ADDIS ABABA INSTITUTE OF TECHNOLOGY SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINERRING

Thermal Comfort for Passenger Train from Addis Ababa to Dire Dawa

BY |

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(Railway stream)

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ABSTRACT

Air conditioning in passenger train is an effort to obtain controlled environmental conditions in accordance with the needs of passenger comfort and the passenger does not feel cold or hot. However, passengers will feel comfortable during the trip despite travel long distances and ravel times. Thermal comfort for the passengers inside the carriage is influenced by external and internal loads.

The thermal environment when the passenger train travel from Dire dawa to Addis Ababa is hot that affect the driver's and passengers' health, performance and comfort. Due to spatial and temporal variation of state variables and boundary conditions in the vehicle cab, the heating, ventilating and air-conditioning (HVAC) does not have to be designed to provide a uniform environment, especially because of individual differences regarding to physiological and psychological.

The objective of this paper focused on the design of a passenger railway air conditioning system through the route of Addis Ababa to Dire Dawa using Hourly Analysis program (HAP) software for estimating the thermal loads analysis and system design ,and Analytical method to estimate a thermal load only.

Finally, the total cooling load using both methods HAP and Analytical methods 49.9kw and 48.74kw respectively, we designed the vehicle HVAC system and select the suitable Roof mounted package unit(RMPU) . Based on the design, conditioned the passenger compartment temperature at 23.5^{o}C with a humidity of 50% during winter or summer can feel comfortable. When the air condition inside the car less than 23 ^{o}C with RH 60% felt uncomfortable.

Keywords: HAP, Thermal comfort, Air conditioning

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NOMENCLATURES

TPU	Thermoplastic polyurethane
A_c	carbody thermal envelope area
A	The apparent solar irradiation
A/C	Air conditioning
A_d	Doorway area
B	Atmospheric extinction coefficient
CAC	car air conditioning
CATIA	Computer aided three dimensional interpolate aided
CLTD	The cooling load temperature differential
CWB	Coincident Wet Bulb Temperature
d	Declination
DB	Dry-bulb
D_f	Doorway flow factor
Dif	Diffuse Radiant Load
$\dot{Q}_{Dir}$	Direct radiation load
DP	Dew Point Temperature
D_t	Doorway open-time factor
E	Effectiveness of doorway protective device
Fm	Density factor
F_{sa}	Lighting special allowance factor
F_{ul}	Lighting use factor
h	Hour angle
H_d	Doorway height
HAP	Hourly analysis program
HBM	Heat Balance Method and
h_i	Enthalpy of infiltration air
h_r	Enthalpy of refrigerated air
HVAC	Heating ventilation and air conditioning
I_{DN}	Direct radiation
MDR	Mean Daily Range

P	Number of doorway passages
PVB	Polyvinyl butyral
Q	Thermal energy transfer rate
Q_R	Transmitive of the surfaces
Q_{Ref}	Reflected radiation loads
RH	Relative humidity
SC	Solar coefficient
SHGC	Solar heat gain coefficient
S_R	Surface area of the roof exposed on direct sun ray
U	carbody thermal transmittance
UA	Factorthe car body heat loss per unit temperature difference
W	Total light wattage,
WB	Wet-bulb
WFM	Weighting Factor Method
β	Altitudes angle
ΔT	Exterior to interior temperature differential
Θ_0	Time door simply stands open,
Θ_d	Daily (or other) time period, h
Θ_p	Door open-close time, seconds per passage
ξ	Wall azimuth angle
ρ_i	Density of infiltration air
ρ_r	Density of refrigerated air

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CHAPTER ONE

INTRODUCTION

1.1 Background

Passengers in a railway travel are adversely affected by infiltration of air unpleasantly laden with dust due to open windows. This is more so in case of high speed passenger carrying trains. Air-conditioning of railway coaches is, therefore, necessary for the maximum comfort and well being of passengers in a railway travel. In keeping with modern trend, air-conditioning of coaches for upper class travellers and lately even for lower class travellers shall be introduce by the Railways corporation to keep the passenger comfort during the travel. .

The term 'Air Conditioning' was coined by Mr. S.W. Cramer in 1906 while he was making efforts in putting the air in a fit condition for the textile industry. The term has since come into use in its broader sense implying control of any or all of the physical or chemical properties of air within any enclosure. 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.” Odour control is another factor concerning comfort which has been subsequently included in the above definition.

Any change in these conditions results in a change in the physiological functions of the body and the body tries to adjust itself to the changing outside conditions. The performance of adjustments takes time and sensation of comfort or discomfort would depend upon the quick or slow adjustment. Often the adjustment may not be reached with consequent increase in discomfort.

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.

In the early stage of automotive history, the vehicle was just seen as an auxiliary tool for locomotion and for the transport of heavy goods [1]. The cabin spaces were open to the environment, requiring the passengers to choose their clothing according to the weather conditions. In the further automotive development and with increasing expectations of the customers in terms of comfort, closed passenger compartments were introduced which required heating and cooling of the interior. The first heating ventilation and air conditioning (HVAC) units consisted of some heat-able clay bricks and ice blocks for heating-up and cooling-down the passenger compartment. The ventilation was implemented by tilting windscreens or vents [2].

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 use roof mounted HVAC units. In addition to the HVAC units and depending on the application, other HVAC components like floor heaters (strip, radiant, or forced-air type), duct heaters, ducting, dampers, diffusers, exhaust fans, booster fans, etc., may need to be integrated into the passenger car design.

All material related to the HVAC equipment selected considering the applicable contractual Reliability, Maintainability, and Safety (including fire safety) criteria.

Nowadays the role of vehicles in the world has totally changed. They are no longer considered as a simple tool for locomotion. Mobility has become a substantial part and advancements in technology have arose new appealing opportunities concerning automotive HVAC systems. Railway HVAC units cannot be seen exclusively as luxury equipment, but have also a significant influence on road and rail safety. Thermal discomfort can be a physical strain and leads to fatigue. Therefore, only a thermal balanced driver is considered to be observant [3].

The proportion of vehicles with air-conditioning has dramatically increased in recent years. An automotive HVAC unit has to provide the most possible amount of comfort under all environmental situations [5]. They can consist of a number of electrical motors to control the air flow inside the passenger compartment and take into account the environmental information of up to a dozen sensors [6]. However, this enormous amount of environmental data as well as the fact that thermal comfort is a complex interaction between those and additional physiological variables, complicates the control and development strategy.

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.

The air-conditioning is usually not activated when there is nobody in the car or when the engine is not running, resulting in occurrence of extreme microclimate conditions . Nevertheless, conditions for human thermal comfort should be the same as in buildings. Standards for thermal comfort suggest that combination of microclimatic parameters of indoor environment (air and radiant temperature, air velocity and relative humidity) and individual parameters (clothing and activity) should be within certain limits, spatially homogeneous and steady state. Vehicle HVAC system obtains desired interior environment by introducing the cooled/heated/dried air into the cab air through the system's outlets, usually placed on the instrument panel and in the front footwell region. From these reasons, obtaining both the uniform and comfortable conditions in the cab is not a simple task. Other restrictive factors will be shape and dimensions of the cab interior, demands for visibility, and other technological features regarding air distribution system. Considering the preferred microclimatic conditions under the different outdoor conditions, there are also inter- and intra-individual differences within the passengers, for example age, gender, mood etc. Furthermore, passenger's clothing is set according to outdoor conditions. Consequently, desired local microclimatic parameters within the cab could be very different among the passengers.

As the system control is based on the change of airflow characteristics (temperature, velocity and direction). Obtaining the preferable environment in warm ambient inside the vehicle cab with proper distribution of conditioned air would lead to improvement of the efficiency of vehicle HVAC system, owing to higher comfortable operative temperature and higher air velocities and to less dependency from size and shape of the cab. In order to determine these values, it is necessary to be familiar with human thermal sensation, thermal characteristics of human body and human thermoregulatory system.

1.2. Railway Coach Air Conditioning System

1.2.1 Air conditioning in trains

The work to design an air conditioning system for railway applications requires considerable experience and first-class engineering quality to meet the many different challenges involved;

Here are some of the main aspects:

- Thermal, acoustic and electrical performances levels expected in the ventilation, heating, cooling and pre-conditioning modes,
- Shocks and vibrations generated during running and/or by other sources ,
- Climatic operating conditions, variations in carriage occupation rates, sudden cut-offs in the electric power supplies, time for putting into service, compliance with maintenance intervals, etc.
- Reliability, maintainability and diagnostics.

1.2.2 Requirements of railway coach air-conditioning system

- Supplying clean fresh air at a controlled uniform temperature,
- Catering, within the confines of the Railway carriages to the continuously changing number of passengers,
- Providing for heating as well as cooling on a train that travels through areas of widely differing climate during its journey,
- Operation of the equipment from power generated, stored and controlled on the train.

1.2.3 A/C Equipment in Railway Coaches

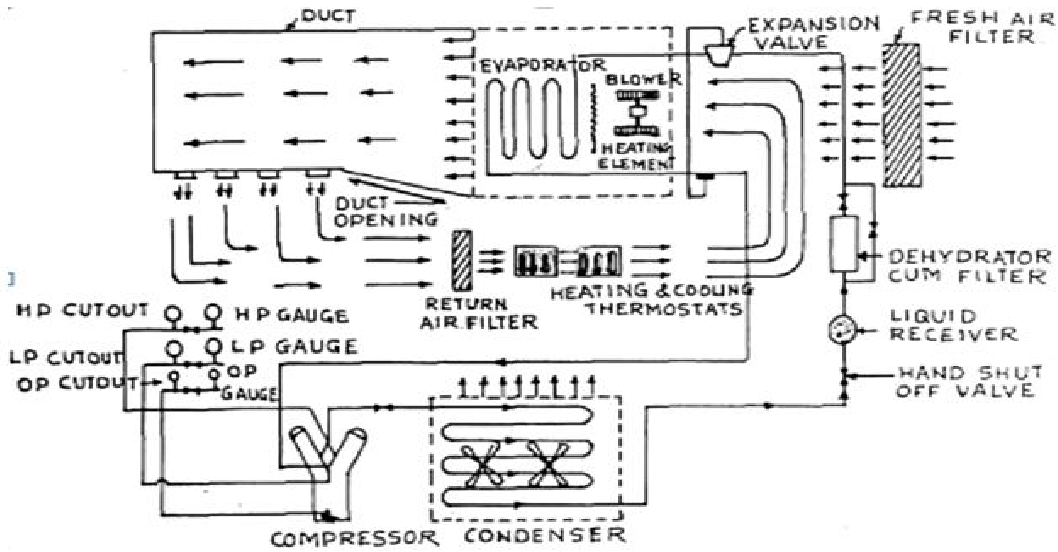


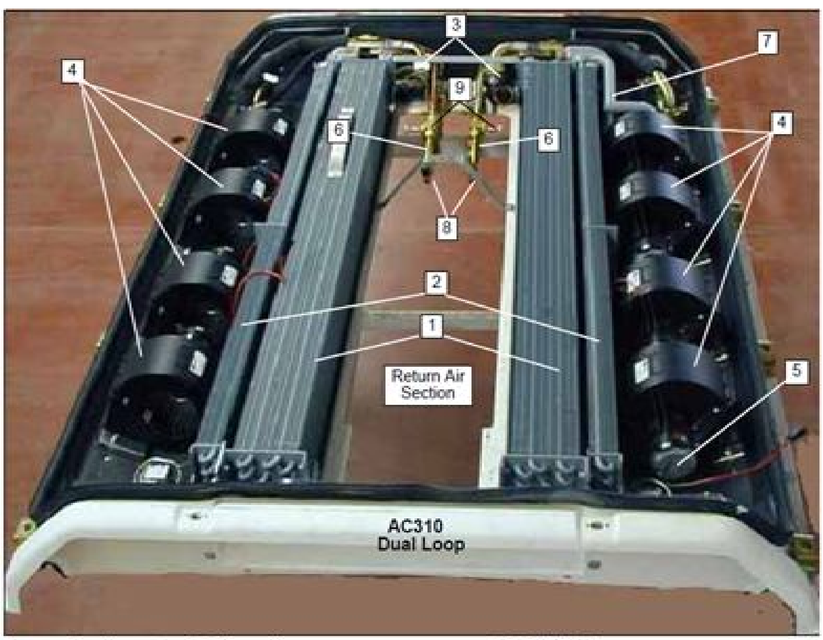
Fig 1:Schematic diagram of air conditioning system for A/C coaches[6]

This consists of the following: -

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

1.2.3.1 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. The liquid line solenoid valve closes when the system is shut down to prevent flooding of coils and the compressor with liquid refrigerant. The evaporator coils provide heat transfer surface for transferring heat from air circulating over the outside of the coil to refrigerant circulating inside the tubes; thus providing cooling. The heating coils provide a heat transfer surface for transferring heat from engine coolant water circulating inside the tubes to air circulating over the outside surface of the tubes, thus providing heating. The fans circulate the air over the coils. The air filters remove dirt particles from the air before it passes over the coils. The thermostatic expansion valve meters the flow of refrigerant entering the evaporator coils. The heat valve controls the flow of engine coolant to the heating coils upon receipt of a signal from the controller.



1. Evaporator Coil Assembly
2. Heater Coil
3. Expansion Valve
4. Evaporator Blower Assembly
5. Evaporator Motor
6. Sight Glass
7. Heating Line
8. Access Port
9. Liquid Line Solenoid

Fig 2: Evaporator section components[7]

1.2.3.2 Compressor

The refrigerant vapour drawn from the evaporator is compressed by means of a multi cylinder reciprocating compressor and compressed to a pressure ranging from 9.85 to 14.79bar according to the load demand. The work done due to compressor raises the temperature of the refrigerant vapour.

1.2.3.3 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.

1.2.3.4 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.

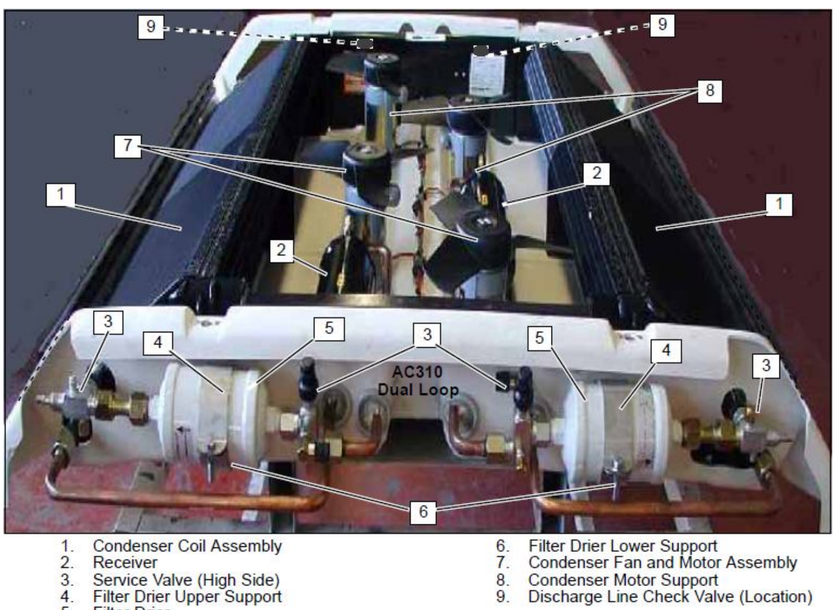


Fig 3: Condenser section components [7]

The condenser coils provide heat transfer surface for condensing refrigerant gas at a high temperature and pressure into a liquid at high temperature and pressure. The condenser fans circulate ambient air across the outside of the condenser tubes at a temperature lower than refrigerant circulating inside the tubes; this results in condensation of the refrigerant into a liquid.

The filter-drier removes moisture and debris from the liquid refrigerant before it enters the thermostatic expansion valve in the evaporator assembly. The service valves enable isolation of the filter-drier for service.

The receiver collects and stores liquid refrigerant. The receiver is also fitted with a pressure relief valve which protects the system from unsafe high pressure conditions.

1.2.3.5 Gauge panel

Gauge panel consists of pressure gauges (HP, LP, and OP) and pressure cutouts to protect the compressor against,

(i) High pressure,

(ii) Low pressure and

(iii) Low oil pressure.

i 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 preset value. The plant cannot be restarted unless the cut-out is reset manually.

ii. 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 a preset value. It protects the system against unduly low evaporator temperatures and formation of frost on the evaporator.

iii. 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.

1.2.3.6 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.

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.

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 standardised for all types of air conditioned coaches

b . Main air duct

The conditioned air from the evaporator unit is blown into the main air duct by means of two centrifugal blower fans driven by a motor with double extended shaft; The air is distributed to each compartment through adjustable diffusers. In our the case, the conditioned air from the main air duct is distributed along the hall through longitudinal apertures suitably set at factory. 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 350metre/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.

1.2.3.7 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 Only copper pipes.

1.2.3.8 Wiring

All wiring has been done by means of multistranded 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.

1.2.3.9 Temperature setting

The temperature inside the air-conditioned compartment is controlled by mercury in glass thermostats with different settings. 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.

1.2.4 Air Flow

The paths for ambient air through the condenser and coach air through the evaporator are illustrated the figure below.

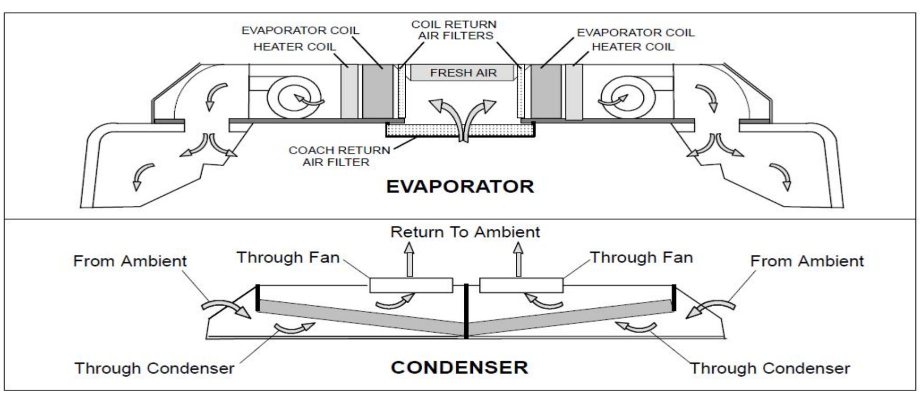


Fig 4: Air flow pattern through Evaporator and condenser[8]

1.2.5 Air Conditioning Refrigeration Cycle

When air conditioning (cooling) is selected by the controller, the unit operates as a vapour compression system using R134a or R407c as a refrigerant.

The main components of the system are the A/C 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 refrigerant leaves the receiver and passes through the receiver outlet/service valve, through a filter-drier where a descecant keeps the refrigerant clean and dry.

From the filter-drier, the liquid refrigerant then flows through the liquid line solenoid valve to the sight-glass and then to the thermostatic expansion valve. The thermal expansion valve reduce 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 (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 bus. 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.

1.3 Thesis Objective

1.3.1 General Objective

In this thesis, a railway air conditioning system is designed when the train travel through hot climate region from Addis Ababa to Dire dawa using Hourly analysis program (HAP) software and Analytical method based on *ASHRAE for the Design and Application of Heating, Ventilation and Air Conditioning Equipment for Rail Passenger Vehicles* and selected the suitable HVAC equipments .

1.3.2 Specific Objectives

The specific objectives of this research are as follows:

- To develop 3D and 2D -Duct work modelling with RMPU using CATIA software,
- To design duct work using duct sizer software,
- Compare the calculated cooling Load in both methods,
- Based on the total cooling load and air flow rate selected suitable HVAC equipments.

1.4 Statement of the Problems

Thermal comfort is crucial for the occupants in a compounded or enclosure such as the passenger compartment of a vehicle. Recent design such as glass in the vehicle styling, tightening fuel economy constraints and the more environmental safe refrigerant and reduce condenser air flow at the idle have made it more difficult to achieve occupant thermal comfort.

The need to improve the thermal comfort with the passenger vehicles is important not only has the passenger comforted but also their health. Slight temperature and air flow imbalance can easily cause respiratory malfunction in humans especially impact of shear flows is combined with large temperature differential (hot-cold or cold-hot).

1.4.1 Sub-problems

The following sub-problems have to be considered:

- Thermal comfort levels may vary in time and location,
- A passenger cabin is a very inhomogeneous and transient environment,

- Environmental variables related to human thermal comfort are not independent from each other and some of them can hardly be calculated,
- Thermal comfort is a highly subjective term and may also depend on psychological parameters.

1.5. Limitation and Scopes of the Study

1.5.1 Limitation of the study

The following limitations have been defined for this thesis:

- Only concepts of air conditioning design will be considered,
- 3-D and 2-D duct work Modelling will be restricted to CATIA,
- Only using the route of Addis Ababa to Dire-dawa,
- There is a direct travel which means no passenger in and out throughout the rout ,the door don't open every stations**.**

1.5.2 Scopes of the study

The scopes of this study will be covered:

- Design the thermal comforts for the occupants in the interior of the passenger trains.
- Using only HAP software and Analytical method to estimate the passenger thermal load and designing the HVAC system in a train.
- To determine the duct size using duct sizer software.
- Using only CATIA software to develop 2D and3D models of duct .
- Estimating the thermal loads through the route of Addis Ababa to Dire dawa and the air conditioning system design to achieve comfortable condition inside the trains with passengers.
- Using analytical method only to estimated the thermal load and compare the result with the HAP output data.

1.6 Thesis Organization

This thesis have total of eight chapters. This chapter has briefly introduced Railway Coach Air Conditioning System and A/C Equipment in Railway Coaches, the objective, the problem statement of the thesis, the limitation and scopes of the thesis.

The second Chapter presents the relevant concepts of railway air condition and human thermal comfort sensation and thermal comfort measurement technologies.

The third Chapter presents Design of Railway Air Condition Theoretical Basis ,general consideration when we designing railway air conditioning system and comfort parameters .

The fourth Chapter deals with the thermal load analysis using mathematical formulation with the aid of ASHRAE hand book for the Design and Application of Heating, Ventilation and Air Conditioning Equipment for Rail Passenger Vehicles.

The Fifth Chapter Deals with thermal load analysis and system design with the aid of Hourly Analysis Program (HAP) software and generate the input and output resulting data.

The sixth chapter Deals with the selection of RMPU and present the design of diffuser and grilles, duct work, selection of suitable fan.

The seventh chapter presented result and discussion of the thesis.

The eighth chapter is the last chapter which Gives ideas for Conclusion,recommendation future works in the study.

CHAPTER TWO

LITERATURE REVIEW

The term 'Air Conditioning' was coined by Mr. S.W. Cramer in 1906 while he was making efforts in putting the air in a fit condition for the textile industry. The term has since come into use in its broader sense implying control of any or all of the physical or chemical properties of air within any enclosure. 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.” Odour control is another factor concerning comfort which has been subsequently included in the above definition[1].

Local designer, manufacturer, and supplier of heating, ventilation and air-conditioning systems Booyco Engineering has been awarded a contract to supply air-conditioning systems for 110 new 19E locomotives being developed by the Union Carriage & Wagon Partnership (UCWP). This forms part of the Transnet Freight Rail main line locomotive investment programme, that aims to get the country’s rolling stock and attractive effort back to standard. “Booyco Engineering will supply a 8.3-kW system rated at an ambient temperature of forty-five degrees Celsius; however, the unit is engineered to accommodate periodic increases in temperatures when the locomotive goes through a tunnel,” says Booyco Engineering technical director Pieter de Koning**.** The Booyco air-conditioning system features a side sliding arrangement that fits behind the driver’s cab. This is supported by a cassette frame, in order to accommodate sliding rails and ducting interfaces. Louvres on the dashboard and on the ceiling pump out cooled air and return the air for filtration. The system is entirely electronically controlled from a panel on the driver’s main control panel.“Booyco Engineering is essentially supplying a climate-control system, as it does not just cater for heating and cooling. It also makes provision for ventilation and demisting. “All the driver has to do is select the required speed and temperature and the system will maintain this automatically. The system features an electronic interface with the locomotive’s central management system, which means that any failures or problems can be reported. The first prototype was delivered in July last year, with final production delivery scheduled for the end of this year. In addition to the air conditioning, Booyco Engineering is also supplying two foot well heaters for each locomotive. These will operate completely independent of the main system. The actual fitment of the system into the 19E locomotive takes Transnet’s fit-on/fit-off maintenance philosophy into account. Fit-on and fit-off time is about 15 minutes. Being an integrated unit, the system simply slides into place, is fastened down and the electrical connections made. This makes for easy maintenance and serviceability. The UCWP is responsible for the overall mechanical design, systems integration, fabrication and assembly of the 19E locomotives. The first locomotive is scheduled to leave the UCWP works in February, 2008, to be followed by extensive commissioning and final acceptance into Transnet's network. The hundred and tenth loco- motive is scheduled to leave the UCWP works in March, 2011, five years from the implementation date to the delivery of the final locomotive. The 19E locomotive will encompass the newest technological developments in the freight rolling stock sector. Features include a single-driver cab locomotive BoBo axle configuration (two axles a bogie and two bogies for each locomotive). The BoBo can negotiate coal Link's tight curves, thereby minimising costly wheel wear. The cab will be equipped with comfort features such as an air conditioner, a refrigerator, and a hot plate. A chemical toilet will also be installed at the rear end of the locomotive. A 25-t axle will make for a total locomotive weight of 100 t. Dual voltage (3 kV dc/25 kV ac) will allow the 19E locomotives to operate on any Transnet Freight network with the exception of the oryx network, which uses a 50kV-ac supply. Booyco Engineering has an established record in the design, engineering, and manufacture of customised and standard cooling solutions for a myriad of applications from mining to military and materials handling[9].

The other passenger railway air conditioning was invented by L M. Anderson and Samuel M. Anderson. This invention relates to the conditioning of air with the air supply and relates more particularly to methods and apparatus for the purification and temperature conditioning of air.

In most buildings and in passenger vehicles for reasons of economy, the air from the occupied space is continuously re-circulated through the conditioner, the air supplied to the occupied space being made up of a small volume of fresh air and a much larger volume of re-circulated air. It has been determined that the coughing of one passenger in a railway passenger car, for example, adds many germs to the air in his or her vicinity, which air is re-circulated through the conditioner and supplied with the germs to the other passengers. The usual air conditioners especially of the dry surface or coil type as distinguished from washers, do not effectively remove bacteria from the air. Another recognized disadvantage of reconditioning re-circulated air is that odours are not removed and this objection applies particularly to the use of coil type conditioners in railway passenger cars.

This invention provides means for removing bacteria and odours from air and for supplying substantially pure air to a conditioner for temperature and where desired, humidity conditioning. The conditioner may be of the coil type although it is preferred that air washers be used for the reason that experience has shown that not only is the air treated by such conditioners more wholesome but more accurate humidity and temperature control may be obtained. An object of this invention:

- To effectively purify air.
- To remove bacteria from air to be conditioned.
- To remove odours from air to be conditioned.
- To provide in an air conditioning system an air purifier and conditioner.

Another and more limited object of the invention is to provide within the limited space available, an air purifier and conditioning system for a passenger vehicle.

Due to the necessity of avoiding using passenger space, the space for air conditioning apparatus is limited. It is customary to mount the air conditioners in one end and in the roof zone of a railway car. It is usual with such arrangements to draw the re-circulated air through a ceiling grille underneath the air conditioner, fresh air being drawn in through the vestibule. According to this invention, it is proposed to purify the re-circulated air on its way to the conditioner. The proper purifying apparatus occupies considerable space so in one embodiment of this invention, the purifier is located overhead the passenger space in the opposite end of the car from the air conditioner and supplies pure re-circulated air to the conditioner.

It is preferred that a spray type conditioner be mounted in one end of the car. This conditioner receives and sprays refrigerated water in summer and may contain the usual heat exchange coils for heating in winter. The fans of the conditioner draw in fresh air through the vestibule grille containing the usual filter and supply it to the conditioner. The conditioner also receives purified re-circulated air for recondition as will now be described.

The purifier may be an electro-static precipitator with this type of purifier, microscopic foreign particles are removed from 'the air and deposited for collection by electro-static action. The fans are associated with the precipitator and draw in re-circulated air from the car through the grille and pass it through the purifier. The ultra-violet sterilizer mounted on the-output side of the precipitator may also be used for purifying the re-circulated air drawn in by the fan. This lamp may be used alone for sterilizing the air or may be used with or alternately with the precipitator as desired by manipulation of the switches which act to connect the precipitator and lamp to the rotary converter or other suitable source of electric energy. The lamp which generates very strong ultra-violet light through the ionization of a gas such as mercury vapour.

The fans which draw in re-circulated air from the passenger space; pass it through the precipitator and if the sterilizer is used, over it, and then discharge the purified air through the duct which extends the length of the car into the chamber enclosing the conditioner and fans. The fans draw in outside air through the grille and the mixed outside and the purified re-circulated air is passed through the conditioner where it is conditioned.

The conditioned air from the conditioner is forced through the duct and discharges there from through the small openings into the passage from which it passes into the passenger space through the openings below the deflector. The ducts may be arranged alongside each other thereby conserving both space and material.

Means may be provided of course for passing some outside air into the purifier and means may also be provided for passing some re-circulated air directly into the conditioner. Likewise the purifier could discharge air directly into the car. It is preferred however, that the air supplied to the purifier be all re-circulated air which is more likely to be contaminated by odours and germs and that the conditioner handle directly introduced outside air when is likely to be substantially free from contamination. With the arrangement illustrated, all foreign matter added within the car to the air is removed with the result that the savings in refrigeration due to the reconditioning of re-circulated air are accomplished, while pure air is supplied to the conditioner. The precipitator removes the small oily particles responsible for odours as well as a large percentage of the bacteria. During an epidemic or when it is desired to take unusual precautions, the ultraviolet lamp may be used with the precipitator. The ultra-violet lamp is believed to be more effective as far as destroying germs is concerned since its light acts upon invisible germs, the minute size of which might prevent them being acted upon by the electro-static precipitator. Of course the ultra-violet lamp could be mounted in advance with respect to air flow of the precipitator.

The fans mounted with the electric purifier are not absolutely necessary as the fans of the conditioner will draw the re-circulated air through the sterilizer.

The sterilizer as well as the conditioner may, of course, be located elsewhere in or on the car than overhead passenger space.

The claimed of this:

a. An air conditioning system for a passenger vehicle comprising an air conditioner located in one end of said vehicle, an electric air purifier located in the other end of said vehicle, means for passing outside air into said conditioner, means for passing air re-circulated from the passenger space, into said purifier, and a pair of overhead longitudinal ducts arranged alongside each other between said conditioner and said purifier, one of said ducts receiving air from said purifier and discharging same into said conditioner, and the other of said ducts receiving air from said conditioner and discharging same into the passenger space of said vehicle.

b. An air conditioning system for a passenger vehicle comprising an air conditioner located in one end of said vehicle, an electric air purifier located in the other end of said vehicle, means for passing outside air into said conditioner, means for passing air re-circulated from the passenger space, into said purifier, a pair of longitudinal ducts, means for passing air from said conditioner into one of said ducts, means for passing air from said purifier through the other of said ducts into said conditioner, and means connecting with the duct from said conditioner for directing the air from said conditioner into the passenger space of said vehicle. L M. Anderson and Samuel M. Anderson[10].

Welstand, J., Haskew, H. stated that “Air conditioning operation is proven to have a significant impact on the emissions and fuel economy; *e.g.* A/C usage can increase NOx emission from 15% to 100%’’. Welstand *et al.* studied the effects of A/C on vehicle emissions and fuel consumption. [11].

Lambert, M. A., and Jones, B. J. stated that “The A/C power consumption of mid-size cars is estimated to be higher than 12% of the total vehicle power during regular commuting’’ [12].

Besides emission and fuel efficiency, passengers' comfort is another major factor that should be considered in the design of new vehicles. In fact, auto manufacturers pay a significant attention to driver and passenger comfort which is directly linked to A/C system. This trend is evidenced by new features such as multi-zone climate control and heated/cooled seats that can be found even in recent compact vehicles. Based on these observations, it is of both environmental and economic interest to seek new methods to improve the efficiency and performance of A/C systems of vehicle [13].

*Fanger, P. O***.** stated that" 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 A/C system''. The function of A/C 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 A/C research and applications as the basis for comfort assessment [14].

Ingersoll, J. 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 [15].

ASHRAE Handbook of Fundamentals ASHRAE Handbook of Fundamentals provides two major thermal load calculation methodologies: Heat Balance Method (HBM) and Weighting Factor Method (WFM) [16].

Kamar, H. stated that" HBM is the most scientifically rigorous available method and can consider more details with less simplifying assumptions. 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. However, when more detailed information of the vehicle body and thermal loads is available, HBM is the preferred choice'' [17].

*Zheng, Y., Mark, B., and Youmans, H***.** 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.

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.

The method accuracy was validated using wind tunnel tests. The results showed the predicted cooling load is very close to the tested value, and the deviation between calculated and tested heat loads is smaller with fresh air mode than that with recirculation mode [18].

Ding, Y. and Zito, R. Ding and Zito also used a lumped model for the cabin and solved the corresponding transient heat transfer differential equation analytically. Their analytical solution can be used as a benchmark for basic problems like the cool-down test. This paper describes a first order differential equation that relates the cabin heat transfer coefficient, discharge panel temperature and discharge volumetric air flow to the average interior temperature. The solution to this equation leads to an overall understanding of automotive air conditioning designs and tests [19].

CHAPTER THREE

DESIGN OF RAILWAY AIR CONDITION THEORETICAL BASIS

3.1 General Considerations Designing Railway Air Condition

ASHRAE Standard 55[20], 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 [21].

The following factors, typically encountered during rail travel, should 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.

3.2 The comfort parameters

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.

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[22].

Other factors influencing thermal comfort are air quality (dust content; microorganism content; gases and vapours; smells; ion content; electrical and electrostatic fields), noise, lighting, colour scheme, etc. While these factors do not have a direct effect on ambient temperature, they may influence the subjective perception of thermal comfort .

The comfort criteria parameters and design features stated herein are based on practices that have been utilized by the rail transit industry and have resulted in acceptable levels of railcar comfort.

3.2.1 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.

3.2.2 Temperature Variation

Depending on the type of service (e.g., urban, commuter, or intercity), conditions that influence the temperature balance in the passenger compartment can change rapidly. Frequent exterior door openings and the entry and outlet of passengers through multiple doorways exposed directly to the passenger occupied areas characterize urban and some commuter rail operations.

3.2.3 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. In any case, these events are considered temporary in nature and may result in a higher percentage of discomfort for some people. The maximum recommended relative humidity (RH) during rated design cooling mode operation is 60%.

3.2.4 Thermal Recovery Rate

The need for a properly designed passenger rail car HVAC system to maintain stable interior environmental comfort throughout its specified range of external ambient conditions and changing passenger loads is 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. Two related specified criteria are the pull-up/pull-down time and the thermal recovery following door open/closed cycles for passenger loading and unloading.

3.2.4.1 Pull-up and Pull-down Time

The pull-up/pull-down time is the time required to raise (heating mode) or reduce (cooling mode) the temperature of the vehicle from an initial average air temperature, before passengers occupy the vehicles, to the specified temperature control range. The initial condition may be specified to be at an energy conserving standby/set-back control temperature (commonly called layover temperature) or the pullup/ pull-down requirement may be specified to start at a maximum heat or cold soak temperature, intended to simulate a vehicle being stored for a long period of time with the HVAC system turned off. The criteria are normally evaluated from the time the HVAC system is activated to the time when the average temperature reaches the design range. Key considerations in calculating the time required to reach the control temperature range is the available heating or cooling capacity, the carbody thermal insulation, and the thermal inertia of the interior fittings, seating, and lining materials.

3.2.4.2 Temperature Recovery during Revenue Operation with Door Cycling

Another condition where thermal recovery is important in passenger rail service is the ability to respond rapidly to unconditioned air entrance and resulting abrupt interior air temperature changes from door openings and passenger ingress and egress during normal revenue service. This condition is most prevalent in multi-door urban transit service, but is also a factor to a lesser extent on commuter and long distance service. The main means of mitigation for door openings is car pressurization as it tends to maintain an outflow of interior conditioned air and minimize the inrush of unconditioned outside air through the doorways. Unlike the pull-down/pull-up condition, the thermal inertia of "pre-conditioned" interior appointments helps to retain interior comfort despite rapid air changes if the door openings are of short duration.

3.2.5 Surface Temperatures

3.2.5 .1 Influence on Thermal Comfort

In addition to the previously discussed temperature and humidity conditions that influence the passenger environment, 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.

3.2.5 .2 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.

3.2.6 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.

3.2.7 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.

3.2.8 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. In our case grilles or diffusers mounted on ceiling. 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.

3.2.9 Heating Air Supply Temperature

With the HVAC equipment working in stable conditions, the supply air temperature be controlled between the interior temperature control point plus -13°C. Discharge temperature swings below set point will increase the possibility of discomfort from normal air circulation. Conversely, significant higher temperature excursions beyond this range have proven to result in thermal stratification that may cause passenger discomfort from the vertical temperature non-uniformity. In addition, the resulting uppermost high temperature thermal strata can also significantly bias the temperature control sensors in or near the ceiling.

3.2.9.1 Cooling Air Supply Temperature

With the HVAC equipment working in stable conditions, supply air temperature during cooling operation should not be more than -4^{0}C below the average vehicle interior temperature. The design, insulation, and thermal isolation of the air distribution ducts and diffusers should be such to prevent surrounding surface temperatures from falling below the conditioned area's dew point temperature to help avoid condensation formation.

3.3 Ventilation

3.3.1 Introduction

This section provides information for the supply of outside ventilation air into air conditioned passenger railcars where smoking is not permitted.

3.3.2 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 carboy 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).Comprehensive research on the air quality on rail passenger vehicles of different types, passenger density, and service schedules is not currently available; therefore, the ventilation rates recommended based on a combination of accepted current best practice for passenger railcar HVAC design and requirements from similar environments contained in ASHRAE Standard 62.1, Ventilation for Acceptable Indoor Air Quality. The suggested ventilation rate recommendations provided herein (Table 1) are expressed in litter per second (l/s) per person at design passenger load. Passenger density on most transit vehicles varies significantly throughout the day from a few passengers during off-peak time to full capacity, including standees, at peak operating periods. The design passenger load, defined in the vehicle technical specification, is the basis for determining the ventilation and cooling capacity sizing of the HVAC equipment. Exceptional passenger loads, above the design capacity, may also be defined in the Technical Specification, but these infrequent short term conditions are generally not considered in the ventilation criteria. Guidance for outside air quantities on other enclosed passenger compartments such as berthing compartments can be found in ASHRAE Standard 62.1.

Table 1 Recommended Ventilation Rates litter per second per Person at Design Conditions

	Urban service	Commuter	Intercity
Minimum Recommended ventilation Rate(l/s)	3.54	4.72	5.664

3.3.3 Variable Ventilation Control

Although past passenger railcar HVAC systems usually have staged cooling and heating controls to maintain the desired interior temperature, supply and ventilation airflow is typically fixed at the design airflow rates. 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.

3.3.4 Air Exhaust

As previously noted, 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 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. In addition, in localized areas where concentrated heat is generated (equipment lockers, food galleys, etc.), or objectionable odours 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. This may require higher ventilation rates than the minimum recommendation provided in Table 1.

Typical exhaust rates for passenger rail car toilet compartments range from 23.6 l/s to 94.4l/s depending on the size of the room. The exhaust of electrical lockers and food preparation apparatus varies significantly depending on the size and design of the equipment. Specific guidance for exhaust design can be found in the ASHRAE Fundamentals handbook.

3.4 Rail Passenger Vehicle Thermal Load Analysis

3.4.1 Introduction

AC 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 is worked out.

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. Compounding these issues are the operational characteristics of rail passenger vehicles.

Door-cycling air infiltration can represent a substantial transient heating or cooling load and should be addressed as appropriate in the heating and cooling load analysis.

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.

3.4.2 Climatic and Environmental Design Criteria

An excellent source of climatologically design data is the *Climatic Design Information* chapter of the ASHRAE *Handbook Fundamentals*. The actual climatologically data conditions existing we got form Ethiopian Metrology Service Agency throughout the track alignment but we take the critical hottest region. In some cases, supplemental surveys of right-of-way environmental characteristics may be utilized.

3.4.2.1 Cooling Ambient Design Conditions

ASHRAE 0.4%, 1.0%, and 2.0% 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.

3.4.2.2 Heating Ambient Design Conditions

Unless otherwise indicated in the project technical specification, ASHRAE 99.6% heating dry-bulb temperature data is recommended for determining the required heating capacity of HVAC equipment.

3.4.2.3 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 dry bulb(DB) temperatures.

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

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

The net heat flux (W) 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.

3.4.2.4 Roof-Mounted and Under frame-Mounted Equipment

Localized under frame and roof temperatures may be higher than those calculated from climatologically 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 radioactive heat gains must be accounted for as required.

3.4.2.5 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.

3.5 Car body Heat Gain

3.5.1 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.

3.5.2 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.

3.5.3 Car body Conductive Heat Gain

3.5.3.1 Empirical Analysis

Empirical data from environmental testing may be available to the car builder for use in calculating car body heat transfer. When available, the use of test data is generally recommended in lieu of car body thermal-performance data obtained through computation. 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 2 for various rail passenger vehicle constructions and applications.

Table 2 Car body Thermal Performance for Various Vehicle Types

[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 Carbody	*24.5*
Light Rail Vehicle, Steel Carboy	*24.5*
Light Rail Vehicle, Double-Articulated, Composite-Clad Carbody	*24.5*
Light Rail Vehicle, Aluminium Car body	*24.5*
Bi-Level Commuter Car, Aluminium Carboy	*33*

Using data from Table 2 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. Carbody conductive heat gain or loss at design temperatures is the product of the UA factor and the exterior/interior temperature differential.

The "UA" value is Calculated as per the following formula:

$$UA = \frac{\dot{Q}}{\{(T2 - T1)\}} \left[\frac{w}{k}\right] \text{--} (1)$$

where: $\dot{Q}$ - total heat applied to the car interior in watts,

T_2 - average car interior temperature (ºC)

T_1 - average Climate Room ambient temperature (ºC)

3.5.3.2 Computational Analysis

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

$$\dot{Q} = UA(\Delta T) \text{--} \textbf{(2)}$$

Where:

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

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

A - carbody thermal envelope area [m^2]

ΔT -exterior to interior temperature differential [K]

This equation assumes an average thermal transmittance for the entire carbody 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 recommends use of the values in Table 2, adjusted as required for specific vehicle construction or car builder experience. Carbody glazing and opaque surface solar heat gain should be accounted

3.5.4 Carbody 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.

3.5.5 Glazing Conductive Heat Gain

If the vehicle UA factor is known or is to be estimated using data from Table 2, 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 carbody 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.

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 [23].

$$h_o = 0.6 + 6.64\sqrt{V} \quad \text{---(3)}$$

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

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[$\frac{\mathrm{w}}{\mathrm{m^2\ k}}$] is recommended for both heating and cooling load calculations.

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 $\frac{\mathrm{W}}{\mathrm{mK}}$. Polycarbonate may be included in the glazing system to provide additional impact resistance; this material has a thermal conductivity of approximately 0.2 $\frac{\mathrm{W}}{\mathrm{mK}}$. 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 $\frac{\mathrm{W}}{\mathrm{mK}}$.

3.6 Glazing Solar Radiation Heat Gain

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. Technical specifications typically do not permit contribution from solar radiation heat gain in determining the vehicle's heating load.

3.7 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 3 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 3 Recommended Heat-Gain Values for Passengers and Operators

Application, Activity	Total Heat[W/person]		% Sensible Heat
	Adult Male	Adjusted,M/Fa	
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

3.8 Outside Air (Ventilation) Heating and Cooling Loads

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

Outside-air load calculations are usually prepared using psychometric software. In this thesis we prepared psychometric chart using cytsoft psychometric software and calculator 1.0(Appendix IV).

3.9 Passenger Door Cycling Air-Infiltration Heating and Cooling Loads

Infiltration due to passenger door cycling can represent significant heating and 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.

ASHRAE *Handbook–Refrigeration*. 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 psychometrics properties of interior and exterior air.

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

$$\dot{Q}_d = \dot{Q}_t D_t D_f (1 - E) \text{--} (4)$$

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 (q):

$$\dot{Q}_t = A_d(h_i - h_r)\rho_r\left(1 - \frac{\rho_i}{\rho_r}\right)^{0.5}(gH_d)^{0.5} F_m \text{---} \mathbf{(5)}$$

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

$$F_m=[\frac{2}{(1+(\rho_r/\rho_i)^{1/3})}]^{1.5} \text{-----(6)}$$

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

$$D_t = \frac{(P\Theta_p + 60\Theta_o)}{3600\Theta_d} \text{-----}\mathbf{(7)}$$

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

The equation above determines total cooling load per passenger door, with stabilized airflow and the door open 100% of the time (fully established flow).

3.10 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:

$\dot{Q}_l$ = $WF_{ul}F_{sa}$ --(8)

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

W - total light wattage[W]

F_{ul} - lighting use factor and ,

F_{sa} - lighting special allowance factor

3.11 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.*

3.12 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.

CHAPTER FOUR

THERMAL LOAD ANALYSIS USING ANALYTICAL METHOD

4.1 Thermal Load Analysis

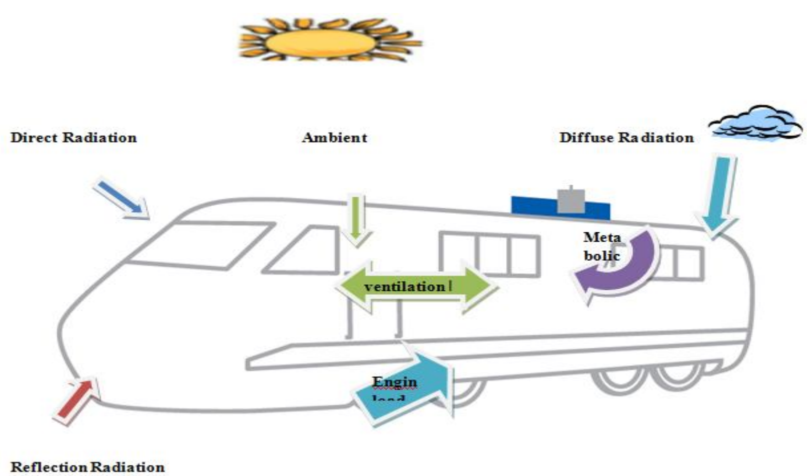


Fig 5: Schematic representation of thermal loads in a typical train cabin.[24]

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.

The use of safety factors in heating and cooling load calculations is not required or recommended. Safety factors may be desired to account for unknowns or uncertainties in the analyses or to provide additional capacity to accommodate specified interior temperature pull-up or pull-down times. If required, these safety factors may be applied during equipment selection.

4.1.1 Cooling Load Analysis

To determined the cooling load analysis' in this thesis first we collected the necessary data, like Dire Dawa weather conditions, location ,elevation ,latitude ,altitude ,daily Mean range, and hourly dry bulb and wet bulb data from Ethiopian Meteorology service Agency and the vehicle dimensions like, type of HVAC units mounted on the train, etc.... .we get from Ethiopian Railways Corporation Rolling Stocks Specifications .

Geographical Information:-

Elevation: 1200m, Latitude: $9^0 35$” N, Longitude: $41^0 45$”E

Environmental and Interior Design Criteria

- ***Ambient (cooling)***

Dry-bulb temperature corresponding to 0.4%, annual cumulative frequency of occurrence and the mean coincident wet-bulb temperature (warm),

Wet-bulb temperature corresponding to 0.4% annual cumulative frequency of occurrence and the mean coincident dry bulb temperature,

Dew-point temperature corresponding to 0.4% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature and the humidity ratio.

Mean daily range, which is the mean of the difference between daily maximum and minimum dry-bulb temperatures for the warmest month (highest average dry-bulb temperature)

DBT = Dry Bulb Temperature = 38°C;

CWB = Coincident Wet Bulb Temperature = 25.9°C;

DP = Dew Point Temperature = 19.8°C;

MDR = Mean Daily Range (K) = 20.3

The above outdoor design condition data for cooling at Dire Dawa locations in Ethiopia obtained from *Ethiopian Standard EBCS 11.*

- ***Interior (cooling)***

23.5°C DB / 50% RH ;

Cooling dry-bulb temperature/mean coincident wet-bulb temperature data are typically used to determine the required cooling capacity of rail HVAC equipment

Vehicle Information:

Vehicle type: Inter-city and commuter, two cabs, *YZ25G Hard-seat Passenger Car* from Ethiopian Railway corporation Rolling stocks specification [Appendix X]

Thermal envelope dimensions (exterior): 25500mm long x 3105mm wide x 3543mm height

Passenger-compartment window and cab side-window tilt from horizontal: 90°

Average vehicle speed: 120km/h

Carbody UA Factor:-

Vehicle type from Table 2:*Vehicle type: Inter-city and commuter , two cabs,*YZ25G Hard-seat Passenger Car UA- factor per metre of car length can be take from ASHARE recommended value for the given passenger rail vehicle is :

$$\frac{\text{UA}}{\text{vehicle length}} = 24.234\frac{\text{w}}{\text{mk}}$$

Carbody UA factor × train length $= 25.5\text{m}\times 24.234\frac{\text{w}}{\text{mk}} = 618\frac{\text{w}}{\text{k}}$

Carbody Conductive Load (including glazing)

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

For Cooling:

$$\dot{Q} = 618\frac{w}{k} \text{x } (38°\text{c} - 23.5°\text{c}) = 8.71\,\mathbf{kw}$$

Carbody Glazing Solar-Radiation Cooling Load

Side Window Area (one side only, including both cabs): = 13.43m^2

Windshield Area: = 2.3m^2

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

May is the hottest month in Dire Dawa according to the data available from Ethiopia Meteorology Agency and at a time of 13:00 .

Assume the sky to be cloudless

- Direct radiation load ($\dot{Q}_{Dir}$):

$$\dot{Q}_{Dir} = \sum_{surfaces} S\tau \dot{I}_{Dir} \cos\theta \quad \text{--}(9)$$

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

τ_R – Transmitive of the surfaces

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

Several solar radiation models are available for calculation of direct radiation from sun. One of the commonly used models for air conditioning calculations is the one suggested by ASHRAE. According to this model, the direct radiation (I_{DN}):

$$I_{DN} = \frac{A}{\exp\left(\frac{B}{\sin\beta}\right)} \quad \text{--}(10)$$

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

  3.556mm for winter,

  5.334mm for summer.

Data collected From Ethiopian Meteorology Agency:

Latitude =9°.58’and longitude = 42°.33’, April is the hottest month according to the data available from Ethiopia Meteorology Agency and at a time of 13:00 pm ,

Reflectivity of the ground = 0.6

Assume the sky to be cloudless;

May 13:00pm ,

Declination, d = 15.2

N-day of the year start from January

Hour angle, h = 43.33

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

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

$\beta = \sin^{-1}(\cos l \cosh \cos d + \sin l \sin d)$--(11)

$= \sin^{-1}(0.736)$

$= 47.4^0$

The angle is taken as +ve if the normal to the surface is to the west of south and –ve if it is to the east of south.

$\theta = \cos^{-1}(\sin\beta\cos\sum + \cos\beta\cos\alpha\sin\sum)$--12)

Altitudes angle $\beta = 47.4^0$

$I_{DN} = \frac{A}{\exp\left(\frac{B}{\sin\beta}\right)} = \frac{1080}{\exp\left(\frac{0.21}{\sin 47.4}\right)} = 812\frac{\mathrm{w}}{\mathrm{m}^2}$

$(\dot{Q}_{Dir,R}) = S_R\, \tau_R I_{DN,R} \cos\theta$ ---(13)

$S_R = L \times W = 25.5\text{m} \times 3.105\text{m} = 79.2\text{m}^2$

τ_R = is the roof surface element transmissivity = 0

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

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

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

In Each surface element of the Glass is vertical,

i.e Front ,Rear, two Sides of glass

$\tau_G = 0.5$ given

Radiant load on a train at the orientation West

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

α= [Π- (90 + 78)]×1 = 12

$\theta_{ver} = \cos^{-1}(\cos 47.4 \cos 12) = 48.54^0$

At wall side

Direct Radiant Load ($\dot{\boldsymbol{Q}}_{Dif}$)

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

= 12.2*0.5×812cos48.54

$\dot{Q}_{Dir,G}$ = 3279.5W

Diffuse Radiant Load ($\dot{\boldsymbol{Q}}_{Dif}$)

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

= 12.2×0.5×54.81w = 334.34W

Reflected radiation loads

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

=12.2×0.5×130= 793.13W

At the face of a train

Direct Radiant Load ($\dot{\boldsymbol{Q}}_{Dif}$)

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

= 3.42×0.5×812cos48.54

$\dot{Q}_{Dir,G}$ = 918.33W

Diffuse Radiant Load ($\dot{\boldsymbol{Q}}_{\text{Dif}}$)

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

$$= 3.42\times0.5\times54.81\text{w} = 93.72\text{W}$$

Reflected radiation loads

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

$$=3.42\times0.5\times130.03= 222.33\text{W}$$

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

$$\dot{Q}_{\text{Di}}=\sum_{surfaces} S\tau\dot{I}_{\text{DN}}\cos\theta_{\text{ver}} = 3279.5\text{W} + 918.33\text{W} = 4197.83\text{W}$$

$$\dot{Q}_{\text{Dif}}=\sum_{surfaces} S\tau\dot{I}_{\text{Dif}} = 334.34\text{W} + 93.725\text{W} = 428.065$$

$$\dot{Q}_{\text{Ref}}=\sum_{surfaces} S\tau\dot{I}_{\text{ref}} = 793.13\text{W} +222.33\text{W} = 1015.46\text{W}$$

$$\dot{Q}_{\text{R}} = \dot{Q}_{\text{Di}} + \dot{Q}_{\text{Dif}} + \dot{Q}_{\text{Ref}} = 4072.94\text{W} + 391.4\text{W} +927.57\text{W}$$

$$\dot{Q}_{\text{R}} = 5641.35\text{W} = 5.04135\text{kW}$$

Total Carbody Glazing Solar-Radiation Instantaneous Cooling Gain (Load) = **5.041kw**,

Cab Glazing Solar-Radiation Cooling Gain (west cab, vehicle centerline east-west):

North: $0.712\text{m}^2\times135.7\,\frac{w}{m^2}\times0.41=39.6\text{w}$

East: **0w**

South: $0.712\text{m}^2\times116.7\,\frac{w}{m^2}\times0.39=32.4\text{w}$

West: $2.3\text{m}^2\times880.245\,\frac{w}{m^2}\times0.41=830.07\text{w}$

Total Cab Glazing Solar-Radiation Instantaneous Cooling Gain (Load) =**902.07w** =**0.902Kw**

Carbody 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.

The following provides an example of one calculation method. Roof opaque-surface solar gains are considered insignificant in this paper due to the extensive roof-mounted equipment and shrouds located along the roof perimeter.

Carbody UA-factor = $618\frac{w}{k}$

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

= $361.05m^2$

Carbody average thermal transmittance =$1.71\frac{w}{m^2k}$

Total glazing area = (2 x $13.43m^2$) + (2 x $2.3m^2$) = $31.46m^2$

Glazing average U-factor (thermal transmittance) is assumed to be $5.394\frac{w}{m^2k}$ for the glazing of this vehicle.

Average opaque-surface transmittance (U):

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

Solving for $U_{opaque\ surfaces}$:

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

$$U_{opaque\ surfaces} = \frac{[\,(1.71\frac{w}{m^2k}\times 361.05m^2) - (5.394\,\frac{w}{m^2k}\times\ 31.46m^2\,)]}{(361.05m^2 - 31.46m^2)} = 1.36\frac{w}{m^2k}$$

Sol-air temperatures calculated 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 May13:00 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.

The May, 13:00 pm LST sol-air temperatures are calculated for vertical east/west surfaces (sidewalls).

A value of 0.20 is assumed for the absorptance/outer-surface heat transfer coefficient. The May13:00 pm LST hourly design day temperature is 38.8^0c ,0.8^0c higher than the 38^0c cooling design temperature. The calculated sol-air temperatures must be reduced by 0.8^0c before determining the conductive gain. Sol-air temperature is based on the design cooling ambient temperature.

Instantaneous cooling gain [$\dot{Q}$ = UA(ΔT)]

North: $1.36\frac{w}{m^2k}$x [(3.543 x3.105) – 2.3)] m^2 x (14.1)°C = 166.81w

South : $1.36\frac{w}{m^2k}$x [(3.543 x3.105) – 2.3)] m^2 x (14.1)°C = 166.81w

East : $1.36\frac{w}{m^2k}$x [(25.5 x3.105) – 13.43)] m^2 x (14.1)°C = 1260.77w

West: $1.36\frac{w}{m^2k}$ x [(25.5 x3.105) – 13.43)] m^2 x (14.1)°C = 1260.77w

Total opaque-surface instantaneous cooling gain = 2.85kw

Part of this gain is transferred to the interior and part is transferred to the surroundings by convection and thermal radiation.

Assume, 50% of the gain is considered a sensible cooling load; this percentage is based on vehicle speed/motion/wind and solar exposure.

Opaque-surface cooling load = 0.5 x 2.85kw=1.427kw

Passenger and Operator Cooling Loads

Inter-city and commuter seated passenger/operator sensible and latent cooling loads are taken from Table 3 and displayed in Table 4

Table 4 Inter-city and commuter Rail Vehicle Passenger/Operator Sensible and Latent Cooling Loads

	Seated	operator
Number of occupants	118	1
Individual sensible Heat Load[w/person]	71.52	73.86
Individual Latent Heat Load[w/person]	45.72	53.03
Individual Total Heat Load[w/person]	117.2	131.9

	Sensible	**Latent**	**total**
Total Occupant Cooling Load [kw]	8.51	5.45	13.96

Ventilation (Outside-Air) Load

we can take ventilation rate is 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.

With the aid of ASHRAE psychometric analysis software, we draw psychometric chart at the given elevation as shown Appendix using this chart and the ventilation loads are calculated and the result as shown in Table 5

Table 5 Ventilation Loads

Ventilation Load	Sensible	Latent	Total
Cooling[kw]	6.34	8.634	14.974

Passenger Door Cycling Air-Infiltration Heating and Cooling Loads

To determine the passenger-door cycling total cooling load (sensible + latent) using algorithms from the ASHRAE Refrigeration Handbook Infiltration Air Load. But for this thesis we ignored such load .

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:

$\dot{Q}_l = WF_{ul}F_{sa}$

Data obtained from [6]

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 lampa = 15 W (for a single Incandescent lamp)

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

W = 960w

$\dot{Q}_l = WF_{ul}F_{sa}$ = 0.96KW

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 6 Show Miscellaneous Internal Heat Gains can be estimated as follows

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

Fan and Fan Motor Heat Gains

In typical rail passenger vehicle applications, the fans and fan motor are located in the supply air stream 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.*

Table 7:Assume that Fan and Fan Motor Heat Gains as shown below

Cooling Load	Sensible	Latent	Total
Fan and Fan Motor Heat Gains[kw]	2	0	2

4.2 Cooling Load Analysis Summary

Table 8 The Cooling Load Analysis Summary

Cooling Loads	**Sensible [kw]**	**Latent [kw]**	**Total [kw]**
Conduction (including glazing)	8. 71	0	8.71
Passengers & Operator	8.573	5.453	13.966
Outside Air (ventilation)	6.34	8.63	14.974
Solar Radiation (glazing)	5.91	0	5.91
Solar Radiation (opaque surfaces)	1.427	0	1.427
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	36.66	14.08	50.74
Vehicle Cooling Load (Net Cooling Load) (Gross Cooling Load minus evaporator unit motor and unit internal sensibleload)	**34.66**	**14.08**	**48.74**

CHAPTER FIVE

THERMAL LOAD ANALYSIS AND SYSTEM DESIGN USING HOURLY ANALYSIS PROGRAM (HAP) SOFTWARE

5.1 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 building energy use and calculating energy costs. HAP uses the ASHRAE-endorsed transfer function method for load calculations and detailed 8,760 hour-by- hour simulation techniques for the energy analysis. This program is released as two separate, but similar products. The “HAP System Design Load” program provides system design and load estimating features.

5.2 HAP System Design Features

HAP estimates design cooling and heating loads for commercial buildings 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.
- Sizes chillers and boilers.

5.3 HAP Energy Analysis Features

HAP estimates annual energy use and energy costs for HVAC and non-HVAC energy consuming systems in a building by simulating building operation for each of the 8,760 hours in a year. Results of the energy analysis are used to compare the energy use and energy costs of alternate HVAC system designs so the best design can be chosen. Specifically, HAP performs the following tasks during an energy analysis:

- Simulates hour-by-hour operation of all heating and air conditioning systems in the building.
- Simulates hour-by-hour operation of all plant equipment in the building.
- Simulates hour-by-hour operation of non-HVAC systems including lighting and appliances.
- Uses results of the hour-by-hour simulations to calculate total annual energy use and energy costs. Costs are calculated using actual utility rate features such as stepped, time-of-day and demand charges, if specified.
- Generates tabular and graphical reports of hourly, daily, monthly and annual data.

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. Before start the program the input data must be gather .The input data such as :

- Climate data for the building/vehicle site.
- Construction material data for walls, roofs, windows, doors, exterior shading devices and floors, and for interior partitions between conditioned and non-conditioned regions.
- Building/vehicle 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, office equipment, appliances and machinery within the building/vehicle.
- Data concerning HVAC equipment, controls and components to be used

5.4 Input Data for HAP

Next, use HAP to enter climate, building and HVAC equipment data. When using HAP, your base of operation is the main program window. From the main program window, first create a new project or open an existing project. Then define the following types of data which are needed for system design work:

a. **Input Weather Data.**

Weather data defines the temperature, humidity and solar radiation conditions the building encounters during the course of a year. These conditions play an important role in influencing loads and system operation. We gather DireDawa weather condition data, from Ethiopia Metrology Service Agency for weather database and weather parameters directly entered. Weather data is entered using the weather input form.

b. Input Space Data.

A space is a region of the building/vehicle space comprised of one or more heat flow elements and served by one or more air distribution terminals. Usually a space represents a single room. However, the definition of a space is flexible. For some applications, it is more efficient for a space to represent a group of rooms or even an entire building/compartment. 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. This space data can be gathering from Ethiopian Railways Corporation Operations and Services Division Equipment Supply and Technical Services Department Rolling Stocks Specifications.

c. Input Air System Data.

An Air System is the equipment and controls used to provide cooling and heating to a region of a building. 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. The input data into the HAP as follows.

5.5 Output Design Data Generated by HAP

Once weather, space, air system and plant data has been entered, HAP can be used to generate system and plant design reports. All the result output data of design reports as shown below summary:-

CHAPTER SIX

TRAIN COACH AIR DISTRIBUTING AND DUCT WORK DESIGN

6.1 Introduction

The rail passenger vehicle HVAC system should be designed to provide a comfortable and pleasant interior thermal environment to the passengers during all expected external ambient environmental conditions [9]. For that purpose, appropriately sized and controlled heating, cooling, and air supply components are integrated into the passenger car design. Based on the cooling load analysis on the previous chapter 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 thesis we use roof mounted HVAC units. In addition to the HVAC units and depending on the application, other HVAC components like floor heaters (strip, radiant, or forced-air type), duct heaters, ducting, dampers, diffusers, exhaust fans, booster fans, etc., may need to be integrated into the passenger car design.

6.2 Vehicle Cooling

To the extent possible, cooling system and air distribution design for the passenger area compartment should minimize drafts and the temperature gradient between air at the diffuser outlet and the return air, while providing enough cooling capacity to compensate for all different cooling loads.

6.3 Vehicle Ventilation and Air Distribution

The HVAC system air ventilation, mixing, treatment, distribution, and exhaust are described in the following sections.

6.3.1 outside (Fresh) Air

Outside air is required to provide car pressurization and to provide ventilation within the car for all operating conditions.

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.

Outside air grille face velocity should be uniform as much as possible and kept to the lowest practical value possible to prevent snow and water ingress as well as pressure drop.

Automatic outside air dampers may be employed for one or any combination of the following purposes:

- To reduce pull-down or pull-up time,
- To close the outside air intakes in the event of an exterior or tunnel smoke condition,
- To reduce the equipment thermal load in extreme temporary conditions, thus preventing it from shut-down, by modulating the outside air volume,
- To adjust the required outside airflow rate based on the number of passengers and/or operation mode, for energy saving,
- To minimize the effects of pressure waves as the result of entering tunnels or passing trains at high speeds.

Dampers that have an ability to control the air volume will require additional inspection and maintenance, as necessary, of the actuating devices and related controls.

6.3.2 Return (Re-Circulated) Air

Return airflow is determined in combination with outside air requirements to achieve the evaporator coil airflow necessary to achieve the design cooling capacity. Return air is usually drawn by the HVAC unit blower through a ceiling mounted grille located below the HVAC unit mixing plenum. Another option is to draw air from smaller grilles or slots distributed in the passenger area. This design has the advantage of reducing the noise and air velocity caused by a single return air opening below the HVAC unit but requires return ducting to be installed in 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 2.04m/s.

Typical return air grille recommended features are the following:

- Rattle free construction,

- Secondary retention hangers to prevent head injury from accidental opening, and,
- Light weight/easily removable design.

6.3.3 Supply Air

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

6.3.3.1 Supply Airflow Rate

The supply (evaporator) airflow rate for this application is calculated using evaporator-coil performance information. With the aid of HAP software , a supply airflow rate of 2001l/s is provided, consisting of 952 l/s outside air and 2001l/s return air. Each of the two HVAC units on this vehicle provides a design supply airflow rate of 1000.5 l/s.

6.4 Exhaust Air

Air 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 without compromising the recommended car interior pressurization criterion. Accordingly, the total exhaust air through the exhaust system should be somewhat lower but in our case air exhausted from the passenger car through passive exhaust system.

6.5 Car Pressurization

To minimize the infiltration of unconditioned air, a positive pressurization of the passenger and cab areas is desirable. Typically, a stationary pressurization of 25pa is sufficient. It is recommended that static exhaust (e.g., damper/duct assembly) or dynamic exhaust (i.e., fan) provisions be incorporated into the design to allow for final adjustment of the car pressurization.

Special consideration should be given to vehicles to be used in high speed inter-city service to minimize passenger discomfort due to the pressure wave created during tunnel entries and passing trains.

6.6 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.

The purpose of an air duct system is to convey conditioned air from one location to another within prescribed limits of space, noise and cost. Although round duct carry the air with less friction and least material, rectangular ducts are suitable for air conditioning systems for the following reasons:

1. Space consideration as they fit easily in building construction
2. for simple reason of ease of fabrication

6.6.1 Duct design methods

The most common methods of air duct system design are

1. Velocity reduction
2. Equal friction rates
3. Static regain and

NB*: from the three methods using only equal friction method in this thesis to design the supply duct.*

a. 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.

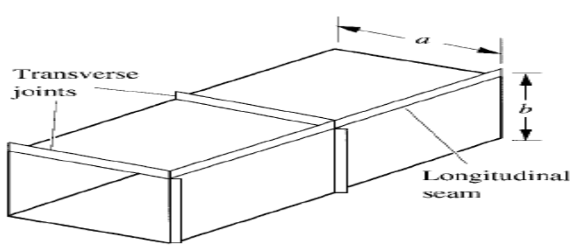


Fig 6: Rectangular duct

From the design duct chart or using Duct sizer software the supply duct can be design as shown the following summarizing duct table 9 and the 2-D duct figure and liable letter as shown the annex part .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 4.35m/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. The aperture of air diffuser has been designed, as shown the next sub topic, to deliver the required quantity of air into the compartment. This diffuser is provided with a knob to deflect the air to the required angle. By the above arrangement the air velocity inside the compartment obtained is between (0.1 m/sec. to 0.215m/sec.) at the face level of the passenger.

Table 9: Supply duct sizing Table

Main Duct Sizing Table											
1	2	3	4	5	6	7	8	9	10	11	12
Section	Length	Flow Rate	Pressure drop per	Duct	Velocity	Velocity	Fittings	Pressure Loss		Section	Cumulative Pressure
		(l/s)	metre	Size	(m/s)	Pressure	pressure loss			Total Pressure Loss	Loss
	(m)		(Pa/m)	(mm)		(Pa)	factor (C)	Fittings	Straight Duct	(Pa)	(Pa)
								(Pa)	(Pa)		
A	1.2	500.25	0.6	350×350	4.35	11.37	7.17	81.52	0.72	82.24	82.24
B	0.9	465.25	0.6	350×325	4.26	10.88	0.25	2.72	0.54	3.26	3.26
C	0.9	430.25	0.6	350×325	4.18	10.48	0.25	2.62	0.54	3.16	3.16
D	0.9	395.25	0.6	325×325	4.09	10.04	0.26	2.64	0.54	3.18	3.18
E	0.9	360.25	0.6	325×300	4.00	9.58	0.26	2.49	0.54	3.03	3.03
F	0.9	325.25	0.6	300×300	3.85	8.90	0.28	2.45	0.54	2.99	2.99
G	0.9	290.25	0.6	300×275	3.79	8.63	0.28	2.42	0.54	2.96	2.96
H	0.9	255.25	0.6	275×275	3.60	7.77	0.26	1.99	0.54	2.53	2.53
I	0.9	220.25	0.6	275×250	3.54	7.50	0.26	1.95	0.54	2.49	2.49
J	0.9	185.25	0.6	250×225	3.51	7.40	0.24	1.80	0.54	2.34	2.34
K	0.9	150.25	0.6	225×225	3.16	6.01	0.28	1.68	0.54	2.22	2.22
M	0.9	115.25	0.6	200×200	3.07	5.68	0.28	1.56	0.54	2.10	2.10
N	0.9	80.25	0.6	200×150	2.74	4.51	0.44	2.00	0.54	2.54	2.54
O	0.9	45.25	0.6	150×125	2.58	3.99	0.45	1.79	0.54	2.33	2.33
RMPU											200.00
supply Diffuser						35					35
							System Total Pressure Drop per one side				352.36
							System Total pressuer Drop pre unit				**352.36**

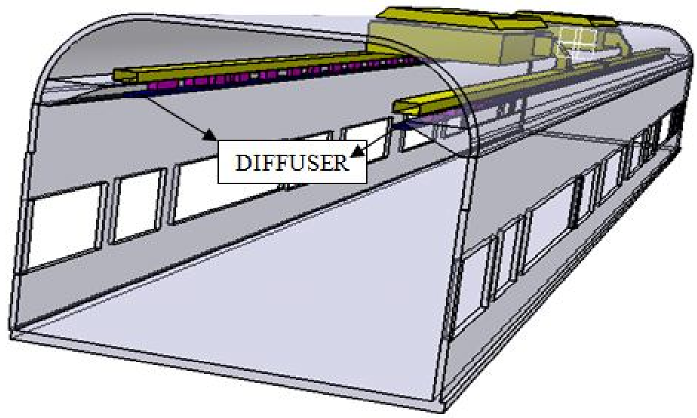


Fig 7a: Front face of the train show 3D- Supply Duct work with RMPU units and diffusers

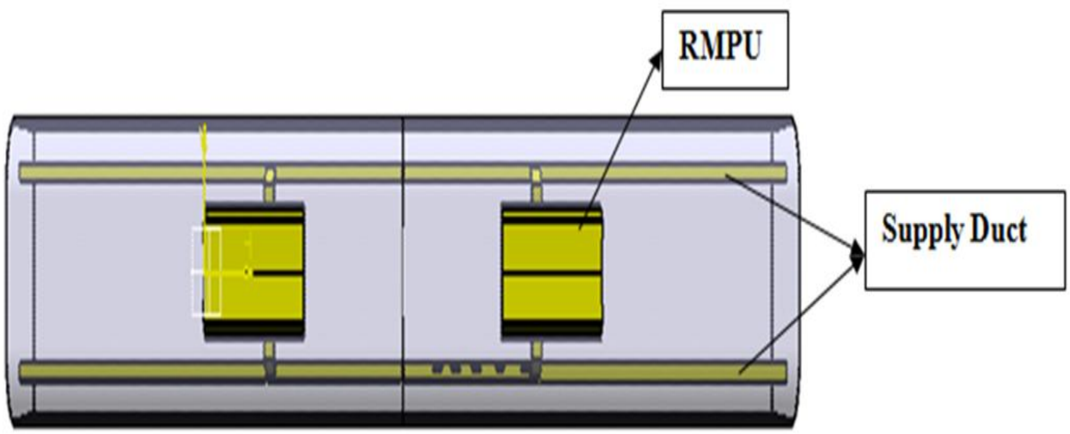


Fig 7b: Top face of the train show 3D- Supply Duct work with RMPU units

6.7 Air Diffusers

6.7.1 Passenger Compartment Diffusers

Air diffusers 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. To minimize uncomfortable drafts, air diffuser should be designed to direct outlet air along the ceiling and the sides of the car.

6.7.2 Cab Diffusers

It is recommended that cab compartment air diffusers be adjustable in volume from full-flow to minimum flow and in direction to provide the driver with the maximum adjustment flexibility. For narrow space cab compartment, spot-cooler type diffusers are favoured in over grille or linear diffusers. For larger cab compartments, a combination of grille type and spot-cooler diffusers is often the best option. To assure proper circulation of air in the cab compartment and return to the passenger area return air grille, cab doors or walls should be fitted with properly sized air transfer grilles. For this project since ceiling diffusers are the best choice for cooling & we choice this used

6.7.3 Air supply Diffuser/Grills

The choice of Diffusers and Grilles:

Choice of the linear grille diffuser or involves the assignment of the claim, performance, and data analysis. Diffusers and grilles should be selected and calculated according to the following criteria:

- The type and style
- Function
- Air volume requirement
- Throw requirement
- Pressure requirement
- Sound requirement

The volume of air per diffuser or lattice is what is needed for cooling, heating, ventilation or requirements of the area served by the unit. The amount of airflow required when related to quit, sound, pressure or design limitations, determines the proper diffuser grille or size.

Supply diffusers are selected based on throw length and noise level depending on application using flow rate.

A linear diffuser type **LDW/LDB 12/8/1 is selected.**

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.

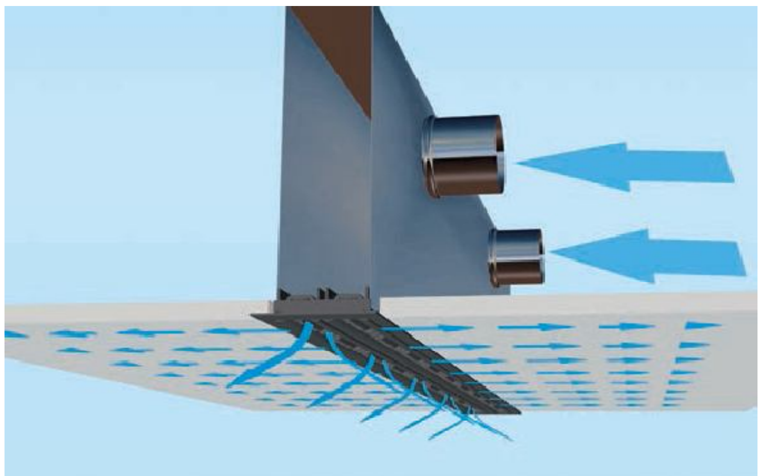


Fig 8: Linear supply Diffuser [25]

Linear supply Diffuser features are:-

- High induction effect for rapid reduction of speed and temperature differences, thus high comfort due to low air speeds in the occupied zone.
- Additional ceiling air curtain to reduce pollution on the ceiling in the diffuser area caused by high induction.
- Possibility to adjust diffusers subsequently to optimize the indoor air flow.
- Extremely low-noise with comparatively high flow rate.
- Flexible border options to match many ceiling systems.

Manufacture's catalogue provide information on the throw, drop, noise level and pressure drop for a specified air field air flow rate and a variety of sizes and types.

Based on the following parameters we can select the suitable linear diffuser:

- The volume flow rate (m^3/h/m);

- supply air temperature(t_{zu});
- Room air temperature (t_{RA}).

The selected Linear diffuser Resulting data:

Model: Linear diffuser type **LDB 12/8/1**

- pressure drop(pa) =35,
- sound power level(dB) =40,
- Extension of jet at which the maximum speed of ambient air was measure (a(C_{max}) in m) =3m,
- Maximum speed of ambient air with uniform distributed thermal load (cm/s)=21.5,
- 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.

6.7.4 Air Return / Exhaust Grills

Grillers for the package of return air from the passengers' compartment to the plenum chamber through this hole.

We use the return air system a Perforated Ceiling Grills exhaust air form the interior of the train through the return and we can use the same as supply diffuser.

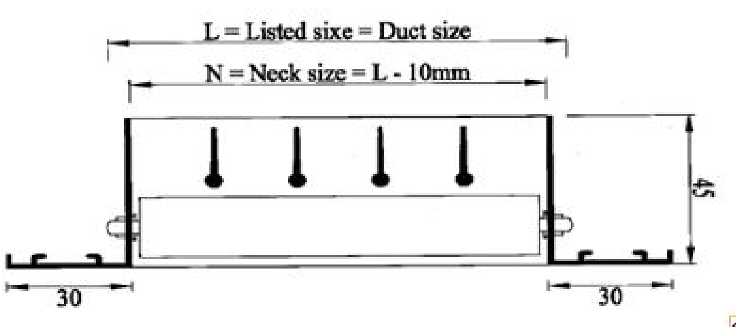


Fig 9:the return air system a Perforated Ceiling Grills[26]

Based on the return flow rate we choice the return Grills .There are six number of grills on the ceiling.

Using grill products catalogue and we selected the following type.

- Maximum security perforated Grille
- Model: AMPSG

Neck Size in mm		
Area factor in m^2		
	l/s	261.48
	m3/s	0.261
	Face vel. m/s	1
600*600	P_T in mmH_2O	0.217
	Thrown in m	3.1-12.7
1.9266	NC	<15

6.8 Selection of Roof Mounted Packaged Unit

6.8.1 Roof Mounted Packaged Train Air/Air

Fig 10: Roof mounted packaged unit

The Roof Mounted Packaged Train Air/Air units are packaged conditioners operating with R134a and R407c, equipped with electrical heater. They are suitable for overhead installation in railway wagon.

Structure: suitable for the typical stress of railway use, made of stainless steel painted with polyester resin base paints. The whole structure is designed and controlled with the F.E.M. method to withstand the stress foreseen in the railway use (in compliance with CEI-IEC-61373 standard).

Scroll compressor suitable for railway use and fitted with vibration damping supports. **Condensing section fan** suitable for railway use with on/off condensation control.

The connections between units, with fixed wall enclosure and external electrical connections are made with cable boxes equipped with mobile IP66 and a double metal sheaths "agraffatura".

In this thesis selected the following suitable Roof Mounted Packaged Train Air/Air units:-

Specifications	Roof mounted units Tramways, commuter trains, coaches, metros
PERFORMANCE	
Cooling capacity	20 -35 kW
Heating capacity	Up to 32 kW
Operating temperature	-30 to + 70 °C
Weight	up to 800kg
POWER SUPPLY	
Unit power supply	3x208V, 60Hz • 3x230V, 50/60Hz • 3x380/400V, 50Hz • 3x460/480V, 60Hz •750V DC
Heater power supply	AC ; see Unit power supply • DC up to 750V nominal
CONTROL	
Control voltage	24 to 110V
Communication	CAN Open, MVB or Ethernet
Control of external heaters	Yes
Temperature and humidity control	Yes, by microprocessor
Air distrubution control by dampers	Yes, by fresh, return and supply air dampers
REFRIGERANT	
Refrigerant circuit	Hermetic
Refrigerant	R134a or R407C

Fig 11: specification of the Roof Mounted Packaged Train Air/Air units:-

The actual weight of the A/C Package unit up to 800 Kg. The Roof mounted A/C package unit will be so designed that it interfaces with the vehicle and does not allow rainwater, washing plant fluid and condensate to enter into the vehicle. The connection of the supply air opens to the main duct and the return air opening shall be through bellows with metallic frame at both the ends. The dimension of RMPU: 2200mm×2000mm×500mm

6.9 Supply fan selection

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 (352.36pa) as show table 9 and the air volumes (500.25l/s) we can select the Fan which is used in railway air conditioning technology [20] as follows:

Application: Railway roof air handling unit integrated with Fans

The Railway Engineers group supplies fans for railway technology. Very often, these fans are integrated in air handling units, which are installed directly in the roof of the trains. Because of the compact dimensions of the units, which are connected with the supply air channel of the roof, the fans have to be designed space-saving and still powerful. At the same time maintenance openings have to be considered, to guarantee an easy access for maintenance.

The selected centrifugal fans having the following results

- Centrifugal fans (ECOFIT) Type:DRA
- Double inlet centrifugal fan with forward curved centrifugal impeller
- Variable mounting positions possible
- Voltage range: 230/400 VAC, 3-phase
- Frequency: 50/60 Hz
- Air volumes up to 15000 m^3/h
- Total pressure increase up to 900 Pa
- Power input: up to 3,000 W

The selected centrifugal fans using the following fields of application:

- Air conditioning of the passenger cabin,
- Electronics cooling
- Cooling of components

Fig 12: centrifugal fan in railway Air conditioning technology [27]

CHAPTER SEVEN

RESULTS AND DISCUSSION

The foremost outcome of this work is that design of railway air conditioning system, as a whole is absolutely essential in order to acquire a better picture of the design of railway air Conditioning system throughout the route. This paper is focused on the design of railway air conditioning ,and selection the suitable HVAC equipments and analysis of air conditioning system when the passenger train travel through the route of Addis Abba to Dire-dawa.

Two different methodologies are presented to achieve the objectives of this work. The first one Using Hourly Analysis Program (HAP) software which is used to estimate the cooling load and system design which is presented in Chapter 6. It identifies the various attributes needing to be considered for the design and selection of air conditioning systems.

It presents the result of the processing in terms of suitability parameters, which is used to show the passengers thermal comfort and capability of the air conditioning for the given application. The contribution of the work regarding HAP can be summarized as:

1. It’s used to estimate Cooling load calculations to determine equipment sizing are specific to the prevailing operating environment, application, and operating conditions. The total cooling load when the train travel through the given root is can be estimated 49.9kw, sensible coil is 34.1kw, and sensible heat ratio is 0.683provides a useful in defining the air conditioning system accurately and precisely.

2. It’s also used to determine the system design of railway air conditioning. In addition to capacity requirements, considerations for equipment selection include the available space, location, accessibility, available power source, reliability, and maintainability *and as shown Air System Sizing Summary for Packaged Rooftop AHU* output data**.**

3.Its determined and show at which month and day the maximum load occurs for packaged rooftop AHU in *appendix II (Hourly Air System Design Maximum Month and Day Loads for Packaged Rooftop AHU):* as we can see the graph the maximum cooling load is 38kw and its occurs at the month of May13 .

The second method which is used to determined only estimate the total cooling load is Analytical method with a Guideline of ASHRAE for the Design and Application of Heating, Ventilation and Air Conditioning Equipment for Rail Passenger Vehicles presented in Chapter 5 in order to achieve the objective of air conditioning system. This method which is used a different load in a passenger train which affect the thermal environment and we can take a number of assumptions and recommended parameters used to calculate the cooling load in the air conditioning system and describe the whole system. The contribution of the work regarding this method can be summarized as:

1. It's used to only estimated Cooling load calculations to determine equipment sizing are specific to the prevailing operating environment, application, and operating conditions. as we can see *Table 8 :the Cooling Load Analysis Summary*. The total estimated cooling load 48.74kw, sensible coil is 34.66kw; sensible heat ratio is 0.71.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.

2. The air conditioning system permanent function is a mathematical model characterizing the structure of the air conditioning system and also helps one to determine the air conditioning system numerical values. This numerical values are used for comparison, and then selection of the suitable air conditioning system.

The outside design temperature 38°C (DBT) and 25.9°C (CWBT) and 0.4%, annual cumulative frequency of occurrence this is the data we get from EBCS which is used to determine the required cooling capacity of rail HVAC equipment and inside design temperature 23.5°C (DBT) and 50% RH were chosen from ASHRAE comfort zone [6].

CHAPTER EIGHT

CONCLUSIONS, RECOMMENDATIONS AND FUTURE WORKS

8.1 Conclusion and Recommendations

In this paper from the beginning till the end every section that is data collection for the project which includes (weather condition, vehicle dimension and location, orientation, and vehicle construction material),as mentioned earlier the cooling load analysis can be estimated using both HAP software and Analytical method with the help of *ASHRAE for the Design and Application of Heating, Ventilation and Air Conditioning Equipment for Rail Passenger Vehicles*. We use HAP software for the analysis purpose because its time consuming and easy to describe the system and easy to evaluate, analysis and summarizing the results with the aid of tables, graphs and charts .

The total cooling load can be estimated with the aid of Hap 49.9kw and the Supply air (l/s) required for the passenger compartment is 2001 l/s. Therefore, we selected the suitable Roof Mounted Package Unit(RMPU) model is satisfied with both cooling capacities and supply air required for the passenger compartment .We have two RMPU units mounted on the roof of the train which having a the maximum total cooling capacity each unit is 35kw we can see the specification of Train Air/Air unit package .Duct is designed using Duct Seizer software based on the flow rate and the assumed pressure loss(Pa/m) & materials that take the required flow rate from the duct to the passenger compartment are selected, like diffusers & grilles based on the flow rate capacity, are carefully thoroughly done & finally fans for the power rated are selected from Fan For Railway Technology supply group catalogue with its suitable motors . The location of duct is selected based on the:

- the vehicle Space consideration ,
- Aesthetic value ,
- Relatively ease of installation.

And finally I recommend that if the Ethiopian railway corporation uses such system using through the hot regions the air conditioning system hopefully it will be successful

8.2 Future works

For this thesis everything almost that should be done has thoroughly been analyzed, may be what we feel is to continue with the heating load (the amount of heat that must be supplied to keep the interior passenger compartment at the desired temperature) design the whole the system. Even though heating load is really not the problem of Dire-Dawa to Addis Ababa route except for the few months, for the sack of completeness the design of this air conditioning system shall be added. Thus with idea mentioned above the air conditioning system will be perfect.

Finally, we suggest that there were limitations of getting data's because of a new filed in Ethiopian there is no actual condition , so for the future if the actual data's will obtain, it better to use theses so that design for the air conditioning system will be safe and the actual data derived from measurements of air inside the car and traversed areas such as temperature and humidity of landing space, passengers room, bath room, and the environment using a digital hygrometer and a laser thermometer to measure body surface temperature in the passengers space in the roof temperature, inner wall and floor . In addition to this to check the distribution of flow pattern of air flow rate and the temperature profile inside the passenger compartments using computational fluid dynamics (CFD) or Fluent software and we can see how it distribute.

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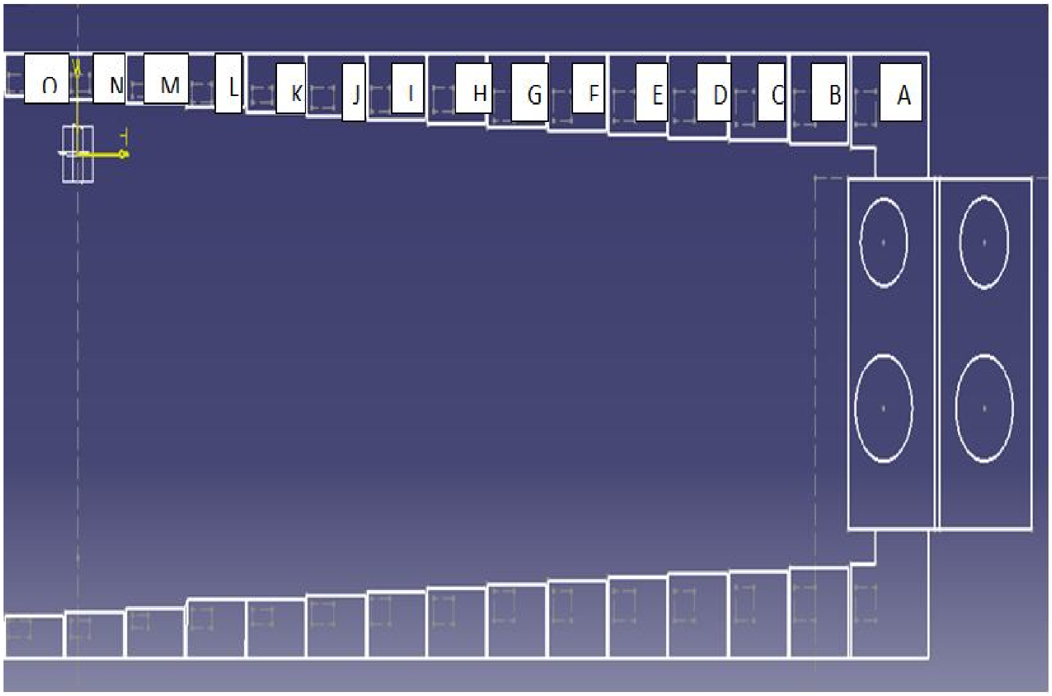
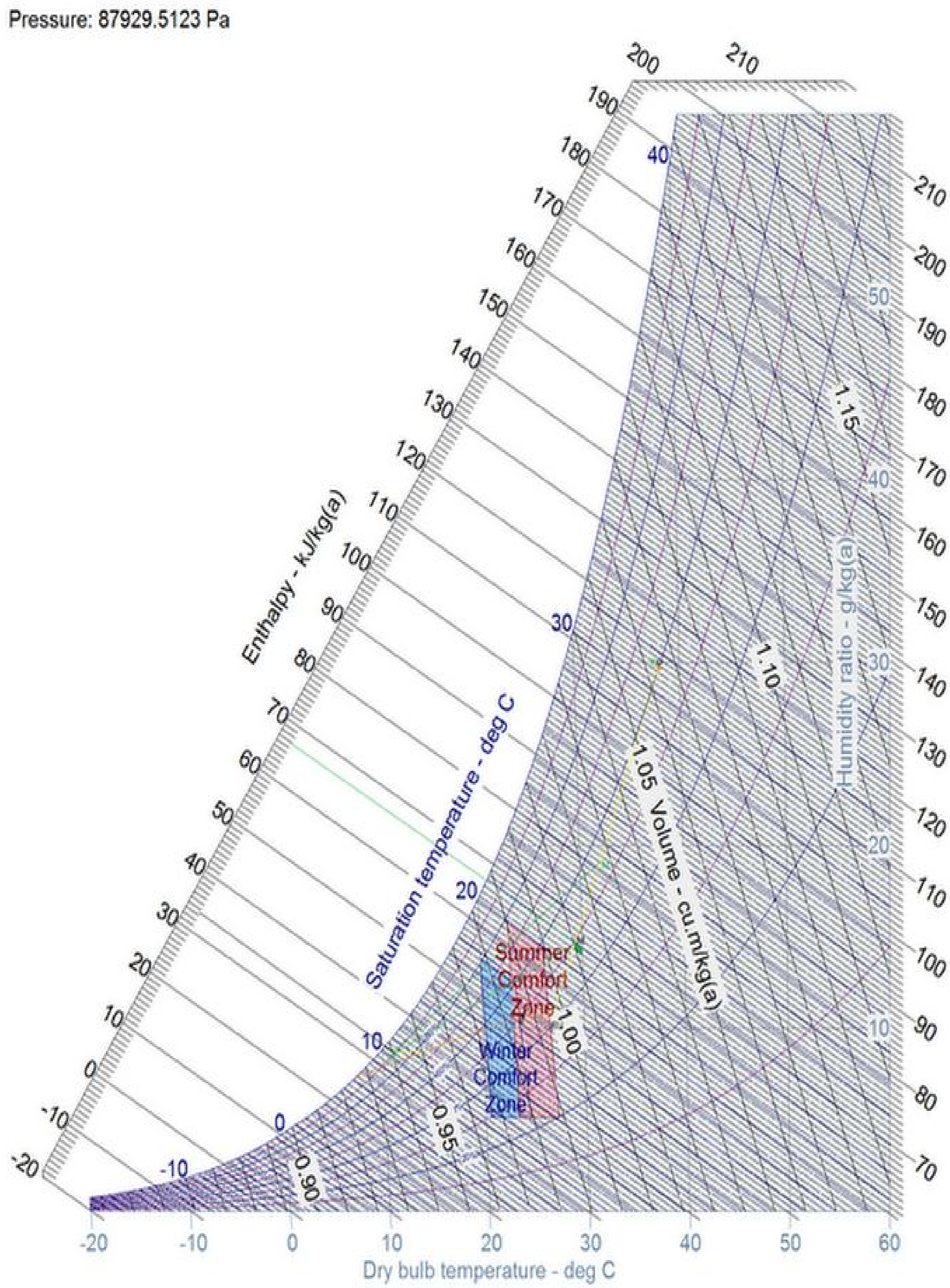
Appendix

Appendix VIII

Appendix 2-D duct work layout



Appendix IX


Pressure: 87929.5123 Pa
Enthalpy - kJ/kg(a)
Saturation temperature - deg C
Volume - cu.m/kg(a)
Humidity ratio - g/kg(a)
Dry bulb temperature - deg C
Summer Comfort Zone
Winter Comfort Zone
1.15
1.10
1.05
1.00
0.95
0.90

Appendix X

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.