heat Gains[kw]	1.2	0	1.2

3.1.2. Cooling Load Analysis Summary

Table 6 Cooling Load Analysis Summary

Cooling Loads	**Sensible [kw]**	**Latent .[kw]**	**Total [kw]**
Conduction (including glazing)	10.31	0	10.31
Passengers & Operator	8.5	5.44	13.94
Outside Air (ventilation)	16.01	23.81	39.82
Passenger-Door Cycling Air	0.825	1.227	2.052
Solar Radiation (glazing)	6.002	0	6.002
Solar Radiation (opaque surfaces)	2.4	0	2.4
Evaporator Fans / Motors (assumed)	2	0	2
Miscellaneous Internal and infiltration Loads(assumed)	2.74	0	2.74
Interior Lighting Heat Gains	0.96	0	0.96
Evaporator Cooling Load	49.745	30.477	80.22
Vehicle Cooling Load (Net Cooling Load) (Gross Cooling Load minus evaporator unit motor and unit internal sensible load)	47.745	30.477	78.22

3.2. Thermal Load Analysis and System Design Using Hourly Analysis Program (HAP) Software

3.2.1. Introduction

Hourly Analysis Program (HAP) is a computer tool which assists engineers in designing HVAC systems. HAP is two tools in one. First it is a tool for estimating loads and designing systems. Second, it is a tool for simulating energy use and calculating energy costs. HAP uses the ASHRAE-endorsed transfer function method for load calculations.Even though Hap has two analysis features in this thesis only using HAP System Design Features and compare the results what we have got in the previous cooling load calculation as discussed below.

3.2.2. HAP System Design Features

HAP estimates design cooling and heating loads in order to determine required sizes for HVAC system components. Ultimately, the program provides information needed for selecting and specifying equipment. Specifically, the program performs the following tasks:

- Calculates design cooling and heating loads for spaces, zones, and coils in the HVAC System
- Determines required airflow rates for spaces, zones and the system.
- Sizes cooling and heating coils.
- Sizes air circulation fans.

Before design calculations can be performed types of information needed include:

- Climate data
- Construction material data for walls, roofs, windows, doors, exterior shading devices and floors, and for interior partitions between conditioned and non-conditioned regions.
- Size and layout data including wall, roof, window, door and floor areas, exposure orientations and external shading features.
- Internal load characteristics determined by levels and schedules for occupancy, lighting systems, equipment
- Data concerning HVAC equipment, controls and components to be used.

3.2.3. Input Data needed for HAP analysis

A. Weather Data.

Weather data defines the temperature, humidity and solar radiation conditions encounters during the course of a year. These conditions play an important role in influencing loads and system operation. To define weather data, a city can be chosen from the program's weather database, or weather parameters can be directly entered. Weather data is entered using the weather input form.

B. Space Data

To define a space, all elements which affect heat flow in the space must be described. Elements include walls, windows, doors, roofs, skylights, floors, occupants, lighting, electrical equipment, miscellaneous heat sources, infiltration, and partitions.

While defining a space, information about the construction of walls, roofs, windows, doors and external shading devices is needed, as well as information about the hourly schedules for internal heat gains. This construction and schedule data can be specified directly from the space input form (via links to the construction and schedule forms), or alternately can be defined prior to entering space data.

C. Input Air System Data

An air system serves one or more zones. Zones are groups of spaces having a single thermostatic control. Examples of systems include central station air handlers, packaged rooftop units, packaged vertical units, split systems, packaged DX fan coils, hydraulic fan coils and water source heat pumps. In all cases, the air system also includes associated ductwork, supply terminals and controls. To define an air system, the components, controls and zones associated with the system must be defined as well as the system sizing criteria.

Input data into the HAP is summarized as shown below:

Design Parameters:

City Name .... **Aisha**
Location .... **Ethiopia**
Latitude .... **10.5** Deg.
Longitude .... **42.3** Deg.
Elevation .... **721.0** m
Summer Design Dry-Bulb .... **40.0** °C
Summer Coincident Wet-Bulb .... **26.4** °C
Summer Daily Range .... **16.8** °K
Winter Design Dry-Bulb .... **26.0** °C
Winter Design Wet-Bulb .... **15.6** °C
Atmospheric Clearness Number **1.00**
Average Ground Reflectance **0.20**
Soil Conductivity .... **1.000** W/ (m-°K)
Consider Daylight Savings Time .... **No**
Daylight Savings Begins .... **January, 2**
Daylight Savings Ends .... **December, 26**
Simulation Weather Data .... **noneN/A**
Current Data is .... **User Modified**
Design Cooling Months .... **January to December**

Design Day Maximum Solar Heat Gains

(The MSHG values are expressed in W/m²)

Month	**N**	**NNE**	**NE**	**ENE**	**E**	**ESE**	**SE**	**SSE**	**S**
January	99.4	99.4	216.8	475.2	688.6	793.9	776.0	662.8	548.2
February	106.1	106.1	344.9	575.5	737.4	786.3	713.7	543.3	390.3
March	114.2	202.8	473.1	653.9	739.0	723.3	598.1	371.7	205.2
April	127.1	351.8	563.5	685.1	701.8	611.1	432.5	189.1	124.9
May	204.0	441.2	608.6	684.9	657.3	530.6	318.2	125.8	124.6
June	252.6	474.4	617.0	668.8	624.6	486.9	267.9	125.6	122.4
July	212.8	442.1	599.7	667.4	637.1	513.6	304.7	128.3	126.6
August	81.3	208.3	325.5	392.0	401.8	353.3	250.2	109.1	78.2
Septemb	71.2	117.2	274.1	380.6	429.0	411.7	342.5	219.3	125.2
October	109.1	109.1	337.6	576.5	718.8	749.7	683.6	527.7	382.8
Novemb	101.1	101.1	211.3	490.9	677.2	765.7	764.2	646.5	543.5
Decemb	96.4	96.4	167.3	446.1	650.2	781.3	791.6	696.4	600.4
Month	**SSW**	**SW**	**WSW**	**W**	**WNW**	**NW**	**NNW**	**HOR**	**Mult**
January	662.7	776.3	793.7	687.9	475.2	217.0	99.4	834.5	1.00
February	541.1	713.9	782.3	730.0	580.6	348.6	106.1	904.8	1.00
March	369.6	599.2	727.7	745.8	650.1	467.6	202.6	933.9	1.00
April	185.4	442.1	623.7	709.4	688.0	563.4	345.1	910.7	1.00
May	125.7	323.4	538.5	664.5	692.0	616.0	447.4	878.0	1.00
June	125.5	272.3	496.5	636.2	682.4	627.1	479.4	855.2	1.00
July	128.3	309.1	519.9	644.6	674.7	605.2	445.4	859.5	1.00
August	107.6	253.3	359.0	409.9	399.8	330.5	207.7	531.8	0.60
Septemb	218.8	343.6	420.9	434.5	381.8	270.7	112.1	545.3	0.60
October	528.9	679.2	756.1	722.0	576.5	333.4	109.1	892.6	1.00
Novemb	650.7	758.4	773.2	685.3	490.6	202.1	101.1	836.2	1.00
Decemb	697.2	778.3	783.8	667.2	444.6	162.0	96.4	803.7	1.00

Mult. = User-defined solar multiplier factor.

Fig.6: Design Weather Parameters & MSHGs

1. General Details:
Floor Area .... **79.2** m²
Avg. Ceiling Height .... **2.5** m
Building Weight .... **346.7** kg/m²
1.1. OA Ventilation Requirements:
Space Usage .... **TRANSPORTATION: Vehicles**
OA Requirement 1 .... **8.0** L/s/person
OA Requirement 2 .... **0.00** L/(s-m²)
Space Usage Defaults .... **ASHRAE Std 62-2001**
2. Internals:
2.1. Overhead Lighting:
Fixture Type .... **Recessed (Unvented)**
Wattage **960.0**
....Watts
Schedule .... **Lighting**
2.2. Task Lighting:
Wattage **0.00**
....W/m²
Schedule .... **None**
2.3. Electrical Equipment:
Wattage .... **480.0** Watts
Schedule .... **None**

2.4. People:
Occupancy **119.0**People
Activity Level .... **Seated at Rest**
Sensible **67.4**W/person
Latent **35.2**W/person
Schedule .... **passeng**
2.5. Miscellaneous Loads:
Sensible .... **1200** W
Schedule .... **Miscellaneous**
Latent .... **0** W
Schedule .... **None**

3. Walls, Windows, Doors:

Exp.	Wall Gross Area (m²)	Window 1 Qty.	Window 2 Qty.	Door 1 Qty.
W	90.3	16	0	0
E	90.3	14	0	2
N	11.0	1	0	0
S	11.0	1	0	0

3.1. Construction Types for Exposure W
Wall Type .... **Light Weight Wall**
1st Window Type .... **side window**
3.2. Construction Types for Exposure E
Wall Type .... **Light Weight Wall**
1st Window Type .... **side window**
1st Window Shade Type .... **casement window reveal**
Door Type .... **door**
3.3. Construction Types for Exposure N
Wall Type .... **Light Weight Wall**
1st Window Type .... **front and rear window**
3.4. Construction Types for Exposure S
Wall Type .... **Light Weight Wall**
1st Window Type .... **front and rear window**
4. Roofs, Skylights:

Exp.	Roof Gross Area (m²)	Roof Slope (deg.)	Skylight Qty.
H	79.2	0	0

4.1. Construction Types for Exposure H
Roof Type .... **Roof**
5. Infiltration:
Design Cooling .... **0.50** ACH
Design Heating .... **0.00** L/s
Energy Analysis .... **0.00** L/s
Infiltration occurs only when the fan is off.
6. Floors:
Type .... **Floor Above Conditioned Space**

7. Partitions:
(No partition data).

Fig 7: Space Input Data

door

Door Details:
Gross Area .... **2.9** m²
Door U-Value .... **1.703** W/(m²-°K)

Glass Details:
Glass Area .... **1.0** m²
Glass U-Value .... **5.340** W/(m²-°K)
Glass Shade Coefficient **0.880**
Glass Shaded All Day? .... **No**

Fig 8: Door Constructions

side window

Window Details:
Detailed Input .... **Yes**
Height .... **0.56** m
Width .... **0.61** m
Frame Type .... **None**
Internal Shade Type .... **None**
Overall U-Value .... **3.237** W/(m²-°K)
Overall Shade Coefficient **0.887**

Glass Details:
Gap Type .... **6mm Air Space**

Glazing	Glass Type	Transmissivity	Reflectivity	Absorptivity
Outer Glazing	3mm clear	0.841	0.078	0.081
Glazing #2	3mm clear	0.841	0.078	0.081
Glazing #3	not used	1.000	0.000	0.000

Fig.9: Window Constructions

Light Weight Wall

Wall Details
Outside Surface Color .... **Light**
Absorptivity **0.450**
Overall U-Value .... **3.018** W/(m²-°K)

Wall Layers Details (Inside to Outside)

Layers	Thickness Mm	Density kg/m³	Specific Ht. kJ / (kg - °K)	R-Value (m²-°K)/W	Weight kg/m²
Inside surface resistance	0.000	0.0	0.00	0.12064	0.0
stainless steel	0.600	7900.0	0.50	0.00001	4.7
Epoxy	60.000	41.5	0.02	0.15206	2.5
stainless steel	0.800	7833.0	0.50	0.00002	6.3
Outside surface resistance	0.000	0.0	0.00	0.05864	0.0
Totals	**61.400**	**-**		**0.33137**	**13.5**

Fig.10: Wall Constructions

Roof

Roof Details

Outside Surface Color .... **Light**
Absorptivity **0.450**
Overall U-Value .... **2.177** W/(m²-°K)

Roof Layers Details (Inside to Outside)

Layers	Thickness Mm	Density kg/m³	Specific Ht. kJ / (kg - °K)	R-Value (m²-°K)/W	Weight kg/m²
Inside surface resistance	0.000	0.0	0.00	0.12064	0.0
stainless steel	6.000	7900.0	0.50	0.00001	47.4
Air space	0.000	0.0	0.00	0.16026	0.0
Epoxy	40.000	41.5	0.02	0.00088	1.7
balsa wood	19.050	544.6	1.21	0.16428	10.4
stainless steel	8.000	7900.0	0.50	0.00010	63.2
Outside surface resistance	0.000	0.0	0.00	0.01324	0.0
Totals	**73.050**	**-**		**0.45941**	**122.6**

Fig.11: Roof Constructions

1. General Details:

Air System Name .... **packaged roof top AHU**
Equipment Type .... **Packaged Rooftop Units**
Air System Type .... **Single Zone CAV**
Number of zones **1**

2. System Components:

Ventilation Air Data:

Airflow Control .... **Constant Ventilation Airflow**
Ventilation Sizing Method .... **ASHRAE Std 62-2001**
Unocc. Damper Position .... **Closed**
Damper Leak Rate .... **2** %
Outdoor Air CO2 Level .... **400** ppm

Central Cooling Data:

Supply Air Temperature .... **14.5** °C
Coil Bypass Factor **0.028**
Cooling Source .... **Air-Cooled DX**
Schedule .... **JFMAMJJASOND**
Capacity Control .... **Temperature Reset by Outdoor Air Schedule**
Max. Supply Temperature .... **15.0** °C
OAT for Min. Supply Temperature .... **30.0** °C
OAT for Max. Supply Temperature .... **-1.1** °C

Supply Fan Data:

Fan Type .... **Forward Curved**
Configuration .... **Draw-thru**
Fan Performance .... **774** Pa
Overall Efficiency .... **50** %

Duct System Data:

Supply Duct Data:

Duct Heat Gain .... **5** %
Duct Leakage .... **5** %

Return Duct or Plenum Data:

Return Air Via .... **Return Air Plenum**
Wall Heat Gain to Plenum .... **20** %
Roof Heat Gain to Plenum .... **70** %
Lighting Heat Gain to Plenum .... **30** %

3. Zone Components:

Space Assignments:

Zone 1: Zone 1	
Passenger coach	x1

Thermostats and Zone Data:

Zone	Cooling T-Stat Occ. (°C)	Cooling T-Stat Unocc. (°C)	Heating T-Stat Occ. (°C)	Heating T-Stat Unocc. (°C)	T-Stat Throttling Range (°C)	Diversity Factor (%)	Direct Exhaust Airflow (L/s)	Direct Exhaust Fan (kW)
1	23.0	29.4	21.0	15.0	1.67	100	0.0	0.0

Thermostat Schedule .... **thermsta**
Unoccupied Cooling is .... **Not Available**

Supply Terminals Data:

Zone	Terminal Type	Min. Airflow	Fan Performance	Fan Efficiency	Design Supply Temperature
1	Diffuser	8.00 L/s/person	-	-	-

Zone Heating Units:
Zone .... **All**
Zone Heating Unit Type .... **None**

Zone Unit Heat Source .... **Electric Resistance**
Zone Heating Unit Schedule .... **JFMAMJJASOND**

4. Sizing Data (Computer-Generated):
System Sizing Data:
Cooling Supply Temperature .... **14.5** °C
Supply Fan Airflow .... **2582.6** L/s
Ventilation Airflow .... **952.0** L/s

Hydronic Sizing Specifications:
Chilled Water Delta-T .... **5.6** °K
Hot Water Delta-T .... **11.1** °K

Safety Factors:
Cooling Sensible .... **10** %
Cooling Latent .... **10** %
Heating .... **10** %

Zone Sizing Data:
Zone Airflow Sizing Method .... **Peak zone sensible load**
Space Airflow Sizing Method .... **Zone L/(s-m²)**

Zone	Supply Airflow (L/s)	Zone Htg Unit (kW)	Reheat Coil (kW)	- (L/s)
1	2582.6	-	-	

5. Equipment Data
Central Cooling Unit - Air-Cooled DX
Estimated Maximum Load .... **74.2** kW
Design OAT .... **41.0** °C
Gross Cooling Capacity .... **50.0** kW
Compressor & OD Fan Power .... **13.00** kW
Conventional Cutoff OAT .... **12.0** °C
Low Temperature Operation .... **Not used**

Fig.12: packaged roof top AHU Input Data

3.2.4. Output Design Data Generated

Once weather, space, air system and plant data has been entered, HAP can be used to generate system and plant design reports. Some of the result output data of design reports summarized as shown below and the rest are attached in the annex.

Air System Information

Air System Name	**packaged roof top AHU**		Number of zones	**1**	
Equipment Class	**PKG ROOF**		Floor Area	**79.2**	m²
Air System Type	**SZCAV**		Location	**Aisha, Ethiopa**	

Sizing Calculation Information

Zone and Space Sizing Method:

Zone L/s	**Peak zone sensible load**	Calculation Months	**Jan to Dec**
Space L/s	**Zone L/(s-m²)**	Sizing Data	**Calculated**

Central Cooling Coil Sizing Data

Total coil load	**74.2**	kW	Load occurs at	**Jun 1600**	
Sensible coil load	**50.1**	kW	OA DB / WB	**40.0 / 26.3**	°C
Coil L/s at Jun 1600	**2583**	L/s	Entering DB / WB	**30.2 / 21.0**	°C
Max block L/s at Jun 1600	**2583**	L/s	Leaving DB / WB	**12.7 / 12.4**	°C
Sum of peak zone L/s	**2719**	L/s	Coil ADP	**12.2**	°C
Sensible heat ratio	**0.676**		Bypass Factor	**0.028**	
m²/kW	**1.1**		Resulting RH	**54**	%
W/m²	**936.4**		Design supply temp.	**14.5**	°C
Water flow @ 5.6 °K rise	**N/A**		Zone T-stat Check	**1 of 1**	OK
			Max zone temperature deviation	**0.0**	°K

Supply Fan Sizing Data

Actual max L/s at Jun 1600	**2583**	L/s	Fan motor BHP	**5.36**	BHP
Standard L/s	**2369**	L/s	Fan motor kW	**4.00**	kW
Actual max L/(s-m²)	**32.61**	L/(s-m²)	Fan static	**774**	Pa

Outdoor Ventilation Air Data

Design airflow L/s	**952**	L/s	L/s/person	**8.00**L/s/person
L/(s-m²)	**12.02**L/(s-m²)			

Fig.13: Air System Sizing Summary for packaged roof top AHU

Air System Information

Air System Name .... **packaged roof top AHU**
Equipment Class .... **PKG ROOF**
Air System Type .... **SZCAV**

Number of zones **1**
Floor Area .... **79.2** m²
Location .... **Aisha, Ethiopa**

Sizing Calculation Information

Zone and Space Sizing Method:
Zone L/s .... **Peak zone sensible load**
Space L/s .... **Zone L/(s-m²)**

Calculation Months .... **Jan to Dec**
Sizing Data .... **Calculated**

Zone Sizing Data

Zone Name	Maximum Cooling Sensible (kW)	Design Air Flow (L/s)	Minimum Air Flow (L/s)	Time of Peak Load	Maximum Heating Load (kW)	Zone Floor Area (m²)	Zone L/(s-m²)
Zone 1	24.3	2583	2583	Jun 1600	0.0	79.2	32.61

Zone Terminal Sizing Data

No Zone Terminal Sizing Data required for this system.

Space Loads and Airflows

Zone Name / Space Name	Mult.	Cooling Sensible (kW)	Time of Load	Air Flow (L/s)	Heating Load (kW)	Floor Area (m²)	Space L/(s-m²)
Zone 1							
Passenger coach	1	24.3	Jun 1600	2583	0.0	79.2	32.61

Fig. 14: Zone Sizing Summary for packaged roof top AHU

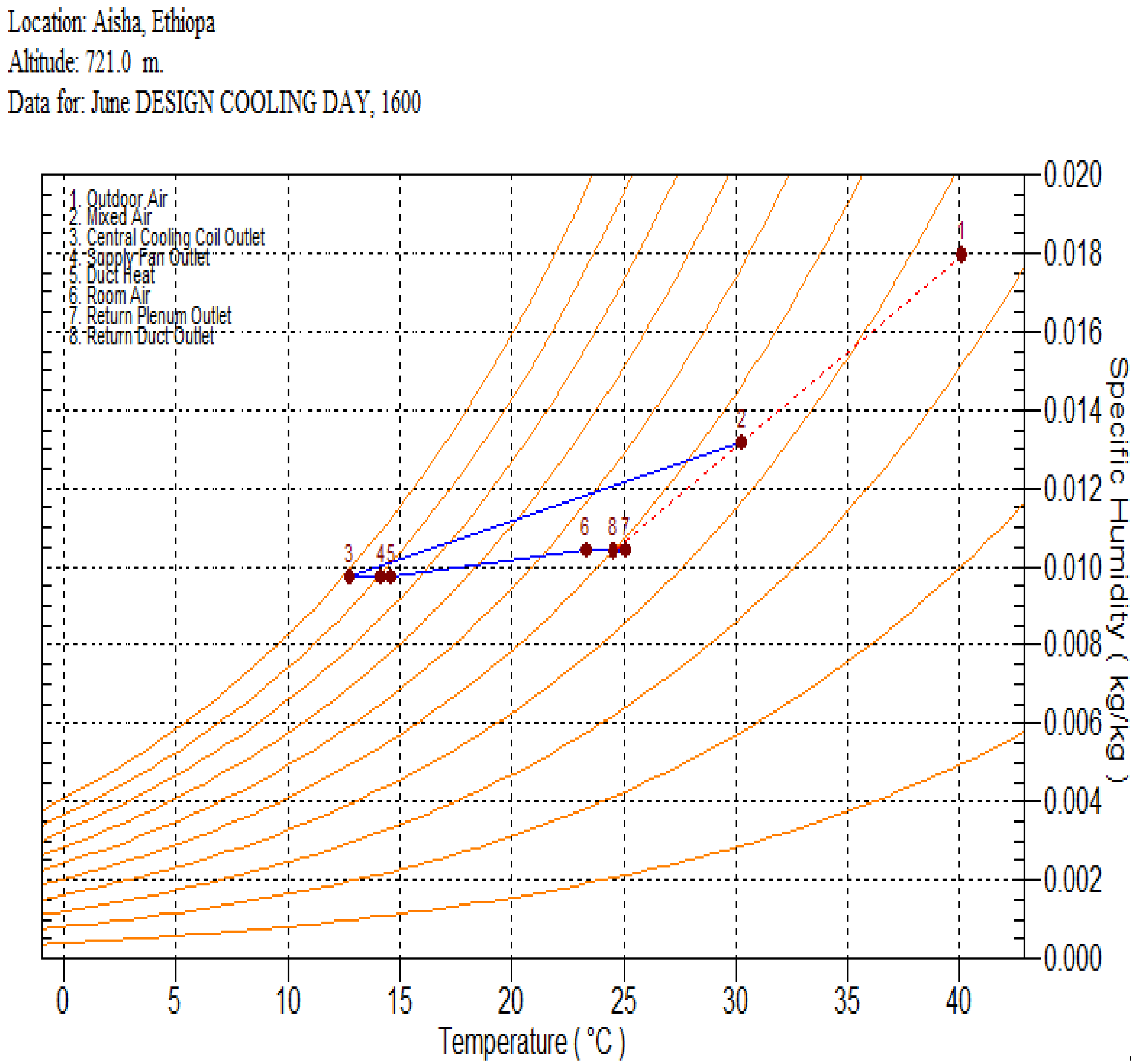


Fig.15: Psychometric Analysis for packaged roof top AHU

CHAPTER FOUR

TRAIN COACH HVAC EQUIPMENT SELECTION AND DUCT WORK DESIGN

4.1. Equipment Selection

The final step in the design process is to use system design reports to select HVAC equipment from product catalogue or electronic catalogue software. System and plant design reports provide information necessary to select all the components of HVAC system including air handlers, packaged equipment, supply terminals, duct systems, piping systems and plant equipment.

HAP is used to model the heat transfer processes in order to determine the following eight equipment sizing values:

- Rooftop unit required cooling capacity.
- Rooftop unit required fan airflow
- Rooftop unit required preheat coil capacity.
- Terminal design airflow rates.
- Terminal minimum airflow rates.
- Terminal fan design airflow rates.
- Terminal reheat coil capacities
- Space required supply airflow rates.

Data provided on the System Sizing Summary report (see fig.13) and the Zone Sizing Summary report (see fig.14) can be used to meet all eight of these objectives.

The Air System Sizing Summary provides data for sizing and selecting the supply fan and central cooling and heating coils. This data includes the design supply airflow rate, the design cooling and heating coil loads, coil selection parameters and useful check figures. The Zone Sizing Summary provides information for sizing space supply diffusers, zone and space ductwork and any zone heating units or reheat coils.

4.1.1. System configuration

There are two typical configurations used in rail transit HVAC systems:

- Unitized or Packaged systems
- Split systems

4.1.1.1. Unitized or Packaged Systems

Unitized or packaged systems are designed such that all refrigeration components and piping are contained within a single module. The number of units on a particular car may vary, depending on the car configuration. In most cases, a single vehicle has two HVAC units, one at each end. Multi-sectioned articulated vehicles often have one HVAC unit per car section. However, smaller vehicles, such as low floor street cars, may have two smaller units per section, due to the limited roof area and necessity to locate other equipment on the roof.

Unitized systems are most commonly installed on the vehicle roof but can also be installed under the floor of the vehicle. Unitized or packaged systems facilitate the installation and removal of the HVAC unit on the vehicle. In addition to the mechanical mounting hardware, only the electrical, ducting, and drain connections are required.

4.1.1.2. Split Systems

Split systems usually consist of separate evaporator and compressor/condenser sections (condensing unit) with connecting piping routed inside the vehicle walls. Evaporator sections are located above the car ceiling while the condensing units are located under the car floor. Electrical controls are packaged in a separate under floor or interior mounted control enclosure, an electrical cabinet incorporated into the evaporator or condenser unit assemblies, or some combination of these locations. The evaporator section typically contains a fan/motor assembly, evaporator coil and heater assembly, along with associated refrigerant and electrical control apparatus. The condenser section typically contains the compressor(s), condenser coil(s), condenser fan assembly and associated refrigeration, and electrical components and refrigerant system access fittings.

In this thesis roof mounted packaged unit is selected and its detail technical specification will be discussed later on in the following sections.

4.1.2. Cooling equipment design

The following recommendations should be considered in designing the HVAC equipment for use in rail transit:

- Superheat must be present at the evaporator outlet and at the compressor inlet and its values should take into account the compressor manufacturer requirements. Typical superheat values are from 5 to 8°C (9 to 15°F) at the evaporator outlet at design conditions where expansion valve bulb is connected.
- Sub-cooling should be present at the expansion valve inlet. Sub-cooling may be achieved as part of the condenser coil design and/or by a separate heat exchanger. It is recommended to provide 5 to 8°C (9 to15°F) of sub-cooling at the expansion valve inlet at design conditions.

4.1.2.1. Refrigerant selection

Any refrigerant selected for the transit HVAC system should be non-toxic and non-flammable as classified in Group A1 of ASHRAE Standard 34.

Until approximately the year 2000, R-22, a hydro-chlorofluorocarbon (HCFC), was the refrigerant of choice for rail transit equipment. In accordance with the Clean Air Act and international regulations, HCFC refrigerants are no longer used for new equipment. The current most viable replacements are blends of hydro-fluorocarbons (HFCs), R-407C and R-410A. Due to unique characteristics and performance of each refrigerant, it should be selected based on the operating environment of each system, primarily the expected ambient temperature ranges.

In this thesis R-134a has been selected and it has the advantage of being a single component refrigerant, thus eliminating the slide in condensing and evaporating temperatures present in blended refrigerants.

4.1.2.2. Compressor selection

In this thesis Hermetic scroll compressor is selected to increased reliability and other factors; such as:

- Relatively quiet, small, and light

- In many instances, more efficient than reciprocating
- More tolerant to liquid slugging
- No seals or gaskets for refrigerant leaks

4.1.2.3. Selection of other mechanical components

A. Expansion Valves

Thermal Expansion Valves (TXVs) with external equalizers are most commonly used in rail transit applications. TXVs should be located to provide easy access for replacement, especially when used in difficult-to-remove above-ceiling mounted evaporator-blower units.

B. Temperature and Humidity Control

Rail car HVAC systems are automatically controlled for year-round comfort. In most cases, the HVAC unit controller also directly controls or provides the information of the vehicle status for the Layover operation.

4.1.2.4. Air Control Equipment

A. Condenser Fans

Condenser fan(s) are typically multi-blade axial type. Fan blade design should be selected to ensure optimum fan efficiency and minimum noise generation. Connection of the fan hub to the motor shaft is normally effected using a tapered bushing type hub, with a key. It is recommended that the condenser fan motor shaft should be of a corrosion resistant material, or treated to prevent corrosion and seizing of the fan hub on the shaft.

B. **Outside Air Intakes**

Outside air is required to provide car pressurization and to provide ventilation within the car for all operating conditions. The combined outside/re circulated air plenum should be designed to allow the outside air to be thoroughly mixed with the return air prior to being treated by the HVAC unit before being delivered to the passenger area.

C. **Evaporator Blowers**

Evaporator blowers are almost universally centrifugal (radial flow) fans directly mounted to the motor shaft. Keyed hubs with double set screws, keyed hubs with threaded shafts for retention nuts, flanged hubs, and tapered hubs are generally used.

D. Air Dampers

When automatic re-circulated air or ventilation air dampers are provided on unitized HVAC equipment, it is recommended that the dampers and actuators be integrated into the unit design.

4.1.3. Vehicle ventilation and air distribution

The HVAC system air ventilation, mixing, treatment, distribution, and exhaust are described as follows

4.1.3.1. Outside Air

Outside air is required to provide car pressurization and to provide ventilation within the car for all operating conditions. When unitized HVAC equipment is located on top of the vehicle roof, the HVAC unit usually has outside air intakes located on the unit itself.

When intermediate outside air intake ducts are used should be constructed of non-corrosive materials (stainless steel, aluminium, fiber composite, etc.), and should be sloped to drain to the outside of the car. Outside air filters should remain dry under all possible operation scenarios and car/HVAC unit orientation and location in the train.

Outside air should be thoroughly mixed with the return air and then be treated by the HVAC unit before being delivered to the passenger area. Outside air is normally drawn into the system along with the re-circulated air by the HVAC main blower/fan. However, when necessary, a separate booster fan may be required to compensate for the pressure losses in the outside air ducting.

Depending on the car and HVAC unit design, the outside air dampers may be installed on the car side or on the HVAC unit side.

4.1.3.2. Return Air

Return airflow is determined in combination with outside air requirements to achieve the evaporator coil airflow necessary to achieve the design cooling capacity. In this thesis return air is drawn by the HVAC unit blower through a ceiling mounted grille located below the HVAC unit mixing plenum instead of incorporating the return air duct to the space critical ceiling area To minimize noise and pressure loss, it is recommended that the average air face velocity at the return air grille not exceed 2.03m/s.

4.1.3.3. Supply Air

The amount of supply air is dictated by the required system capacity (sensible and latent) and the acceptable supply air temperature.

4.1.3.4. Exhaust Air

Air should be exhausted from the passenger car through passive (static air exhaust) or active (forced air ventilation) exhaust systems or a combination of both in sufficient quantity to facilitate the entrance of the outside air. Accordingly, the total exhaust air through the exhaust system should be somewhat lower than the outside airflow rate. The difference between both airflows (sometimes referred to as “pressurization air”) represents the quantity of air being exhausted through various natural leakage points of the car such as door seals.

To the extent possible, exhaust grilles should be located at areas on the car body where the outside static pressure disturbances are at a minimum to prevent large fluctuations of the car interior static pressure.

4.1.4. Air Diffusers selection

Air diffusers should be arranged to provide uniform distribution of air throughout the car. Provisions for closing off sections of the air diffusers near return air inlets to prevent supply air from being drawn directly into the return outlet are sometimes required. The diffusers should be designed to provide uniform airflow distribution and mixing of the conditioned air with surrounding air throughout the car.

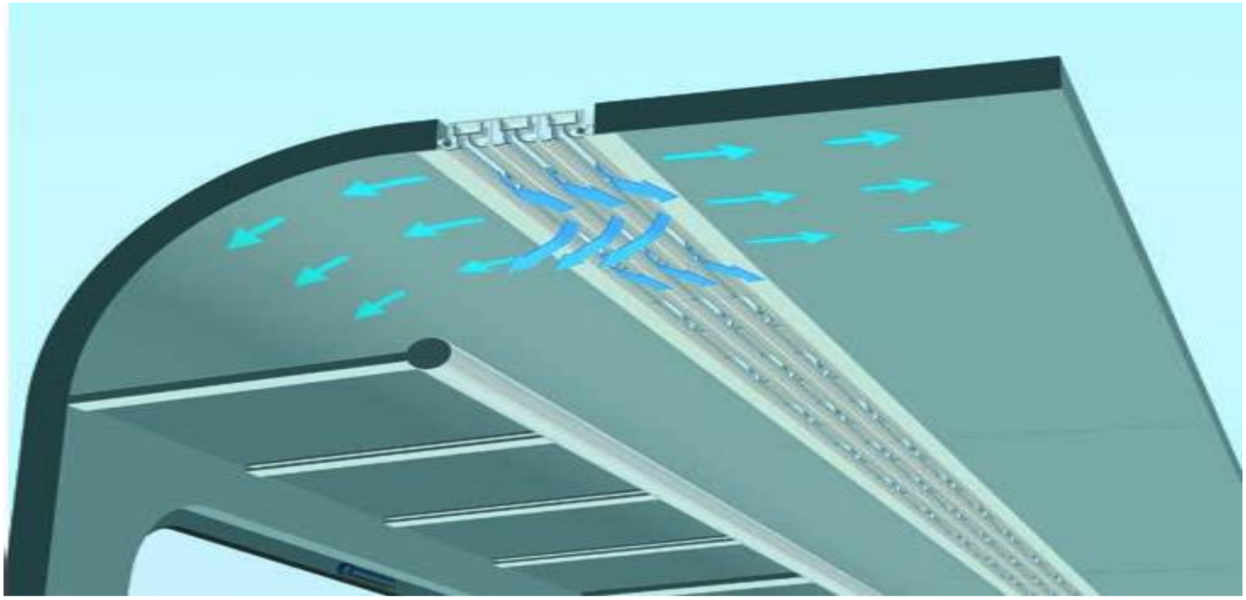


Fig 16: Linear supply Diffuser [31]

The Function a linear diffuser as follows:-

- The clean air supply is split into a large number of individual jets. Additionally, a part of the supply air is diffused directly along the ceiling through a slot in the diffuser border profile. With this air curtain, dust particles from tobacco smoke to textile abrasions are prevented from depositing on the ceiling which considerably reduces the need for frequent repainting or repair
- High cooling capacity, high air exchange and thorough rising with fresh air, noiseless and without draft phenomena even with low ceiling heights fast reduction of temperature and velocity difference b/n supplied air and room air within the shortest distance from the diffuser.

Selection depends on:

- Requirement of indoor environment control
- Shape, size and ceiling height
- Volume flow rate per unit floor area
- Volume flow rate per outlet
- Throw
- Noise level
- Total pressure drop

- Cost and Appearance

The selected linear diffuser Resulting data:

Model: Linear diffuser type **LDB 12/8/2 [31]**

- pressure drop(pa) =14
- sound power level(dB) =28
- Extension of jet at which the maximum speed of ambient air was measure (a(C_{max}) in m) =2m,
- Maximum speed of ambient air with uniform distributed thermal load (cm/s)=23
- Height of measuring point(h) =1.7m
- Height of Room/ cabin(H) =2.8m

The diffusers shall be designed to deliver equalized airflow throughout the car and meet the temperature variation requirements specified.

4.1.5. Selection of Roof Mounted Packaged Unit

The function of the roof mounted packaged unit is to provide air at a quantity, temperature, and humidity to offset the sensible and latent heat gains to the space (in the cooling mode) and the heat losses (in the heating mode), while maintaining the required temperature and humidity in the space.

Structure suitable for the typical stress of railway use, made of stainless steel painted with polyester resin base paints. Can be inspected completely from above by means of removable panels for all the maintenance operations and fitted with access to the main cooling elements to perform regulations and controls. The whole structure is designed and controlled with the F.E.M. method to withstand the stress foreseen in the railway use.

The packaged conditioners are operating with R134a and suitable for overhead installation in railway wagon.

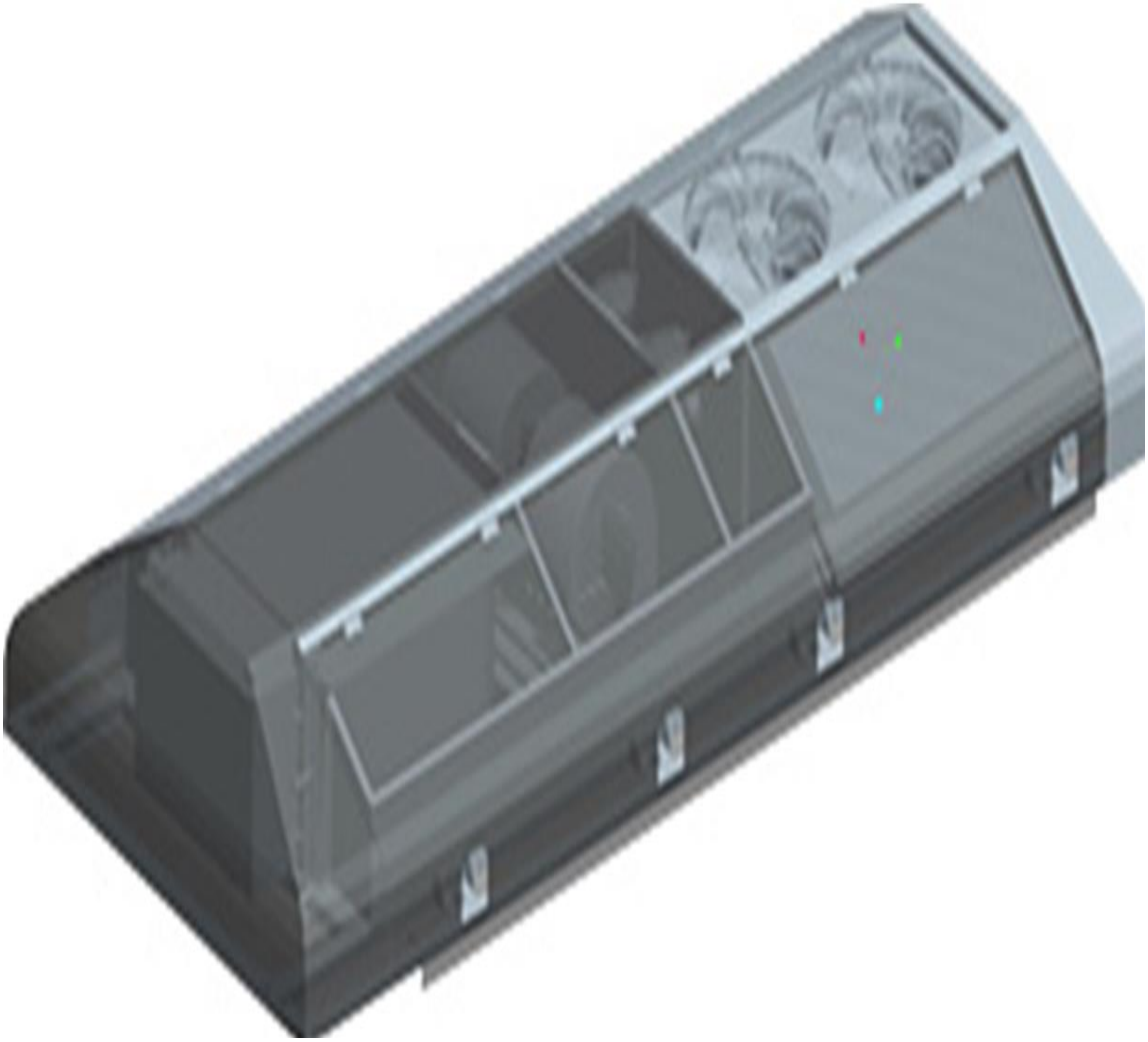


Fig 17: Roof mounted packaged unit [catalogue]

Scroll compressor suitable for railway use and fitted with vibration damping supports and fitted with a 24 V cc resistance carter.

Condensing section fan with three phase, 380V/3ph/50 Hz motor, suitable for railway use with on/off condensation control.

Dual intake directly coupled centrifuge air treatment fans with 220 V a.c. supplied single phase motor suitable for railway use. Finned pack heat exchanger with copper pipes and aluminium fins.

Cooling circuit made of copper pipes includes: anti-vibration connection joints to the compressor, thermostatic expansion valve for cooling operation, liquid solenoid valve, hot gas valve, liquid receiver, intake accumulator, liquid indicator, high and low pressure switches, pressure switches for controlling condensation, pressure transducer for controlling hot gas injection, insulation of the suction lines with material complying with railway specifications, double-circuit refrigerator to get the full redundancy in case of failure single circuit and to work under degraded ambient.

General technical data		
Total cooling capacity	kW	42
Compressor		
Total power input	kW	13.8
Air treatment fans		
Total air flow	m^3/h	7400
External head pressure	Pa	300
Power input	kW	1.8
Condensing section fans		
Total air flow	m^3/h	20000
Power input	kW	0.84
Electrical resistance		
Total number	n.	2
Total power supply	kW	8
Power modulation control	Steps	2
Total power input	kW	11.6
Power supply	380V/50Hz/3Ph	
Dimension		
Length	Mm	2360
Width	Mm	2150
Height	Mm	325
Weight	241Kg	

Fig 18: specification of the Roof Mounted Packaged Train Air units

4.1.6. Selection of Air Return / Exhaust Grills

Grillers for the package of return air from the passenger compartment to the plenum chamber through the hole. In this case there will be ten pieces of return grill on the ceiling, Transfer grill type-AGS with fixed inverted vee blades with visible fixing screw, boarder counter punched is selected with the following specification depending on the return air flow. neck size 600x600mm, volume-256L/s, jet velocity-1.5m/s and with pressure drop of 7pa.

4.2. Ductwork Design

All duct attachments to the car body should have a thermal break. Outside air and supply air ducts should be thermally insulated on the outside to minimize the heat transfer and to prevent condensation. In addition to thermal insulation, a short section at the beginning of the supply air duct may also need to be equipped with acoustical lining to absorb the noise generated by the HVAC unit blower.

Conditioned air is supplied to the vehicle by blower fans supplied as part of the HVAC units. The conditioned air should be carried throughout the vehicle within a distribution duct. Provisions should be made to provide uniform air distribution throughout the duct such as diagonal splitter, baffles, turning vanes, or other similar approaches. To minimize pressure drops and noise, the air distribution ducts are typically sized to maintain air velocity below 6.1m/s.

The most common methods of air duct system design are:

A. Velocity reduction
B. Equal friction rates
C. Static regain and

NB*: for this thesis only equal friction method is used to design the supply duct.*

4.2.1.1. Constant friction loss method

The equal friction method, as its name implies, is based on maintaining the same pressure drop per unit of duct length (or friction rate) throughout the system. The duct size is based on the flow rate through a particular section of duct, and design value for the friction rate. Each section is sized using the design friction rate criterion, and the total pressure drop for each run is simply the sum of the pressure drop of each individual section. The duct sections pressure drop includes straight duct friction loss, pressure losses through fittings such as elbows, take-offs, and registers and /or diffusers. In the sections entering and leaving the HVAC unit, pressure losses associated with the flow transitions entering the leaving the unit (the system effect) are also included. The unit fan speed is selected to provide the design flow and produce enough pressure difference to overcome pressure losses in the supply and return branches having the greatest pressure drop. Note that duct systems designed using the equal friction method is not self-balancing. Balancing dampers must be installed in lower pressure loss branches to balance the system. This method provides a design that is better balanced, but balance dampers must still be installed since it is not possible to provide a truly self-balanced system using this method. Duct section air velocity is also displayed to check for potential noise problems. Duct dimensions are based on the interior dimensions; if duct liner is used, the thickness of the duct liner must be subtracted. Pressure loss data are based on smooth duct. Adjustment factors must be applied to lined duct, duct board, and flex duct [30]

Determination of the supply duct

From earlier analysis we have the supply flow rate and take the assumed resistance or pressure drop (pa/m) in the main duct is 0.6.

Using Duct sizer software the supply duct is designed as shown the following table 6 .The cross section of the main air duct has been designed in such a way that air velocity inside the duct shall not be higher than 6.1m/s in order to reduce turbulence and noise due to air motion in the duct. For the same reason the main air duct has been connected to evaporator outlet by means of an intermediate transition duct made of fire resistant canvas to prevent transmission of noise produced by the blower unit.

Table 7: Supply duct sizing Table

Duct Sizing Table											
1	2	3	4	5	6	7	8	9	10	11	12
Section	**Length (m)**	**Flow Rate (l/s)**	**Pressure drop per metre (Pa/m)**	**Duct Size (mm)**	**Velocity (m/s)**	**Velocity Pressure (Pa)**	**Fittings pressure loss factor or ζ (zeta) factor**	**Pressure Loss**		**Total Pressure Loss (Pa)**	**Cumulative Pressure Loss (Pa)**
								Fittings (Pa)	**Straight Duct (Pa)**		
A	1.2	645.75	0.6	425x350	4.64	12.91	7.17	92.56	0.72	93.28	93.28
B	0.97	600.75	0.6	400x350	4.57	12.58	0.25	3.11	0.58	3.69	3.69
C	0.97	555.75	0.6	375x350	4.52	12.25	0.25	3.03	0.58	3.61	3.61
D	0.97	510.75	0.6	350x350	4.44	11.85	0.25	2.94	0.58	3.52	3.52
E	0.97	465.75	0.6	325x350	4.36	11.44	0.25	2.85	0.58	3.43	3.43
F	0.97	420.75	0.6	325x325	4.25	10.83	0.25	2.74	0.58	3.32	3.32
G	0.97	375.75	0.6	325x300	4.11	10.13	0.26	2.60	0.58	3.18	3.18
H	0.97	330.75	0.6	325x275	3.95	9.38	0.26	2.42	0.58	3.00	3.00
I	0.97	285.75	0.6	325x250	3.76	8.52	0.26	2.24	0.58	2.82	2.82
J	0.97	240.75	0.6	300x225	3.62	7.85	0.26	2.04	0.58	2.62	2.62
K	0.97	195.75	0.6	300x200	3.43	7.09	0.26	1.87	0.58	2.45	2.45
L	0.97	150.75	0.6	300x175	3.21	6.21	0.27	1.68	0.58	2.26	2.26
M	0.97	105.75	0.6	300x125	2.94	5.18	0.28	1.46	0.58	2.04	2.04
N	0.97	60.75	0.6	300x100	2.55	3.91	0.37	1.44	0.58	2.02	2.02
	Duct total pressure drop										131.24
		RMPU pressure drop									200
		SUPPLY DIFFUSER pressure drop									14
		System total pressure drop per unit									345.24

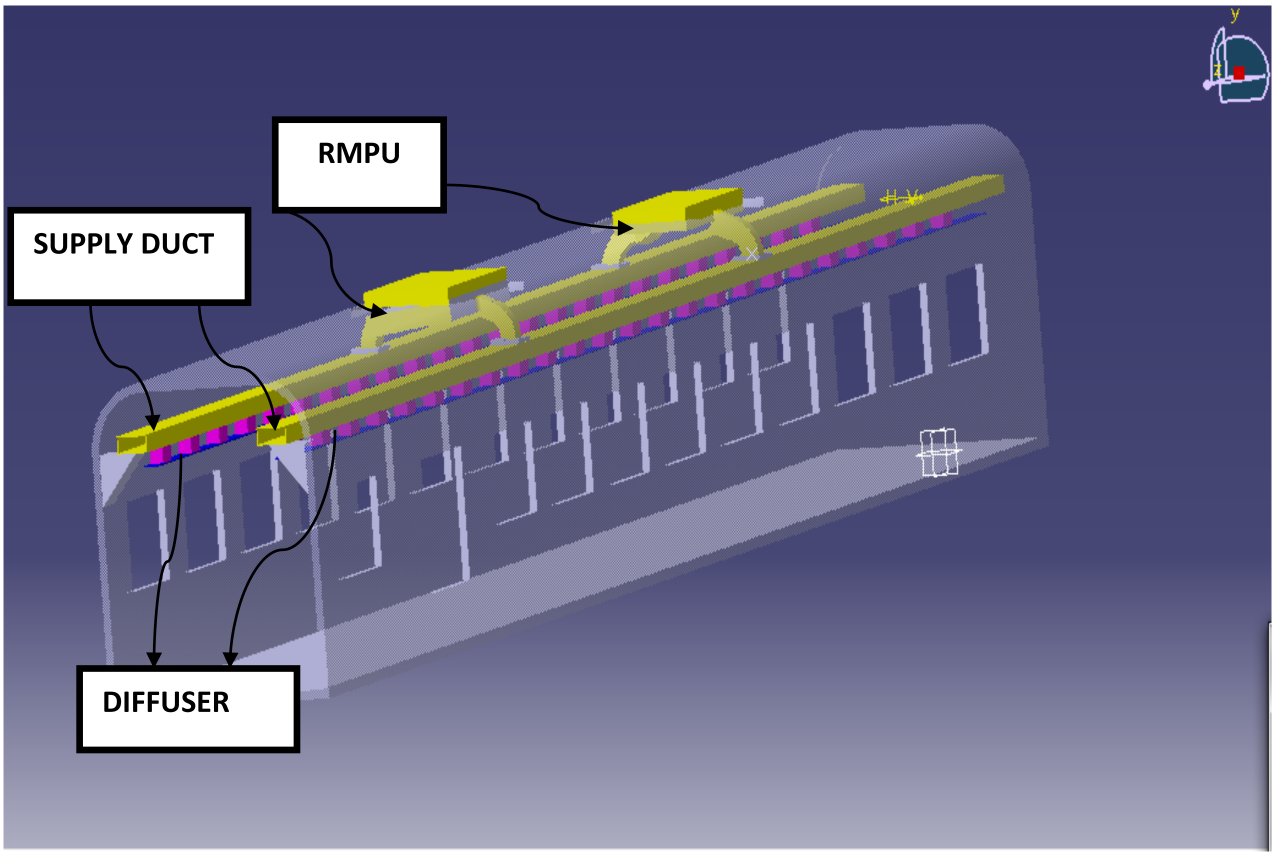


Fig 19: 3D- Supply Duct work with RMPU units and diffusers

4.2.2 Supply fan selection

The fan is an essential and one of the most important components of almost all air conditioning systems. Thus a basic understanding of fan performance characteristics is essential in the design of air conditioning systems.

The centrifugal fan is most commonly used in air conditioning systems as it can efficiently move large quantities of air over a large range of pressures. The centrifugal fan with forward-curved blades is widely used in low-pressure air conditioning systems.

While we select a fan we have to take in to account the required amount of supply air flow and total pressure drop across the air distributing system.

Supply fan static pressure consists of pressure losses:

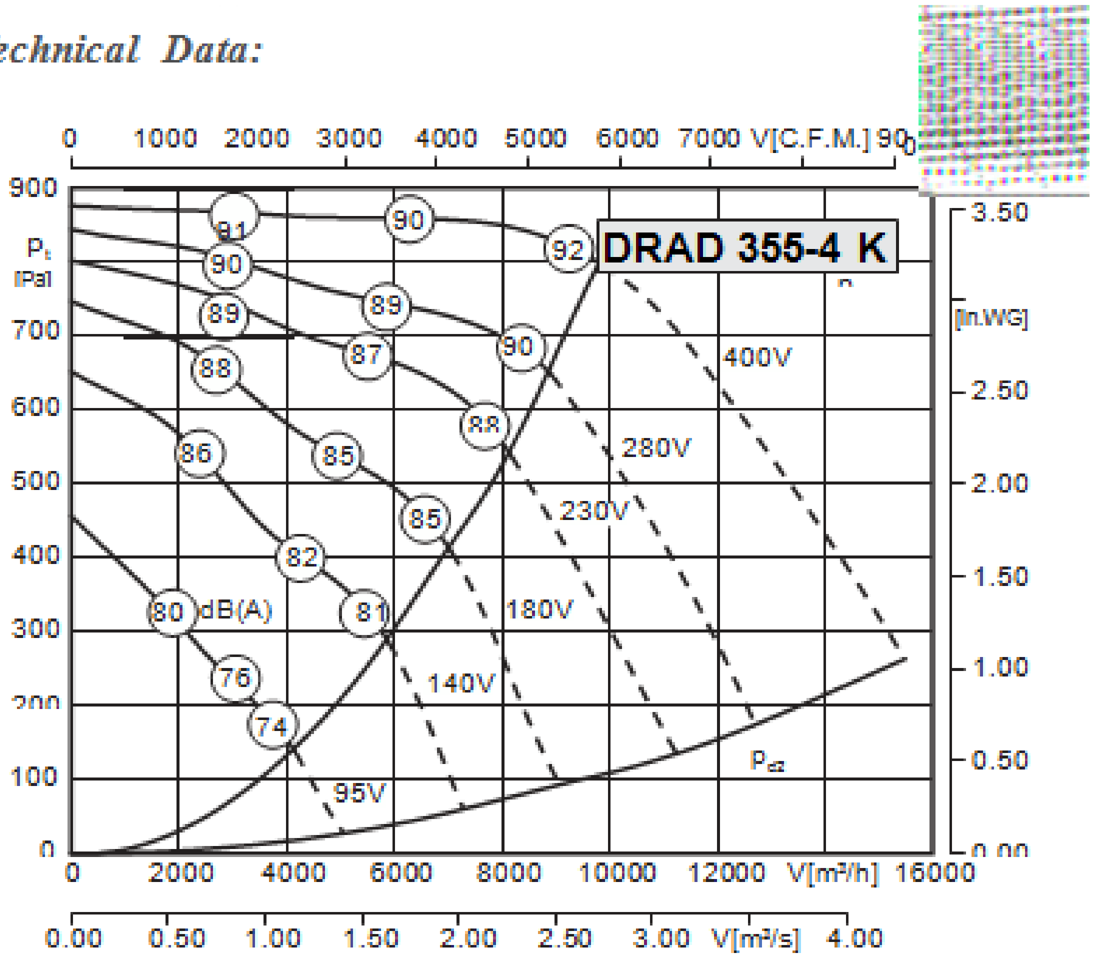
- Resistance of air conditioning unit including filters, inlet baffles, air washer, eliminator;
- Resistance at outlet terminals,
- Resistance at intake terminals,
- Resistance at the suction end of the fan,
- Resistance in the supply duct ,

Based on the total static discharge pressure loss (345.24pa) as shown on table 7 and with air flow rate of (645.75l/s) we can select the Fan which is used in railway air conditioning technology as follows:

Application: Railway roof air handling unit integrated with Fans

- Centrifugal fans Type: DRAD
- Double inlet centrifugal fan with forward curved centrifugal impeller
- Variable mounting positions possible
- Voltage range: 400 V, 3-phase
- Frequency: 50 Hz
- Air volumes up to 15000 m^3/h
- Total pressure increase up to 700 Pa
- Power input: up to 5KW

Technical Data:



Type :	DRAD 355-4K	Art.-No. :	C10-35500
U	400V (50Hz)	pfa min	700 Pa
P_1	5.0 kW	1	2 %
I_N	10 A	I_A / I_N	4.6
n	1380 min^{-1}		IP54

- Casing made of galvanized sheet steel
- Forward curved impeller
- Asynchronous external rotor motor
- Protection class IP54
- Insulation class F
- Side cable connection
- Motor protection by thermal contacts
- Speed is variable using auto transformers

Fig.20: Air supply fan specification [32]

CHAPTER FIVE

RESULTS AND DISCUSSION

Two different methodologies are presented to achieve the objectives of this work. The first one is the analytical method and the second one is the Hourly Analysis Program (HAP) software in order to estimate the cooling load and system design which is presented in Chapter 3 which in turn identifies the various attributes to be considered for the design and selection of air conditioning systems.

The analytical method which estimates the total cooling load based on ASHRAE for the design and application of heating, ventilation and air conditioning equipment for rail passenger vehicle while travelling through the route Dire Dawa to Djibouti and summarized as shown on table-6.

As we can see from the table-6 of cooling load analysis summary the total load is estimated as 78.22KW, sensible coil is 47.745kw and sensible heat ratio is 0.61.This method builds a flexible and comprehensive model, which has the capability to consider the interactions between various types of load for the total air conditioning system.

And the numerical values obtained from the analysis are used for the comparison and then the selection of the suitable air conditioning system.

The HAP software also identifies the various attributes to be considered for the design and selection of air conditioning systems using system design reports generated. The following table summarizes the sizing objectives and locations of data which we are already used through the design process.

Table 8: Location of System Sizing Data on Design Reports

Objective:	Figure
Report: Air System Sizing Summary	
1. Rooftop Cooling Capacities	Central Cooling Coil Sizing Data (*see fig.13*)
2. Rooftop Supply Fan Airflow	Supply Fan Sizing Data (*see fig.13*)
Report: Zone Sizing Summary	
3. Supply Terminal Airflow Rate	Zone Sizing Data *(see fig.14)*
4. Supply Terminal Minimum Airflow Rates	Zone Sizing Data *(see fig.14)*
5. Space Airflow Rates	Space Loads and Airflows *(see fig.14)*

The System Sizing Summary report contains data used to select the packaged rooftop unit. The Central Cooling Coil Sizing Data table lists the peak coil capacities, coil entering and leaving conditions and a number of useful check figures as :

Total coil load = 74.2 kW

Sensible coil load = 50.1 kW

Coil L/s at Jun 1600 = 2583 L/s

Max L/s at Jun 1600 = 2583 L/s

Sensible heat ratio = 0.676

Entering DB / WB = 30.2 / 21.0 °C

Leaving DB / WB = 12.7 / 12.4 °C

From the Supply Fan Sizing Data the required airflow rates and motor power for the supply fan is: actual max L/s at Jun 1600 = 2583L/s, Standard L/s = 2369 L/s, actual max L/(s m²) = 32.61L/(s-m²) and fan motor power is 4.00kW. Together this data is used in packaged rooftop unit selection as already done in the previous chapter.

The Zone Sizing Summary report contains data used to select terminal equipment and used in duct design calculations to size ductwork for the system based on air flow rate of 2583L/s as discussed in the previous chapter.

And from the hourly air system design day loads for packaged roof top unit summery report we have observed that maximum cooling load occurs at 16:00 hours of June.

CHAPTER SIX

CONCLUSION, RECOMMENDATION AND FUTURE WORKS

6.1. Conclusion and Recommendations

As mentioned above in this paper we followed two ways of thermal load analysis methods so as to estimate the cooling load inside the passenger compartment and later the necessary HVAC equipment selection based on the estimated load.

Since the total cooling load and air flow required to the passenger compartment is calculated as 74.2KW and 2583L/s respectively using HAP software, two roof-mounted packing units (RMPU) were selected from the manufacturers catalogue through the design process with a maximum cooling capacity of each 42KW so that we can get the required thermal comfort. In addition, duct is designed using duct sizer software based on the flow rate and assumed pressure drop, diffusers and grills also selected from their respective selection charts. Finally air supply fan also selected based on the required flow rate and over all pressure drop across the system with its driving motor. As a result we can achieve the desired pleasant and comfortable interior thermal environment to the passengers during all expected external ambient environmental conditions through the route Dire-Dawa to Djibouti.

All the HVAC equipment selection above is done by considering the applicable contractual Reliability, Maintainability, and Safety criteria. And finally i would like to recommend the design to be incorporated in the future for locally manufactured rail vehicles, hopefully it will be successful.

6.2. Future works

Because of the rail way operation is not started yet in Ethiopia, particularly through the selected route there were a scarcity of actual data (such as body surface temperature variation), so in the future it will be better to use actual data on site derived from direct measurements of air parameters inside the car and traversed areas of passenger compartment, bath room, and the environment using a digital hygrometer and a laser thermometer and will be incorporated in the HVAC design so that we can get more efficient and effective outcome. In addition, it will be better to use computational fluid dynamics (CFD) software so that we can check the distribution of air flow pattern and temperature profile inside the passenger compartment.

References

1. ASHRAE Guideline 23P, Guideline for the Design and Application of Heating, Ventilation and Air Conditioning Equipment for Rail Passenger Vehicles First Public Review Draft
2. Duckshin Park, Soonbark Kwon, Youngmin Cho, Eunyoung Park, Seyoung Kim, Miyoung Jung ,” Analysis of Indoor Air Quality and Development of New Air Cleaner for Railroad Passenger Cabin“,Korea Railroad Research Institute, 360-1 Woulam-dong, Uiwang-si, Kyonggi-do, 437-757, Korea.
3. Sârbu, I., “Thermal comfort evaluation models”, in Theoretical considerations, No. 2.(43), Universit y of Ti mi soara, 2007.
4. Devonshire, J. M., Sayer, J. R., “The effects of infrared-reflective and antireflective glazing on thermal comfort”, University of Michigan, March 2002.
5. “Predicting human thermal comfort in a transient non uniform thermal environment. In, Proceedings of the Fifth International Meeting on Thermal Manikins and Modeling, 5IMM, Strasbourg, France, 2003 (National Institute for Working Life, Stockholm, Sweden), also in Eur. J. Appl. Physiol., 2004, 92(6), 721–727.
6. Daniel Trzebie ski, Ireneeuz Szczgiee, Thermal analysis of car air cooler, Silesian University of Technology, Institute of Thermal Technology, Journal of KONES Powertrain and Transport, Vol. 18, No. 2 2011
7. Fanger, P. O., “Thermal Comfort: Analysis and Applications in Environmental Engineering,” McGraw-Hill, New York, 1972.
8. Ingersoll, J., Kalman, T., Maxwell, L., and Niemiec, R., “Automobile Passenger Compartment Thermal Comfort Model - Part II: Human Thermal Comfort Calculation,” SAE Technical Paper 920266, 1992, doi: 10.4271/920266
9. Taeyoung Han, Linjie Huang and Sean Kelly., “Virtual Thermal Comfort Engineering”, SAE, paper series 2001-010588.
10. Radu MUSAT1, Elena HELEREA1, Parameters and Models of the Vehicle Thermal Comfort , Acta Universitatis Sapientiae Electrical and Mechanical Engineering, 1 (2009) 215-226
11. David Huang K., Sheng-Chung Tzeng, Tzer-Ming Jeng , Wing-Ding Chiang., “Air-conditioning system of an intelligent vehicle-cabin”, Graduate School of the Vehicular Engineering, Dayeh University, Changhua, Taiwan.

12. Dusan Fiala , K. J. Lomas, and M. Stohrer., "Dynamic Simulation of Human Heat Transfer and Thermal Comfort", Building Sciences Group (fbta) University of Karlsruhe, Germany.
13. Tzu-Ping-Lin., Ruey-Lung-Hwang., KuoTsang Huang, "Passenger thermal perceptions, thermal comfort requirement, and adaptations in short and long haul vehicles", Department of Leisure planning, National Formosa University, Huwei, Taiwan-Oct 2009
14. ASHRAE Handbook of Fundamental, American Society of Heating, Refrigerating, and Air Conditioning, Atlanta, GA, 1988.
15. Kamar, H., Senawi, M., and Kamsah, N., "Computerized Simulation of Automotive Air-Conditioning System: Development of Mathematical Model and Its Validation," International Journal of Computer Science Issues, 9(2):23-34, 2012.
16. Pedersen, C. O., Fisher, D. E., and Liesen, R. J., "Development of a Heat Balance Procedure for Calculating Cooling Loads," ASHRAE Transactions, 103(2):459-468, 1997.
17. Zheng, Y., Mark, B., and Youmans, H., "A Simple Method to Calculate Vehicle Heat Load," SAE Technical Paper 2011-01-0127, 2011, doi:10.4271/2011-01-0127.
18. Arici, O., Yang, S., Huang, D., and Oker, E., "Computer Model for Automobile Climate Control System Simulation and Application," International Journal of Applied Thermodynamics, 2(2):59-68, 1999.
19. Ding, Y., Zito, R., "Cabin Heat Transfer and Air Conditioning Capacity," SAE Technical Paper 2001-01-0284, 2001, doi:10.4271/2001-01-0284
20. Alexandrov, A., Kudriavtsev, V., and Reggio, M., "Analysis of Flow Patterns and Heat Transfer in Generic Passenger Car Mini-Environment," Presented at 9th Annual Conference of the CFD Society of Canada, Canada, May 27-29, 2001.
21. Khayyam, H., Kouzani, A. Z., and Hu, E. J., "Reducing Energy Consumption of Vehicle Air Conditioning System by an Energy Management System," Presented in IEEE The 4th International Green Energy Conference, China, 2009
22. Wei, K. C., Dage, G. A., "An Intelligent Automotive Climate Control System," IEEE International Conference on Systems, Man and Cybernetics. Intelligent Systems for the 21st Century, 4:2977-2982, 1995, doi: 10.1109/ICSMC. 1995.538237.
23. Barnaby, C. S., Spitler, J. D., and Xiao, D., "The Residential Heat Balance Method for Heating and Cooling Load Calculations," ASHRAE Transactions, 111(1):308-319, 2005.

24. PRIIA 305 Next-Generation Equipment Committee Bi-Level Passenger Rail Cars HVAC System, Amtrak 2012
25. Operation and service for 68ac353-102micromax Rooftop Air Conditioning Unit, Carrier Transicold Division, Carrier Corporation Transport Air Conditioning Group Syracuse, N.Y. 13221 U.S A, www.carrier.transicold.com, 2010,
26. Gabriel Haller, Thermal Comfort in Rail Vehicles, Climatic Wind Tunnel Vienna Professional article September 2006
27. Dawit G. Thermal Comfort and air conditioning system design for Passenger Train, 2013.
28. Refrigeration and Air conditioning, useful training material for mechanical engineering students/college or as reference for engineer, E IIT, Kharagpur, India, 2008
29. Mohammad Ali Fayazbakhsh, Majid Bahrami," Comprehensive Modeling of Vehicle Air Conditioning Loads Using Heat Balance Method," Simon Fraser University, 2013
30. Bhatia A., B.E., HVAC Ducting - Principles and Fundamentals, PDH online Course M246 (4PDH), 2012
31. LTG Aktiengesellschaft Air Technology Solutions for Humans and Products catalogue,105 Corporate Drive, Suite ESpartanburg, SC 29303 USA, www.LTG-INC.net,
32. Rosenberg the air movement group and ventilation GMBH (online)Available from: www.rosenberg-gmbh .com
33. PRIIA 305 Next-Generation Equipment Committee Bi-Level Passenger Rail Cars HVAC System, Amtrak 2012
34. Michael Boduch, Warren Fincher, standards of human comfort (relative and absolute), the university of Texas at Austin, school of architecture
35. Amit K. Ahirrao.1, H. K. Narahari2, S.Umesh3, Vivek Kumar4, Effect of vent shape on Thermal comfort of passengers in a car, 1-M.Sc.[Engg.] Student, 2-Professor and Head, 3-Senior Lecturer, Department of AAE, 4-Manager (CFD), Tata Motors Ltd Pune, M. S. Ramaiah School of Advanced Studies, Bangalore 560 058
36. Roberto de Lieto Vollaro. Indoor Climate Analysis for Urban Mobility Buses: a CFD Model for the Evaluation of thermal Comfort. International Journal of Environmental Protection and Policy. Vol. 1, No. 1, 2013, pp. 1-8. doi: 10.11648/j.ijepp.20130101.11

37. Daniel Trzebie ski, Ireneeuz Szczgiee, Thermal analysis of car air cooler, Silesian University of Technology, Institute of Thermal Technology, Journal of KONES Powertrain and Transport, Vol. 18, No. 2 2011
38. Youssef Riachi, Denis Clodic. A Numerical Model for Simulating Thermal Comfort Prediction in Public Transportation Buses. International Journal of Environmental Protection and Policy. Vol. 2, No. 1, 2014, pp. 1-8. doi: 10.11648/j.ijepp.20140201.11
39. Tim Berlitz, Gerd Matschke, interior airflow simulation in railway rolling stock, Research and Technology Centre of the Deutsche Bahn AG Division for aerodynamics and air-conditioning FTZ , Völckerstraße 5, 80939 München
40. Dragan RUZIC. Improvement of thermal comfort in a passenger car by localized air distribution, Acta Technica corviniensis- Bulletin of Engineering Tome IV (year 2011),ISSN 2067 - 3809

Appendixes

Appendix I

Cooling Design Temperature Profiles

Location: Aisha, Ethiopa

(Dry and Wet Bulb temperatures are expressed in °C)

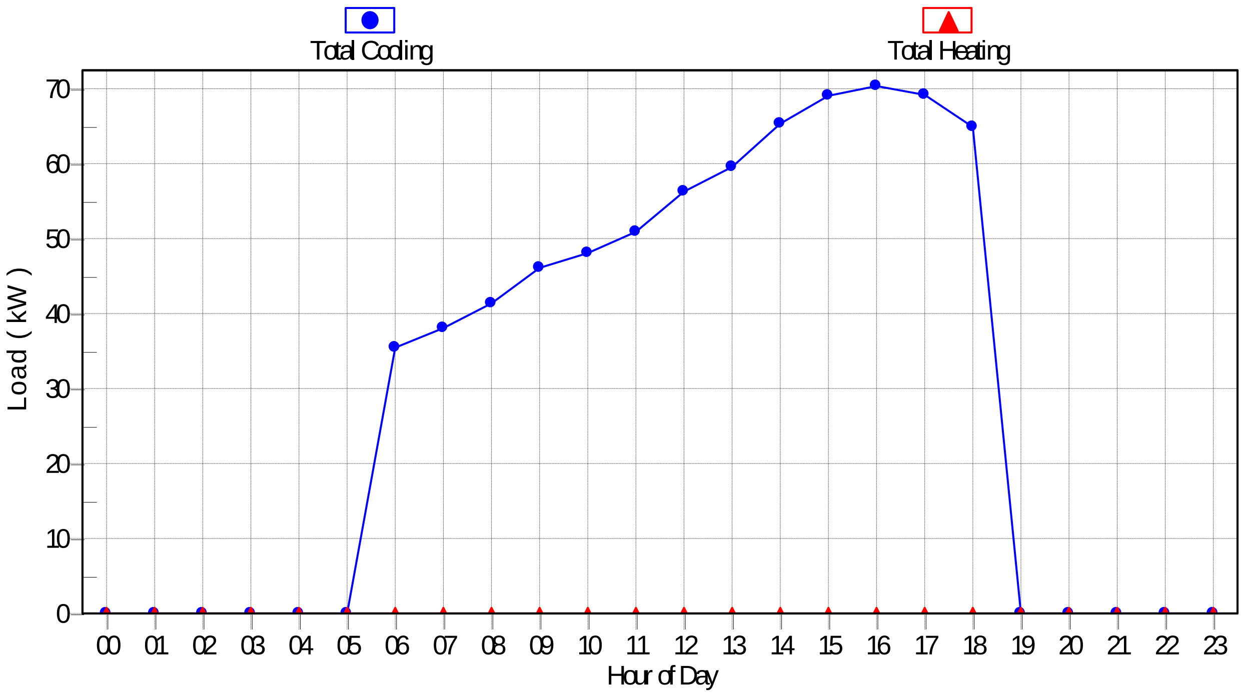
Hr	January		February		March		April		May		June	
	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB
0000	21.7	19.5	22.8	20.2	24.5	21.4	25.0	22.1	25.6	22.7	27.2	23.4
0100	20.7	19.2	21.8	19.9	23.4	21.2	24.0	21.8	24.6	22.4	26.2	23.2
0200	19.8	19.0	20.9	19.6	22.6	20.9	23.2	21.6	23.7	22.2	25.4	22.9
0300	19.0	18.7	20.1	19.4	21.8	20.7	22.3	21.3	22.9	22.0	24.5	22.7
0400	18.3	18.0	19.4	19.2	21.1	20.5	21.6	21.2	22.2	21.8	23.9	22.6
0500	17.8	17.5	18.9	18.7	20.6	20.3	21.1	20.9	21.7	21.4	23.4	22.4
0600	17.6	17.4	18.8	18.5	20.4	20.1	21.0	20.7	21.5	21.3	23.2	22.4
0700	18.0	17.7	19.1	18.8	20.8	20.4	21.3	21.0	21.9	21.6	23.5	22.5
0800	18.8	18.5	19.9	19.3	21.6	20.6	22.2	21.3	22.7	21.9	24.4	22.7
0900	20.3	19.1	21.4	19.8	23.1	21.1	23.7	21.7	24.2	22.3	25.9	23.1
1000	22.5	19.8	23.6	20.4	25.3	21.7	25.8	22.3	26.4	22.9	28.1	23.6
1100	25.0	20.5	26.1	21.1	27.8	22.3	28.4	23.0	28.9	23.6	30.6	24.2
1200	27.9	21.3	29.0	21.9	30.7	23.1	31.2	23.7	31.8	24.3	33.4	24.9
1300	30.6	22.1	31.7	22.6	33.4	23.8	33.9	24.4	34.5	24.9	36.1	25.5
1400	32.6	22.6	33.7	23.2	35.4	24.3	35.9	24.8	36.5	25.4	38.2	25.9
1500	33.9	22.9	35.1	23.5	36.7	24.6	37.3	25.2	37.8	25.7	39.5	26.2
1600	34.4	23.1	35.6	23.6	37.2	24.7	37.8	25.3	38.3	25.8	40.0	26.3
1700	33.9	22.9	35.1	23.5	36.7	24.6	37.3	25.2	37.8	25.7	39.5	26.2
1800	32.8	22.6	33.9	23.2	35.5	24.3	36.1	24.9	36.7	25.5	38.3	25.9
1900	30.9	22.1	32.0	22.7	33.7	23.9	34.2	24.4	34.8	25.0	36.5	25.5
2000	28.7	21.5	29.8	22.1	31.5	23.3	32.1	23.9	32.6	24.5	34.3	25.0
2100	26.5	20.9	27.7	21.5	29.3	22.7	29.9	23.3	30.4	23.9	32.1	24.5
2200	24.7	20.4	25.8	21.0	27.5	22.3	28.0	22.9	28.6	23.5	30.3	24.1
2300	23.0	19.9	24.1	20.6	25.8	21.8	26.4	22.4	26.9	23.1	28.6	23.7

Cont….

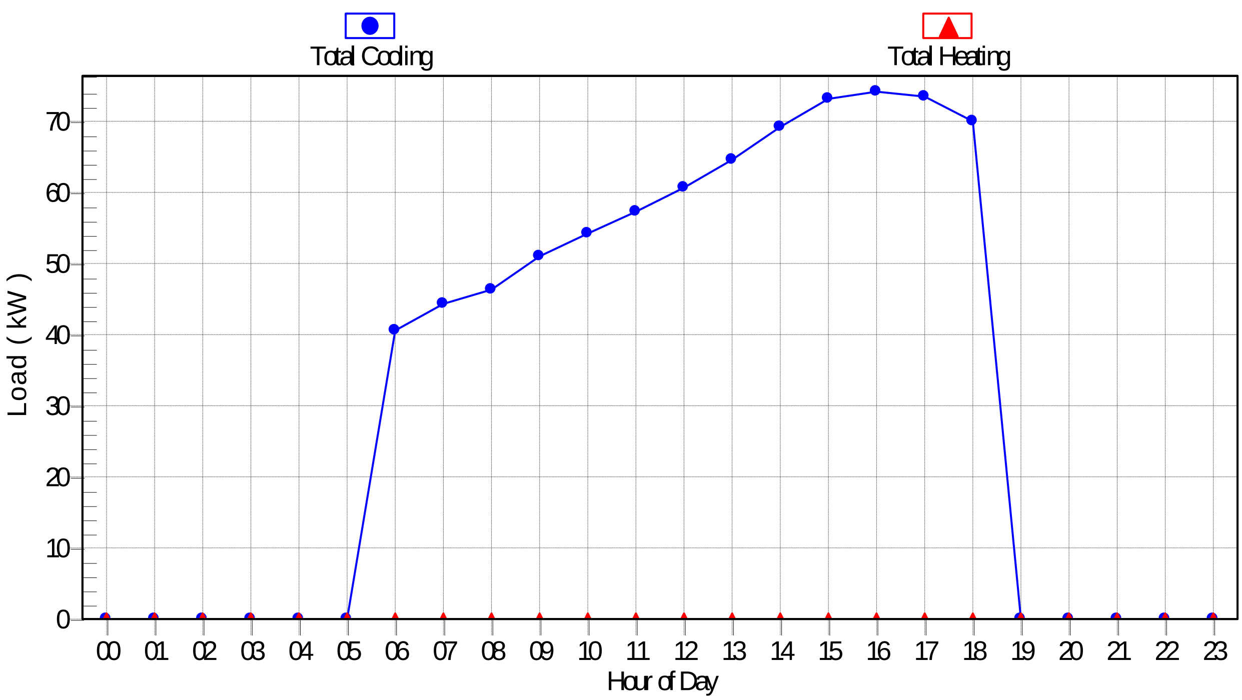
Hr	July		August		September		October		November		December	
	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB
0000	27.2	23.2	26.7	23.0	26.1	22.7	25.0	22.1	23.3	21.4	22.2	20.2
0100	26.2	22.9	25.7	22.8	25.1	22.4	24.0	21.8	22.3	21.2	21.2	19.9
0200	25.4	22.7	24.8	22.6	24.3	22.2	23.2	21.6	21.5	20.9	20.4	19.6
0300	24.5	22.5	24.0	22.3	23.4	22.0	22.3	21.3	20.7	20.4	19.5	19.3
0400	23.9	22.4	23.3	22.2	22.8	21.8	21.6	21.2	20.0	19.7	18.9	18.6
0500	23.4	22.2	22.8	22.0	22.3	21.7	21.1	20.9	19.5	19.2	18.4	18.1
0600	23.2	22.2	22.6	22.0	22.1	21.6	21.0	20.7	19.3	19.0	18.2	17.9
0700	23.5	22.3	22.9	22.1	22.4	21.7	21.3	21.0	19.6	19.4	18.5	18.3
0800	24.4	22.5	23.8	22.3	23.3	21.9	22.2	21.3	20.5	20.2	19.4	19.1
0900	25.9	22.9	25.4	22.7	24.8	22.3	23.7	21.7	22.0	21.1	20.9	19.8
1000	28.1	23.4	27.6	23.2	27.0	22.9	25.8	22.3	24.2	21.7	23.1	20.4
1100	30.6	23.9	30.2	23.8	29.5	23.6	28.4	23.0	26.7	22.3	25.6	21.1
1200	33.4	24.6	33.1	24.5	32.3	24.3	31.2	23.7	29.6	23.1	28.4	21.9
1300	36.1	25.2	35.8	25.1	35.0	24.9	33.9	24.4	32.2	23.8	31.1	22.6
1400	38.2	25.6	37.9	25.6	37.0	25.4	35.9	24.8	34.3	24.3	33.2	23.2
1500	39.5	25.9	39.3	25.9	38.4	25.7	37.3	25.2	35.6	24.6	34.5	23.5
1600	40.0	26.0	39.8	26.0	38.9	25.8	37.8	25.3	36.1	24.7	35.0	23.6
1700	39.5	25.9	39.3	25.9	38.4	25.7	37.3	25.2	35.6	24.6	34.5	23.5
1800	38.3	25.6	38.1	25.6	37.2	25.5	36.1	24.9	34.4	24.3	33.3	23.2
1900	36.5	25.3	36.2	25.2	35.4	25.0	34.2	24.4	32.6	23.9	31.5	22.7
2000	34.3	24.8	34.0	24.7	33.2	24.5	32.1	23.9	30.4	23.3	29.3	22.1
2100	32.1	24.3	31.7	24.2	31.0	23.9	29.9	23.3	28.2	22.7	27.1	21.5
2200	30.3	23.9	29.8	23.8	29.1	23.5	28.0	22.9	26.4	22.3	25.3	21.0
2300	28.6	23.5	28.1	23.4	27.5	23.0	26.4	22.4	24.7	21.8	23.6	20.6

Appendix II, Hourly Air System Design Day Loads for packaged roof top AHU

Data for May



Data for June



Appendix III, Space Design Load Summary for packaged roof top AHU

TABLE 1.1.A. COMPONENT LOADS FOR SPACE " Passenger coach " IN ZONE " Zone 1 "						
	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jun 1600 COOLING OA DB / WB 40.0 °C / 26.3 °C OCCUPIED T-STAT 23.0 °C			HEATING DATA AT DES HTG HEATING OA DB / WB 26.0 °C / 15.6 °C OCCUPIED T-STAT 21.0 °C		
		Sensible	Latent		Sensible	Latent
SPACE LOADS	**Details**	**(W)**	**(W)**	**Details**	**(W)**	**(W)**
Window & Skylight Solar Loads	16 m²	2479	-	16 m²	-	-
Wall Transmission	181 m²	9732	-	181 m²	0	-
Roof Transmission	79 m²	839	-	79 m²	0	-
Window Transmission	16 m²	736	-	16 m²	0	-
Skylight Transmission	0 m²	0	-	0 m²	0	-
Door Loads	6 m²	513	-	6 m²	0	-
Floor Transmission	0 m²	0	-	0 m²	0	-
Partitions	0 m²	0	-	0 m²	0	-
Ceiling	0 m²	0	-	0 m²	0	-
Overhead Lighting	96 W	89	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	119	6514	4189	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	1200	0	-	0	0
Safety Factor	10% / 10%	2210	419	10%	0	0
>> Total Zone Loads	**-**	**24312**	**4608**	**-**	**0**	**0**

TABLE 1.1.B. ENVELOPE LOADS FOR SPACE " Passenger coach " IN ZONE " Zone 1 "						
				COOLING	COOLING	HEATING
	Area	U-Value	Shade	TRANS	SOLAR	TRANS
	(m²)	(W/(m²-°K))	Coeff.	(W)	(W)	(W)
W EXPOSURE						
WALL	85	3.018	-	5778	-	0
WINDOW 1	5	3.237	0.887	247	1200	0
E EXPOSURE						
WALL	80	3.018	-	3279	-	0
WINDOW 1	5	3.237	0.887	216	659	0
DOOR	4	1.703	-	91	-	0
DOOR GLASS	2	5.340	0.880	149	273	0
N EXPOSURE						
WALL	8	3.018	-	388	-	0
WINDOW 1	3	3.237	0.887	136	448	0
S EXPOSURE						
WALL	8	3.018	-	288	-	0
WINDOW 1	3	3.237	0.887	136	172	0
H EXPOSURE						
ROOF	79	2.177	-	839	-	0

Appendix IV, Air System Design Load Summary for packaged roof top AHU

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jun 1600 COOLING OA DB / WB 40.0 °C / 26.3 °C			HEATING DATA AT DES HTG HEATING OA DB / WB 26.0 °C / 15.6 °C		
ZONE LOADS	Details	Sensible (W)	Latent (W)	Details	Sensible (W)	Latent (W)
Window & Skylight Solar Loads	16 m²	2479	-	16 m²	-	-
Wall Transmission	181 m²	9732	-	181 m²	0	-
Roof Transmission	79 m²	839	-	79 m²	0	-
Window Transmission	16 m²	736	-	16 m²	0	-
Skylight Transmission	0 m²	0	-	0 m²	0	-
Door Loads	6 m²	513	-	6 m²	0	-
Floor Transmission	0 m²	0	-	0 m²	0	-
Partitions	0 m²	0	-	0 m²	0	-
Ceiling	0 m²	0	-	0 m²	0	-
Overhead Lighting	96 W	89	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	119	6514	4189	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	1200	0	-	0	0
Safety Factor	10% / 10%	2210	419	10%	0	0
>> Total Zone Loads	**-**	**24312**	**4608**	**-**	**0**	**0**
Zone Conditioning	-	24955	4608	-	-112	0
Plenum Wall Load	20%	2779	-	0	0	-
Plenum Roof Load	70%	1958	-	0	0	-
Plenum Lighting Load	30%	29	-	0	0	-
Return Fan Load	2583 L/s	0	-	2583 L/s	0	-
Ventilation Load	952 L/s	16413	19454	952 L/s	-5341	0
Supply Fan Load	2583 L/s	3998	-	2583 L/s	-3998	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	5%	1216	-	5%	0	-
>> Total System Loads	**-**	**51348**	**24062**	**-**	**-9451**	**0**
Central Cooling Coil	-	50100	24062	-	-9445	0
>> Total Conditioning	**-**	**50100**	**24062**	**-**	**-9445**	**0**
Key:	**Positive values are clg loads Negative values are htg loads**			**Positive values are htg loads Negative values are clg loads**		

Appendix V

Ventilation Sizing Summary for Packaged Rooftop AHU

1. Summary

Ventilation Sizing Method .... **ASHRAE Std 62-2001**
Design Ventilation Airflow Rate .... **952** L/s
Uncorrected Outdoor Airflow Rate .... **952** L/s

2. ASHRAE Std 62 Analysis Details

	Sizing Data For: Terminal boxes full open
Critical Space	Passenger coach
Critical Space Outdoor Air Fraction ('Z')	0.369
Uncorrected Outdoor Air Fraction ('X')	0.369
Corrected Outdoor Air Fraction ('Y')	0.369
Required System Ventilation Airflow (L/s)	952

3. Space Ventilation Analysis Table

Zone Name	**Mult.**	**Floor Area (m²)**	**Maximum Occupants**	**Maximum Supply Air (L/s)**	**Required Outdoor Air (L/s/person)**	**Required Outdoor Air (L/(s-m²))**	**Required Outdoor Air (L/s)**	**Uncorrected Outdoor Air (L/s)**	**Uncorrected Outdoor Air Ratio 'Z'**	**Corrected Outdoor Air (L/s)**
Zon1										
Passenger coach	1	79.2	119.0	2582.6	8.00	0.00	0.0	952.0	0.369	952.0
Total				**2582.6**				**952.0**	**0.369**	**952.0**

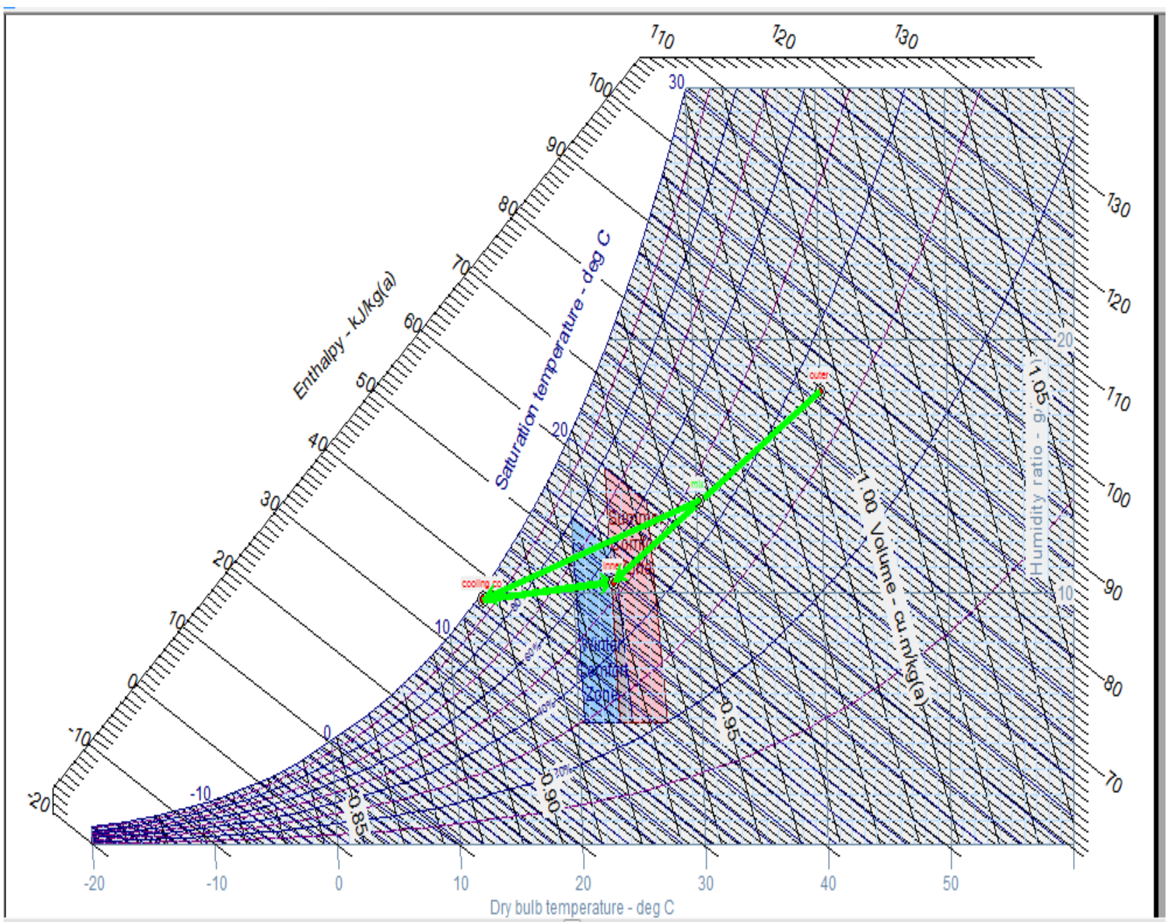
Appendix VI

Zone Design Load Summary for packaged roof top AHU

Zone 1	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jun 1600 COOLING OA DB / WB 40.0 °C / 26.3 °C			HEATING DATA AT DES HTG HEATING OA DB / WB 26.0 °C / 15.6 °C		
	OCCUPIED T-STAT 23.0 °C			OCCUPIED T-STAT 21.0 °C		
ZONE LOADS	Details	Sensible (W)	Latent (W)	Details	Sensible (W)	Latent (W)
Window & Skylight Solar Loads	16 m²	2479	-	16 m²	-	-
Wall Transmission	181 m²	9732	-	181 m²	0	-
Roof Transmission	79 m²	839	-	79 m²	0	-
Window Transmission	16 m²	736	-	16 m²	0	-
Skylight Transmission	0 m²	0	-	0 m²	0	-
Door Loads	6 m²	513	-	6 m²	0	-
Floor Transmission	0 m²	0	-	0 m²	0	-
Partitions	0 m²	0	-	0 m²	0	-
Ceiling	0 m²	0	-	0 m²	0	-
Overhead Lighting	96 W	89	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	119	6514	4189	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	1200	0	-	0	0
Safety Factor	10% / 10%	2210	419	10%	0	0
>> Total Zone Loads	**-**	**24312**	**4608**	**-**	**0**	**0**

Appendix VII

psychometric chart at 721m elevation



Appendix VIII

1 Technical Description of YZ25G Hard-seat Passenger Car

1.1 General Technical Specification

Conditions of Use:

- Maximum configuration: 20car per trainset
- Ambient temperature: -40^{0}C ~ +40^{0}C
- Maximum relative humidity ≤95%
- Platform height ,Suitable for platform height with 300mm 、500mm and1250mm, distance between platform edge and railway center is 1750mm.
- Maximum line gradient ≤30‰

1.2 Main technical specification

Railway gauge 1435mm

Maximum operation speed 120km/h

Emergency brake distance on straight line (with 30% overload and with initial speed 120km/h) ≤800m

Minimum negotiable curve

- Single car 100m
- Coupling 145m

Ride index W≤2.5

Noise (120Km/h) ≤68dB（A）

1.3 *Main dimension*

Carbody length 25500mm

Carbody width Approx 3105mm

Height between rail top to car top(empty car) 4433mm

Vehicle center distance 18000mm

Height from rail top to coupler center (empty car) 880+10mm or 880-5mm

Height from rail top to inter car crossing pedal (empty car) 1333mm

Height from rail top to floor surface (empty car) 1283mm

Carbody center plate to rail top (empty car) 780mm

1.4 *Carbody steel structure*

Body steel structure will use overall carrying beam cylindrical structure, side wall will use flat plate. Thickness of sheets and profiles which ≤ 6mm will use nickelchromium weathering steel, thickness which ≤ 2.5mm will use 05CuPCrNi, thickness which between 3 ~ 6mm will use 09CuPCrNi-B (Q295GNHL), plate thickness more than 6mm will use ordinary carbon steel. Air conditioning seat, toilet and wash room iron floor whichever easily corrosive will use stainless steel.