EXECUTIVE BUS AIR CONDITIONING

FINAL ASSIGNMENT

Presented as a Meaning for Gaining Engineering Degree Holder

Mechanical Engineering Department

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FACULTY OF ENGINEERING

SANATA DHARMA UNIVERSITY

YOGYAKARTA

2003

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6				
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8				

AUTHENTICITY

I hereby informed that myself except a number of reviewers and references write this book.

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ABSTRACT

EXECUTIVE BUS AIR CONDITIONING

The most air-conditioned vehicle is the automobile, for which between 5 and 10 million systems are sold annually. The major contributor to the cooling load in many of these vehicles is heat from solar radiation, and in the case of public transportation, heat from people. The loads are also characterized by rapid changes and by a high intensity per unit volume in comparison to building air conditioning. Air conditioning embraces more than cooling. The definition of comfort air conditioning is the process of treating air to control simultaneously its temperature, humidity, cleanliness, and distribution to meet the comfort requirements of the occupants of the conditioned space.

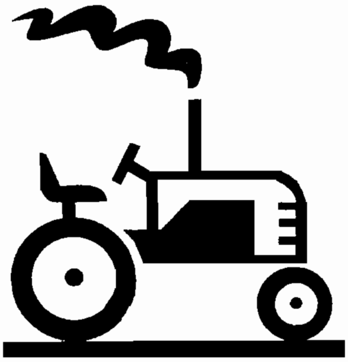
The air conditioning system on the super executive bus is packaged in a rooftop unit and uses a single compressor, driven by the bus engine, which called, is direct system. The evaporator blowers draw return air from inside the bus, through the return air filter, into the return air chamber. Fresh air also enters to the return air chamber through the fresh air filter at the front of the unit. The air mixes and passes through the evaporator coils, where it is cooled and dehumidified. The conditioned air is then discharged into the ducts on each side of the bus, to be supplied to the cabin.

MOTTO

- ❖ **Practice makes perfect**
- ❖ **Keep your heart with all diligence**
- ❖ **You will never know till you have tried**

PRESENTATION

I presented this book to my :



- *Beloved mommy and daddy*
- *Beloved grandmother and grandfather*
- *Dearest sis' and bros : Nofita, Roby, Romy, and Gustia.*
- *Dearest darling, Jason.*
- *Dian (Rembang), the real phlegmatic and best advisor.*
- *Friends in deeds and needs : Rony, Stevie, Ary (sikil), Aris (bulus), Lius, Lilik, and Prisca.*
- *Dearest best friends : Anny, Alice, Kibar, and Herman (Richie).*

PREFACE

Thesis is the fundamental project for every major in Sanata Dharma University as a meaning for gaining degree holder. The complete volume presents a course of study which provides air conditioning and refrigeration systems. Both of these are subjects of extensive study in themselves and it is well for students to be apprised of them.

Although a detailed study is beyond the scope of this book, the author believes that opening door encourages further investigation.

In conclusion the author thanks all those who have contributed to the complete of this thesis :

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The author hopes that all of you will give the benefit of your comments and suggestions.

Yogyakarta, September 2003

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CONTENTS

LIST OF FIGURES

LIST OF TABLES

CHAPTER ONE

INTRODUCTION

A. SCOPE OF AIR CONDITIONING AND REFRIGERATION SYSTEMS

The development of effective heating, ventilating, and air conditioning (abbreviated HVAC), however, was begun barely 100 years ago. Central heating systems were developed in the nineteenth century and summer air conditioning by mechanical refrigeration has grown only in the last 50 years. A refrigerating system removes heat from one part of its surroundings and discharges heat from a warmer part.

To the average person the term air conditioning usually means cooling, but this is only part of the story. The definition of comfort air conditioning is "the process of treating air to control simultaneously its temperature, humidity, cleanliness, and distribution to meet the comfort requirements of the occupants of the conditioned space".[1]

Most heating and cooling systems have at least the following components :[2]

1. A heating or cooling source.
2. A distribution system, such as ducts or piping.
3. Equipment for moving the air or water (fans or pumps).

1) Stoecker, Refrigeration and Air Conditioning, p.1
2) Edward. G. Pita, Air Conditioning Principles and Systems, p.5

4. Devices for transferring heat between the fluid and room. Examples are convectors and diffusers.

Air conditioning is used for 2 purposes, comfort or process control. Comfort refers to providing air conditions that create satisfaction for people. Process control refers to air conditions that are required to carry out or improve some operation or process. For example, certain values of humidity are required for proper operation of computers. In nuclear research facilities, very close control of air quality is required.

B. AIR CONDITIONING SYSTEMS AND EQUIPMENT

There are large number of variations in the types of air conditioning systems and the ways they can be used to control the environment in a building. Load changes, zoning requirements, space available, and costs are some of the variables that determine which type of system is to be used.

AC systems can be classified in 2 ways :

1. By cooling/heating fluid distributed, whether it is air or water.
2. According to whether the equipment is packaged together (unitary system) or separate (central system).

This classification is not according to how the system functions, but how the equipment is arranged. A unitary system is one the refrigeration and air conditioning components are factory selected and assembled in a package. This includes refrigeration equipment, fans, coils, filters, dampers and controls. A central or remote system is one where the components are all separate. Each is selected by the designer and installed and connected by the contractor. Unitary equipment is usually located in or close to the space to be conditioned. Central equipment is usually remote from the space, and each of the components may or may not be remote from each other, depending on the desirability.

Unitary or central systems can both in theory be all-air, all-water, or air-water systems, but practically, unitary systems are generally all-air systems, and limited largely to the more simple types such as single zone with or without reheat or multi zone. This is because they are factory assembled on a volume basis. Unitary systems and equipment can be divided into the following groups :

1. Room units[3]

Room units are available in two types : window units and through-the-wall units. The window unit fits in the sash opening of an existing window, resting on the sill. The through-the-wall unit fits in an outside wall opening, usually under the window sill.

Room units are available up to about 3 tons of refrigeration capacity. Their advantages are low cost and simplicity of installation and operation. Room units have no flexibility in handling high latent heat gains or changed sensible heat ratios, and therefore do not give good humidity control. Sounds level are higher than with remote equipment. Air cleaning

3) Ibid, p.279

quality is minimal because the filter remove only large particles, in order that the resistance to air flow be low.

2. **Unitary conditioners (self-contained unit or packaged units)**

This type of unit is installed in or near the conditioned space. Unitary conditioners are available in vertical or horizontal arrangements, according to the space available for the equipment. Units are available that have all components packaged except the condenser. This is a popular arrangement in private residential applications. The condenser is located outdoors and compressor, coil, and fan package in an attic or basement. Another common arrangement is a split system. The condenser and compressor are in one package, located outdoors, and the fan and the cooling coil are another package located indoors. This is especially popular with small residential heat pumps. Units are available in sizes up to about 50 tons.

3. **Rooftop units**[4]

This type of unitary equipment is designed to be located outdoors, and is generally installed on roofs. Usually, all of the refrigeration, cooling, and air handling equipment is assembled together, although the compressor and condenser may be remote. Heating equipment may be incorporated in the unit. Rooftop units may be used with ductwork and air outlets. They must have weatherproofing features not required with

4) Ibid, p.280

equipment located indoors. All electrical parts must be moisture proof, and the casing and any other exposed parts must be corrosion protected.

The advantages of rooftop units are that they do not use valuable building space and they are relatively low in cost. Units are available with multi zone arrangement, thereby offering zone controls, but humidity control is limited.

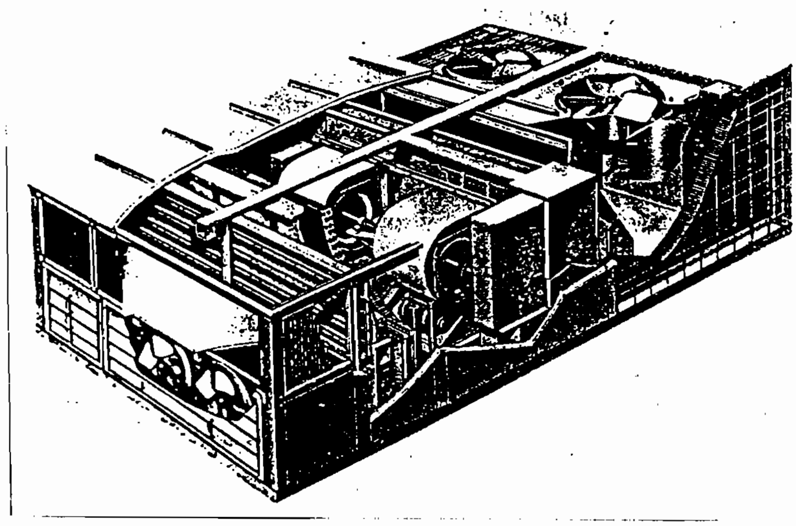


Figure 1. A rooftop unit
(Source : W. F. Stoecker, Refrigeration and Air Conditioning, p.2)

4. Air handling unit (AHU)

The air-handling unit refers to the combination of coils, fan, filters, dampers, and casing. It is also sometimes called the central air condition apparatus. There are basically two arrangements : single zone units and multi zone units. In small and medium-capacities air handling units are factory made in sections-fan section, coil sections, mixing box, filter section-in numerous sizes. Casing are usually made of galvanized sheet metal. The casing should be insulated to prevent energy losses. When cooling and dehumidifying, drain pans must be included under the coil to collect condensed moisture, and a piping drain connection must be provided which is run to a waste drain.

Fluid distribution still can be classified as following :

a) All-air systems[5]

Single zone system is the simplest all-air systems. A unit conditions and then distributes a constant volume of air through one duct to a group of rooms. The equipment provides a complete year-round air conditioning system to control both temperature and humidity. Not all the components are used in all circumstances, such as supply air fan, cooling coils and reheat coils.

b) All-water (hydronic) systems

Hydronic systems distribute hot or chilled water from the central plant to each space. No air is distributed from the central plant. Hydronic terminal units such as fan-coil units heat or cool the room air. Ventilation air can be brought through the outside wall and the terminal unit.

All-water systems for commercial use can be considerably less expensive and take up much less space than all-air systems. Water has a much higher specific heat and density than air. This means that considerably less volume of water needs to be circulated for the same

5) Ibid, p.277

amount of heat transfer. The result is that the cross-sectional are of piping is much smaller than the ductwork would be for the same job.

On the other hand, all-water systems have certain disadvantages. The multiplicity of fan-coil units means a great deal of maintenance work and costs. Control of ventilation air quantities is not precise with the small fans in the units. Control of humidity is limited.

c) Air-water combination systems[6]

Combination air-water systems distribute both chilled and/or hot water and conditioned air from a central system to the individual rooms. Terminal units in each room cool or heat the room. Most of the energy is carried in the water. Usually the air quantities distributed are only enough for ventilation and carried at high velocities. Fan-coil units can be used as the room terminal units, arranged to receive the centrally distributed air, or the air can be supplied directly to the room. However, the most common air-water system uses terminal units called induction units.

The central air delivered to each unit is called primary air. As it flows through the unit at high velocity, it induces room air (secondary air) through the unit and across the water coil. Therefore, no fans and motors are required in this type of unit, reducing maintenance greatly.

The primary air quantity in the induction system may be only about 25% or less than the total of the air volume rate of a conventional all-

6) Ibid, 278

air system. There are buildings with air-water systems requiring at outdoor temperatures as low as 30 F.

C. REFRIGERATION SYSTEMS AND EQUIPMENT

An environmental control system that includes cooling and dehumidification will require a means of removing heat from the conditioned spaces. Because heat flows only from a higher to a lower temperature, a fluid with a temperature lower than the room design temperature must be available, to which the excess room heat can be transferred. Refrigeration produces this low temperature fluid. Vapor compression and absorption refrigeration systems are both used widely for producing refrigeration required for air conditioning.

1) Vapor compression refrigeration system

The vapor compression cycle is the most widely used refrigeration cycle in practice. In this cycle a vapor is compressed then condensed to a liquid, following which the pressure is dropped so that fluid can evaporate at a low pressure. The major equipment components in the vapor compression refrigeration system functions are compressor, evaporator, condenser, and flow control devices.

a. Compressor

Positive displacement compressors function by reducing the volume of gas in the confined space, thereby raising its pressure.

Reciprocating, rotary, and screw compressors are positive displacement types.

1. Reciprocating compressor[7]

This is the most widely used type, available in sizes from fractional horsepower and tonnage up to a few hundred tons. Construction is similar to the reciprocating engine of a vehicle, with pistons, cylinders, valves, connecting rods, and crankshafts. Open compressors have an exposed shaft to which the electric motor or other driver is attached externally. Hermetic compressors are manufactured with both compressor and motor sealed in casing. In this way there is no possibility of refrigerant loss from leaking around the shaft. The motor is cooled by refrigerant in a hermetic compressor.

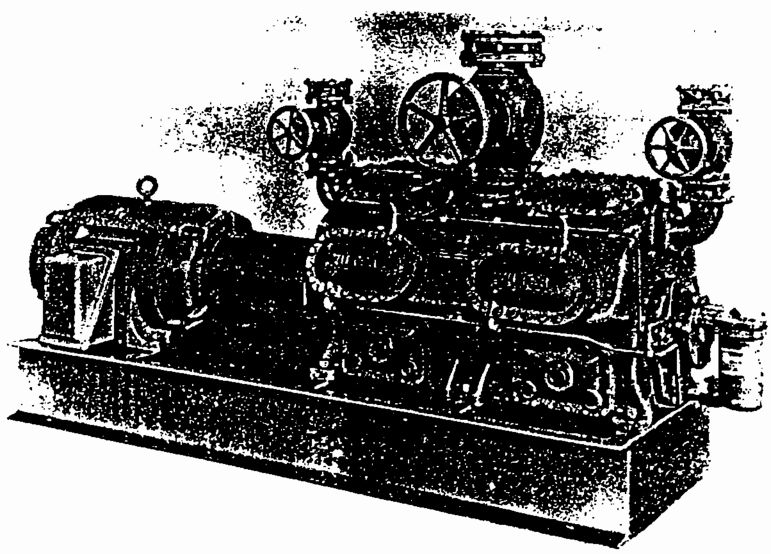


Figure 2. A 16-cylinder reciprocating compressor for ammonia
(Source : W. F. Stoecker, Refrigeration and Air Conditioning, p. 206)

7) Ibid, p.296

2. Rotary compressor[8]

This type has a rotor eccentric to the casing, as the rotor turns it reduces gas volume and increases its pressure. Advantages of this compressor are that it has few parts, is of a simple construction, and can be relatively quiet and vibration free. Small rotary compressors are often used in household refrigerators and window air conditioners.

3. Screw (helical rotary) compressor

Two meshing helical shaped screws rotate and compress the gas as the volume between the screws decreases toward the discharge end. This type of compressor has become popular in recent years due to its reliability, efficiency, and cost. It is generally used in the larger size ranges of positive displacement compressors, in capacities up to about 1000 tons.

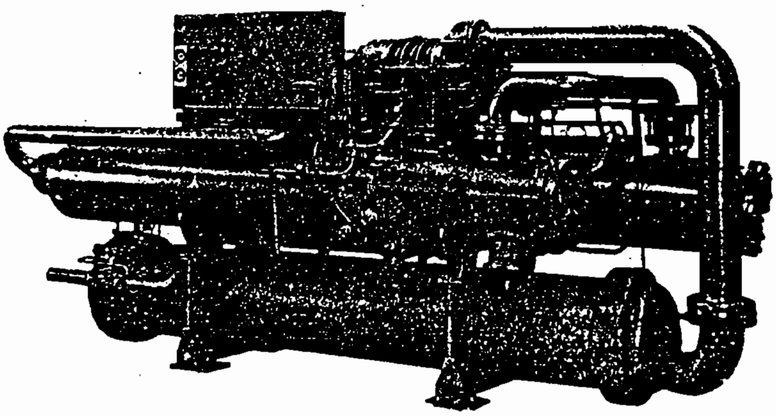


Figure 3. A water-chilling package that uses a screw compressor
(Source : W. F. Stoecker, Refrigeration and Air Conditioning, p. 221)

8) Ibid, p.297

4. Centrifugal compressor

This type of compressor has vane impellers rotating inside a casing, similar to a centrifugal pump. The impellers increase the velocity of the gas, which is then converted into a pressure increase by decreasing the velocity. The nature of the centrifugal compressor makes it suitable for very large capacities, up to 10.000 tons, the impellers can be rotated at speeds up to 20.000 RPM, enabling it to handle large quantities of refrigerant.

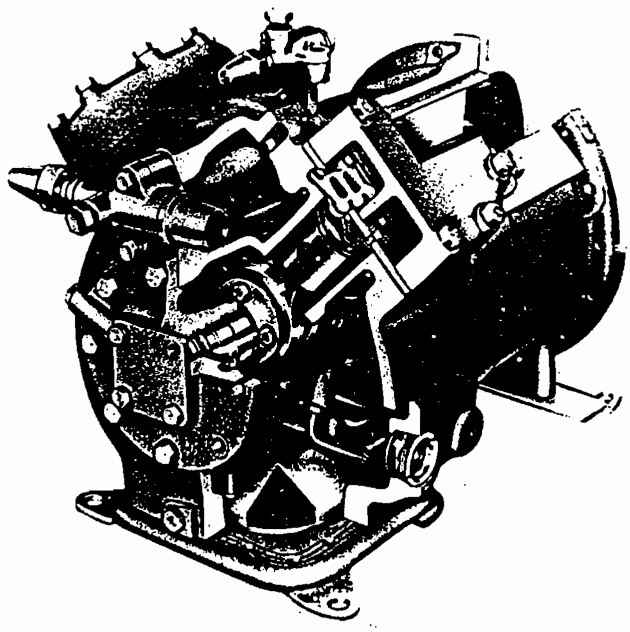


Figure 4. Cutaway view of a hermetically sealed compressor
(Source : W. F. Stoecker, Refrigeration and Air Conditioning, p. 207)

b. Evaporator

These may be classified into two types for air conditioning service-dry expansion (DX) evaporators or flooded evaporators. In the dry expansion type, refrigerant flow through tubing and there is no liquid storage of refrigerant in the evaporator. In the flooded type of evaporator, a liquid pool of refrigerant is maintained.

Evaporator for cooling water or other liquids are called chillers. In the shell and tube type a bundle of straight tubes in enclosed in a cylindrical shell. The chiller may be either the flooded type, with water circulating through the tubes and refrigerant through the shell or dry expansion. The shell can be made in one piece or can be constructed with bolted removable ends, called heads. This construction is more expensive, however flooded chillers are generally used on the larger systems.[9]

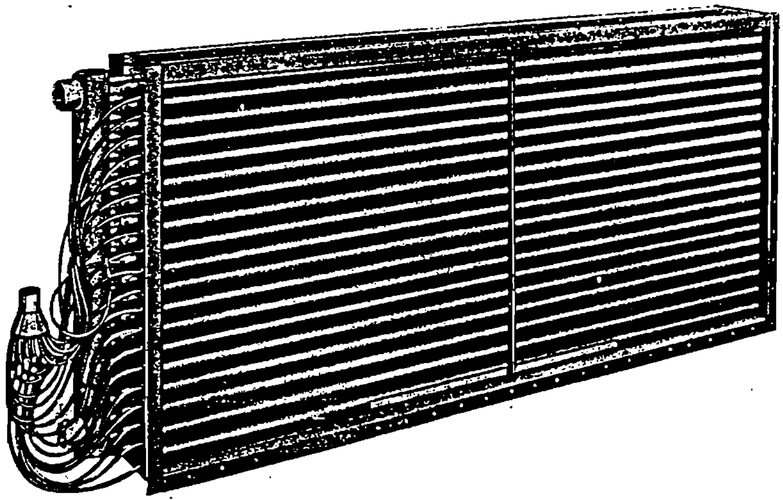


Figure 5. Air cooling evaporator
(Source : W. F. Stoecker, Refrigeration and Air Conditioning, p. 252)

9) Ibid, p.295

c. Condenser

The condenser rejects from the system the energy gained in the evaporator and the compressor. Atmospheric air or water is the two most convenient heat sinks to which the heat can be rejected.[10]

In the air-cooled condenser, the refrigerant circulates through a coil and air flows across the outside of the tubing. Natural convection effects may cause the air motion when the air is heated, the condenser can include a fan to increase the airflow rate, resulting in greater capacity. Air-cooled condensers are normally installed outdoors. They are available in sizes up to about 50 tons.

Water-cooled condensers are usually of shell and tube construction, similar to shell and tube evaporators. Usually, however, natural sources of water are not sufficient, and the water must be re-circulated through a cooling tower.

Evaporative condensers reject the heat to the atmosphere as do air cooled condensers, but by spraying water on the coils. Some heat is transferred to the water as well as the air, increasing the capacity of the condenser. Evaporative condensers can be installed indoors as well as outdoors, by using the ductwork to discharge the exhaust air outside.

Higher condensing pressure result in more power use, and extremely high pressures can damage the equipment. On the other

10) Ibid, p.299

hand, if the pressure is too low, the flow control device will not operate satisfactorily.

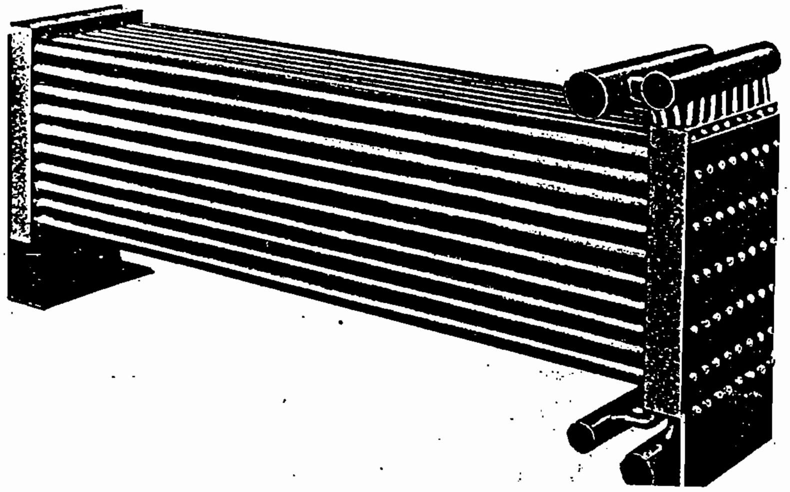


Figure 6. Water-cooled condenser with a cleanable tubes
(Source : W. F. Stoecker, Refrigeration and Air Conditioning, p. 244)

d. Flow control devices

The restricting device that causes the pressure drop of the refrigerant also regulates the refrigerant flow according to the load.
The capillary tube is a very small diameter tube of considerable length, which thus causes the required pressure drop. It is used often in small units (e.g., domestic refrigerators and window air conditioners) because of its low cost and simplicity.[11]

The thermostatic expansion valve (TEV) is widely used in dry expansion systems. The small opening between the valve seat and the

11) Ibid, p.301

disc results in the required pressure drop. It also does an excellent job of regulating flow according to the need.

A low side float valve is a flow control device that is used with the flooded chillers. If too much liquid refrigerant accumulates because flow is no adequate, the float rises and a connecting linkage opens the valve, allowing more flow.

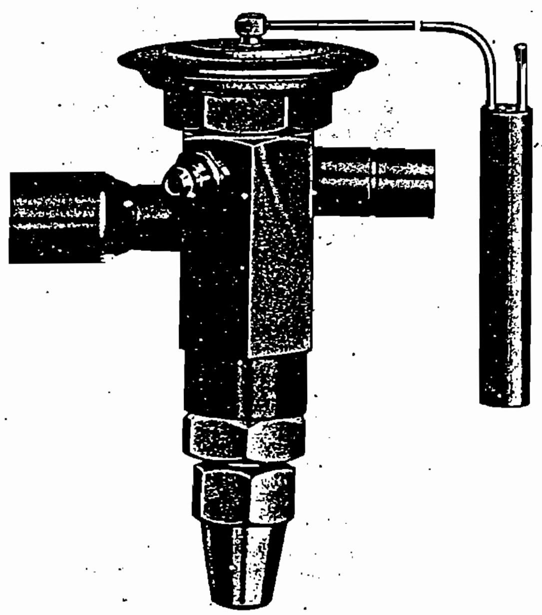


Figure 7. Cutaway view of thermostatic expansion valve
(Source : W. F. Stoecker, Refrigeration and Air Conditioning, p. 274)

2) Absorption Refrigeration System

Absorption refrigeration machines are often used for large air conditioning systems. The absence of a compressor usually has the advantages of less vibration, noise, and weight than with a vapor compression machine.[12]

The absorption system first absorbs the low-pressure vapor in an appropriate absorbing liquid. Embodied in the absorption process is the conversion of vapor into liquid, since this process is akin to condensation, heat must be rejected during the process. The next step is elevating the pressure of the liquid with a pump and the final step releases the vapor from the absorbing liquid by adding heat.

There is a requirement for some work in the absorption cycle to drive the pump, but the amount of work for a given quantity of refrigeration is minor compared with that needed in the vapor compression cycle.

3) Refrigerants

Although many other substances have been used as refrigerants in the vapor compression systems, those that are called the fluorinated hydrocarbons have become universally adopted in refrigeration equipment for air conditioning. Important characteristics of refrigerants for a safety standpoint :

12) Ibid, p.311

1) Nontoxic
2) Noninflammable
3) Non-explosive
4) Non-corrosive
5) Low boiling point

Each has different pressure-temperature boiling point characteristics, so that a suitable refrigerant can be chosen for the application and equipment available. All refrigerants are identified by a standard code numbering system.[13]

13) Ibid, p.320

CHAPTER TWO

AIR CONDITIONING OF SUPER EXECUTIVE BUS

A. PRINCIPLE OF AIR CONDITIONING OF SUPER EXECUTIVE BUS

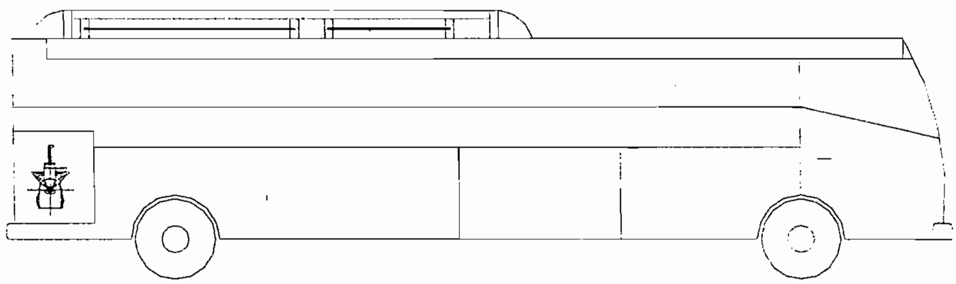
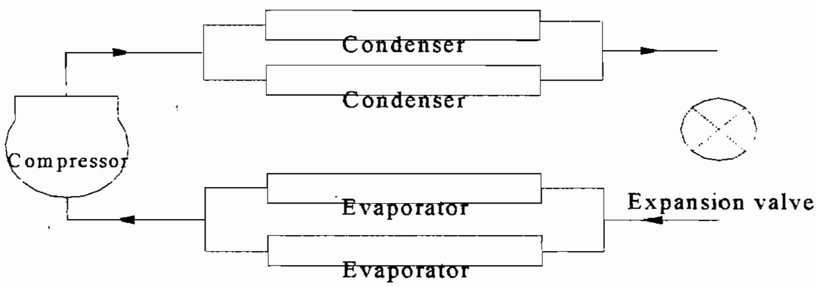


Figure 9. Sketch of Super Executive Bus
(Source : SRLT Bus and Coach Air Conditioner)

The air conditioning system of super executive bus is packaged in a rooftop unit and uses a single compressor, driven by the bus engine which, mostly called a direct system. The cooling unit and condensing units are integrated. The cooling unit consists of the evaporator, expansion valve, sight glass, blower motor, blower resistor, reheat core, etc. The condensing unit consists of the condenser and super cooler (one unit), condenser motor, condenser fan dryer, receiver, etc. The driver controls the air conditioning system with the digital driver dash panel. The dash panel also provides information to the driver on how the system is operating.

The super executive bus using the vapor compression cycle as refrigeration cycle. The processes constituting the vapor compression cycle are :

The evaporator blowers draw return air from inside the bus through the return air filter into the return air chamber. Fresh air also enters to the return air chamber through the fresh air filter at the front of the unit. The air mixes and passes through the evaporator coils where it is cooled and dehumidified. The air then passes through the heater coils, where it may be heated during cool operation times. Leaving the evaporator, the refrigerant is a gas at low pressure and low temperature. In order to be able to use it again to achieve the refrigerating effect continuously (vaporized), it must be compresses to a high pressure and high temperature gas by using a compressor. The refrigerant leaves the compressor as a gas at high temperature and pressure. In order to change it to a liquid, the condenser coils transfer the heat from the refrigerant into the ambient air. Incoming gas/liquid mixture from the condenser coils is store in high pressure at receiver tank to ensure that only liquid refrigerant passes on the TX (thermal expansion) valves. The ball valve enables the drier to be isolated from the system for removal, inspection or replacement without loss of refrigerant from the system. Before entering solenoid valve, the refrigerant must be filtered and removed from moisture and foreign particles. The function of solenoid valve here is to eliminate liquid refrigerant migration from the low-pressure side of the system to compressor sump during off cycle, protecting the compressor against damaging liquid slugging. The solenoid valve is in a closed position when AC system is off. The conditioned air is then discharged into the ducts on each side of the bus to be supplied to the cabin. The central condenser fans draw ambient air into the unit through the side grilles. The air passes through the condenser coils and exits through the top of the unit and vice versa.



B. DESIGN CONDITIONS

The first step in creating a heating, ventilating, and air conditioning system is to design it. The specifications are written descriptions of materials, equipments, and other matters. Inside conditions are those that provide comfort. Outdoors design conditions based on reasonable maximum temperatures. Based on the given data, location for the warmest city in Java is Jakarta at 6S and 107E with the outdoor design conditions in the summertime hours. [14] Other given data based on the above conditions are :

14) Wiranto Arismunandar, Penyegaran Udara, p. 35

Warmest month of the year	: May
Dry bulb temperature (T_o)	: 35 °C = 95 °F
Wet bulb temperature	: 31,8 °C = 89,24 °F
Relative humidity	: 80 %
Enthalpy (h_o)	: 105,85 kJ / kg dry air
Humidity ratio (Wh_o)	: 0,0288 kg moisture / kg dry air
	: 200 grains / lb of d.a
Specific volume	: 0,92 m^3 / kg dry air

Design room conditions are :

Dry bulb temperature (T_i)	: 22 °C = 72 °F
Wet bulb temperature	: 17 °C = 62,6 °F
Relative humidity	: 60 %
Enthalpy (h_i)	: 48 kJ / kg dry air
Humidity ratio (Wh_i)	: 0,010 kg moisture / kg dry air
	: 72,5 grains / lb of d.a
Specific volume	: 0,85 m^3 / kg dry air

C. COOLING LOAD CALCULATIONS

Cooiing load calculations are usually based on inside and outdoor design condition (temperature and humidity). The interior of the bus gains heat from a number of sources. If the temperature and humidity of the rooms are to be maintained at a comfortable level, heat must be extracted to offset these heat gains. The net amount of heat that is removed is called the cooling load.

It is also convenient to arrange the heat gains into two different set of two groups : sensible and latent heat gains. Sensible heat gains results in increasing the air temperature and latent heat gains are due to addition of water vapor, thus increasing humidity.[15]

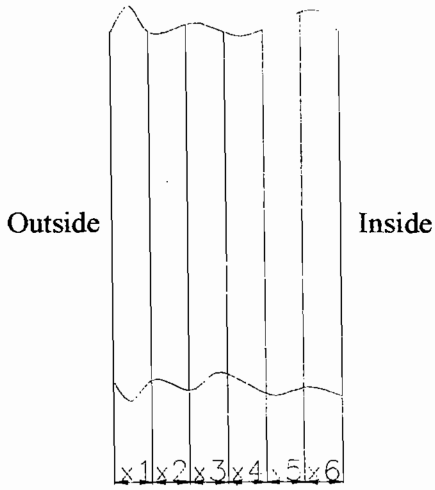
Construction of the bus is as follows :

1. Wall

Table 2.1. Dimensions of wall

No.	Description of construction	Thickness, x (inch)	Conductivity, k (Btu in / hr ft^2 °F)
1.	Painted films	0,157	0,43
2.	Sheet metal	0,043	26,2
3.	Metal lath	1,575	25,9
4.	Foam	1,18	0,40
5.	Triplex	0,157	1,94
6.	Interior finish	0,197	0,092

15) Edward G. Pita, op. cit, p. 93



Assumptions :

V_o = 80 km/hr = 22,2 m/s = 49,72 mph

V_i = 50 cm/s = 0,5 m/s = 1,12 mph

f_o = 15,3 Btu/hr ft^2 °F

f_i = 1,7 Btu/hr ft^2 °F

The film coefficient (f_o, f_i) depends on the outer surface finishing and air surface velocity. Thus, overall heat transfer coefficient for wall (U_w) :[16]

$$\frac{1}{U_w} = \frac{1}{f_o} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{x_4}{k_4} + \frac{x_5}{k_5} + \frac{x_6}{k_6} + \frac{1}{f_i}$$

$$\frac{1}{U_w} = \frac{1}{15{,}3} + \frac{0{,}157}{0{,}43} + \frac{0{,}043}{26{,}2} + \frac{1{,}575}{25{,}9} + \frac{1{,}18}{0{,}40} + \frac{0{,}157}{1{,}94} + \frac{0{,}197}{0{,}092} + \frac{1}{1{,}7} \qquad (2.1)$$

$$U_w = 0{,}159 \text{ Btu/hr ft}^2\ °\text{F} = 0{,}903 \text{ W / m}^2\ °\text{C}$$

16) Ir. G. Harjanto, AC and Refrigeration, p. 26

where :

V_o = speed of the bus (mph)

V_i = air velocity inside the conditioned space (mph)

f_o = outside wall surface film coefficient

f_i = inside wall surface film coefficient

a) Area of walls (A) for :

1.a.1 Front

$$A_{wf} = p \times l$$
$$A_{wf} = 2{,}5 \times 2 = 5\ m^2 = 53{,}76\ ft^2 \qquad (2.2a)$$

1.a.2 Left side

$$A_{wl} = p \times l$$
$$A_{wl} = 11{,}35 \times 2 = 22{,}7\ m^2 = 244{,}086\ ft^2 \qquad (2.2b)$$

1.a.3 Right side

$$A_{wr} = p \times l$$
$$A_{wr} = 11{,}35 \times 2 = 22{,}7\ m^2 = 244{,}086\ ft^2 \qquad (2.2c)$$

1.a.4 Back

$$A_{wb} = p \times l$$
$$A_{wb} = 2{,}5 \times 2 = 5\ m^2 = 53{,}76\ ft^2 \qquad (2.2d)$$

b) Heat gains through walls (Q_w) :

$$Q_w = A_w \times U_w \times (T_o - T_i) \qquad (2.3)$$

where :

A_w = area of wall, ft^2

U_w = overall heat transfer coefficient for wall, Btu/hr ft^2 °F

T_o = outdoor temperature, °F

T_i = inside temperature, °F

1.b.1 Front

$$Q_{wf} = (A_{wf} - A_{gf}) \times U_w \times (T_o - T_i)$$
$$Q_{wf} = 16{,}13 \times 0{,}159 \times (95 - 72) \quad (2.3a)$$
$$Q_{wf} = 58{,}987 \text{ Btu/hr}$$

1.b.2 Left side

From left side of wall, it has 2 doors and 6 windows. Heat gain thru left side of the wall (Q_{wsl}) :

$$Q_{wsl} = \{A_s - (n_g \times A_{gsl})\} \times U_w \times (T_o - T_i)$$
$$Q_{wsl} = \{244{,}086 - (6 \times 6{,}882)\} \times 0{,}159 \times (95 - 72) \quad (2.3b)$$
$$Q_{wsl} = 741{,}618 \text{ Btu/hr}$$

1.b.3 Right side

From right side only have 1 door and 6 windows. The heat gain from right side is :

$$Q_{wsr} = \{A_s - (n_g \times A_{gsr})\} \times U_w \times (T_o - T_i)$$
$$Q_{wsr} = \{244{,}086 - (6 \times 6{,}882)\} \times 0{,}159 \times (95 - 72) \quad (2.3c)$$
$$Q_{wsr} = 741{,}618 \text{ Btu/hr}$$

1.b.4 Back

$$Q_{wb} = (A_{wb} - A_{gb}) \times U_w \times (T_o - T_i)$$
$$Q_{wb} = 29{,}57 \times 0{,}159 \times (95 - 72) \quad (2.3d)$$
$$Q_{wb} = 108{,}138 \text{ Btu/hr}$$

Total sensible heat gains from walls (Q_{SW}) is :

$$Q_{SW} = Q_{wf} + Q_{wsr} + Q_{wsl} + Q_{wb}$$
$$Q_{SW} = (58{,}987 + 741{,}618 + 741{,}618 + 108{,}138) \text{ Btu/hr}$$
$$Q_{SW} = 1650{,}361 \text{ Btu/hr}$$

2. Glass

Front face of the bus using double glass material with the overall heat transfer coefficient (k_{gf}) is 2,2 kCal / m^2 hr °C = 2,58 W / m^2 °C = 0,45 Btu/hr ft^2 °F.[17]

Both sides of the bus and backside using single clear glass with the overall heat transfer coefficient (k_{gs}) is 5,5 kCal / m^2 hr °C = 6,45 W / m^2 °C = 1,14 Btu/hr ft^2 °F.

a) Area of glass (A) for :

2.a.1 Front

$$A_{gf} = p \times l$$
$$A_{gf} = 2{,}5 \times 1{,}4 = 3{,}5 \text{ m}^2 = 37{,}63 \text{ ft}^2$$

17) Wiranto Arismunandar, op. cit, p. 44

2.a.2 Left side

$$A_{gl} = n \times p \times l$$
$$A_{gl} = 6 \times 0{,}8 \times 0{,}8 = 0{,}64\ m^2 = 41{,}292\ ft^3$$

2.a.3 Right side

$$A_{gr} = n \times p \times l$$
$$A_{gr} = 6 \times 0{,}8 \times 0{,}8 = 0{,}64\ m^2 = 41{,}292\ ft^3$$

2.a.4 Back

$$A_{gb} = p \times l$$
$$A_{gb} = 2{,}5 \times 0{,}9 = 2{,}25\ \ m^2 = 24{,}19\ \ ft^2$$

b) Heat gain thru glass (Q_g) :

$$Q_g = A_g \times k_g \times (T_o - T_i) \tag{2.4}$$

where :

A_g = area of glass, ft^2

k_g = overall coefficient of heat transfer, Btu/hr ft^2 °F

2.b.1 Front

$$Q_{gf} = A_{gf} \times k_{gf} \times (T_o - T_i)$$
$$Q_{gf} = 37{,}63 \times 0{,}45 \times (95 - 72)$$
$$Q_{gf} = 389{,}471\ \text{Btu/hr}$$

2.b.2 Left side

$$Q_{gl} = A_{gl} \times k_{gs} \times (T_o - T_i)$$
$$Q_{gl} = 41{,}292 \times 1{,}14 \times (95 - 72)$$
$$Q_{gl} = 1082{,}676\ \text{Btu/hr}$$

2.b.3 Right side

$$Q_{gr} = A_{gr} \times k_{gs} \times (T_o - T_i)$$
$$Q_{gr} = 41{,}292 \times 1{,}14 \times (95 - 72)$$
$$Q_{gr} = 1082{,}676 \text{ Btu/hr}$$

2.b.4 Back

$$Q_{gb} = A_{gb} \times k_{gs} \times (T_o - T_i)$$
$$Q_{gb} = 24{,}19 \times 1{,}14 \times (95 - 72)$$
$$Q_{gb} = 634{,}262 \text{ Btu/hr}$$

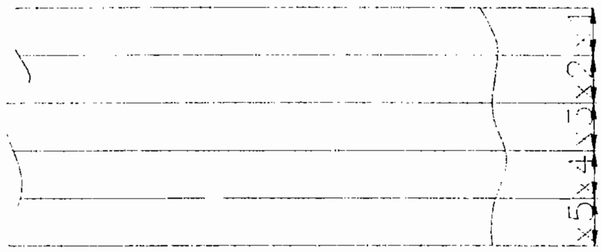
Total sensible heat gain from windows (Q_{SG}) :

$$Q_{SG} = Q_{gf} + Q_{gl} + Q_{gr} + Q_{gb}$$
$$Q_{SG} = (389{,}471 + 1082{,}676 + 1082{,}676 + 634{,}262) \text{ Btu/hr}$$
$$Q_{SG} = 3189{,}085 \text{ Btu/hr}$$

3. Floor

Table 2.2. Dimensions of floor

No.	Description of construction	Thickness, x (inch)	Conductivity, k (Btu in / hr ft^2 °F)
1.	Metal lath	2,362	25,9
2.	Sheet metal	0,11	26,2
3.	Triplex	0,709	1,94
4.	Rubber pad	0,317	0,089
5.	Carpet	0,557	0,395



Overall heat transfer coefficient for floor (U_F) : .

$$\frac{1}{U_F} = \frac{1}{f_o} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{x_4}{k_4} + \frac{x_5}{k_5} + \frac{1}{f_i}$$

$$\frac{1}{U_F} = \frac{1}{15{,}3} + \frac{2{,}362}{25{,}9} + \frac{0{,}11}{26{,}2} + \frac{0{,}709}{1{,}94} + \frac{0{,}317}{0{,}089} + \frac{0{,}557}{0{,}395} + \frac{1}{1{,}7}$$

$$U_F = 0{,}164 \text{ Btu/hr ft}^2 \text{ °F} = 0{,}931 \text{ W / m}^2 \text{ °C}$$

a) Area of floor (A_F) :

$$A_F = p \times l$$

$$A_F = 11{,}35 \times 2{,}5 = 28{,}375 \text{ m}^2 = 305{,}108 \text{ ft}^2$$

b) Sensible heat gains from floor (Q_{SF}) :

$$Q_{SF} = A_F \times U_F \times (T_o - T_i)$$

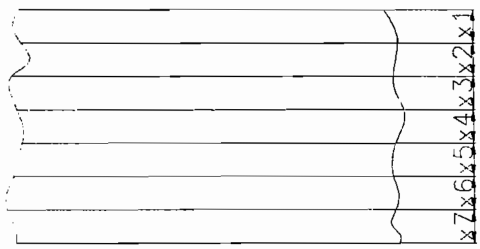
$$Q_{SF} = 305{,}108 \times 0{,}164 \times (95 - 72)$$

$$Q_{SF} = 1150{,}867 \text{ Btu/hr}$$

4. Roof

Table 2.3. Dimensions of roof

No.	Description of construction	Thickness, x (inch)	Conductivity, k (Btu in / hr ft^2 °F)
1.	Painted films	0,157	0,43
2.	Sheet metal	0,257	26,2
3.	Metal lath	1,575	25,9
4.	Foam	1,18	0,40
5.	Triplex	0,0985	1,94
6.	Fibrous pad	0,394	2,08
7.	Carpet	0,118	0,395



Overall heat transfer coefficient for roof (U_R) :

$$\frac{1}{U_R} = \frac{1}{f_o} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{x_4}{k_4} + \frac{x_5}{k_5} + \frac{x_6}{k_6} + \frac{x_7}{k_7} + \frac{1}{f_i}$$

$$\frac{1}{U_R} = \frac{1}{15,3} + \frac{0,157}{0,43} + \frac{0,257}{26,2} + \frac{1,575}{25,9} + \frac{1,18}{0,40} + \frac{0,0985}{1,94} + \frac{0,394}{2,08} + \frac{0,118}{0,395} + \frac{1}{1,7}$$

$$U_R = 0{,}218 \text{ Btu/hr ft}^2 \text{ °F} = 1{,}238 \text{ W / m}^2 \text{ °C}$$

a) Area of roof (A_R) :

$$A_R = p \times l$$
$$A_R = 11{,}35 \times 2{,}5 = 28{,}375 \ m^2 = 305{,}108 \ ft^2$$

b) Sensible heat gains from roof (Q_{SR}) :

$$Q_{SR} = A_R \times U_R \times (T_o - T_i)$$
$$Q_{SR} = 305{,}108 \times 0{,}218 \times (95 - 72)$$
$$Q_{SR} = 1529{,}812 \ \text{Btu/hr}$$

5. Occupants

The heat gain from people is composed of two parts, sensible heat and the latent heat resulting from perspiration. The rate of the heat gain from people depends on their physical activity.[18]

For this designing, considering the number of passengers are same as number of seats. It consists of 2 bus drivers, 1 usher and 26 passengers.

Table 2.4. Heat Gains from Occupants

No.	Occupations	Degree of activity	Sensible heat, Q_S Btu/hr	Latent heat, Q_L Btu/hr
1.	1st bus driver	Seated	200	300
	2nd bus driver	Seated at rest	180	150
2.	Usher	Seated, very light work	195	205
3.	Passenger	Seated at rest	180	150

18) Edward G. Pita, op. cit, p. 111

The heat storage effect factor CLF (cooling load factor) applies to the sensible heat gain from people. The design conditions assume that the air conditioning is switch on during occupied hours. That means storage should be included. Thus, value of CLF = 1.

a) Sensible heat gains from occupants are :

5.a.1. First driver :

$$Q_{S1} = Q_S \times n \times CLF$$
$$Q_{S1} = 200 \times 1 \times 1 \tag{2.5}$$
$$Q_{S1} = 200 \text{ Btu/hr}$$

5.a.2. Second driver :

$$Q_{S2} = Q_S \times n \times CLF$$
$$Q_{S2} = 180 \times 1 \times 1$$
$$Q_{S2} = 180 \text{ Btu/hr}$$

5.a.3. Usher :

$$Q_{S3} = Q_S \times n \times CLF$$
$$Q_{S3} = 195 \times 1 \times 1$$
$$Q_{S3} = 195 \text{ Btu/hr}$$

5.a.4. Passengers :

$$Q_{S4} = Q_S \times n \times CLF$$
$$Q_{S4} = 180 \times 26 \times 1$$
$$Q_{S4} = 4680 \text{ Btu/hr}$$

Total sensible heat gains from occupants (Q_{SO}) :

$$Q_{SO} = Q_{S1} + Q_{S2} + Q_{S3} + Q_{S4}$$
$$Q_{SO} = 200 + 180 + 195 + 4680$$
$$Q_{SO} = 5255 \text{ Btu/hr}$$

b) The latent heat gains from occupants are :

5.b.1. First driver :

$$Q_{L1} = Q_L \times n$$
$$Q_{L1} = 300 \times 1 \qquad (2.6)$$
$$Q_{L1} = 300 \text{ Btu/hr}$$

5.b.2. Second driver :

$$Q_{L2} = Q_L \times n$$
$$Q_{L2} = 150 \times 1$$
$$Q_{L2} = 150 \text{ Btu/hr}$$

5.b.3. Usher :

$$Q_{L3} = Q_L \times n$$
$$Q_{L3} = 205 \times 1$$
$$Q_{L3} = 205 \text{ Btu/hr}$$

5.b.4. Passengers :

$$Q_{L4} = Q_L \times n$$
$$Q_{L4} = 150 \times 26$$
$$Q_{L4} = 3900 \text{ Btu/hr}$$

Total latent heat gains from occupants :

$$Q_{LO} = Q_{L1} + Q_{L2} + Q_{L3} + Q_{L4}$$
$$Q_{LO} = 300 + 150 + 205 + 3900$$
$$Q_{LO} = 4555 \text{ Btu/hr}$$

6. Equipments

The heat gain from equipment may sometimes be found directly from the manufacturer or the nameplate data. The super executive bus has six 15 W fluorescent lighting fixture in use, 66 W TV set, two ½ HP fan-coil, two ½ HP blower motor and ⅛ HP fan motor. All are operate during occupied hours. Ballast factor (BF) accounts for heat losses in the ballast in fluorescent lamps, other special losses. A typical value of BF is 1,25 for fluorescent lighting.[19]

Heat gains from lighting (Q_l) :

$$\begin{aligned} Q_l &= W \times BF \times 3{,}4 \times CLF \\ Q_l &= 90\ W \times 1{,}25 \times 3{,}4 \times 1 \\ Q_l &= 382{,}5\ \text{Btu/hr} \end{aligned} \tag{2.7}$$

Heat gains from TV set (Q_e) :

$$\begin{aligned} Q_e &= W \times 3{,}4 \times CLF \\ Q_e &= 66\ \ W \times 3{,}4 \times 1 \\ Q_e &= 224{,}4\ \text{Btu/hr} \end{aligned} \tag{2.8}$$

Table 2.5. Heat Gains from Appliances

No.	Appliances	Heat output, Q_a Btu/hr
1.	2 x ½ HP fan-coil	3640
2.	2 x ½ HP blower motor	3640
3.	1/8 HP fan motor	580
		7860

19) Ibid, p. 108

Total sensible heat gain from equipment (Q_E) :

$$Q_E = Q_l + Q_c + Q_a$$
$$Q_E = (382,5 + 224,4 + 7860)\,\text{Btu/hr}$$
$$Q_E = 8466,9\ \text{Btu/hr}$$

7. Infiltration

Air quality must also be maintained to provide healthy, comfortable indoor environment. The super executive bus has sealed windows and therefore has no infiltration loss, except for entrances.[20]

Assume that flow rate of outside air due to the entry (v_{OA}) is 0,25 m/s and each entry is about minutes. Volumetric flow rate of outside air ($\overset{\circ}{Q}_{OA}$) :

$$\overset{\circ}{Q}_{OA} = v_{OA} \cdot A_d$$
$$\overset{\circ}{Q}_{OA} = 0,25 \cdot 1,8 \qquad (2.9)$$
$$\overset{\circ}{Q}_{OA} = 0,45\ \text{m}^3/\text{s} = 953,4\ \text{CFM}$$

Each day is about 5 times per entry, for a day (s) each entry is :

$$s = 5\ \text{times} \times 5\ \text{minutes}$$
$$s = 25\ \text{minutes}$$

Thus, outdoor air quantity (Q_{OA}) a day for each 12 hours per journey is :

$$Q_{OA} = s \times \frac{\overset{\circ}{Q}_{OA}}{12\text{hr}}$$
$$Q_{OA} = 25 \times \frac{953,4}{12\ \text{hrs}} \qquad (2.10)$$
$$Q_{OA} = 1986,25\ \text{CFM/hr}$$
$$\dot{Q}_{OA} = 33,104\ \text{CFM}$$

20) Ibid, p. 114

Previous data shown :

$T_o = 35\ °C = 95\ °F$

$T_i = 22\ °C = 72\ °F$

$Wh_o = 200$ grains / lb of d.a

$Wh_i = 72{,}5$ grains / lb of d.a

$Q_{OA} = 33{,}104$ CFM

Sensible heat due to infiltration (Qs_{INFIL}) :[21]

$$
\begin{aligned}
Qs_{INFIL} &= 1{,}1 \times Q_{OA} \times TC \\
Qs_{INFIL} &= 1{,}1 \times 33{,}104 \times (95 - 72) \qquad (2.11) \\
Qs_{INFIL} &= 837{,}5312\ \text{Btu/hr}
\end{aligned}
$$

The latent heat difference between the desired inside humidity conditions and the outdoor infiltration air humidity is given by equation :[22]

$$
\begin{aligned}
Ql_{INFIL} &= 0{,}68 \times Q_{OA} \times (Wh_o - Wh_i) \\
Ql_{INFIL} &= 0{,}68 \times 33{,}104 \times (200 - 72{,}5) \qquad (2.12) \\
Ql_{INFIL} &= 2870{,}1168\ \text{Btu/hr}
\end{aligned}
$$

8. Ventilation

Ventilation is defined as supplying air by natural or natural mechanical means to a space. Sources of pollution exist in both the internal and external dilution. Indoor air quality is controlled by removal or

21) Ibid, p. 49
22) Ibid, p. 50

contaminant or by dilution. Normally, ventilation air is made up of outdoor air and recirculated air. The outdoor air is provided for dilution. Ventilation imposes a significant load on heating and cooling equipment and thus is a major contribution to energy use. Space occupancies and the choice of ventilation rates should be considered carefully. Design conditions prohibited smoking inside the conditioned space.[23] Ventilation air quantity for non smoking occupancy (CFM) :

$$\begin{aligned} CFM &= n \times 7{,}5\,CFM / person \\ CFM &= 29 \times 7{,}5\,CFM / person \\ CFM &= 217{,}5\,CFM \end{aligned} \tag{2.13}$$

Sensible heat from ventilation air (Qs_V) :

$$\begin{aligned} Qs_V &= 1{,}1 \times CFM \times (T_o - T_i) \\ Qs_V &= 1{,}1 \times 217{,}5 \times (95 - 72) \\ Qs_V &= 5502{,}75\ \text{Btu/hr} \end{aligned} \tag{2.14}$$

Latent heat from ventilation air (Ql_V) :

$$\begin{aligned} Ql_V &= 0{,}68 \times CFM \times (Wh_o - Wh_i) \\ Ql_V &= 0{,}68 \times 217{,}5 \times (200 - 72{,}5) \\ Ql_V &= 18857{,}25\ \text{Btu/hr} \end{aligned} \tag{2.15}$$

9. Solar radiation

Radiant energy from the sun passes through transparent materials such as glass and becomes a heat gain to room. Its values vary with time,

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23) Stoecker, op. cit, p. 62

orientation, and storage effect. The heat gain can be found from the following equation :[24]

$$Q_{TH} = SHGF \times A_g \times SC \times CLF \qquad (2.16)$$

where :

Q_{TH} = solar radiation heat gain through glass, Btu/hr

SHGF = solar heat gain factor, Btu/hr ft^2

A = area of glass, ft^2

CLF = cooling load factor for glass

SC = shade coefficient

Outdoors design conditions are in February located at 0°N (SHGF = 67 Btu/hr ft^2) for summertime hours by neglecting the glass orientation (CLF = 1). Previous data :

$$A_{gf} = 37{,}63\ ft^2$$

$$A_{gb} = 24{,}19\ ft^2$$

$$A_{gs} = 41{,}292\ ft^2$$

a) Solar radiations through each side of glass :

9.a.1. Front :

$$Q_{THf} = SHGF \times A_{gf} \times SC \times CLF$$
$$Q_{THf} = 67 \times 37{,}63 \times 1 \times 1$$
$$Q_{THf} = 2521{,}21\ Btu/hr$$

24) Ibid, p.102

9.a.2. Left side :

$$Q_{THl} = SHGF \times A_{gs} \times SC \times CLF$$
$$Q_{THl} = 67 \times 41{,}292 \times 1 \times 1$$
$$Q_{THl} = 2766{,}564 \text{ Btu/hr}$$

9.a.3. Right side :

$$Q_{THr} = SHGF \times A_{gs} \times SC \times CLF$$
$$Q_{THr} = 67 \times 41{,}292 \times 1 \times 1$$
$$Q_{THr} = 2766{,}564 \text{ Btu/hr}$$

9.a.4. Back :

$$Q_{THb} = SHGF \times A_{gb} \times SC \times CLF$$
$$Q_{THb} = 67 \times 24{,}19 \times 1 \times 1$$
$$Q_{THb} = 1620{,}73 \text{ Btu/hr}$$

b) Total solar radiation through glass (Q_{TH}) :

$$Q_{TH} = Q_{THf} + Q_{THl} + Q_{THr} + Q_{THb}$$
$$Q_{TH} = 2521{,}21 + 2766{,}564 + 2766{,}564 + 1620{,}73$$
$$Q_{TH} = 9675{,}068 \text{ Btu/hr}$$

10. Air leakage loss

The supply duct normally has 50 °F DB to 60 °F DB air flowing through it. The duct may pass through an unconditioned space having a temperature of, say, 90 °F DB and up. This results in a heat gain to duct before it reaches the space to be conditioned air.[25]

25) Ibid, p.129

Supply air duct heat gain : 5 %

Supply air duct leakage : 5 %

Total : 10 %

Room sensible heat (RSH) :

$$RSH = Q_{SG} + Q_{SW} + Q_{SR} + Q_{SF} + Q_{SO} + Q_{E} + Q_{TH} + Q_{SV} + Q_{Sinfil}$$
$$RSH = \begin{pmatrix} 3189{,}085 + 1650{,}361 + 1529{,}812 + 1150{,}867 + 5255 + \\ 8466{,}9 + 9675{,}068 + 5502{,}75 + 837{,}5312 \end{pmatrix}$$
$$RSH = 37257{,}3742 \text{ Btu/hr}$$

Heat gain to supply air duct (Q_D) :

$$Q_D = \text{Total supply air duct} \times RSH$$
$$Q_D = 10\,\% \times 37257{,}3742 \tag{2.17}$$
$$Q_D = 3725{,}73742 \text{ Btu/hr}$$

Room latent heat (RLH) :

$$RLH = Q_{LO} + Q_{LV} + Q_{IINFIL}$$
$$RLH = 4555 + 18857{,}25 + 2870{,}1168$$
$$RLH = 26282{,}367 \text{ Btu/hr}$$

Outdoor air sensible heat (OASH) :[26]

$$OASH = 1{,}1 \times CFM \times (T_o - T_i)$$
$$OASH = 1{,}1 \times 217{,}5 \times (95 - 72) \tag{2.18}$$
$$OASH = 5502{,}75 \text{ Btu/hr}$$

Outdoor air latent heat (OALH) :

$$OALH = 0{,}68 \times CFM \times (Wh_o - Wh_i)$$
$$OALH = 0{,}68 \times 217{,}5 \times (200 - 72{,}5) \tag{2.19}$$
$$OALH = 18857{,}25 \text{ Btu/hr}$$

26) Carrier, Handbook of Air Conditioning System Design, p.1-150

Given data :

OASH	= 5502,75 Btu/hr
OALH	= 18857,25 Btu/hr
RSH	= 37257,3742 Btu/hr
RLH	= 26282,367 Btu/hr
Q_D	= 3725,73742 Btu/hr
CFM	= 217,5 CFM
$T_i = T_{rm}$	= 22 °C = 72 °F
BF	= 0,05

Room sensible heat factor (RSHF) :

$$RSHF = \frac{RSH}{RSH + RLH}$$

$$RSHF = \frac{37257,3742}{37257,3742 + 26282,367} \tag{2.20}$$

$$RSHF = 0,586$$

Total sensible heat (TSH) :

$$TSH = RSH + OASH + Q_D$$

$$TSH = 37257,3742 + 5502,75 + 3725,73742 \tag{2.21}$$

$$TSH = 46485,862 \text{ Btu/hr}$$

Total latent heat (TLH) :

$$TLH = RLH + OALH$$

$$TLH = 26282,367 + 18857,25 \tag{2.22}$$

$$TLH = 45139,617 \text{ Btu/hr}$$

Grand total heat (GTH) :

$$GTH = TSH + TLH$$
$$GTH = 46485{,}862 + 45139{,}617 \quad (2.23)$$
$$GTH = 91625{,}479 \text{ Btu/hr}$$

Grand sensible heat factor (GSHF) :

$$GSHF = \frac{TSH}{GTH}$$
$$GSHF = \frac{46485{,}862}{91625{,}479} \quad (2.24)$$
$$GSHF = 0{,}507$$

Effective sensible heat factor (ESHF) :[27] (2.25)

$$ESHF = \frac{RSH + (BF \times OASH)}{RSH + (BF \times OASH) + RLH + (BF \times OALH)}$$
$$ESHF = \frac{37257{,}3742 + (0{,}05 \times 5502{,}75)}{37257{,}3742 + (0{,}05 \times 5502{,}75) + 26282{,}367 + (0{,}05 \times 18857{,}25)}$$
$$ESHF = 0{,}579$$

On the psychrometric chart, ESHF of 0,579 intersects the saturation curve at apparatus dew point (T_{adp}) 43 °F.

Dehumidified air quantity (CFM_{DA}) :

$$CFM_{DA} = \frac{RSH + (BF \times OASH)}{1{,}08 \times (1 - BF)(T_{rm} - T_{adp})}$$
$$CFM_{DA} = \frac{37257{,}3742 + (0{,}05 \times 5502{,}75)}{1{,}08 \times (1 - 0{,}05)(72 - 43)} \quad (2.26)$$
$$CFM_{DA} = 1261{,}43 \text{ CFM}$$

CFM_{DA} is also CFM_{SA} when no air is to be physically bypassed around the cooling coil.

27) Ibid, p. 1-129

Return air quantity (CFM_{RA}) :

$$CFM_{RA} = CFM_{SA} - CFM_{OA}$$
$$CFM_{RA} = (1261{,}43 - 217{,}5) \tag{2.27}$$
$$CFM_{RA} = 1043{,}93 \text{ CFM}$$

Supply air temperature (T_{SA}) :[28]

$$T_{SA} = T_{rm} - \frac{RSH}{1{,}08 \times CFM_{SA}}$$
$$T_{SA} = 72 - \frac{37257{,}3742}{1{,}08 \times 1261{,}43} \tag{2.28}$$
$$T_{SA} = 44{,}5\ °F$$

Entering dry-bulb temperature (T_{edb}) :

$$T_{edb} = \frac{(CFM_{OA} \times T_o) + (CFM_{RA} \times T_{rm})}{CFM_{SA}}$$
$$T_{edb} = \frac{(217{,}5 \times 95) + (1043{,}93 \times 72)}{1262{,}43} \tag{2.29}$$
$$T_{edb} = 76\ °F$$

To determine the entering wet-bulb temperature (T_{ewb}), the T_{edb} crosses the straight line plotted between the outdoor and room design conditions.

$T_{ewb} = 68$ °F.

Leaving dry-bulb temperature (T_{ldb}) :

$$T_{ldb} = T_{adp} + BF \times (T_{edb} - T_{adp})$$
$$T_{ldb} = 43 + 0{,}05 \times (76 - 43) \tag{2.30}$$
$$T_{ldb} = 45\ °F$$

28) Ibid, p. 1-130

Leaving wet-bulb temperature (T_{lwb}) can be obtained by drawing a straight line between the adp and the entering conditions to the apparatus (the GHSF line). $T_{lwb} = 44{,}6$ °F.

Effective room sensible heat (ERSH) :[29]

$$\begin{aligned} ERSH &= RSH + (BF \times OASH) + Q_D \\ ERSH &= 37257{,}3742 + (0{,}05 \times 5502{,}75) + 3725{,}73742 \\ ERSH &= 41258{,}249 \text{ Btu/hr} \end{aligned} \tag{2.31}$$

Effective room latent heat (ERLH) :

$$\begin{aligned} ERLH &= RLH + (BF \times OALH) \\ ERLH &= 26282{,}367 + (0{,}05 \times 18857{,}25) \\ ERLH &= 27225{,}229 \text{ Btu/hr} \end{aligned} \tag{2.32}$$

Effective room total heat (ERTH) :

$$\begin{aligned} ERTH &= ERSH + ERLH \\ ERTH &= 41258{,}249 + 27225{,}229 \\ ERTH &= 68483{,}478 \text{ Btu/hr} \end{aligned} \tag{2.33}$$

The cooling load calculations are usually based on inside and outdoor design conditions of temperature and humidity. A safety factor to be added to the cooling load should be considered as strictly a factor of probable error in the survey or estimate, and should usually be 15 %.[30]

The refrigeration capacity for super executive bus (QC) :

29) Ibid, p. 1-150
30) Ir. G. Harjanto, op. cit, p. 26

$$QC = ERTH + (15\% \times ERTH)$$
$$QC = 68483{,}478 + 10272{,}522 \tag{2.34}$$
$$QC = 78756\ \text{Btu/hr} = 23\ \text{kW} = 6{,}56\ \text{TR}$$

The ton of refrigeration (TR) is a unit measuring heat rate that is used often in refrigeration work.

Table 2.6. Cooling Load Calculations Form

Project : Super Executive Bus   Room : Cabin   Check : Mr. Greg
Location : Jakarta, Java   Lat. : 6S and 107E   Calc. by : Nadia

		DB ^0F	WB ^0F	RH %	W gr/lb	
						Daily range : -
						Ave. : -
Design	Outdoor	95	92.3	80	200	Day : -
Conditions	Room	72	59.36	60	71.5	Time : 12.00 PM

Conduction	Dir	U	A ft^2	DT T_o	DT T_i	RSHG Btu/hr
Glass	Front	0.45	37.63	95	72	389.47
	Left	1.14	41	95	72	1082.68
	Right	1.14	41	95	72	1082.68
	Back	1.14	24.19	95	72	634.26
Wall	Front	0.159	53.76	95	72	196.60
	Left	0.159	244	95	72	892.62
	Right	0.159	244	95	72	892.62
	Back	0.159	53.76	95	72	196.60
Roof		0.218	305	95	72	1529.81
Floor		0.164	305	95	72	1150.87

Radiation	Dir	SHGF	A ft^2	SC	CLF	RSHG Btu/hr	RLHG Btu/hr
Glass	Front	67	37.63	1	1	2,521	
	Left	67	41	1	1	2,767	
	Right	67	41	1	1	2,767	
	Back	67	24.19	1	1	1,621	
Lighting						382.5	
Equipments						8084.4	
Occupants						5255	4555
Infiltration						838	2870.117
Ventilation						5502.75	18857.25
SA duct gain : 5%							
SA duct leakage : 5%							
Total SA duct : 10%						3,726	
					TSH & TLH	46485.86	45139.62

RSH	Room Sensible Heat	=	37257.37 Btu/hr
RLH	Room Latent Heat	=	26282.37 Btu/hr
RTH	Room Total Heat	=	63539.74 Btu/hr
OASH	Outdoor air Sensible Heat	=	5502.75 Btu/hr
OALH	Outdoor Air Latent Heat	=	18857.25 Btu/hr
TSH	Total Sensible Heat	=	46485.86 Btu/hr
TLH	Total Latent Heat	=	45139.62 Btu/hr
GTH	Grand Total Heat	=	91625.48 Btu/hr
RSHF	Room Sensible Heat Factor	=	0.586
GSHF	Grand Sensible Heat Factor	=	0.507
BF	Bypass Factor	=	0.05
CFM_{da}	Dehumidified Air Quantity	=	1261.43 CFM
T_{adp}	Apparatus Dewpoint Temperature	=	43 ^{0}F
T_{SA}	Supply Air Temperatur	=	44.5 ^{0}F
T_{edb}	Entering Dry-bulb Temperature	=	76 ^{0}F
T_{ldb}	Leaving Dry-bulb Temperature	=	45 ^{0}F
ESHF	Effective Sensible Heat Factor	=	0.579
ERSH	Effective Room Sensible Heat	=	41258.25 Btu/hr
ERLH	Effective Room Latent Heat	=	27225.23 Btu/hr
ERTH	Effective Room Total Heat	=	68483.48 Btu/hr
QC	Refrigeration Capacity	=	**6.563 TR**

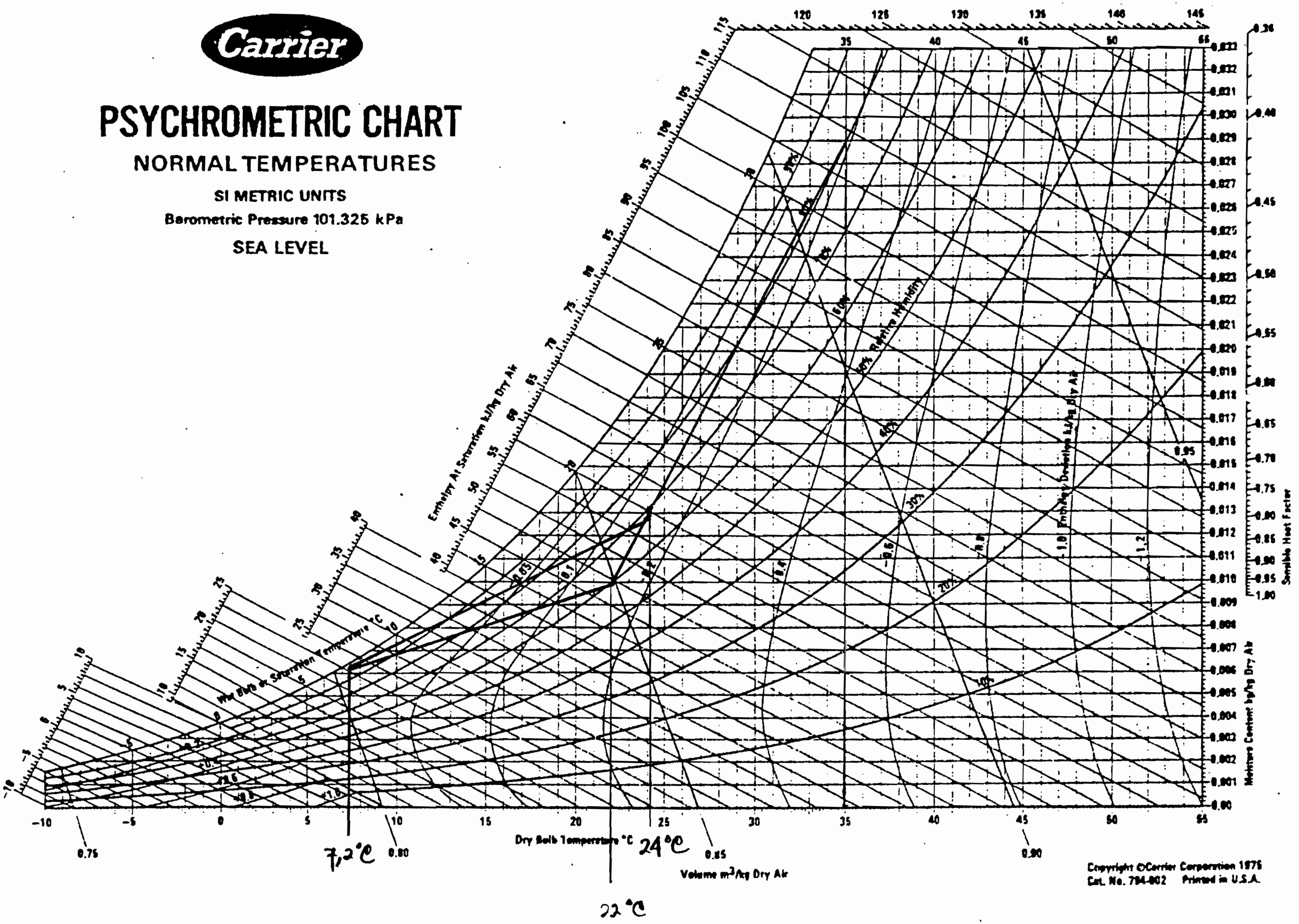


Figure 10. Psychrometric Chart

(Source : Edward G. Pita, Air Conditioning Principles and Systems, p. 458)

CHAPTER THREE

MAIN COMPONENTS OF AC SYSTEM

A. REFRIGERANT SELECTION

Refrigerants are the working fluids in air conditioners, chillers and refrigerators. The majority are chlorinated and fluorinated hydrocarbons (CFCs), although a number of other organic and inorganic compound can be used. The refrigerant selected depends on the type of application. The author selects the R-134a considering the properties of this refrigerant are as follow .[31] :

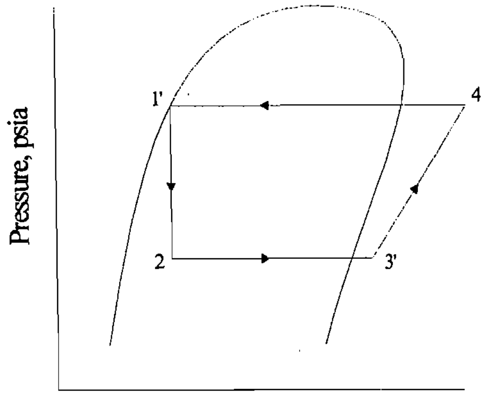
1. Hydroflourocarbon refrigerant.
2. Non-ozone depleting.
3. Colorless in gas and liquid states.
4. Faint ether-like odor.
5. Non-flammable.
6. Non-toxic when in its natural state.

Physical properties of R-134a :

- Chemical formula : CF_3CH_2F
- Boiling point at 1 atm : -15,1°F (-26,2°C)
- Ozone depletion potential (ODP) : 0

31) SLRT Bus and Coach Air Conditioner, p. 15

- Greenhouse warming potential (GWP) : 0,285
- Flammable range, % volume in air : none



The diagram above represents the refrigerant in pressure-enthalpy scale that is called Mollier diagram. It is a graphic presentation of the data contained in thermodynamic tables. Each corresponds to a different physical state of the refrigerant. The zone on the left represents sub cooled liquid refrigerant. Sub cooling of the liquid in the condenser is a normal occurrence and serves the desirable function of ensuring that 100% liquid will enter the expansion device. And the zone on the right represents refrigerant in superheated vapor state. Superheating of the vapor usually occurs in the evaporator and is recommended as a precaution against droplets of liquid being carried over into the compressor.

The vapor compression cycle is the most widely used refrigeration cycle in practice. The normal vapor compression cycle consists of four basic processes:

1. Process 3'-4 represents the action of the compressor. Compression line drawn on a slope (constant entropy) from the end of the evaporation line to

the condensation line in the superheated vapor zone. Even though the compressor may be of the reciprocating type, where flow is intermittent rather than steady. Knowledge of the work of the compression is important because it may be one of the largest operating costs of the system.

2. The heat rejection is transferred from the refrigerant in process 4-1', which is $h_4 - h_1$'. Condensation line drawn horizontally (constant pressure) at the appropriate condensing temperature from the saturated liquid line into the superheated vapor zone. The value of heat rejection is used in sizing the condenser and calculating the required flow quantities of the condenser cooling fluid.
3. The throttling process, 1'-2, is one of constant enthalpy and therefore expansion line drawn vertically from the end of the condensation line to the evaporation line.
4. Evaporation line drawn horizontally (constant pressure) at the appropriate evaporating temperature from the saturated liquid line to the saturated vapor line. The refrigerant effect is the heat transferred in process 2-3', or $h_2 - h_3$'. Knowledge of the magnitude of the term is necessary because forming this process is the ultimate of the entire system.

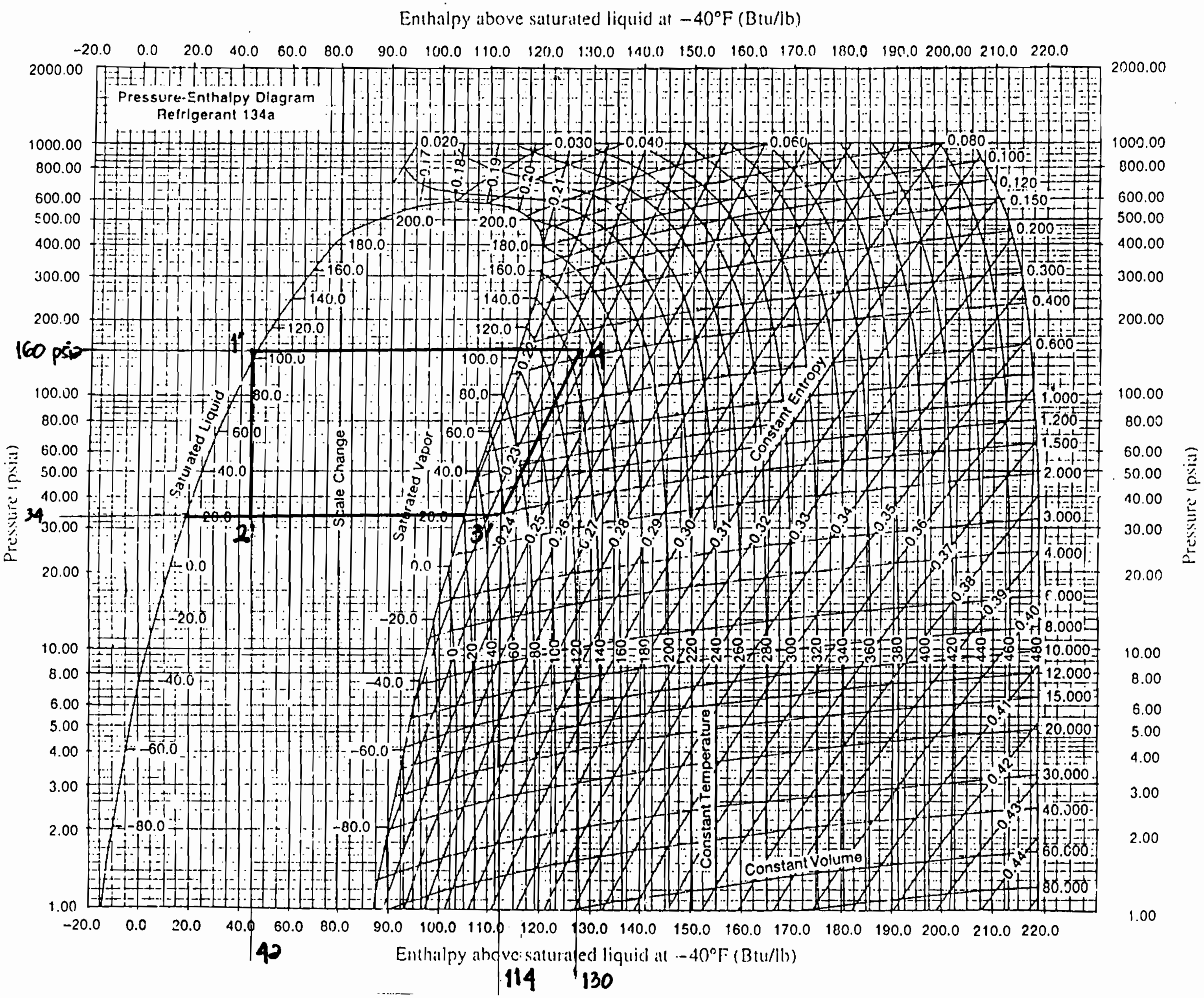


Figure 11. Pressure-Enthalpy Diagram for R134a

(Source : Ari Rabl, Heating and Cooling for Buildings, p. 799)

The comparative cycle performance of R-134a shown on the ph diagram (Mollier diagram) are :

- Evaporating temperature = 20°F
- Condensing temperature = 110°F
- Suction superheat = 30°F
- Sub cooling = 10°F

From the ph diagram of R134a, can be obtained the following results :

- State 1 :

  Pressure 1' (P_1') = 160 psia

  Enthalpy 1' (h_1') = 42 Btu/lb

- State 2 :

  Pressure (P_2) = 34 psia

  Enthalpy 2 (h_2) = 42 Btu/lb

- State 3 :

  Pressure (P_3') = 34 psia

  Enthalpy 3' (h_3') = 114 Btu/lb

- State 4 :

  Pressure (P_4) = 160 psia

  Enthalpy 4 (h_4) = 130 Btu/lb

The values for each of the refrigerant properties can be followed through each process.

a. Refrigerating effect (RE) :

$$RE = h_3' - h_1'$$
$$RE = (114 - 42)\,\text{Btu/lb} \qquad (3.35)$$
$$RE = 72\ \text{Btu/lb}$$

b. Refrigerant flow rate (w) :

$$w = \frac{200}{RE}$$
$$w = \frac{200}{72} \qquad (3.36)$$
$$w = 2{,}78\ \text{lb/min} = 1{,}26\ \text{kg/ton.min}$$

c. Theoretical horsepower of the compressor (W_{hp}) :

$$W_{hp} = 4{,}717 \times \frac{(h_4 - h_3')}{(h_3' - h_1')}$$
$$W_{hp} = 4{,}717 \times \frac{(130 - 114)}{(114 - 42)} \qquad (3.37)$$
$$W_{hp} = 1{,}048\ \text{HP / ton} = 1{,}048 \times 6{,}56 = 6{,}875\ \text{HP}$$

d. Coefficient of performance (COP) :

$$COP = \frac{h_3' - h_1'}{h_4 - h_3'}$$
$$COP = \frac{114 - 42}{130 - 114} \qquad (3.38)$$
$$COP = 4{,}5$$

e. Heat rejected in condenser (Q_{cond}) :

$$Q_{cond} = 200 \times \frac{(h_4 - h_1')}{(h_3' - h_2)}$$
$$Q_{cond} = 200 \times \frac{(130 - 42)}{(114 - 42)} \qquad (3.39)$$
$$Q_{cond} = 244{,}4\ \text{Btu / ton.min}$$

B. COMPRESSOR SELECTION

When designing internal combustion engines, a considerable part of the manufacturing costs is required for the valve gear, starter, and either carburetor with ignition mechanism or pump injection. The cost for these components does not vary greatly with the number of cylinders. For this reason, it is advantageous to increase the output of an internal combustion engine by increasing the number of cylinders.[32]

The compression of the suction vapor from the evaporator to the condenser pressure can be achieved by mechanical compression or ejector compression or by a process combination of absorption or vapor, pumping and desorption. For mechanical compressor, fundamentally, there are two types of machines :[33]

1. Positive displacement machines, viz., reciprocating, rotary, and screw compressors.
2. Non-positive displacement machines, viz., centrifugal compressors.

As standard, a Bitzer 4NF compressor is supplied to the super executive bus. It is one of the types of reciprocating compressors. In standard installations, the bus engine directly drives the compressor via an electromagnetic clutch. The advantages of reciprocating compressors are :

1) Ensuring positive admission and delivery preventing undesired reversal of flow within the machine by the use of the valves.

32) Chlumsky, Reciprocating and Rotary Compressors, p. 123
33) Arora, Refrigeration and Air Conditioning, p. 179

2) Intermittent operation.

3) Subjecting the fluid to non-flow processes and work is transferred by virtue of a hydrostatic force on the moving boundary.

Gained data from the refrigeration cycle and some specifications of Bitzer 4NF compressor are as follow :

- Suction pressure (Ps) = 34 psia = 2,4 kg/cm^2
- Discharge pressure (Pd) = 160 psia =11,3 kg/cm^2
- Specific volume of suction gas (V_s) = 1,5 ft^3/lb = 0,09 m^3/kg
- Specific volume of discharge gas (V_d) = 0,38 ft^3/lb = 0,0228 m^3/kg
- Compressor horsepower (HP) = 6,875 HP/compressor = 5,1 kW/compressor
- Number of cylinders (i) = 4
- Compressor speed (N) = 1850 rpm

Theoretical compression processes considered of constant temperature and constant entropy, and actual compression process is the one that is known as polytropic compression. Therefore, the polytropic index of compression (n) is 1,2.[34]

The clearance or gap between the I.D.C. (inner dead center) position of piston and cylinder head is necessary in reciprocating compressors to provide

34) Ibid, p. 185

for thermal expansion and machining tolerances. This factor (ε) is normally 5%.

Volumetric efficiency (η_v) is the term defined in the case of positive displacement compressors to account for the difference in the displacement in-built in the compressor and actual volume of the suction vapor sucked and pumped. Hence, the volumetric efficiency (η_v) :

$$\eta_v = 1 - \varepsilon\left[\left(\frac{Pd}{Ps}\right)^{\frac{1}{n}} - 1\right] \times 100\,\%$$

$$\eta_v = 1 - 0{,}05\left[\left(\frac{160}{34}\right)^{\frac{1}{1.2}} - 1\right] \times 100\,\% \tag{3.40}$$

$$\eta_v = 1 - 0{,}05(2{,}62) \times 100\,\%$$

$$\eta_v = 87\,\%$$

The effect of leakage pass piston rings and under the suction valve elements (η_l) is normally 5%. Normally, the volumetric efficiency is higher than the overall volumetric efficiency at about 4% - 6% (taken 5%) considering :[35]

1) Cooling during compression at cylinder walls and valves.
2) Leakage loss pass piston rings and under suction valve elements.

Thus, overall volumetric efficiency (η_{vt}) :

$$\eta_{vt} = \eta_v - \eta_l$$

$$\eta_{vt} = 87 - 5 \tag{3.41}$$

$$\eta_{vt} = 82\,\%$$

35) Ibid, p. 189

The adiabatic efficiency (η_{ad}) depends on the compression ratio and may other factors such as the size and disposition of valves, cylinder dimensions, heat transfer, etc. Bitzer 4NF compressor is a single-acting compressor. Therefore, its adiabatic efficiency is about 0,8 – 0,95 (taken 0,90).

Theoretical power consumption for each compressor (W_t) :

$$W_t = \frac{HP}{\eta_{ad}}$$

$$W_t = \frac{6,875}{0,90} \tag{3.42}$$

$$W_t = 7,64\,\text{kW}$$

b.1. Cylinder and Piston

The purpose of piston is to compress the gas admitted into the cylinder chamber into the high-pressure side. The most frequently used material for cylinders are both normal and high-grade gray cast iron of pearlitic structure with bending strength of 28 to 46 kg/mm^2. The main demands laid on cast iron for cylinder are :[36]

1) Sufficient strength.
2) Perfect impermeability in particular refrigerating compressor.
3) Wear resistance.
4) Highest resistance against the action to the gas.

36) Chlumsky, op. cit., p. 141

The chemical composition of gray cast iron for cylinder is usually :

- 2,9 - 3,3 % C (carbon)
- 1,2 – 1,6 % Si (silicone)
- 0,6 – 0,9 % Mn (manganese)
- Up to 0,35 % P (phosphor)
- Max. 0,1 % S (sulfur)
- 0,2 – 0,35 % Cr (chromium)
- 0,15 – 030 % Ni (nickel)

The piston is designed to come as close as possible to the cylinder head without touching it in order to press practically all the gas into the high pressure side. The main requirements of a piston are :[37]

1) Good sealing of the cylinder
2) The weight of the piston and the entire crank mechanism is minimum, in order to reduce the inertia forces and to improve the mechanical efficiency.

The most frequently used material for piston manufacture is cast iron, but large machines are made of cast steel with thin wall.

Previous data :

$$w = 1{,}26 \text{ kg/ton.min}$$

$$V_r = 0{,}7 \text{ m}^3\text{/kg}$$

37) Ibid, p. 161

Actual piston displacement of the compressor (Vp) :

$$Vp = \frac{w \times V_r}{\eta_{vt}}$$

$$Vp = \frac{1{,}26 \times 0{,}7}{0{,}82} \tag{3.43}$$

$$Vp = 1{,}08\ m^3/\ min$$

where :

Vp = actual piston displacement of the compressor (m^3/min)

V_r = specific volume of R134a (m^3/kg)

w = refrigerant flow rate (kg/ton.min)

Bore of the compressor (Bp) :

$$Bp = \sqrt{\frac{Vp}{i \times \frac{\pi}{4} \times Q \times \eta_{vt}}} \tag{3.44}$$

where :

Q = comparison of piston stroke and the cylinder diameter (taken 0,8)

$$Bp = \sqrt[3]{\frac{Vp}{i \times \frac{\pi}{4} \times N \times Q \times \eta_{vt}}}$$

$$Bp = \sqrt[3]{\frac{1{,}08}{4 \times \frac{3{,}14}{4} \times 1850 \times 0{,}8 \times 0{,}82}}$$

$$Bp = 0{,}0654\ m = 65\ mm$$

Stroke of the piston (L) :

$$L = 0{,}8 \times Bp$$
$$L = 0{,}8 \times 0{,}0654 \qquad (3.45)$$
$$L = 0{,}05232\ m = 52{,}32\ mm$$

For low-pressure compressor or stages, the wall thickness of cylinder (x_c) is calculated from thin-walled cylinders :

$$x_c = \frac{P_d \times Bp}{2 \times \sigma_c} + a \qquad (3.46)$$

where :

P_d = gas pressure inside the cylinder (kg/cm^2)

σ_c = tensile stress of cylinder material (kg/cm^2)

a = the allowance for weakness of the wall due to inaccurate casting, and for rusting, reboring, etc.

For cast iron cylinders, the permissible tensile stress is 150 – 250 kg/cm^2 (taken 250 kg/cm^2) and the allowance (a) range from 5 – 10 mm according to the size of the cylinder (taken 8 mm = 0,8 cm).[38]

$$x_c = \frac{P_d \times Bp}{2 \times \sigma_c} + a$$
$$x_c = \frac{11{,}3 \times 6{,}5}{2 \times 250} + 0{,}8$$
$$x_c = 0{,}95\ cm \approx 9{,}5\ mm$$

38) Ibid, p. 166

Wall thickness of cylinder head (x_{ch}) :

$$x_{ch} = Bp \times \sqrt{\frac{0{,}1 \times P_d}{\sigma_c}}$$
$$x_{ch} = 6{,}5 \times \sqrt{\frac{0{,}1 \times 11{,}3}{250}} \qquad (3.47)$$
$$x_{ch} = 0{,}44 \text{ cm} = 4{,}4 \text{ mm}$$

Clearance space (Lc) :

$$Lc = 0{,}005L + 0{,}5$$
$$Lc = (0{,}005 \times 52{,}32) + 0{,}5 \qquad (3.48)$$
$$Lc = 0{,}762 \text{ mm}$$

Length of piston stroke (Lst) :

$$Lst = (0{,}8 \text{ to } 1{,}5) \times Bp$$
$$Lst = 0{,}8 \times 65 \qquad (3.49)$$
$$Lst = 52 \text{ mm}$$

Length of cylinder (L_{cyl}) is the uniform of whole length from clearance space and piston stroke.

$$L_{cyl} = Lc + Lst + L$$
$$L_{cyl} = (0{,}762 + 52 + 52{,}32) \text{ mm} \qquad (3.50)$$
$$L_{cyl} = 105{,}082 \text{ mm} \approx 105 \text{ mm}$$

Mean piston speed (v_{ps}) :[39]

$$v_{ps} = \frac{L \times N}{60}$$
$$v_{ps} = \frac{0{,}05232 \text{ m} \times 1850 \text{ rpm}}{60} \qquad (3.51)$$
$$v_{ps} = 1{,}6 \text{ m/s}$$

To minimize friction and wear, the mean piston speed is selected in the range of 1,5 to 5 m/s.

39) Arora, op. cit., p. 194

Radial clearance (C_r) :

$$C_r = 0{,}0035 \times Bp$$
$$C_r = 0{,}0035 \times 65 \qquad (3.52)$$
$$C_r = 0{,}2275 \text{ mm}$$

Piston diameter (D_p) :

$$D_p = Bp - C_r$$
$$D_p = 65 - 0{,}2275 \qquad (3.53)$$
$$D_p = 64{,}8 \text{ mm}$$

Thickness of piston head (x_h) :

$$x_h = 0{,}43 D_p \times \sqrt{\frac{0{,}1 \times P_d}{\sigma_p}} \qquad (3.54)$$

where :

σ_p = permissible tensile stress on piston head (350 – 560 kg/cm^2),

taken 350 kg/cm^2.

$$x_h = 0{,}43 D_p \times \sqrt{\frac{0{,}1 \times P_d}{\sigma_p}}$$
$$x_h = 0{,}43 \times 6{,}48 \times \sqrt{\frac{0{,}1 \times 11{,}3}{350}}$$
$$x_h = 0{,}158 \text{ cm} = 1{,}58 \text{ mm}$$

b.2. Piston Rings

Piston rings are needed for sealing the piston of machines handling vapors and gases. The rings are made with an outer diameter larger than the cylinder diameter by approximately the thickness of the ring.

The super executive bus using the commonest types of piston rings, that is oil scraper rings. Those are mostly made of pearlitic-sorbitic cast iron. For highly stressed ring, it is usually alloyed with chromium and molybdenum. The Brinell hardness varies between 170 – 220 kg/mm^2 or even more, depending upon the purpose, material of the cylinder, size of rings, and method of casting. The bending strength for material is range from 35 – 45 kg/cm^2.[40]

The number of piston sealing rings varies within a wide range depending upon the pressure to be sealed distortion from the circular shape of the compressed ring, radial thickness of the rings, nature of the gas to be compressed, and other circumstances. For sealing pressures up to 5 atg, it fitted with 2 – 4 rings, for 5 – 30 atg the piston ring fitted with 3 – 6 rings.

Radial thickness of the ring (x_{tr}) :[41]

$$\begin{aligned} x_{tr} &= \frac{Bp}{25} \\ x_{tr} &= \frac{65}{25} \\ x_{tr} &= 2{,}6 \text{ mm} \end{aligned} \tag{3.55}$$

Internal diameter of piston ring (ID_p) :

$$\begin{aligned} ID_p &= Bp - x_{tr} \\ ID_p &= 65 - 2{,}6 \\ ID_p &= 62{,}4 \text{ mm} \end{aligned} \tag{3.56}$$

40) Chlumsky, op. cit., p. 175
41) Ibid, p. 178

Normal radial crack on piston ring (N_{rp}) :

$$N_{rp} = 0{,}0015 \times ID_p$$
$$N_{rp} = 0{,}0015 \times 62{,}4 \qquad (3.57)$$
$$N_{rp} = 0{,}0936 \text{ mm}$$

Depth of circular groove (H_{gr}) :

$$H_{gr} = N_{rp} + x_{tr}$$
$$H_{gr} = (0{,}0936 + 2{,}6) \text{mm} \qquad (3.58)$$
$$H_{gr} = 2{,}6936 \text{ mm}$$

Width of circular groove (W_{gr}) :

$$W_{gr} = N_{rp} + Wr_{tot} \qquad (3.59)$$

where :

Wr_{tot} = total width of ring (taken 3 mm)

$$W_{gr} = N_{rp} + Wr_{tot}$$
$$W_{gr} = (0{,}0936 + 3) \text{mm}$$
$$W_{gr} = 3{,}0936 \text{ mm}$$

b.3. Piston Rod

The piston rod transmits the force from the crosshead to one piston or several pistons in line, or inversely, transmits the force from the piston to the crosshead. The piston rod is made of hard, tough carbon steel, or of heat-treated alloy steel.[42] Construction and conditions of piston rods used in super executive bus :

42) Ibid, p. 182

- Tensile strength = 60 kg/mm^2
- Internal diameter (ID_r) = 6 mm
- Outer diameter (OD_r) = 10 mm

Length of piston rod (L_r) :

$$L_r = \frac{L1 + Dp}{2} \tag{3.60}$$

$$\begin{aligned} L1 &= 0{,}45 \times Dp \\ L1 &= 0{,}45 \times 64{,}8 \\ L1 &= 29{,}16 \text{ mm} \end{aligned} \tag{3.61}$$

$$\begin{aligned} L_r &= \frac{L1 + Dp}{2} \\ L_r &= \frac{29{,}16 + 64{,}8}{2} \\ L_r &= 46{,}98 \text{ mm} \end{aligned}$$

Force acting upon the piston rod (G) :

$$\begin{aligned} G &= A_p \times Ppress \\ G &= \left(\frac{\pi}{4} \times Dp^2\right) \times Ppress \\ G &= \left(\frac{3{,}14}{4} \times 6{,}48^2\right) \times 11{,}3 \\ G &= 372{,}5 \text{ kg} \end{aligned} \tag{3.62}$$

b.4. Crosshead

The crosshead guides the piston rod and transfers the side thrust to the guides. The side thrust comprises one component of the force acting along the connecting rod and in horizontal machines weight of a part of the piston rod or of the piston. The crosshead at the same time transmits the axial force from the piston rod to the connecting rod.[43] The force is transmitted to the pin about the axis of which the connecting rod oscillates.

Dimensions of crosshead :

- Material of crosshead : carbon steel
- Tensile strength = 45 – 60 kg/mm^2
- Radius of crosshead (re) = 30 mm

Length of crosshead (Lbg) :

$$\mathrm{Lbg} = (4-6)\times \mathrm{re}$$
$$\mathrm{Lbg} = 5\times 30 \tag{3.63}$$
$$\mathrm{Lbg} = 150\ \mathrm{mm}$$

b.5. Crank Shaft

The crankshaft is the most expensive parts of a reciprocating compressor.

They are 2 types of crankshafts :

1) An overhung crank
2) Cranked shaft

Both usually forged of carbon steel of a tensile strength of 50 – 60 kg/mm^2. The material should not posses a fibrous structure, for on machining the fibers are cut, reducing the notch strength of the material.[44]

43) Ibid, p. 186
44) Ibid, p. 206

Diameter of crank shaft (D_{cs}) :

$$D_{cs} = 0{,}15 \times Bp\sqrt{P_d}$$
$$D_{cs} = 0{,}15 \times 6{,}5\sqrt{11{,}3} \qquad (3.64)$$
$$D_{cs} = 3{,}28\ \text{cm} = 32{,}8\ \text{mm}$$

Crank thickness (x_{cs}) :

$$x_{cs} = (0{,}5 - 0{,}7) \times D_{cs}$$
$$x_{cs} = 0{,}6 \times 32{,}8 \qquad (3.65)$$
$$x_{cs} = 19{,}68\ \text{mm}$$

Width of crank (b_{cs}) :

$$b_{cs} = 1{,}25 \times D_{cs}$$
$$b_{cs} = 1{,}25 \times 32{,}8 \qquad (3.66)$$
$$b_{cs} = 41\ \text{mm}$$

Radius of fillet (r_f) :

$$r_f = \frac{D_{cs}}{15}$$
$$r_f = \frac{32{,}8}{15} \qquad (3.67)$$
$$r_f = 2{,}2\ \text{mm}$$

b.6. Valves

The mechanism that governs the moment of and duration of entry of gas into the cylinder and of its discharge from it is the valve gear. The principal requirements for compressor valve gear are :[45]

1) Good gas tightness.

2) Larger possible through flow area at the minimum periphery.

45) Ibid, p. 222

3) Small flow resistances involving smooth changes of direction of flow and smooth changes of cross section.
4) Small lift and small mass allowing a high compressor speed.
5) Small clearance space.
6) Quiet and smooth operation even at high speed.
7) Low price and ruggedness.

b.7. Motor Driven

Refrigerating capacity control with reciprocating compressors running at constant speed consists in controlling the quantity of the gas delivered to match the fluctuating load. In efficient control system, the power consumption should be proportional to the amount of gas delivered, such as on-off control.[46]

Simple on-off or start-stop control in conjunction with a thermostat is to advantage in small unitary equipment, such as refrigerators, air conditioners, water cooler, etc. It is particularly suited where there is a sudden large demand followed by periods of small or no demand on capacity. Switching off is preferable to running at partial load because of the low part-load efficiency of squirrel-cage induction motors employed in the super executive refrigerant compressor.

Power requirement driven the compressor (W_d) :

$$W_d = \frac{W_t}{\eta_t \cdot \eta_m} \tag{3.68}$$

46) Arora, op. cit., p. 196

where :

W_t = theoretical power consumption for each compressor (kW)

η_t = transmission efficiency, usually range from 0,92 – 0,98 (taken 0,97).

η_m = mechanical efficiency, range from 0,85 – 0,96 (taken 0,85).

$$W_d = \frac{W_t}{\eta_t \cdot \eta_m}$$

$$W_d = \frac{7{,}64\ \text{kW}}{0{,}97 \times 0{,}85}$$

$$W_d = 9{,}27\ \text{kW}$$

b.9. Lubrication on Compressor

Adequate lubrication of the cylinder is essential for all mechanical compressors (with the exception of the special oil-free compressors) and of the driving mechanism, which is most frequently a crank mechanism. The selection of the lubricating system depends upon many factors, such as :[47]

- Size of the compressor
- Nature of the gas handled
- Arrangement of the compressor
- Speed of the compressor, etc.

In single-acting vertical compressors without crossheads, the cylinders are lubricated with the same oil as used for the bearings, crank pin and

47) Chlumsky, op. cit., p. 469

gudgeon pin. The compressor has a closed crankcase and the oil is either thrown onto the cylinder wall by splashing action of a pin fastened to the cover of the connecting rod end, of where the force-feed lubrication is used, the oil is thrown from the shaft and connecting rod. One disadvantage of splash lubrication is the variation of lubrication rate due to the fluctuation of the oil level in the crankcase.

For super executive bus, the compressor is lubricated with an ester-based oil specially designed for use with R-134a refrigerant.

Lubricants viscosity on compressor (ν_l) :

$$\upsilon_l = \frac{100}{V_m^{0.4}} \tag{3.69}$$

where :

v_m = mean velocity of piston, usually 2 m/s.

$$\upsilon_l = \frac{100}{V_m^{0,4}}$$

$$\upsilon_l = \frac{100}{2^{0,4}}$$

$$\upsilon_l = 75{,}76 \text{ kg / m} \cdot \text{s}$$

C. Condenser Selection

It is ultimately in the condenser that heat is rejected in a vapor-compression refrigeration machine. The vapor at discharge from the compressor is superheated. Desuperheating of the vapor takes place in the discharge line and in the first few coils of the condenser. It is followed by the condensation of the vapor at the saturated discharge temperature or condensing temperature.[48]

Given data for condenser :

- Temperature of entering air (Ta_{in}) = 30°C
- Temperature of leaving air (Ta_{exit}) = 35°C
- Temperature difference (ΔTa) = 5°C
- Temperature of refrigerant in the condenser (Tr) = 45°C

The LMTD is the logarithmic average temperature difference between the two fluid streams at inlet and outlet of the heat exchangers.

Log mean temperature difference (LMTD) :[49]

$$\begin{aligned} \text{LMTD} &= \frac{(Tr - Ta_{in}) - (Tr - Ta_{exit})}{\ln\dfrac{(Tr - Ta_{in})}{(Tr - Ta_{exit})}} \\ \text{LMTD} &= \frac{(45-30)-(45-35)}{\ln\dfrac{(45-30)}{(45-35)}} \\ \text{LMTD} &= \frac{5}{0{,}405} \\ \text{LMTD} &= 12{,}35\,°\text{C} \end{aligned} \tag{3.70}$$

Designing air temperature (T_{ad}) :

$$\begin{aligned} T_{ad} &= Tr - \text{LMTD} \\ T_{ad} &= 45 - 12{,}35 \\ T_{ad} &= 32{,}65\,°\text{C} \end{aligned} \tag{3.71}$$

48) Arora, op. cit., p. 219
49) Stoecker, op. cit, p 247

Dimensions of condenser :

- Material of tubes = copper
- Thermal conductivity of condenser tubes (k_t) = 400 W/m°C
- Inner diameter of tube (ID) = 1,125 in = 0,0286 m = 28,575 mm
- Outer diameter of tube (OD) = 1,375 in = 0,035 m = 34,925 mm
- Thickness of tube (x) = 0,25 in = 0,00635 m
- Number of passes (p) = 2
- Number of tubes (n_t) = 40 tubes
- Number of tubes each pass (n_p) = 20 tubes
- Tube spacing (S_t) = 3,2 mm = 0,0032 m

Heat rejectcd on the side of condenser tubes (Qt_{cond}) :

$$
\begin{aligned}
Qt_{cond} &= Q_{cond} \times QC \\
Qt_{cond} &= 244{,}4\ \text{Btu / ton.min} \times \frac{6{,}56\ \text{TR}}{2} \qquad (3.72)\\
Qt_{cond} &= 801{,}632\ \text{Btu/min} = 48097{,}92\ \text{Btu/hr} \\
Qt_{cond} &= 14{,}1\ \text{kW} \approx 14\ \text{kW}
\end{aligned}
$$

The properties of R134a at 45°C :[50]

- Density of liquid (ρ_{lr}) = 1124,3 kg/m^3
- Specific latent enthalpy (h_{fg}) = 273,785 kJ/kg
- Conductivity of refrigerant (k_r) = 0,0697 W/m°C

50) Ari Rabl, Heating and Cooling for Buildings, p. 795

- Dynamic viscosity (μ_r) = 0,000267 kg/m.s
- Prandtl number (Pr_r) = 13,468

The properties of air at 35°C :[51]

- Density of air (ρ_a) = 1,192 kg/m^3
- Thermal conductivity of air (k_a) = 0,027 W/m°C
- Specific heat (cp_a) = 1,006 kJ/kg°C
- Dynamic viscosity (μ_a) = 0,00001998 kg/m.s
- Prandtl number (Pr_a) = 0,706

Outer area of tubes (air-side coefficient) :

In refrigerant condenser with forced convection, air flows outside the tubes in cross-flow to the refrigerant. The tubes are arranging in-line.

Resistance of copper tubes (h_{tubes}) :

$$h_{tubes} = \frac{x \cdot Ao}{k_t \cdot Am}$$

$$h_{tubes} = \frac{(0,035 - 0,0286)/2}{400} \times \frac{34,925}{(28,575 + 34,925)/2} \qquad (3.73)$$

$$h_{tubes} = 0,0000088 \ m^2.°C/W$$

The value above will prove to be negligible in comparison of other resistances.

Fouling factor ($\frac{1}{h_{ff}}$) = 0,000176 m^2.°C/W.[52]

51) Ir. E. Jasjfi, J. P. Holman, Perpindahan Kalor, p. 589
52) Stoecker, op. cit., p. 249

The flow rate of air (m_a) needed to carry the heat away from the condenser with a temperature rise from 30°C to 35°C is :

$$m_a = \frac{Q_{cond}}{cp_a \times \Delta Ta}$$

$$m_a = \frac{14\,kW}{1{,}006 \times (35-30)} \tag{3.74}$$

$$m_a = 2{,}78\ kg/s$$

The volume flow rate of air (Qa) :[53]

$$Qa = \frac{m_a}{\rho_a}$$

$$Qa = \frac{2{,}78}{1{,}192} \tag{3.75}$$

$$Qa = 2{,}33\ m^3/s$$

The air velocity through the tubes (v_a) :

$$v_a = \frac{Qa}{n_p \times \frac{\pi}{4} \times ID^2}$$

$$v_a = \frac{2{,}33}{20 \times \frac{3{,}14}{4} \times (0{,}0286)^2} \tag{3.76}$$

$$v_a = 181{,}44\ m/s$$

Reynolds number of air (Re_a) :[54]

$$Re_a = \frac{v_a \times \rho_a \times ID}{\mu_a}$$

$$Re_a = \frac{181{,}44 \times 1{,}192 \times 0{,}0286}{0{,}00001998} \tag{3.78}$$

$$Re_a = 309584{,}9514$$

53) Ibid, p. 250
54) Ibid, p. 237

According to Mc Adams, Reynolds number for turbulent flow 3000<Re<3.000.000. The value above from the calculation indicates that the flow is turbulent.

The Nusselt number (Nu_a) can be calculated as :

$$Nu_a = 0{,}023 \times Re_a^{0.8} \times Pr_a^{0,4}$$
$$Nu_a = 0{,}023 \times 309584{,}9514^{0.8} \times 0{,}706^{0,4} \qquad (3.79)$$
$$Nu_a = 494{,}2$$

Air-side heat transfer coefficient (h_a) :

$$h_a = \frac{Nu_a \times k_a}{OD}$$
$$h_a = \frac{494{,}2 \times 0{,}027}{0{,}035} \qquad (3.80)$$
$$h_a = 381{,}24\ W/m^2 \cdot {}^\circ C$$

Resistance of tubes (R_{tubes}) :[55]

$$R_{tubes} = \frac{\ln \frac{r_o}{r_i}}{2 \times \pi \times k_t}$$
$$R_{tubes} = \frac{\ln \frac{0{,}0175}{0{,}0143}}{2 \times 3{,}14 \times 400} \qquad (3.81)$$
$$R_{tubes} = 0{,}0000792$$

55) Ibid, p. 249

Refrigerant-side coefficient :

The average numbers of tubes per row (N_v) :

$$N_v = \frac{n_t}{n_p}$$
$$N_v = \frac{40}{20} \tag{3.82}$$
$$N_v = 2$$

Heat transfer of refrigerant side (h_r) :

$$h_r = 0{,}725\left(\frac{g \times h_{fg} \times \rho_r^{\,2} \times k_r^{\,3}}{\mu_r \times N_v \times ID \times \Delta T}\right)^{1.4}$$
$$h_r = 0{,}725\left(\frac{10 \times 273{,}785 \cdot 10^3 \times 1124{,}3^2 \times 0{,}0697^3}{0{,}000267 \times 2 \times 0{,}0286 \times 5}\right)^{1.4} \tag{3.83}$$
$$h_r = 1434{,}95\ \mathrm{W/m^2\,°C}$$

Overall heat-transfer coefficient (U_{tot}) :

$$\frac{1}{U_{tot}} = \frac{1}{h_a} + (R \times Ao) + \frac{Ao}{h_{ff}} + \frac{Ao}{h_r \times Ai}$$
$$\frac{1}{U_{tot}} = \frac{1}{381{,}24} + 0{,}0000088 + \frac{0{,}035}{0{,}0286} \times 0{,}000176 + \frac{0{,}035}{0{,}0286 \times 1434{,}95}$$
$$\frac{1}{U_{tot}} = 0{,}0026230196 + 0{,}0000088 + 0{,}0002153 + 0{,}0008528354 \tag{3.84}$$
$$\frac{1}{U_{tot}} = 0{,}003699955$$
$$U_{tot} = 270{,}3\ \mathrm{W/m^2 \cdot °C}$$

Total outside area of tubes (Ao_t) :[56]

$$Ao_t = \frac{Q_{cond}}{U_{tot} \times LMTD}$$
$$Ao_t = \frac{14000\text{ W}}{270{,}3 \times 12{,}35} \tag{3.85}$$
$$Ao_t = 4{,}2\ \text{m}^2$$

Length of condenser tubes (Lc_t) :

$$Lc_t = \frac{Ao_t}{n_t \times \pi \times OD}$$
$$Lc_t = \frac{4{,}2}{40 \times 3{,}14 \times 0{,}035} \tag{3.86}$$
$$Lc_t = 0{,}96\ \text{m}$$

Total inside area of tubes (Ai_t) :

$$Ai_t = \pi \times ID \times Lc_t \times n_t$$
$$Ai_t = 3{,}14 \times 0{,}0286 \times 0{,}96 \times 40 \tag{3.87}$$
$$Ai_t = 3{,}45\ \text{m}^2$$

Total length of condenser tubes (Lc_{tot}) :

$$Lc_{tot} = n_t \times Lc_t$$
$$Lc_{tot} = 40 \times 0{,}96 \tag{3.88}$$
$$Lc_{tot} = 38{,}4\ \text{m}$$

c.1. Pressure Drop in Condenser

As the fluid flows inside the tubes through a condenser, a pressure drop occurs both in straight tubes and in the U-bends or head of the heat exchanger. Some drop in pressure is also attributable to entrance and exit losses.

56) Ibid, p. 250

Since the pressure drop in the straight tubes in a condenser may represent only 50 – 80% of the total pressure drop, experimental or catalog data on the pressure drop as a function of flow are desirable.[57]

Friction factor by air inside area of tubes (f_a) :

$$f_a = \frac{1}{(1{,}82 \times \log \mathrm{Re}_a - 1{,}64)^2}$$

$$f_a = \frac{1}{(1{,}82 \times \log 309584{,}9514 - 1{,}64)^2} \qquad (3.89)$$

$$f_a = 0{,}0143$$

Pressure drop of air flowing in tubes (ΔP_a) :

$$\Delta P_a = f_a \times \frac{Lc_{tot}}{ID} \times \frac{v_a^{\ 2}}{2g} \times \rho_a$$

$$\Delta P_a = 0{,}0143 \times \frac{38{,}4}{0{,}0286} \times \frac{181{,}44^2}{2 \times 10} \times 1{,}192 \qquad (3.90)$$

$$\Delta P_a = 37671{,}6 \text{ kg/m}^2 = 37{,}7 \text{ kPa}$$

Previous data :

Q_{cond} = 14 kW

cp_r = 1,484 kJ/kg°C (specific heat of R-134a at 45°C)

The flow rate of refrigerant (m_r) :

$$m_r = \frac{Q_{cond}}{cp_r \times \Delta T_r}$$

$$m_r = \frac{14 \text{ kW}}{1{,}484 \times (55 - 40)} \qquad (3.91)$$

$$m_r = 0{,}63 \text{ kg/s}$$

57) Ibid, p. 237

The volume flow rate of refrigerant (Q_r) :

$$Q_r = \frac{m_r}{\rho_{lr}}$$

$$Q_r = \frac{0,63}{1124,3} \tag{3.92}$$

$$Q_r = 0,00056 \text{ m}^3/\text{s} \approx 0,56 \text{ L/s}$$

c.2. Extended Surface (Fins)

Most air-cooling coils consists of tubes with fins attached to the outside of the tubes to increase the area on the air side where the convection coefficient is generally much lower than on the refrigerant side. Refrigerant flows inside the tubes, and air flows over the outside of the tubes and the fins.[58]

The bar fin is not a common shape but the dominant type of finned surface is the rectangular plate fin mounted on cylindrical tubes. The fin effectiveness of the rectangular plate fins is often calculated by using properties of the corresponding annular fin. It has the same area and thickness as the plate fin.

Given data for condenser fins :

- Fins material = aluminum
- Thermal conductivity of fins (k_f) = 202 W/m °C
- Fins thickness (x_f) = 0,001 m = 1 mm
- Height of fins (h_f) = 0,02 m = 20 mm

58) Ibid, p. 239

- Width of fins (w_f) = 0,02 m = 20 mm
- Fins spacing (S_f) = 0,01 m = 10 mm
- Number of fins (N_f) = 250 fins

Surface area of fins per each length of condenser tubes (A_f) :

$$A_f = N_f \times \left[4(x_f \times w_f) + \left((w_f' \times h_f') - \left(\frac{\pi}{4} \times OD^2\right)\right) \times 2\right]$$

$$A_f = 250 \times \left[4(1 \times 20) + \left((40 \times 40) - \left(\frac{3,14}{4} \times 34,925^2\right)\right) \times 2\right] \tag{3.93}$$

$$A_f = 341245,92 \ \ mm^2/\ tube$$

Total area of fins in condenser (A_{ftot}) :

$$A_{ftot} = A_f \times n_t$$

$$A_{ftot} = 0,34 \times 40 \tag{3.94}$$

$$A_{ftot} = 13,6 \ \ m^2$$

Total surface area of tubes (A_{tubes}) :

$$A_{tubes} = (\pi \times OD \times Lc_t) - (\pi \times OD \times x_f \times N_f)$$

$$A_{tubes} = (3,14 \times 34,925 \times 960) - (3,14 \times 34,925 \times 1 \times 250) \tag{3.95}$$

$$A_{tubes} = 77861,795\, mm^2 = 0,078 \ \ m^2$$

Surface area of fins per length of tubes (A_{tf}) :

$$A_{tf} = A_f + A_{tubes}$$

$$A_{tf} = 0,34 + 0,078 \tag{3.96}$$

$$A_{tf} = 0,418\, m^2$$

Total area of condenser fin coils (A_{tot}) :

$$A_{tot} = A_{ftot} + (A_{tubes} \times n_t)$$

$$A_{tot} = 13,6 + (0,078 \times 40) \tag{3.97}$$

$$A_{tot} = 16,72 \ \ m^2$$

The ratio of the actual rate of heat transfer to that which would be transferred if the fins were at the temperature of tubes (T) is called the fin effectiveness.

Harper and Brown found that the fin effectiveness for bar fin could be represented by :[59]

$$M = \sqrt{\frac{h_a}{ky}} \tag{3.98}$$

where :

M = bar fin effectiveness per length (m^{-1})

h_a = air-side heat transfer coefficient ($W/m^2.°C$)

k_f = thermal conductivity of fins (W/m °C)

y = half thickness of fin (m)

$$M = \sqrt{\frac{h_a}{ky}}$$

$$M = \sqrt{\frac{381{,}24}{202 \times 0{,}5 \cdot 10^{-3}}}$$

$$M = 61{,}44 \;\; m^{-1}$$

$$\begin{aligned}(r_e - r_i)M &= (0{,}0175 - 0{,}0143) \times 61{,}44 \\ (r_e - r_i)M &= 0{,}2\end{aligned} \tag{3.99}$$

The annular fins having the same area as the plate fin has an external radius (r_e) of 17,5 mm and internal radius (r_i) of 14,3 mm.

For $(r_e - r_i)M = 0{,}2$ and $r_e/r_i = 1{,}22$ the fin effectiveness (η_f) is 0,98.

59) Ibid, p. 241

c.3. Condenser Fans and Motors

Ultimately it is the fan, which moves the air through the entire duct system and conditioned space. Fans may be classified into two main types based on the direction of air flow through the fan :

1) Centrifugal fans : the direction of air flow is radial (centrifugal) relative to the fan shaft.
2) Axial flow fans : the direction of air flow is axial relative to the fan shaft.

The super executive bus applies the axial flow fans considering its applications and performances are as follow :

a) Low in cost and pressure
b) Little or no ductwork
c) Typical applications are as wall or window-installed exhaust fans

Specifications of fan and motor :

- Material of fans = aluminum
- Type of propeller = axial blade
- Number of fans (N_{cf}) = 2 pieces
- Number of blades (N_b) = 6 pieces
- Outer diameter of fan (OD_f) = 457 mm
- Inner diameter of fan (ID_f) = 152 mm
- Speed of motor (N_m) = 1750 rpm

The super executive bus has 2-off heavy-duty squirrel cage motors with the capacity is 0,5 HP (0,37 kW). Types of these motor are as follow :

- Long life
- Permanent magnet
- Replaceable brush type fan motors

Some advantages of squirrel cage motor are :

- Simple in construction
- Easy to start
- Low cost
- Efficient and reasonably good power factor

Previous data :

Temperature of entering air (Ta_{in})	= 30°C
Temperature of leaving air (Ta_{exit})	= 35°C
Temperature difference (ΔTa)	= 5°C
Specific heat of air at 30°C (cp_a)	= 1,006 kJ/kg°C
Density of air at 30°C (ρ_a)	= 1,165 kg/m^3
Q_{cond}	= 14 kW

Mass flow rate of air through the fans ($\dot{m}_{af}$) :

$$\dot{m}_{af} = \frac{Q_{cond}}{cp_a \times \Delta Ta}$$

$$\dot{m}_{af} = \frac{14\ \text{kW}}{1{,}006 \times (35 - 30)}$$

$$\dot{m}_{af} = 2{,}78\ \text{kg/s}$$

Rate of air needed thru fans (Q_{af}) :

$$Q_{af} = \frac{\dot{m}_{af}}{\rho_a}$$

$$Q_{af} = \frac{2{,}78}{1{,}165}$$

$$Q_{af} = 2{,}39\ \text{m}^3/\text{s}$$

Air velocity of fans (v_{af}) :

$$v_{af} = \frac{4 \times Q_{af}}{\pi \times (OD_f^{\,2} - ID_f^{\,2})}$$

$$v_{af} = \frac{4 \times 2{,}39}{3{,}14 \times (0{,}457^2 - 0{,}152^2)} \tag{3.100}$$

$$v_{af} = 16{,}4\ \text{m/s}$$

Blade spacing (S_{fb}) :

$$S_{fb} = \frac{\pi \times OD_f}{N_b}$$

$$S_{fb} = \frac{3{,}14 \times 457}{6} \tag{3.101}$$

$$S_{fb} = 239{,}2\ \text{mm} \approx 0{,}24\ \text{m}$$

Blade width (w_{fb}) :

$$w_{fb} = \frac{Q_{af}}{\pi \times OD_f \times v_{af}}$$
$$w_{fb} = \frac{2{,}39}{3{,}14 \times 0{,}457 \times 16{,}4} \tag{3.102}$$
$$w_{fb} = 0{,}102 \text{ m} = 102 \text{ mm}$$

Angular velocity of fan (ω_f) :

$$\omega_f = \frac{\pi \times OD_f \times N_m}{60}$$
$$\omega_f = \frac{3{,}14 \times 0{,}457 \times 1750}{60} \tag{3.103}$$
$$\omega_f = 41{,}85 \text{ rad/s}$$

D. Expansion Valve Selection

The purpose of the expansion devices is twofold : it must reduce the pressure of the liquid refrigerant, and it must regulatc the flow of the refrigerant to the evaporator.[60]

The super executive bus using the superheat-controlled (thermostatic) expansion valve (TEV), which is the most popular type of expansion device for moderate-sized refrigeration systems. The name maybe is misleading because control is actuated not by the temperature in the evaporator but the magnitude of superheat of the suction gas leaving the evaporator.

The superheat of the suction gas operates the TEV as follows. A bulb filled with a fluid is strapped t the suction line, and thus senses the suction gas temperature. This bulb is connected to the valve by a tube in a manner so that the pressure of the fluid in the bulb tends to open the valve more, against a

60) Ibid, p. 260

closing spring temperature. If the load in the system increases, the refrigerant in the evaporator picks up more heat and the suction gas temperature rises. The pressure of the fluid in the bulb increases as its temperature rises, and it opens the valve more. This increases the refrigerant flow needed to handle the increased load. The reverse of all these events occurs when the refrigeration load decreases.

It is important that the refrigerant vapor leaving the evaporator be a few degrees above the saturation temperature (called superheat) to insure that no liquid enters the compressor, which might result in its damage.

The type of TEV employed in the super executive bus is an externally equalized TEV. Some advantages of the externally equalized TEV are :[61]

1) Automatically operation.
2) Increasing the amount of superheat of refrigerant vapor.
3) Reducing the effectiveness of the evaporator if there is a larger pressure drop in the evaporator.
4) Operating over a wide range of evaporator temperatures.
5) Regulating the rate of flow of liquid refrigerant in proportion to the rate of evaporation in the evaporator.

Dimensions of TEV :

- Material of tubes = steel
- Outside diameter of tube (OD_{TEV}) = 0,0272 m

61) Edward G. Pita, op. cit., p. 303

- Inside diameter of tube (ID_{TEV}) = 0,0214 m
- Liquid density of R143a in condenser (ρ_{lr}) = 1124,3 kg/m^3
- Dynamic viscosity of R134a in condenser (μ_r) = 0,000267 kg/m.s
- Refrigerant flow rate ($\dot{w}$) = 2,78 lb/min = 0,021 kg/s

The volume flow rate of refrigerant inside of tube (Qx_r) :[62]

$$Qx_r = \frac{w}{\rho_{lr}}$$

$$Qx_r = \frac{0{,}021}{1124{,}3}$$

$$Qx_r = 1{,}8 \cdot 10^{-5} \text{ m}^3/\text{s}$$

Area of inside of tube (Ax_i) :

$$Ax_i = \frac{\pi}{4} \times ID_{TEV}{}^2$$

$$Ax_i = \frac{3{,}14}{4} \times (0{,}0214)^2$$

$$Ax_i = 0{,}00036 \text{ m}^2$$

Mass rate of refrigerant flow inside of tube (vx_r) :

$$vx_r = \frac{w}{Ax_i}$$

$$vx_r = \frac{0{,}021}{0{,}00036}$$

$$vx_r = 58{,}3 \text{ kg} / \text{m}^2 \cdot \text{s}$$

62) Arora, op. cit, p. 247

Reynolds number (Re_x) :

$$Re_x = \frac{vx_r \times ID_{TEV}}{\mu_r}$$

$$Re_x = \frac{58{,}3 \times 0{,}0214}{0{,}000267}$$

$$Re_x = 4672{,}7$$

Mc Adams proposed the Reynolds number for turbulent region is 3000<Re<3.000.000. Thus, the value above is shown in turbulent flows.

Pressure difference (ΔP) :

$$\Delta P = P_{cond} - P_{evap}$$

$$\Delta P = (11{,}3 - 2{,}4)\,kg/cm^2 \qquad (3.104)$$

$$\Delta P = 8{,}9\ kg/cm^2$$

Area of orifice (A_{ori}) :

$$A_{ori} = \frac{Qx_r}{0{,}7 \times \left(\frac{2 \times \Delta P}{\rho_{lr}}\right)^{1/2}}$$

$$A_{ori} = \frac{1{,}8 \cdot 10^{-5}}{0{,}7 \times \left(\frac{2 \times 89}{1124{,}3}\right)^{1/2}} \qquad (3.105)$$

$$A_{ori} = 0{,}000064\ m^2$$

Diameter of orifice (D_{ori}) :

$$D_{ori} = \sqrt{\frac{4 \times A_{ori}}{\pi}}$$

$$D_{ori} = \sqrt{\frac{4 \times 0{,}000064}{3{,}14}} \qquad (3.106)$$

$$D_{ori} = 0{,}009\ m$$

E. Evaporator Selection

Evaporators are heat exchangers within the tubes of which boiling refrigerant flows to cool either water (for use in cooling coils) or building supply air directly. The three heat-transfer resistances in evaporators are :[63]

1) Refrigerant side for the transfer of heat from solid surface to the liquid refrigerant.
2) Metal wall.
3) Cooled-medium side which could be due to air, water, brine or any other fluid or a wetted surface on cooling and dehumidifying coil.

Of primary interest here is the heat transfer from a solid surface of the evaporating refrigerant. However, the mechanism of boiling is so complex because of the influence of such factors as surface tension, saturation temperature, latent heat and nature of the solid surface, in addition to the usual transport properties, that it is difficult to predict the heat transfer coefficient.

Refrigerant also boils to cool air in direct expansion (DX) evaporators. The thermal performance of DX evaporators (usually shell-and-finned-tube heat exchangers) can be enhanced by using fins. The super executive bus using the dry expansion (DX) cooling coils by considering the advantages are as follow :

1) Wetting all the interior surfaces of the evaporator.
2) Maintaining a high coefficient of heat transfer.

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63) Ibid, p. 252

3) Higher compressor efficiency.

4) Higher compressor volumetric efficiency.

5) Less danger of damage to the compressor caused by slugs of liquid.

Calculated values :

- Evaporator pressure (P_e) = 34 psia = 2,4 kg/cm^2
- Cooling capacity on evaporator (Q_e) = 3,28 TR = 11,55 kW
- Refrigerant temperature (T_r) = 4°C
- Temperature of outside surface of tubes (T_{os}) = 35°C
- Temperature of inside surface of tubes (T_{is}) = 20°C
- Refrigerating effect (RE) = 72 Btu/lb

Dimensions of evaporator :

- Material of tubes = copper
- Thermal conductivity of tube metal (k_t) = 400 W/m°C
- Inner diameter of tube (ID) = 1,125 in = 0,0286 m = 28,575 mm
- Outer diameter of tube (OD) = 1,375 in = 0,035 m = 34,925 mm
- Thickness of tube (x_e) = 0,25 in = 0,00635 m
- Number of tubes in evaporator (n_{te}) = 40 tubes
- Number of tubes each pass (n_{tp}) = 20
- Number of pass (n_{pe}) = 2
- Tube spacing (S_e) = 32 mm = 0,032 m

The properties of R-143a at 4°C :

- Density of vapor (ρ_{vr}) = 16,536 kg/m^3
- Density of liquid (ρ_{lr}) = 1280,3 kg/m^3
- Dynamic viscosity (μ_r) = 0,000192 m^2/s
- Thermal conductivity of refrigerant (k_r) = 0,070 W/m°C
- Prandtl number (Pr_r) = 3,72
- Specific heat (cp_r) = 1,353 kJ/kg°C
- Enthalpy (h_{fg}) = 195,718 kJ/kg

The properties of air at 35°C :

- Density of air (ρ_a) = 1,192 kg/m^3
- Thermal conductivity of air (k_a) = 0,027 W/m°C
- Prandtl number of air (Pr_a) = 0,706
- Cinematic viscosity (ν_a) = 2,33 x 10^{-5} m^2/s
- Dynamic viscosity (μ_a) = 1,88 x10^{-5} kg/m.s
- Specific heat (cp_a) = 1,006 kJ/kg°C

Outer area of tubes (air-side coefficient) :

Mass flow rate of air ($\overset{\circ}{m}_a$) :

$$\dot{Q}_e = \dot{m}_a \times cp_a \times \Delta T$$

$$\dot{m}_a = \frac{Q_e}{cp_a \times (T_a - T_r)}$$

$$\dot{m}_a = \frac{11,55}{1,006 \times (22 - 4)}$$

$$\dot{m}_a = 0,64 \text{ kg/s}$$

Mass velocity of air (G) :

$$G = \frac{\dot{m}_a}{\rho_a}$$

$$G = \frac{0,64}{1,192}$$

$$G = 0,537 \text{ m}^3/\text{s}$$

The air velocity (v_{ae}) is range from 2 m/s to 3 m/s, assuming the air velocity to be taken is 2,5 m/s.[64]

Reynolds number for air (Re_a) :

$$Re_a = \frac{v_{ae} \cdot \rho_a \cdot OD}{\mu_a}$$

$$Re_a = \frac{2,5 \times 1,192 \times 0,035}{1,88 \cdot 10^{-5}}$$

$$Re_a = 5547,87$$

For Re > 2300 indicates the flow is turbulent.

The Nusselt number (Nu_a) :

$$Nu_a = 0,023 \times Re_a^{0.8} \times Pr_a^{0.3}$$

$$Nu_a = 0,023 \times 5547,87^{0.8} \times 0,706^{0.3}$$

$$Nu_a = 20,5$$

64) Ibid, p. 260

Heat transfer coefficient for air (h_a) flowing over exterior surface of tubes is :

$$h_a = \frac{Nu_a \times k_a}{OD}$$

$$h_a = \frac{20{,}5 \times 0{,}027}{0{,}035}$$

$$h_a = 15{,}8\ W / m^2\ °C$$

Refrigerant-side coefficient :

Mass flow rate of refrigerant ($\overset{\circ}{m}_r$) :

$$Q_e = \overset{\circ}{m}_r \times cp_r \times (T_{os} - T_r)$$

$$\overset{\circ}{m}_r = \frac{Q_e}{cp_r \times (T_{os} - T_r)}$$

$$\overset{\circ}{m}_r = \frac{11{,}55}{1{,}353 \times (20 - 4)}$$

$$\overset{\circ}{m}_r = 0{,}534\ kg/s$$

The volume flow rate of refrigerant (Q_r) :

$$Q_r = \frac{\overset{\circ}{m}_r}{\rho_{vr}}$$

$$Q_r = \frac{0{,}534}{1280{,}3}$$

$$Q_r = 0{,}00042\ m^3/s$$

The velocity of refrigerant thru the evaporator tubes (v_r) :

$$v_r = \frac{Q_r}{n_p \times \frac{\pi}{4} \times ID^2}$$

$$v_r = \frac{0{,}00042}{20 \times \frac{3{,}14}{4} \times 0{,}0286^2}$$

$$v_r = 0{,}033\ \ m/s$$

Reynolds number for R-134a (Re_r) :

$$Re_r = \frac{\rho_{lr} \times v_r \times ID}{\mu_r}$$

$$Re_r = \frac{1280{,}3 \times 0{,}033 \times 0{,}0286}{0{,}000192}$$

$$Re_r = 6293{,}5$$

The value above indicates the turbulent flows (Re>2300).

Partial steam of refrigerant (X_r) :

$$X_r = \frac{h_3' - h_2}{h_4 - h_1'}$$

$$X_r = \frac{114 - 42}{130 - 42}$$

$$X_r = 0{,}82$$

Condensing coefficient for vapor condensing on the outside tubes (h_{ct}) :

$$h_{ct} = \frac{0{,}06 \times k_r}{OD} \times \left(Re_r \times X_r\right)^{0.87} \times Pr_r^{\ 0.4} \times \left(\frac{\rho_{vr}}{\rho_{lr}}\right)^{0.28}$$

$$h_{ct} = \frac{0{,}06 \times 0{,}070}{0{,}035} \times \left(6293{,}5 \times 0{,}82\right)^{0.87} \times 3{,}72^{0.4} \times \left(\frac{1280{,}3}{16{,}536}\right)^{0.28}$$

$$h_{ct} = 1165{,}4 \ W / m^2 \cdot {}^\circ C$$

The Nusselt number of refrigerant (Nu_r) :

$$Nu_r = 0{,}023 \times Re_r^{\ 0.8} \times Pr_r^{\ 0.3}$$

$$Nu_r = 0{,}023 \times 6293{,}5^{0.8} \times 3{,}72^{0.3}$$

$$Nu_r = 37{,}33$$

The refrigerant-side heat transfer coefficient of inner side of evaporator tubes (h_r) :

$$h_r = \frac{Nu_r \times k_r}{ID}$$
$$h_r = \frac{37{,}33 \times 0{,}070}{0{,}0286}$$
$$h_r = 91{,}37 \ W / m^2 \ °C$$

Resistance of heat transfer of evaporator tubes (h_{tubes}) :

$$\frac{1}{h_{tubes}} = \frac{x \cdot A_o}{k_t \cdot A_m}$$
$$\frac{1}{h_{tubes}} = \frac{(0{,}035 - 0{,}0286)/2}{400} \times \frac{34{,}925}{(28{,}575 + 34{,}925)/2}$$
$$\frac{1}{h_{tubes}} = 0{,}0000088 \ m^2 \cdot °C / W$$

Several different agencies have established standards for the fouling factor to be used. One trade association species 0,000176 m^2.K/W, which means that the evaporator should leave the factory with a $1/U_o$ value 0,000176 A_o/A_i less than the minimum required to meet the quoted capacity of the evaporator.[65]

$$(\frac{1}{h_{ff}}) = 0{,}000176 \ m^2.°C/W$$

Total resistances to heat transfer in the evaporator (U_{oe}) :

$$\frac{1}{U_{oe}} = \frac{1}{h_{ct}} + \frac{1}{h_{tube}} + \frac{A_o}{h_{ff}} + \frac{A_o}{h_r A_i}$$
$$\frac{1}{U_{oe}} = \frac{1}{1165{,}4} + 0{,}0000088 + \left(\frac{0{,}035}{0{,}0286} \times 0{,}000176\right) + \frac{0{,}035}{0{,}0286 \times 91{,}37}$$
$$\frac{1}{U_{oe}} = 0{,}00086 + 0{,}0000088 + 0{,}000215 + 0{,}0134$$
$$U_{oe} = 71{,}43 \ W/m \cdot °C$$

65) Stoecker, p. 247

The LMTD of evaporator ($LMTD_e$) :

$$LMTD_e = \frac{(T_{os} - T_r) - (T_{is} - T_r)}{\ln \frac{(T_{os} - T_r)}{(T_{is} - T_r)}}$$

$$LMTD_e = \frac{(35 - 4) - (20 - 4)}{\ln \frac{(35 - 4)}{(20 - 4)}}$$

$$LMTD_e = 22,7\ °C$$

Overall area of outer tubes of evaporator (A_{oe}) :

$$A_{oe} = \frac{Q_e}{U_{oe} \times LMTD}$$

$$A_{oe} = \frac{11550\ W}{71,43 \times 22,7}$$

$$A_{oe} = 7,12\ m^2$$

Length of evaporator tubes (L_{evap}) :

$$L_{evap} = \frac{A_{oe}}{n_t \times \pi \times OD}$$

$$L_{evap} = \frac{7,12}{40 \times 3,14 \times 0,035}$$

$$L_{evap} = 1,62\ m$$

Total length of evaporator tubes (Lt_{evap}) :

$$Lt_{evap} = L_{evap} \times n_t$$

$$Lt_{evap} = 1,62 \times 40$$

$$Lt_{evap} = 64,8\ m \approx 65\ m$$

e.1. Pressure Drop in Evaporator

The pressure of the refrigerant drops as it flows through tube-type evaporators. The effect of pressure drop on system performance is that the compressor must pump from a lower suction pressure, which increases the power requirement. On the other hand, a high refrigerant velocity can be achieved if more pressure drop is permitted, and this high velocity improves the heat-transfer coefficient friction factor by refrigerant inside area of tubes (f_r) :

$$f_r = \frac{1}{(1{,}82 \times \log \mathrm{Re}_r - 1{,}64)^2}$$

$$f_r = \frac{1}{(1{,}82 \times \log 6293{,}5 - 1{,}64)^2}$$

$$f_r = 0{,}036$$

Pressure drop of refrigerant flowing in tubes (ΔP_r) :

$$\Delta P_r = f_r \times \frac{Lt_{evap}}{ID} \times \frac{v_r^{\,2}}{2g} \times \rho_r$$

$$\Delta P_r = 0{,}036 \times \frac{65}{0{,}0286} \times \frac{0{,}033^2}{2 \times 10} \times \frac{1280{,}3}{16{,}536}$$

$$\Delta P_r = 0{,}345\ \mathrm{kg/m^2} = 0{,}345\ \mathrm{Pa}$$

Thus, the refrigerant side pressure drop is ignored.

CHAPTER FOUR

AUXILIARY COMPONENTS OF AC SYSTEM AND MAINTAININGS

A. DUCT CONSTRUCTION

The function of a duct system is to transmit air from the air handling unit apparatus to the space to be conditioned. To fulfill this function in a practical manner, the system must be designed within the prescribed limits of available space, friction loss, velocity, sound level, heat and leakage losses and gains.

Duct designs are divided into two generic classes. Low-velocity systems have velocities below about 2500 ft/min (13 m/s), whereas high velocity systems have velocities up to 4500 ft/min (23 m/s). High-velocity systems can use smaller ducts if space is a problem, but fan power levels are higher. Low-velocity systems are use if fan-operating costs are to be lower and if adequate building space for larger ducts exits.

The most commonly used material for general HVAC ducts is galvanized steel sheet metal. In recent years, molded glass fiber ducts have also come into use. When air is being carried is corrosive, more corrosion resistant materials are used, such as stainless steel, copper, or aluminum. The details of recommended duct construction can be found in SMACNA publications. Glass fiber ducts are recommended for low-pressure systems. [66]

From Table 7, an initial velocity of 1400 FPM is selected considering AC of super executive bus is a commercial comfort air conditioning with friction rate 0,15 in. wg per 100 feet equivalent length, gained the total air quantity is 2300 CFM.[67]

Duct area (A_D) :

$$A_D = \frac{\text{total air quantity}}{\text{duct velocity}}$$

$$A_D = \frac{2300 \text{ CFM}}{1400 \text{ FPM}} \tag{4.107}$$

$$A_D = 1{,}643 \text{ ft}^2$$

CFM capacity (% CFM) :

$$\% \text{CFM} = \frac{\text{CFM of each terminals}}{\text{total air quantity}}$$

$$\% \text{CFM} = \frac{177}{2300} \tag{4.108}$$

$$\% \text{CFM} = 0{,}077 = 7{,}7\,\%$$

Friction loss (f_L) :

$$f_L = \text{total length} \times \text{friction rate}$$

$$f_L = \left(\frac{10{,}5}{0{,}3048}\right)\text{ft} \times \left(\frac{0{,}15}{100}\right)\text{in.wg} \tag{4.109}$$

$$f_L = 0{,}052 \text{ in.wg}$$

66) Edward G. Pita, op. cit., p. 216
67) Carrier, op. cit., p. 2-47

Using a 75% regain coefficient, velocity in last section is 200 fpm.[68]

$$\text{Regain} = 0{,}75 \times \left[\left(\frac{Va}{4000}\right)^2 - \left(\frac{Vd}{4000}\right)^2\right] \qquad (4.110)$$

where :

Va = initial velocity, fpm

Vd = duct velocity, fpm

$$\text{Regain} = 0{,}75 \times \left[\left(\frac{Va}{4000}\right)^2 - \left(\frac{Vd}{4000}\right)^2\right]$$

$$\text{Regain} = 0{,}75 \times \left[\left(\frac{1400}{4000}\right)^2 - \left(\frac{200}{4000}\right)^2\right]$$

$$\text{Regain} = 0{,}09 \text{ in.wg}$$

Therefore, the total static pressure at fan discharge (Ps_{tot}) :

$$Ps_{tot} = f_L + \text{friction rate} - \text{regain}$$

$$Ps_{tot} = (0{,}052 + 0{,}15) - 0{,}09 \qquad (4.111)$$

$$Ps_{tot} = 0{,}112 \text{ in.wg}$$

B. Maintenance Schedule

To ensure consistent high performance from air conditioning equipment, the following maintenance and service program must be followed.

68) Ibid, p. 2-48

b.1. Initial servicing

The initial services must follow the installation and commissioning of the system. The bus operator should carry out the initial 14 days service. Other service procedures should be approved by service center or dealer network.[69]

- After the first 14 days operation :
    1. Clean the return air filter.
    2. Check compressor belt tension and wear, and adjust if necessary.
    3. Check the visible oil or gas leaks on refrigeration hoses and fittings.
    4. Check the operation of all dash switches.
    5. Check the operation of the fan and blower motors.

- After the first 1.000 hours of operation or 100.000 km :
    1. Perform the 14 days services checks.
    2. Replace the filter drier.

b.2. Regular maintenance

The weekly service should de carried out by the coach operator, while the other services should be performed by authorized service center.

69) SRLT Bus and Coach Air Conditioner, p. 29

- Every week :

  1. Clean the return air filter.

- Every month :

  1. Clean the return air filter.
  2. Check the tension and wear of compressor belt.
  3. Check for visible oil or gas leaks on hoses and fittings.
  4. Check the operation of all dash switches, fan motors and blowers.
  5. Clean evaporator and condenser coils if necessary.

- Every six months :

  1. Perform monthly service checks.
  2. Check the unit for leaks.
  3. Check the unit gas charge and top up as necessary.
  4. Check all the hose and pipe joints and connections for tightness and leaks. Tighten where necessary.
  5. Check the operation of climate controller.
  6. Check the compressor oils level.
  7. Check the condition of the compressor pulley bearing. Clean and grease if necessary.

8. Check the operation of compressor belt tensioning device. Check compressor belts for wear and correct tension.
9. Check the tightness of all electrical connections and tighten as necessary.
10. Check the engine pulley center bolt and air conditioning pulley bolts for correct tension.
11. Check the record refrigeration operation.

- Every 12 months :
  1. Perform the six months service checks.
  2. Change the filter drier. Top up the system with refrigerant after repairing any leaks, as necessary.
  3. Remove and clean oil strainer at compressor oil inlet port.
  4. Check air gap on compressor clutch and adjust if necessary.
  5. Check brush length of condenser and evaporator motors. Replace brushes if necessary.
  6. Vacuum clean air inlet side of both evaporator coils to remove any dirt or lint built up.
  7. Back wash evaporator and condenser coils using low-pressure water. Take care not to damage the coil fins.

Table 4.7. Refrigeration Components

Compressor	
Condition :	**Possible causes :**
Compressor crank shaft seized.	Total compressor failure.
Excessive vibration from compressor.	Total compressor failure.
Compressor case is excessively hot.	Total compressor failure.
Compressor case is excessively cold.	Faulty TX valve.
Frosting present on suction hose.	Faulty TX valve (open).
Noise or vibration from compressor.	Partial compressor failure.
Compressor belt slipping.	① Incorrect adjustment of the tension device. ② Misalignment resulting from a fault in mounting bracket.
Compressor has shifted on the mounting bracket.	① Bracket mounts have failed due to stress. ② Bracket has deformed due to a stress.
TX Valve	
Normal operation : Liquid line to valve is warm and valve outlet to distributor tubes is cold. The valve body is half warm and half cold.	
Condition :	**Possible causes :**
Blocked inlet strainer	Faulty filter drier or sytem contamination.
TX valve is internally stuck open-inlet and outlet temperatures are both warm to touch.	System contamination.
TX valve is internally stuck closed-inlet is warm and outlet side is very cold. Suction pipe at TX sensor bulb is frosting.	System contamination.
Equaliser tube blocked from suction pipe.	System contamination.
TX valve is partially restricted-inlet side is slightly warm and outlet side is slightly cold. The suction pipe at TX sensor bulb maybe cool to warm.	System contamination.

High Pressure Relief Valve	
Condition :	**Possible causes :**
Valve opens before reaching at 420 psi or does not re-seat after opening.	Faulty pressure relief valve.
Filter Drier	
Normal operation : Similar warm temperature on the drier inlet and outllet connections.	
Condition :	**Possible causes :**
Different temperature on the drier inlet and outlet connections (eg. inlet is warm and outlet is cool to cold).	Faulty filter drier

Table 4.8. Air Flow Components

Ducting	
Air flow through the duct is impeded.	① Loose insulation within the ducting is causing a restriction. ② Intens within the duct (ie. roof structure, cables, piping and/or hoses) may be restricting air flows. ③ The vent system may incorrect or below the minimum requirement. ④ Excessive air leakage caused by poor duct sealing.
Filter	
Blokage in filters.	Dust and particle build-up.
Evaporator coils	
Blokage between coils fins.	Dust and particle build-up.

Motors	
Motor is not rotating while unit power is on.	① Power supply is excessively low or there is no power. ② Bearing has seized. ③ Armature has seized to casing or magnets. ④ Motor windings have been destroyed. ⑤ Brushes are worn out. ⑥ Commutator is worn or pitted.
Motor is noisy while operating.	① Bearings require replacement. ② Brushes have worn down and require replacement. ③ Commutator surface pitted or worn. ④ Fan blade or blower wheel damaged.
Motor speed is slow causing poor air flow.	① Worn bearing. ② Fan blade or blower wheel damaged. ③ Incorrect brush replacement. ④ Commutator is worn or pitted.
Motor shaft rotation direction is incorrect.	① Incorrect connection at motor. ② Incorrect connection at solid state relay.
Motor casing is hot.	① Bearing worn. ② Fan blade or blower wheel damaged. ③ Incorrect brush replacement. ④ Excessive load (check duct, filters and coil for restrictions).

Blower wheels/housing	
Loose blower wheel on motor shaft.	① Grub screw securing the blower wheel is loose. ② Incorrect replacement blower wheel.
Damaged blower wheel.	Incorrect alignment with housing, refer to maintenance section for correct alignment.
Damaged blower housing.	Excessive force placed on housing.
Evaporator blowers rotating incorrectl	① Blower wheel has been installed on the shaft in the wrong orientation (fins facing the wrong way). ② Motor shaft is rotating in the wrong direction (motor fault).

Table 4.9. Operational Components

Hoses and piping	
Hose leakages.	① Loose connections. ② Crack due to over-bending or kinking.
Pipe leakages.	① Corrosion due to acid attack or electrolysis may result in pipe cracking. ② Excessive force placed on the pipe or fittings may cause mechanical rupture. ③ Incorrect fitting tightness. ④ A defective pressure relief valve can lead to pipes bursting due to excessive pressure.

Flange seal	
Refrigerant leakage.	① Loose securing bolts. ② Leakage from double flange seal due to thickness of fibreglass base.
Drains	
Leakage from drain piping.	Loose connections or cracks in the drain hoses.
Unit sealing	
Water or air leakage from the rooftop unit.	Deterioration of seal due to heat or external chemical influence.
Compressor belt	
Belt wear.	Natural wear due to friction.
Incorrect belt tension.	① Incorrect adjustment of the tensioning device. ② Misalignment resulting from the mounting bracket fault.
Compressor mounting bracket	
Deformation or failure due to stress.	A compressor mounting bracket problem may relate back to an installation fault.
Clutch	
No operation.	① Check coil amperage and voltage at the connection using DC clamp-meter or similar. ② An above normal amperage rading may indicate a fault in the coil. ③ No amperage reading may indicate an open circuit in the winding. ④ A fault coil must be replaced.

Clutch slips or does not engage.	① Check the air gap with a feeler gauge. The correct air gap is 0,5 $\pm$ 0,1 mm. ② An incorrect air gap can cause erratic engagement and disengagement and clutch rattle/chatter. Adjust with shims/spacers if possible. ③ A fault may be caused by defective clutch pulley or front plate. If so, replace the defective item.
Unsual noise.	① Check the bearing. ② Remove the belt and rotate the clutch pulley by hand. Listen for suspicious bearing noise and feel if the bearing is 'sticking'. ③ A faulty bearing must be replaced.

CHAPTER FIVE

SUMMARY AND CONCLUSIONS

A. SUMMARY

The super executive bus is packaged in a rooftop unit by using R134a as the air conditioning unit refrigerant. It is carrying 26 passengers with 6,563 TR (ton refrigeration) cooling capacity. Conditions and calculations of main components :

a.1.1.Compressor

- Type = Bitzer 4NF reciprocating piston type
- Suction pressure (Ps) = 34 psia = 2,4 kg/cm^2
- Discharge pressure (Pd) = 160 psia = 11,3 kg/cm^2
- Theoretical power consumption for each compressor (W_t) = 7,64 kW
- Power requirement driven the compressor (W_d) = 9,37 kW
- Material of cylinder = high grade gray cast iron
- Bore of compressor (Bp) = 65 mm
- Length of cylinder (L_{cyl}) = 105 mm
- Material of piston = cast iron
- Piston diameter (Dp) = 64,8 mm
- Stroke of piston (L) = 52,32 mm
- Material of piston rings = alloyed with chromium and molybdenum

- Internal diameter of piston (ID_p) = 62,4 mm
- Depth of circular groove (H_{gr}) = 2,7 mm
- Width of circular groove (w_{gr}) = 3,1 mm
- Material of piston rod = hard, tough carbon steel
- Internal diameter of piston rod (ID_r) = 6 mm
- Outer diameter of piston rod (OD_r) = 10 mm
- Length of piston rod (L_r) = 46,98 mm
- Material of crosshead = carbon steel
- Length of crosshead (Lbg) = 150 mm
- Material of crankshaft = carbon steel
- Diameter of crankshaft (D_{cs}) = 32,8 mm
- Crank thickness (x_{cs}) = 19,68 mm
- Width of crank (b_{cs}) = 41 mm

a.1.2.Condenser

- Type = fin coils
- Material of tubes = copper
- Temperature of entering air (Ta_{in}) = 30°C
- Temperature of leaving air (Ta_{exit}) = 35°C
- Temperature of refrigerant (Tr) = 45°C

- Inner diameter of tube (ID) = 28,575 mm
- Outer diameter of tube (OD) = 34,925 mm
- Number of tubes (n_t) = 40 tubes
- Heat rejected (Q_{cond}) = 14 kW
- The flow rate of air (m_a) = 2,78 kg/s
- The volume flow rate of air (Qa) = 2,33 m^3/s
- Total outside area of tubes (Ao_t) = 4,2 m^2
- Length of condenser tubes (Lc_t) = 960 mm
- Total length of condenser tubes (Lc_{tot}) = 38,4 m
- Material of fins = aluminum fin coils
- Height of fins (h_f) = 20 mm
- Width of fins (w_f) = 20 mm
- Fins spacing (S_f) = 10 mm
- Number of fins (N_f) = 250 fins

a.1.3.Expansion Valve

- Type = externally equalized
- Material of tubes = steel
- Outer diameter of tube (OD_{TEV}) = 0,0272 m
- Inside diameter of tube (ID_{TEV}) = 0,0214 m

- Area of orifice (A_{ori}) = 0,000064 m^2
- Diameter of orifice (D_{ori}) = 0,009 m

a.1.4.Evaporator

- Type = fin coils
- Material of tubes = copper
- Cooling capacity on evaporator (Q_e) = 3,28 TR = 11,55 kW
- Refrigerant temperature (Tr) = 4°C
- Temperature of outside surface tubes (T_{os}) = 35°C
- Temperature of inside surface of tubes (T_{is}) = 45°C
- Inner diameter of tube (ID) = 28,575 mm
- Outer diameter of tube (OD) = 34,925 mm
- Number of tubes (n_{te}) = 40 tubes
- Tubes spacing (S_e) = 32 mm
- Overall area of outer tubes (Ao_e) = 7,12 m^2
- Length of evaporator tubes (L_{evap}) = 1620 mm
- Total length of evaporator tubes (Lc_{tot}) = 65 m

Conditions and calculations of auxiliary components :

a.2.1.Condenser Fans and Motors

- Material of fans = aluminum
- Type of propeller = axial blade
- Number of fans (N_{cf}) = 2
- Number of blades (N_b) = 6
- Outer diameter of fan (OD_f) = 457 mm
- Inner diameter of fan (ID_f) = 152 mm
- Blade width (w_{fb}) = 102 mm
- Blade spacing (S_{fb}) = 239,2 mm
- Speed of motor (N_m) = 1750 rpm

a.2.2.Ducting

- Material of duct = glass fiber
- Total air quantity = 2300 CFM
- Initial velocity = 1400 FPM
- Duct area (A_D) = 1,643 ft^2

B. CONCLUSIONS

From the calculations and some assumptions, can be concluded that the executive bus air-conditioning system :

1. It is designed in a vapor compression cycle.
2. Cooling load depends on the construction and material of bus, temperature, location, occupants, time selection, and other means which gain heat.
3. The use of other refrigerant except R134a may result in damage to the AC unit.

By all means, the author seeks comments and suggestions from all of you considering this final assignment still beyond perfections.

Graceful appreciation is hereby extended to those who is generously contributed their knowledge herein, and so willingly contributed their experiences and times to this project. Hopefully, this book will benefit for those actively engaged in designing air conditioning systems.

REFERENCES

Arismunandar, W., Saito, H., 2002, *Penyegaran Udara*, Jakarta, sixth edition, PT. Pradnya Paramita.

Harjanto, G., 2002, *AC & Refrigeration*, Yogyakarta.

Jasjfi, E., Holman, J., P., 1994, *Perpindahan Kalor*, Jakarta, sixth edition, PT. Gelora Aksara Pratama.

Stoecker, W., F., Jones, J., W., 1982, *Refrigeration and Air Conditioning*, Singapore, McGraw-Hill Book Company.

Pita, E., G., 1981, *Air Conditioning Principles and Systems*, United State of America, John Wiley & Sons. Inc.

Carrier, W., H., 1911, *Handbook of Air Conditioning System Design*, McGraw-Hill Book Company Inc.

Arora, C., P., 1986, *Refrigeration and Air Conditioning*, New Delhi, third edition, McGraw-Hill Publishing Company Limited.

Chlumsky, V., 1965, *Reciprocating and Rotary Compressors*, Czechoslovakia, English edition, Publishers of Technical Literature.

-------, 1997, *SRLT and Coach Rooftop Air Conditioning System Service Guide*, Australia, English edition, Air International Transit Pte. Ltd.

APPENDIXES

LIST OF TABLES

Table 1. Transmission Factor from Windows

Kaca	Tanpa penutup	Dengan penutup dalam ruangan
Kaca biasa	0,95	
Kaca ganda		
- Kaca biasa	0,70	0,50
- Menyerap di luar	0,6	0,40
Kaca setengah cermin	0,4	-

Source : Wiranto Arismunandar, Penyegaran Udara, p. 43

Table 2. Heat Output from Motor Driven Equipment (Btu/hr)

Motor Horsepower	Location of Equipment with Respect to Air Stream or Conditioned Space		
	Motor and Driven Machine In	Motor Out Driven Machine In	Motor In Driven Machine Out
1/9	580	320	260
1/6	710	430	280
¼	1000	640	360
1/3	1290	850	440
½	1820	1280	540
¾	2680	1930	750
1	3220	2540	680
1-1/2	4770	3820	950
2	6380	5100	1280
3	9450	7650	1800
5	15600	12800	2800
7-1/2	22500	19100	3400
10	30000	25500	4500
15	44500	38200	6300
20	58500	51000	7500
25	72400	63600	8800

Source : Edward G. Pita, Air Conditioning Principles and Systems, p. 114

Table 3. Ventilation Requirements for Occupants

	Estimated Persons per 1000 ft^2 Floor Area	Required Ventilation, Air per Person	
		Minimum CFM	Recommended CFM
RESIDENTIAL			
Single unit dwellings			
- Living areas, bedrooms.	5	5	7-10
- Kitchens, baths, toilet rooms.	-	20	30-50
Multiple unit dwellings			
- Living areas, bedrooms.	7	5	7-10
- Kitchens, baths, toilet rooms.	-	20	30-50
COMMERCIAL			
Public rest rooms	100	15	20-25
Merchandising			
- Sales floors (basement and ground floors)	30	7	10-15
- Sales floor (upper floor)	20	7	10-15
Dining rooms	70	10	15-20
Kitchens	20	30	35
Cafeterias	100	30	35
Hotels, motels			
- Bedrooms	5	7	10-15
- Living rooms	20	10	15-20
- Baths, toilets	-	20	30-50
Beauty shops	50	25	30-35
Barber shops	25	7	10-15
Parking garages	-	1,5	2-3
Theatres			
- Lobbies	150	20	25-30
- Auditoriums (no smoking)	150	5	5-10
- Auditoriums (smoking permitted)	150	10	10-20
Bowling alleys (seating area)	70	15	20-25
Gymnasiums and arenas			
- Playing floors	70	20	25-30
- Locker rooms	20	30	40-50
- Spectator areas	150	20	25-30
Swimming pools	25	15	20-25
Offices			
- General office space	10	15	15-25
- Conference rooms	60	25	30-40
INSTITUTIONAL			
Schools			
- Classrooms	50	10	10-15
- Auditoriums	150	5	5-7,5
- Gymnasiums	70	20	25-30
- Libraries	20	7	10-12
- Locker rooms	20	30	40-50

Hospitals			
- Single, dual bedrooms	15	10	15-20
- Wards	20	10	15-20
- Operating rooms, delivery rooms	-	20	-

Source : Edward G. Pita, Air Conditioning Principles and Systems, p. 118

Table 4. Shading Coefficients

Type of Glass	Thickness, mm	Shading Coefficients				
		No Indoor Shading	Venetian Blinds		Roller Shades	
			Medium	Light	Dark	Light
Single glass						
- Regular sheet	3	1,00	0,64	0,55	0,59	0,25
- Plate	6-12	0,95	0,64	0,55	0,59	0,25
- Heat absorbing	6	0,70	0,57	0,53	0,40	0,30
	10	0,50	0,54	0,52	0,40	0,28
Double glass						
- Regular sheet	3	0,90	0,57	0,51	0,60	0,25
- Plate	6	0,83	0,57	0,51	0,60	0,25
- Reflective	6	0,2-0,4	0,2-0,33	-	-	-

Source : Stoecker, Refrigeration and Air Conditioning, p. 76

Table 5. Space per Occupants

Type of Space	Occupancy
Residence	2-6 occupants
Office	10-15 m^2 per occupant
Retail	3-5 m^2 per occupant
School	2,5 m^2 per occupant
Auditorium	1,0 m^2 per occupant

Source : Stoecker, Refrigeration and Air Conditioning, p. 73

Table 6. Cooling Load Temperature for Calculating Load from Flat Roofs

escription of onstruction	Weight lb/ft²	U-value Btu/(h·ft²·°F)	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
			Without Suspended Ceiling																							
sheet with 1-in. -in.) insulation	7 (8)	0.213 (0.124)	1	2	-3	-3	-5	-3	6	19	31	49	61	71	78	[illegible]	77	[illegible]	[illegible]	[illegible]	30	18	12	8	5	3
wood with 1-in. ation	8	0.170	6	3	0	-1	-3	-3	-2	4	14	27	39	[illegible]	62	70	74	74	70	62	51	38	28	29	14	9
l.w. concrete	18	0.213	9	5	2	0	-2	-3	-3	1	9	20	32	44	55	64	70	73	71	66	57	45	34	25	18	13
h.w. concrete with 1-in. -in.) insulation	29	0.206 (0.122)	12	8	5	3	0	-1	-1	3	11	20	30	41	51	59	65	66	66	62	54	45	36	29	22	17
wood with 2-in. ation	19	0.109	3	0	-3	-4	-5	-7	-6	-3	5	16	27	39	49	57	61	64	52	57	48	37	26	18	11	7
l.w. concrete	24	0.158	22	17	13	9	6	3	1	1	3	7	15	23	33	43	51	58	62	64	62	57	50	42	35	28
n. wood with 1-in. lation	13	0.130	29	24	20	16	13	10	7	6	6	9	13	20	27	34	42	48	53	55	56	54	49	44	39	34
l.w. concrete	31	0.126	35	30	26	22	18	14	11	9	7	7	9	13	19	25	33	39	46	50	53	54	53	49	45	40
h.w. concrete with 1-in. -in.) insulation	52 (52)	0.200 (0.120)	25	22	18	15	12	9	8	8	10	14	20	[illegible]	33	40	46	50	53	53	52	48	43	38	34	30
n. wood with 2-in. lation	13	0.093	[illegible]	26	23	19	16	13	10	9	8	9	13	17	23	29	36	41	46	49	51	[illegible]	[illegible]	43	39	35
f terrace system	75	0.106	34	31	28	25	22	19	16	14	13	13	15	18	22	26	31	36	40	44	45	46	45	43	40	37
h.w. concrete with 1-in. 2-in.) insulation	75 (75)	0.192 (0.117)	31	28	25	22	20	17	15	14	14	16	18	22	26	31	36	40	43	45	45	44	42	40	37	34
wood with 1-in. 2-in.) insulation	17 (18)	0.106 (0.078)	38	36	33	30	28	25	22	20	18	17	16	17	18	21	24	28	32	36	39	41	43	43	42	40
			With Suspended Ceiling																							
l Sheet with 1-in. 2-in.) insulation	9 (10)	0.134 (0.092)	2	0	-2	-3	-4	-4	-1	9	23	37	50	62	71	77	78	74	67	56	42	28	18	12	8	5
. wood with 1-in. lation	10	0.115	20	15	11	8	5	3	2	3	7	13	21	30	40	48	55	60	62	61	58	51	44	37	30	25
. l.w. concrete	20	0.134	19	14	10	7	4	2	0	0	4	10	19	29	39	48	56	62	65	64	61	54	46	38	30	24
. h.w. concrete with 1-in. lation	30	0.131	28	25	23	20	17	15	13	13	14	16	20	25	30	35	39	43	46	47	46	44	41	38	35	32
. wood with 2-in. lation	10	0.083	25	20	16	13	10	7	5	5	7	12	18	25	33	41	48	53	57	57	56	52	46	40	34	29
l.w. concrete	26	0.109	32	28	23	19	16	13	10	8	7	8	11	16	22	29	36	42	48	52	54	54	51	47	42	37
n. wood with 1-in. lation	15	0.096	34	31	29	26	23	21	18	16	15	15	16	18	21	25	30	34	38	41	43	44	44	42	40	37
. l.w. concrete	33	0.093	39	36	33	29	26	23	20	18	15	14	14	15	17	20	25	29	34	38	42	45	46	45	44	42
. h.w. concrete with . (or 2-in.) insulation	53 (54)	0.128 (0.090)	30	29	27	26	24	22	21	20	20	21	22	24	27	29	32	34	36	38	38	38	37	36	34	33
in. wood with 2-in. lation	15	0.072	35	33	30	28	26	24	22	20	18	18	18	20	22	25	28	32	35	38	40	41	41	40	39	37
of terrace system	77	0.082	30	29	28	27	26	25	24	23	22	22	22	23	23	25	26	28	29	31	32	33	33	33	33	32
. h.w. concrete with 1-in. 2 in.) insulation	77 (77)	0.125 (0.088)	29	28	27	26	25	24	23	22	21	21	22	23	25	26	28	30	32	33	34	34	34	33	32	31
. wood with 1-in 2-in.) insulation	19 (20)	0.082 (0.064)	35	34	33	32	31	29	27	26	24	23	22	21	22	22	24	25	27	30	32	34	35	36	37	36

Source : Edward G. Pita, Air Conditioning Principles and Systems, p. 97

Table 7. Sensible Heat Cooling Load for People

Total Hours in Space	Hours after Each Entry Into Space																							
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
2	0.49	0.58	0.17	0.13	0.10	0.08	0.07	0.06	0.05	0.04	0.04	0.03	0.03	0.02	0.02	0.02	0.02	0.01	0.01	0.01	0.01	0.01	0.01	0.01
4	0.49	0.59	0.66	0.71	0.27	0.21	0.16	0.14	0.11	0.10	0.08	0.07	0.06	0.06	0.05	0.04	0.04	0.03	0.03	0.03	0.02	0.02	0.02	0.01
6	0.50	0.60	0.67	0.72	0.76	0.79	0.34	0.26	0.21	0.18	0.15	0.13	0.11	0.10	0.08	0.07	0.06	0.06	0.05	0.04	0.04	0.03	0.03	0.03
8	0.51	0.61	0.67	0.72	0.76	0.80	0.82	0.84	0.38	0.30	0.25	0.21	0.18	0.15	0.13	0.12	0.10	0.09	0.08	0.07	0.06	0.05	0.05	0.04
10	0.53	0.62	0.69	0.74	0.77	0.80	0.83	0.85	0.87	0.89	0.42	0.34	0.28	0.23	0.20	0.17	0.15	0.13	0.11	0.10	0.09	0.08	0.07	0.06
12	0.55	0.64	0.70	0.75	0.79	0.81	0.84	0.86	0.88	0.89	0.91	0.92	0.45	0.36	0.30	0.25	0.21	0.19	0.16	0.14	0.12	0.11	0.09	0.08
14	0.58	0.66	0.72	0.77	0.80	0.83	0.85	0.87	0.89	0.90	0.91	0.92	0.93	0.94	0.47	0.38	0.31	0.26	0.23	0.20	0.17	0.15	0.13	0.11
16	0.62	0.70	0.75	0.79	0.82	0.85	0.87	0.88	0.90	0.91	0.92	0.93	0.94	0.95	0.95	0.96	0.49	0.39	0.33	0.28	0.24	0.20	0.18	0.16
18	0.66	0.74	0.79	0.82	0.85	0.87	0.89	0.90	0.92	0.93	0.94	0.94	0.95	0.96	0.96	0.97	0.97	0.97	0.50	0.40	0.33	0.28	0.24	0.21

Source : Edward G. Pita, Air Conditioning Principles and Systems, p. 97

Table 8. Heat Gain from People

DEGREE OF ACTIVITY	TYPICAL APPLICATION	Metabolic Rate (Adult Male) Btu/hr	Average Adjusted Metabolic Rate* Btu/hr	ROOM DRY-BULB TEMPERATURE									
				82 F		80 F		78 F		75 F		70 F	
				Btu/hr		Btu/hr		Btu/hr		Btu/hr		Btu/hr	
				Sensible	Latent	Sensible	Latent	Sensible	Latent	Sensible	Latent	Sensible	Latent
Seated at rest	Theater, Grade School	390	350	175	175	195	155	210	140	230	120	260	90
Seated, very light work	High School	450	400	180	220	195	205	215	185	240	160	275	175
Office worker	Offices, Hotels, Apts., College	475	450	180	270	200	250	215	235	245	205	285	165
Standing, walking slowly	Dept., Retail, or Variety Store	550											
Walking, seated	Drug Store	550	500	180	320	200	300	220	280	255	245	290	210
Standing, walking slowly	Bank	550											
Sedentary work	Restaurant†	500	550	190	360	220	330	240	310	280	270	320	230
Light bench work	Factory, light work	800	750	190	560	220	530	245	505	295	455	365	385
Moderate dancing	Dance Hall	900	850	220	630	245	605	275	575	325	525	400	450
Walking, 3 mph	Factory, fairly heavy work	1000	1000	270	730	300	700	330	670	380	620	460	540
Heavy work	Bowling Alley‡, Factory	1500	1450	450	1000	465	985	485	965	525	925	605	845

*Adjusted Metabolic Rate is the metabolic rate to be applied to a mixed group of people with a typical percent composition based on the following factors:
Metabolic rate, adult female = Metabolic rate, adult male × 0.85
Metabolic rate, children = Metabolic rate, adult male × 0.75

†Restaurant—Values for this application include 60 Btu per hr for food per individual (30 Btu sensible and 30 Btu latent heat per hr).

‡Bowling—Assume one person per alley actually bowling and all others sitting, metabolic rate 400 Btu per hr; or standing, 550 Btu per hr.

Source : Carrier, Handbook of Air Conditioning System Design, p. 1-100

Table 9. Heat Gain from Lights

TYPE	HEAT GAIN* Btu/hr
Fluorescent	Total Light Watts × 1.25† × 3.4
Incandescent	Total Light Watts × 3.4

Source : Carrier, Handbook of Air Conditioning System Design, p. 1-101

Table 10. Apparatus Dewpoints

ROOM CONDITIONS DB (F)	RH (%)	WB (F)	W (gr/lb)	EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*									
90	20	62.7	42.0	ESHF	1.00	.96	.92	.90	.88	.86	.84	.82	.81
				ADP	43.5	41	39	37	35	32	29	24	22
	25	65.1	52.7	ESHF	1.00	.96	.92	.88	.84	.82	.80	.78	.75
				ADP	49.6	48	46	44	41	39	36	32	22
	30	67.3	63.6	ESHF	1.00	.92	.87	.83	.80	.76	.74	.72	.70
				ADP	54.5	52	50	48	46	42	38	34	24
	35	69.3	74.2	ESHF	1.00	.92	.85	.81	.76	.73	.71	.69	.66
				ADP	58.8	57	55	53	50	48	45	42	33
	40	71.2	84.8	ESHF	1.00	.92	.83	.78	.74	.69	.66	.63	.62
				ADP	62.4	61	59	57	55	52	48	44	40
	45	73.0	95.5	ESHF	1.00	.92	.82	.76	.70	.66	.62	.60	.58
				ADP	65.8	65	63	61	59	56	52	49	43
	50	74.9	106.4	ESHF	1.00	.92	.78	.68	.64	.60	.58	.56	.54
				ADP	68.9	68	66	63	61	58	56	53	47
	55	76.7	117.5	ESHF	1.00	.92	.76	.68	.64	.57	.54	.52	.50
				ADP	71.6	71	69	67	66	62	59	57	50
	60	78.4	128.4	ESHF	1.00	.86	.68	.60	.56	.52	.50	.48	.46
				ADP	74.2	73	71	69	67	64	62	59	50
	65	80.0	139.6	ESHF	1.00	.75	.68	.62	.55	.50	.47	.45	.43
				ADP	76.8	75	74	73	71	69	66	64	59
	70	81.6	151.0	ESHF	1.00	.78	.66	.60	.52	.47	.43	.41	.39
				ADP	79.0	76	77	76	74	72	69	66	58
85	20	59.6	35.8	ESHF	1.00	.98	.95	.92	.90	.88	.87	.86	.84
				ADP	39.4	38	36	34	32	30	28	26	22
	25	61.7	44.8	ESHF	1.00	.98	.93	.90	.86	.84	.82	.80	.78
				ADP	45.2	44	42	40	37	35	32	28	20
	30	63.7	54.1	ESHF	1.00	.94	.89	.85	.81	.79	.77	.75	.73
				ADP	50.2	48	46	44	40	38	35	31	22
	35	65.5	62.9	ESHF	1.00	.92	.86	.82	.78	.74	.72	.70	.69
				ADP	54.1	52	50	48	45	41	38	32	27
	40	67.4	71.7	ESHF	1.00	.92	.84	.79	.76	.73	.69	.67	.65
				ADP	57.9	56	54	52	50	48	44	40	32
	45	69.1	81.1	ESHF	1.00	.92	.83	.77	.72	.68	.64	.62	.61
				ADP	61.2	60	58	56	54	51	46	41	36
	50	70.8	90.1	ESHF	1.00	.92	.80	.73	.68	.64	.61	.59	.57
				ADP	64.2	63	61	59	57	54	51	48	39
	55	72.3	99.4	ESHF	1.00	.92	.83	.73	.67	.60	.57	.56	.54
				ADP	66.9	66	65	63	61	57	54	52	47
	60	73.9	108.8	ESHF	1.00	.92	.76	.67	.61	.56	.54	.52	.50
				ADP	69.5	69	67	65	63	60	58	55	49
	65	75.5	118.2	ESHF	1.00	.88	.69	.61	.56	.53	.50	.48	.47
				ADP	71.9	71	69	67	65	63	61	58	54
	70	77.0	127.6	ESHF	1.00	.81	.63	.55	.51	.49	.47	.45	.43
				ADP	74.0	73	71	69	67	66	64	62	55
82	35	63.2	57.0	ESHF	1.00	.92	.88	.84	.80	.76	.74	.72	.71
				ADP	51.6	49	48	46	43	39	36	31	27
	40	65.0	65.1	ESHF	1.00	.90	.87	.82	.78	.74	.71	.69	.67
				ADP	55.2	53	52	50	48	45	41	38	31
	45	66.7	73.5	ESHF	1.00	.91	.87	.80	.75	.72	.68	.65	.63
				ADP	58.5	57	56	54	52	50	46	41	53
	50	68.3	81.9	ESHF	1.00	.90	.80	.74	.70	.64	.62	.60	.59
				ADP	61.5	60	58	56	54	50	47	42	37
	55	69.8	90.2	ESHF	1.00	.90	.83	.74	.68	.64	.61	.58	.56
				ADP	64.2	63	62	60	58	56	54	50	44
	60	71.3	98.5	ESHF	1.00	.92	.76	.68	.63	.59	.56	.54	.52
				ADP	66.7	66	64	62	60	58	55	52	44
	65	72.8	107.0	ESHF	1.00	.86	.71	.63	.58	.54	.52	.51	.49
				ADP	69.1	68	66	64	62	60	58	56	51
	70	74.2	115.5	ESHF	1.00	.80	.71	.65	.60	.54	.51	.48	.46
				ADP	71.2	70	69	68	67	65	63	60	56
81	35	62.5	55.2	ESHF	1.00	.94	.89	.84	.81	.77	.75	.73	.71
				ADP	50.8	49	47	45	43	39	36	32	21
	40	64.2	63.2	ESHF	1.00	.94	.87	.82	.78	.75	.72	.69	.67
				ADP	54.4	53	51	49	47	45	41	36	23
	45	65.9	71.2	ESHF	1.00	.96	.91	.83	.78	.74	.70	.67	.64
				ADP	57.6	57	56	54	52	50	47	43	36
	50	67.5	79.0	ESHF	1.00	.90	.84	.80	.74	.70	.66	.62	.60
				ADP	60.5	59	58	57	55	53	50	45	38
	55	69.0	87.4	ESHF	1.00	.90	.77	.71	.66	.62	.60	.58	.56
				ADP	63.2	62	60	58	56	53	51	47	35
	60	70.5	95.4	ESHF	1.00	.92	.77	.68	.63	.59	.56	.54	.53
				ADP	65.8	65	63	61	59	56	53	50	46
	65	71.9	103.7	ESHF	1.00	.85	.76	.71	.66	.60	.56	.52	.50
				ADP	68.2	67	66	65	64	62	60	56	52
	70	73.3	111.9	ESHF	1.00	.80	.71	.61	.55	.52	.48	.47	.46
				ADP	70.3	69	68	66	64	62	58	56	52
80	20	56.4	30.4	ESHF	1.00	.98	.95	.93	.91	.89	.88	.87	.86
				ADP	35.4	34	32	30	28	26	24	22	20
	25	58.3	38.0	ESHF	1.00	.96	.93	.90	.88	.86	.84	.82	.81
				ADP	40.9	39	37	35	33	31	28	24	21
	30	60.0	45.6	ESHF	1.00	.96	.91	.88	.85	.83	.80	.78	.76
				ADP	45.7	44	42	40	38	36	32	28	21
	35	61.8	53.5	ESHF	1.00	.94	.88	.85	.82	.79	.77	.73	.72
				ADP	49.8	48	46	44	42	40	37	29	24
	40	63.5	61.2	ESHF	1.00	.94	.90	.84	.80	.76	.73	.70	[illegible]
				ADP	53.5	52	51	49	47	44	41	36	[illegible]
	45	65.1	68.9	ESHF	1.00	.96	.87	.81	.76	.73	.70	.67	[illegible]
				ADP	56.8	56	54	52	50	48	45	41	[illegible]
	50	66.7	76.7	ESHF	1.00	.89	.80	.74	.70	.66	.64	.62	.61
				ADP	59.7	58	56	54	52	49	46	42	[illegible]
	55	68.2	84.6	ESHF	1.00	.89	.82	.74	.69	.65	.61	.59	[illegible]
				ADP	62.3	61	60	58	56	54	50	47	[illegible]
	60	69.6	92.2	ESHF	1.00	.91	.83	.72	.66	.62	.59	.57	[illegible]
				ADP	64.8	64	63	61	59	57	55	53	[illegible]
	65	71.1	100.4	ESHF	1.00	.85	.76	.71	.63	.58	.55	.52	[illegible]
				ADP	67.2	66	65	64	62	60	58	54	[illegible]
	70	72.4	108.2	ESHF	1.00	.78	.71	.65	.61	.55	.52	.49	[illegible]
				ADP	69.4	68	67	66	65	63	61	58	[illegible]

Source : Carrier, Handbook of Air Conditioning System Design, p. 1-145

Table 10. Apparatus Dewpoints (continued)

DB (F)	RH (%)	WB (F)	W (gr/lb)		Effective Sensible Heat Factor and Apparatus Dewpoint*								
79	35	61.0	51.5	ESHF	1.00	.96	.91	.89	.85	.82	.78	.75	.73
				ADP	48.9	48	46	45	43	41	37	32	26
	40	62.7	59.2	ESHF	1.00	.97	.90	.84	.80	.76	.74	.71	.69
				ADP	52.7	52	50	48	46	43	41	36	29
	45	64.3	66.7	ESHF	1.00	.91	.83	.78	.75	.72	.70	.67	.65
				ADP	55.9	54	52	50	48	46	44	39	32
	50	65.9	74.2	ESHF	1.00	.89	.80	.75	.71	.68	.66	.63	.61
				ADP	58.9	57	55	53	51	49	47	42	33
	55	67.4	81.9	ESHF	1.00	.96	.82	.74	.69	.66	.63	.60	.58
				ADP	61.4	61	59	57	55	53	51	47	41
	60	68.8	89.3	ESHF	1.00	.90	.76	.69	.64	.61	.57	.55	.54
				ADP	63.9	63	61	59	57	55	51	47	41
	65	70.2	97.0	ESHF	1.00	.84	.71	.64	.59	.56	.54	.52	.51
				ADP	66.3	65	63	61	59	57	55	51	48
	70	71.6	104.8	ESHF	1.00	.81	.71	.65	.58	.54	.52	.50	.48
				ADP	68.5	67	66	65	63	61	59	57	53
78	35	60.3	50.0	ESHF	1.00	.96	.91	.87	.83	.79	.77	.75	.73
				ADP	48.2	47	45	43	41	37	35	31	22
	40	61.9	57.3	ESHF	1.00	.93	.87	.82	.79	.77	.73	.71	.69
				ADP	51.7	50	48	46	44	42	38	34	25
	45	63.5	64.6	ESHF	1.00	.95	.86	.81	.76	.74	.70	.68	.66
				ADP	55.0	54	52	50	48	46	42	39	34
	50	65.0	71.9	ESHF	1.00	.94	.83	.76	.73	.70	.67	.64	.62
				ADP	57.9	57	55	53	51	49	47	42	36
	55	66.6	79.2	ESHF	1.00	.96	.83	.75	.70	.65	.62	.60	.59
				ADP	60.5	60	58	56	54	51	48	44	41
	60	67.9	86.4	ESHF	1.00	.90	.82	.76	.69	.64	.60	.57	.55
				ADP	63.0	62	61	60	58	56	53	49	42
	65	69.3	93.8	ESHF	1.00	.85	.77	.71	.67	.62	.58	.54	.52
				ADP	65.2	64	63	62	61	59	57	53	48
	70	70.6	101.2	ESHF	1.00	.71	.66	.62	.59	.55	.52	.50	.48
				ADP	67.5	65	64	63	62	60	58	55	48
77	35	59.6	48.3	ESHF	1.00	.96	.91	.87	.83	.79	.77	.75	.74
				ADP	47.3	46	44	42	40	36	33	28	24
	40	61.2	55.5	ESHF	1.00	.96	.89	.84	.81	.78	.76	.73	.70
				ADP	50.9	50	48	46	44	42	40	36	27
	45	62.7	62.4	ESHF	1.00	.94	.86	.81	.77	.74	.72	.69	.66
				ADP	54.1	53	51	49	47	45	43	39	27
	50	64.2	69.7	ESHF	1.00	.94	.84	.77	.73	.70	.68	.65	.63
				ADP	57.0	56	54	52	50	48	46	42	37
	55	65.6	76.6	ESHF	1.00	.95	.83	.75	.70	.67	.63	.61	.59
				ADP	59.6	59	57	55	53	51	48	44	37
	60	6[illegible].1	83.6	ESHF	1.00	.89	.82	.77	.73	.67	.62	.58	.56
				ADP	62.0	61	60	59	58	56	53	48	43
	65	68.5	90.8	ESHF	1.00	.84	.72	.64	.60	.57	.55	.54	.53
				ADP	64.4	63	61	59	57	55	53	51	48
	70	69.8	97.9	ESHF	1.00	.79	.66	.60	.55	.53	.51	.50	.49
				ADP	66.5	65	63	61	59	57	55	53	49

DB (F)	RH (%)	WB (F)	W (gr/lb)		Effective Sensible Heat Factor and Apparatus Dewpoint*								
76	35	58.9	46.7	ESHF	1.00	.96	.91	.87	.84	.81	.79	.77	.74
				ADP	46.3	45	43	41	39	37	34	31	21
	40	60.4	53.7	ESHF	1.00	.96	.89	.84	.81	.78	.76	.72	.70
				ADP	49.9	49	47	45	43	41	39	32	22
	45	61.9	60.4	ESHF	1.00	.94	.86	.81	.77	.74	.71	.69	.67
				ADP	53.2	52	50	48	46	44	40	37	31
	50	63.4	67.4	ESHF	1.00	.93	.83	.77	.73	.69	.67	.65	.63
				ADP	56.2	55	53	51	49	46	43	40	32
	55	64.9	74.0	ESHF	1.00	.94	.82	.75	.70	.67	.65	.62	.60
				ADP	58.7	58	56	54	52	50	48	44	38
	60	66.2	80.9	ESHF	1.00	.90	.77	.70	.66	.62	.60	.58	.57
				ADP	61.1	60	58	56	54	52	49	46	43
	65	67.6	87.6	ESHF	1.00	.84	.72	.65	.61	.58	.56	.54	.53
				ADP	63.4	62	60	58	56	54	52	48	43
	70	68.9	94.6	ESHF	1.00	.80	.67	.60	.56	.54	.52	.51	.50
				ADP	65.5	64	62	60	58	56	54	52	49
75	20	53.2	25.7	ESHF	1.00	.98	.96	.94	.92	.90	.89		
				ADP	31.5	30	28	26	24	22	20		
	25	54.8	32.1	ESHF	1.00	.95	.92	.90	.88	.86	.84		
				ADP	36.9	34	32	30	28	25	21		
	30	56.5	38.5	ESHF	1.00	.97	.93	.90	.87	.85	.82	.80	.79
				ADP	41.4	40	38	36	34	32	28	24	20
	35	58.1	45.2	ESHF	1.00	.96	.91	.87	.84	.80	.78	.76	.75
				ADP	45.5	44	42	40	38	34	31	27	22
	40	59.6	51.8	ESHF	1.00	.96	.89	.84	.81	.79	.76	.73	.71
				ADP	49.1	48	46	44	42	40	37	32	24
	45	61.1	58.2	ESHF	1.00	.94	.87	.81	.77	.75	.72	.69	.67
				ADP	52.2	51	49	47	45	43	40	35	21
	50	62.6	65.0	ESHF	1.00	.92	.84	.78	.74	.71	.69	.66	.64
				ADP	55.2	54	52	50	48	46	44	40	34
	55	64.0	71.5	ESHF	1.00	.94	.87	.78	.73	.69	.65	.63	.61
				ADP	57.8	57	56	54	52	50	47	44	39
	60	65.3	77.9	ESHF	1.00	.90	.77	.71	.66	.63	.61	.59	.58
				ADP	60.1	59	57	55	53	51	49	46	43
	65	66.7	84.8	ESHF	1.00	.84	.72	.65	.61	.59	.57	.55	.54
				ADP	62.4	61	59	57	55	53	51	48	44
	70	68.0	91.2	ESHF	1.00	.80	.73	.68	.61	.57	.54	.52	.51
				ADP	64.5	63	62	61	59	57	55	52	49
72	35	55.9	40.8	ESHF	1.00	.98	.93	.89	.86	.83	.81	.79	.77
				ADP	42.8	42	40	38	36	34	31	28	22
	40	57.3	46.7	ESHF	1.00	.95	.92	.87	.84	.81	.77	.75	.73
				ADP	46.3	45	44	42	40	38	34	30	23
	45	58.7	52.7	ESHF	1.00	.94	.87	.82	.79	.76	.74	.71	.69
				ADP	49.5	48	46	44	42	40	38	32	22
	50	60.1	58.8	ESHF	1.00	.92	.38	.81	.77	.73	.70	.68	.66
				ADP	52.4	51	50	48	46	43	40	37	30
	55	61.4	64.4	ESHF	1.00	.93	.83	.77	.72	.68	.66	.64	.63
				ADP	54.9	54	52	50	48	45	42	39	36
	60	62.7	70.2	ESHF	1.00	.89	.79	.72	.68	.65	.63	.61	.60
				ADP	57.3	56	54	52	50	48	46	42	39

Source : Carrier, Handbook of Air Conditioning System Design, p. 1-146

Table 10. Apparatus Dewpoints (continued)

DB (F)	RH (%)	WB (F)	W (gr/lb)		EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*								
72	65	64.0	76.3	ESHF	1.00	.84	.73	.67	.63	.61	.59	.58	
				ADP	59.5	58	56	54	52	50	48	47	
	70	65.2	82.3	ESHF	1.00	.80	.69	.62	.59	.56	.54	.53	
				ADP	61.6	60	58	56	54	51	48	44	
70	20	49.9	21.6	ESHF	1.00	.98	.96	.94	.93				
				ADP	27.6	26	24	22	21				
	25	51.5	27.0	ESHF	1.00	.97	.94	.92	.90	.88			
				ADP	33.7	31	29	27	25	22			
	30	53.0	32.8	ESHF	1.00	.98	.94	.91	.88	.86	.84	.82	
				ADP	37.1	36	34	32	30	27	25	20	
	35	54.4	38.0	ESHF	1.00	.97	.93	.89	.86	.84	.82	.80	.78
				ADP	41.1	40	38	36	34	32	30	27	22
	40	55.8	43.5	ESHF	1.00	.95	.90	.86	.83	.80	.78	.76	.74
				ADP	44.5	43	41	39	37	35	32	29	22
	45	57.1	49.1	ESHF	1.00	.93	.87	.82	.79	.77	.75	.73	.71
				ADP	47.7	46	44	42	40	38	36	33	27
	50	58.5	54.8	ESHF	1.00	.92	.84	.80	.76	.74	.71	.69	.67
				ADP	50.5	49	47	45	43	41	38	35	25
	55	59.7	60.1	ESHF	1.00	.93	.83	.77	.73	.71	.68	.66	.64
				ADP	53.1	52	50	48	46	44	42	38	32
	60	60.9	65.5	ESHF	1.00	.89	.79	.73	.69	.66	.64	.62	.61
				ADP	55.4	54	52	50	48	46	43	40	36
	65	62.2	71.1	ESHF	1.00	.93	.78	.71	.66	.63	.61	.59	.58
				ADP	57.7	57	55	53	51	49	47	44	40
	70	63.4	76.9	ESHF	1.00	.90	.74	.66	.61	.59	.57	.56	.55
				ADP	59.8	59	57	55	53	51	49	47	45
	75	64.5	82.5	ESHF	1.00	.88	.70	.62	.57	.55	.53	.52	.51
				ADP	61.7	61	59	57	55	53	51	49	44
	80	65.7	88.0	ESHF	1.00	.87	.73	.65	.60	.54	.51	.49	.48
				ADP	63.5	63	62	61	60	58	56	53	49
	85	66.8	93.7	ESHF	1.00	.71	.56	.52	.50	.48	.47	.46	.45
				ADP	65.3	64	62	61	60	59	58	57	54
	90	67.9	99.3	ESHF	1.00	.66	.56	.50	.47	.45	.43	.42	.41
				ADP	66.9	66	65	64	63	62	61	60	56
	95	69.0	105.0	ESHF	1.00	.60	.47	.42	.39	.38	.37		
				ADP	68.5	68	67	66	65	64	62		
65	60	56.6	55.0	ESHF	1.00	.95	.84	.77	.73	.70	.68	.66	.65
				ADP	50.6	50	48	46	44	42	39	36	34
	65	57.7	59.7	ESHF	1.00	.92	.85	.80	.73	.69	.66	.64	.62
				ADP	52.9	52	51	50	48	46	44	41	37
	70	58.9	64.5	ESHF	1.00	.89	.80	.76	.69	.65	.62	.60	.58
				ADP	55.0	54	53	52	50	48	46	43	37
	75	59.9	69.2	ESHF	1.00	.88	.78	.72	.65	.61	.58	.56	.55
				ADP	56.9	56	55	54	52	50	48	45	41
	80	51.0	73.8	ESHF	1.00	.75	.68	.63	.60	.58	.55	.53	.52
				ADP	58.7	57	56	55	54	53	51	48	46
	85	62.0	78.6	ESHF	1.00	.71	.63	.58	.55	.52	.50	.49	
				ADP	60.3	59	58	57	56	54	52	50	
	90	63.0	83.2	ESHF	1.00	.70	.58	.53	.50	.48	.46	.45	
				ADP	61.9	61	60	59	58	57	55	53	
	95	64.0	88.0	ESHF	1.00	.69	.51	.46	.43	.42	.41		
				ADP	63.5	63	62	61	60	59	58		
60	60	52.3	46.2	ESHF	1.00	.94	.89	.81	.77	.74	.72	.70	.68
				ADP	46.0	45	44	42	40	38	36	34	28
	65	53.3	50.0	ESHF	1.00	.91	.86	.78	.74	.70	.69	.67	.65
				ADP	48.1	47	46	44	42	40	39	36	31
	70	54.3	53.9	ESHF	1.00	.89	.83	.74	.70	.67	.65	.63	.62
				ADP	50.1	49	48	46	44	42	40	37	34
	75	55.3	57.8	ESHF	1.00	.79	.74	.71	.68	.64	.62	.60	.59
				ADP	52.0	50	49	48	47	45	43	40	37
	80	56.3	61.7	ESHF	1.00	.85	.76	.70	.66	.61	.59	.57	.56
				ADP	53.8	53	52	51	50	48	46	44	41
	85	57.2	65.5	ESHF	1.00	.75	.67	.63	.57	.56	.54	.53	
				ADP	55.4	54	53	52	50	49	47	45	
	90	58.2	69.4	ESHF	1.00	.72	.62	.57	.54	.52	.50	.49	
				ADP	57.0	56	55	54	53	52	50	47	
	95	59.1	73.5	ESHF	1.00	.69	.55	.49	.47	.46	.45		
				ADP	58.5	58	57	56	55	54	52		
55	60	47.9	38.4	ESHF	1.00	.93	.89	.85	.80	.77	.75	.73	.71
				ADP	41.3	40	39	38	36	34	32	29	23
	65	48.8	41.4	ESHF	1.00	.91	.86	.83	.78	.74	.72	.70	.68
				ADP	43.3	42	41	40	38	36	34	31	24
	70	49.7	44.6	ESHF	1.00	.90	.84	.80	.74	.71	.69	.67	.66
				ADP	45.2	44	43	42	40	38	36	33	31
	75	50.6	48.0	ESHF	1.00	.89	.82	.74	.69	.66	.65	.64	.63
				ADP	47.1	46	45	43	41	39	37	36	34
	80	51.5	51.2	ESHF	1.00	.88	.79	.74	.67	.64	.62	.61	.60
				ADP	48.8	48	47	46	44	42	40	39	37
	85	52.4	54.5	ESHF	1.00	.77	.70	.66	.63	.60	.58	.57	
				ADP	50.4	49	48	47	46	44	42	40	
	90	53.2	57.7	ESHF	1.00	.76	.67	.61	.58	.55	.54	.53	
				ADP	52.0	51	50	49	48	46	44	41	
	95	54.2	61.2	ESHF	1.00	.69	.58	.54	.51	.49			
				ADP	53.6	53	52	51	50	48			

Source : Carrier, Handbook of Air Conditioning System Design, p. 1-147

Table 11. Typical Bypass Factors (For Finned Coils)

Depth Of Coils (rows)	Without Sprays		With Sprays*	
	8 fins/in	14 fins/in	8 fins/in	14 fins/in
	Velocity (FPM)			
	300-700	300-700	300-700	300-700
2	.42-.55	.22-.38		
3	.27-.40	.10-.23		
4	.19-.30	.05-.14	.12-.22	.03-.10
5	.12-.23	.02-.09	.08-.14	.01-.08
6	.08-.18	.01-.06	.06-.11	.01-.05
8	.03-.08		.02-.05	

* The bypass factor with spray-coils is decreased because the spray provides more surfaces for contacting the air.

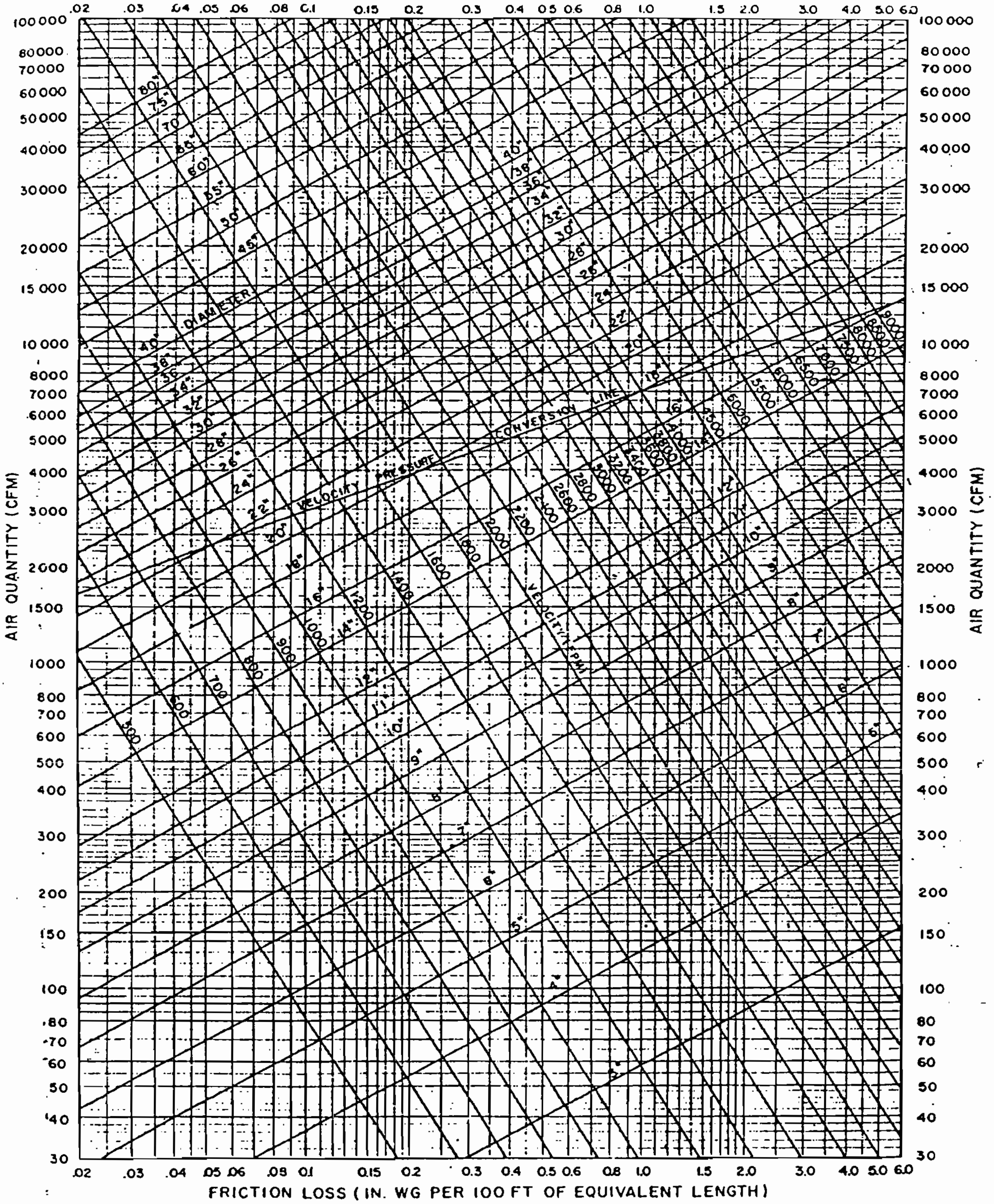
Source : Carrier, Handbook of Air Conditioning System Design, p. 1-127

Table 12. Typical Bypass Factors (For Various Applications)

Coil Bypass Factor	Type of Application	Example
0,30 to 0,50	A small total load or a load that is somewhat larger with a low sensible heat factor (high latent load).	Residence
0,20 to 0,30	Typical comfort application with a relatively small total load or a low sensible heat factor with a somewhat larger load.	Residence, small retail shop, and factory.
0,10 to 0,20	Typical comfort application.	Dept. store, bank, and factory.
0,05 to 0,10	Applications with high internal sensible loads or requiring a large amount of outdoor air for ventilation.	Dept. store, restaurant, and factory.
0 to 0,10	All outdoors air application.	Hospital, operating room, and factory.

Source : Carrier, Handbook of Air Conditioning System Design, p. 1-127

Table 13. Friction Loss for Round Duct



Source : Carrier, Handbook of Air Conditioning System Design, p. 2-33

Table 14. Duct Dimensions, Section Area, Circular Equivalent Diameter, and Duct Class

SIDE	6		8		10		12		14		16		18		20		22	
	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.
10	.39	8.4	.52	9.8	.65	10.9												
12	.45	9.1	.62	10.7	.77	11.9	.94	13.1										
14	.52	9.8	.72	11.5	.91	12.9	1.09	14.2	1.28	15.3								
16	.59	10.4	.81	12.2	1.02	13.7	1.24	15.1	1.45	16.3	1.67	17.5						
18	.66	11.0	.91	12.9	1.15	14.5	1.40	16.0	1.63	17.3	1.87	18.5	2.12	19.7				
20	.72	11.5	.99	13.5	1.26	15.2	1.54	16.8	1.81	18.2	2.07	19.5	2.34	20.7	2.61	21.9		
22	.78	12.0	1.08	14.1	1.38	15.9	1.69	17.6	1.99	19.1	2.27	20.4	2.57	21.7	2.86	22.9	3.17	24.1
24	.84	12.4	1.16	14.6	1.50	16.6	1.83	18.3	2.14	19.8	2.47	21.3	2.78	22.6	3.11	23.9	3.43	25.1
26	.89	12.8	1.26	15.2	1.61	17.2	1.97	19.0	2.31	20.6	2.66	22.1	3.01	23.5	3.35	24.8	3.71	26.1
28	.95	13.2	1.33	15.6	1.71	17.7	2.09	19.6	2.47	21.3	2.86	22.9	3.25	24.4	3.60	25.7	4.00	27.1
30	1.01	13.6	1.41	16.1	1.82	18.3	2.22	20.2	2.64	22.0	3.06	23.7	3.46	25.2	3.89	26.7	4.27	28.0
32	1.07	14.0	1.48	16.5	1.93	18.8	2.36	20.8	2.81	22.7	3.25	24.4	3.68	26.0	4.12	27.5	4.55	28.9
34	1.13	14.4	1.58	17.0	2.03	19.3	2.49	21.4	2.96	23.3	3.43	25.1	3.89	26.7	4.37	28.3	4.81	29.7
36	1.18	14.7	1.65	17.4	2.14	19.8	2.61	21.9	3.11	23.9	3.63	25.8	4.09	27.4	4.58	29.0	5.07	30.5
38	1.23	15.0	1.73	17.8	2.25	20.3	2.76	22.5	3.27	24.5	3.80	26.4	4.30	28.1	4.84	29.8	5.37	31.4
40	1.28	15.3	1.81	18.2	2.33	20.7	2.88	23.0	3.43	25.1	3.97	27.0	4.52	28.8	5.07	30.5	5.62	32.1
42	1.33	15.6	1.86	18.5	2.43	21.1	2.98	23.4	3.57	25.6	4.15	27.6	4.71	29.4	5.31	31.2	5.86	32.8
44	1.38	15.9	1.95	18.9	2.52	21.5	3.11	23.9	3.71	26.1	4.33	28.2	4.90	30.0	5.55	31.9	6.12	33.5
46	1.43	16.2	2.01	19.2	2.61	21.9	3.22	24.3	3.88	26.7	4.49	28.7	5.10	30.6	5.76	32.5	6.37	34.2
48	1.48	16.5	2.09	19.6	2.71	22.3	3.35	24.8	4.03	27.2	4.65	29.2	5.30	31.2	5.97	33.1	6.64	34.9
50			2.16	19.9	2.81	22.7	3.46	25.2	4.15	27.6	4.84	29.8	5.51	31.8	6.19	33.7	6.87	35.5
52			2.22	20.2	2.91	23.1	3.57	25.6	4.30	28.1	5.00	30.3	5.72	32.4	6.41	34.3	7.14	36.0
54			2.29	20.5	2.98	23.4	3.71	26.1	4.43	28.5	5.17	30.8	5.90	32.9	6.64	34.9	7.38	36.8
56			2.38	20.9	3.09	23.8	3.83	26.5	4.55	28.9	5.31	31.2	6.08	33.4	6.87	35.5	7.62	37.4
58			2.43	21.1	3.19	24.2	3.94	26.9	4.68	29.3	5.48	31.7	6.26	33.9	7.06	36.0	7.87	38.0
60			2.50	21.4	3.27	24.5	4.06	27.3	4.84	29.8	5.65	32.2	6.50	34.5	7.26	36.5	8.12	38.6
64			2.64	22.0	3.46	25.2	4.24	27.9	5.10	30.6	5.91	33.1	6.87	35.5	7.71	37.6	8.59	39.7
68					3.63	25.8	4.49	28.7	5.37	31.4	6.26	33.9	7.18	36.3	8.12	38.6	9.03	40.7
72					3.83	26.5	4.71	29.4	5.69	32.3	6.60	34.8	7.54	37.2	8.50	39.5	9.52	41.8
76					4.09	27.4	4.91	30.0	5.86	32.8	6.83	35.4	7.95	38.2	8.90	40.4	9.98	42.8
80					4.15	27.6	5.17	30.8	6.15	33.6	7.22	36.4	8.29	39.0	9.21	41.1	10.4	43.8
84							5.41	31.5	6.41	34.5	7.54	37.2	8.55	39.6	9.75	42.3	10.8	44.6
88							5.58	32.0	6.64	34.9	7.87	38.0	8.94	40.5	10.1	43.1	11.2	45.4
92							5.79	32.6	6.91	35.6	8.12	38.6	9.39	41.5	10.4	43.8	11.7	46.3
96							5.90	33.0	7.14	36.2	8.40	39.2	9.70	42.1	10.8	44.5	12.1	47.2
100									7.40	36.9	8.50	39.5	9.80	42.5	11.3	45.5	12.3	47.6
104									7.60	37.4	8.90	40.5	10.3	43.5	11.6	46.2	13.0	48.8
108									7.90	38.0	9.20	41.2	10.6	44.0	12.0	47.0	13.4	49.6
112									8.10	38.6	9.50	41.8	10.9	44.7	12.3	47.5	13.8	50.3
116											9.80	42.4	11.3	45.5	12.6	48.1	14.3	51.3
120											10.0	42.8	11.5	46.0	13.1	49.1	14.4	51.5
124											10.3	43.5	11.9	46.7	13.4	49.6	15.0	52.4
128											10.6	44.1	12.1	47.1	13.8	50.4	15.5	53.3
132													12.5	47.9	14.1	50.9	15.8	53.9
136													12.8	48.5	14.5	51.6	16.2	54.5
140													13.0	48.8	14.7	52.0	16.5	55.0
144													13.3	49.4	15.2	52.9	16.8	55.6

(0.625)

*Circular equivalent diameter (d_e). Calculated from $d_e = 1.3 \frac{(ab)^{.625}}{(a+b)^{.25}}$

†Large numbers in table are duct class.

Source : Carrier, Handbook of Air Conditioning System Design, p. 2-34

Table 14. Duct Dimensions, Section Area, Circular Equivalent Diameter, and Duct Class (Continued)

SIDE	24		26		28		30		32		34		36		38		40	
	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.
10																		
12																		
14																		
16																		
18																		
20																		
22																		
24	3.74	26.2																
26	4.03	27.2	4.40	28.4														
28	4.33	28.2	4.74	29.5	5.10	30.6												
30	4.68	29.3	5.07	30.5	5.44	31.6	5.86	32.8										
32	4.94	30.1	5.37	31.4	5.79	32.6	6.23	33.8	6.68	35.0								
34	5.24	31.0	5.69	32.3	6.15	33.6	6.60	34.8	7.06	36.0	7.54	37.2						
36	5.58	32.0	5.94	33.0	6.52	34.6	6.99	35.8	7.46	37.0	7.95	38.2	8.46	39.4				
38	5.86	32.8	6.38	34.2	6.87	35.5	7.34	36.7	7.87	38.0	8.37	39.2	8.89	40.4	9.43	41.6		
40	6.15	33.6	6.71	35.1	7.22	36.4	7.71	37.6	8.29	39.0	8.81	40.2	9.34	41.4	9.89	42.6	10.5	43.8
42	6.45	34.4	7.03	35.9	7.58	37.3	8.12	38.6	8.68	39.9	9.21	41.1	9.80	42.4	10.4	43.6	11.0	44.8
44	6.75	35.2	7.34	36.7	7.91	38.1	8.50	39.5	9.07	40.8	9.61	42.0	10.3	43.4	10.8	44.6	11.4	45.8
46	7.03	35.9	7.63	37.4	8.25	38.9	8.85	40.3	9.48	41.7	10.1	43.0	10.7	44.3	11.3	45.6	11.9	46.8
48	7.30	36.6	7.95	38.2	8.59	39.7	9.25	41.2	9.89	42.6	10.5	43.9	11.1	45.2	11.8	46.5	12.4	47.8
50	7.58	37.3	8.25	38.9	8.90	40.4	9.61	42.0	10.3	43.5	10.9	44.8	11.6	46.1	12.2	47.4	13.0	48.8
52	7.87	38.0	8.55	39.6	9.25	41.2	9.98	42.8	10.7	44.3	11.4	45.7	12.1	47.1	12.7	48.3	13.5	49.7
54	8.16	38.7	8.85	40.3	9.61	42.0	10.4	43.6	11.0	45.0	11.8	46.5	12.6	48.0	13.2	49.2	14.0	50.6
56	8.42	39.3	9.16	41.0	9.94	42.7	10.7	44.3	11.4	45.8	12.2	47.3	13.0	48.8	13.7	50.1	14.5	51.5
58	8.63	39.8	9.48	41.7	10.3	43.4	11.0	45.0	11.8	46.6	12.6	48.1	13.4	49.6	14.2	51.0	15.0	52.4
60	8.89	40.4	9.75	42.3	10.5	44.0	11.4	45.8	12.2	47.3	13.0	48.9	13.8	50.4	14.6	51.8	15.5	53.3
64	9.43	41.6	10.3	43.5	11.2	45.4	12.1	47.2	12.9	48.7	13.8	50.4	14.7	52.0	15.5	53.4	16.5	55.0
68	9.98	42.8	10.9	44.7	11.8	46.6	12.8	48.4	13.7	50.2	14.6	51.8	15.6	53.5	16.5	55.0	17.5	56.6
72	10.4	43.8	11.5	45.9	12.4	47.8	13.5	49.7	14.4	51.5	15.4	53.2	16.4	54.9	17.4	56.5	18.3	58.0
76	10.8	44.9	12.0	47.0	13.1	49.0	14.1	50.8	15.1	52.7	16.2	54.6	17.3	56.3	18.3	57.9	19.3	59.5
80	11.5	46.0	12.6	48.0	13.7	50.1	14.7	52.0	15.8	53.9	17.0	55.8	18.1	57.6	19.2	59.3	20.3	61.0
84	12.0	46.9	13.2	49.2	14.2	51.1	15.4	53.2	16.5	55.0	17.7	57.0	18.9	58.9	20.1	60.7	21.2	62.4
88	12.5	47.9	13.7	50.1	14.8	52.2	16.1	54.3	17.3	56.3	18.5	58.2	19.7	60.1	20.9	62.0	22.1	63.7
92	12.9	48.7	14.2	51.1	15.5	53.4	16.7	55.4	18.0	57.4	19.2	59.4	20.5	61.3	21.8	63.2	23.0	65.0
96	13.3	49.5	14.8	52.2	15.9	54.0	17.2	56.2	18.6	58.5	19.7	60.2	21.1	62.2	22.7	64.5	24.0	66.3
100	13.9	50.6	15.0	52.5	16.7	55.3	17.9	57.3	19.2	59.4	20.6	61.5	21.6	63.0	23.4	65.5	24.8	67.5
104	14.6	51.8	15.8	53.9	17.1	56.0	18.6	58.5	19.9	60.5	21.4	62.6	22.7	64.5	24.1	66.5	25.6	68.5
108	14.8	52.1	16.2	54.6	17.6	56.8	19.2	59.4	20.5	61.4	22.0	63.5	23.5	65.7	24.8	67.5	26.5	69.7
112	15.1	52.7	16.8	55.5	18.3	58.0	19.7	60.1	21.1	62.3	22.5	64.3	24.5	67.0	25.7	68.7	27.1	70.5
116	15.8	53.9	17.3	56.4	18.9	58.9	20.3	61.1	22.0	63.6	23.5	65.7	24.8	67.5	26.2	69.4	[illegible]	71.9
120	16.2	54.6	17.8	57.1	19.4	59.6	20.9	62.0	22.7	64.5	24.2	66.7	26.1	69.2	27.2	70.6	29.0	73.0
124	16.6	55.2	18.4	58.1	19.8	60.3	21.6	63.0	23.2	65.4	25.2	68.0	26.5	69.8	28.2	71.9	29.8	74.0
128	17.1	56.0	18.8	58.8	20.3	61.1	22.3	64.0	23.7	66.0	25.6	68.6	27.3	70.8	28.7	72.6	30.2	74.5
132	17.4	56.5	19.3	59.5	20.8	61.8	22.6	64.4	24.5	67.0	26.3	69.5	28.2	72.0	29.8	74.0	32.0	76.6
136	17.9	57.3	19.7	60.2	21.4	62.7	23.0	65.0	25.1	67.9	26.9	70.3	28.7	72.6	30.5	74.8	32.6	77.3
140	18.5	58.2	20.3	61.0	22.3	64.0	24.1	66.5	25.9	69.0	27.5	71.1	29.4	73.5	31.5	76.0	33.4	78.3
144	18.8	58.7	20.6	61.5	22.7	64.5	24.8	67.5	26.3	69.5	28.2	72.0	29.9	74.1	32.0	76.6	34.0	79.0

*Circular equivalent diameter (d_e). Calculated from $d_e = 1.3 \frac{(ab)^{.625}}{(a+b)^{.25}}$

†Large numbers in table are duct class.

Source : Carrier, Handbook of Air Conditioning System Design, p. 2-35

Table 14. Duct Dimensions, Section Area, Circular Equivalent Diameter, and Duct Class (Continued)

SIDE	42		44		46		48		50		52		54		56		58	
	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.
42	11.5	45.9																
44	13.0	46.9	12.6	43.1														
46	12.5	47.9	13.1	49.1	13.8	50.3												
48	13.0	48.9	13.7	50.2	14.3	51.3	15.1	52.6										
50	13.5	49.8	14.3	51.2	14.9	52.3	15.7	53.6	16.3	54.7								
52	14.1	50.8	14.8	52.2	15.5	53.3	16.2	54.6	17.0	55.8	17.6	56.9						
54	14.6	51.8	15.4	53.2	16.1	54.3	16.8	55.6	17.6	56.8	18.3	57.9	19.2	59.4				
56	15.1	52.7	15.9	54.1	16.7	55.3	17.4	56.5	18.2	57.8	18.9	58.9	19.6	60.0	20.5	61.3		
58	15.7	53.7	16.5	55.0	17.2	56.2	18.0	57.5	18.8	58.8	19.6	60.0	20.4	61.2	21.1	62.3	22.0	63.5
60	16.2	54.6	17.0	55.9	17.8	57.1	18.6	58.5	19.5	59.8	20.3	61.0	21.1	62.2	21.8	63.3	22.5	64.3
64	17.3	56.4	18.1	57.7	19.0	59.0	19.8	60.3	20.7	61.6	21.6	62.9	22.4	64.1	23.2	65.3	24.4	66.9
68	18.3	58.0	19.3	59.5	20.1	60.8	21.1	62.1	21.9	63.4	22.9	64.8	23.8	66.1	24.7	67.3	25.5	68.4
72	19.4	59.6	20.3	61.1	21.4	62.6	22.2	63.9	23.1	65.2	24.2	66.6	25.1	67.9	26.1	69.2	27.1	70.5
76	20.4	61.2	21.4	62.7	22.4	64.1	23.4	65.6	24.5	67.0	25.5	68.4	26.4	69.6	27.5	71.0	28.9	72.8
80	21.4	62.7	22.4	64.1	23.5	65.7	24.6	67.2	25.7	68.7	26.8	70.1	28.1	71.8	28.8	72.7	30.1	74.3
84	22.4	64.1	23.5	65.7	24.7	67.3	25.8	68.8	26.9	70.3	28.1	71.8	29.1	73.1	30.2	74.5	31.5	76.0
88	23.3	65.4	24.5	67.0	25.7	68.7	26.9	70.3	28.1	71.8	29.4	73.4	30.6	74.9	31.7	76.3	32.7	77.5
92	24.3	66.8	25.6	68.5	26.8	70.1	28.1	71.8	29.3	73.3	30.6	74.9	31.9	76.5	33.1	77.9	34.2	79.2
96	25.2	68.0	26.7	70.0	27.6	71.1	29.4	73.5	30.2	74.5	31.8	76.4	33.2	78.0	33.9	78.9	35.7	80.9
100	26.0	69.1	27.1	70.5	29.0	72.9	30.2	74.5	31.6	76.1	32.7	77.5	33.8	78.7	35.3	80.7	36.6	82.0
104	27.1	70.5	28.4	72.2	29.4	74.0	31.1	75.5	32.7	77.5	34.0	79.0	35.8	81.0	37.1	82.5	38.5	84.1
108	28.0	71.7	29.5	73.6	30.6	74.9	32.3	77.0	33.3	78.2	35.3	80.5	36.6	82.0	38.5	84.0	39.8	85.5
112	29.2	73.2	30.2	74.5	31.9	76.5	33.1	78.0	34.9	80.0	36.6	82.0	38.0	83.5	39.8	85.5	40.8	86.5
116	30.0	74.2	32.0	76.6	32.7	77.5	34.0	79.0	35.9	81.2	38.0	83.5	39.8	85.5	41.0	86.7	42.4	88.2
120	30.7	75.0	32.7	77.5	33.6	78.5	35.8	81.0	37.4	82.9	39.4	85.0	40.9	86.6	41.9	87.7	43.6	89.4
124	31.5	76.0	33.6	78.5	34.4	79.5	36.5	81.8	38.5	84.1	40.7	86.1	41.5	87.3	43.3	89.1	44.6	90.5
128	32.1	76.8	34.0	79.0	36.2	81.5	37.5	83.0	39.2	84.8	41.4	87.2	42.9	88.7	44.6	90.5	46.6	92.5
132	33.2	78.0	34.9	80.0	36.9	82.3	38.8	84.4	40.7	86.4	42.7	88.5	44.1	90.0	46.0	91.9	48.0	93.9
136	34.0	79.0	35.6	80.8	38.0	83.5	39.7	85.4	41.7	87.5	43.8	89.7	44.8	90.7	47.4	[illegible]	49.7	95.5
140	35.3	80.5	37.0	82.4	38.8	84.4	40.5	86.7	42.4	88.2	44.9	90.8	46.5	92.4	48.6	94.4	50.3	96.1
144	35.8	81.1	37.8	83.3	40.0	85.7	41.4	87.2	44.1	90.0	45.6	91.5	47.8	93.7	49.7	95.5	51.5	97.2

SIDE	60		64		68		72		76		80		84		88		92	
	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.
42																		
44																		
46																		
48																		
50																		
52																		
54																		
56																		
58																		
60	23.5	65.7																
64	25.0	67.7	26.7	70.0														
68	26.5	69.7	28.3	72.1	30.2	74.4												
72	28.0	71.7	29.9	74.1	31.8	76.4	33.8	78.8										
76	29.5	73.6	31.6	76.1	33.5	78.4	35.7	80.9	27.7	83.2								
80	31.0	75.4	33.2	78.1	35.2	80.4	37.4	82.8	39.6	85.3	41.7	87.5						
84	32.5	77.2	34.8	79.9	37.0	82.4	39.2	84.8	41.4	87.2	43.7	89.6	46.0	91.9				
88	34.0	79.0	36.3	81.6	38.6	84.2	41.1	86.8	43.4	89.2	45.7	91.6	48.0	93.9	50.5	96.3		
92	35.6	80.8	27.9	83.4	40.3	86.0	42.9	88.7	45.3	91.2	47.7	93.6	50.1	95.9	52.7	98.3	55.1	100.5
96	37.0	82.4	29.8	85.5	42.1	87.9	44.6	90.5	47.5	93.4	49.8	95.6	51.9	97.6	55.2	100.6	57.8	103.0
100	38.4	83.9	41.2	87.0	44.3	90.2	47.5	93.4	50.2	96.0	51.9	97.6	53.3	98.9	56.7	102.0	60.1	105.0
104	40.3	86.0	42.8	88.6	46.1	92.0	48.2	94.0	51.5	97.2	53.6	99.2	57.3	102.5	59.5	104.5	62.4	107.0
108	41.7	87.5	44.1	90.0	46.9	92.8	50.1	95.9	53.0	98.6	55.6	101.0	58.5	103.6	61.0	105.8	64.7	109.6
112	43.3	88.1	45.3	91.2	48.9	94.7	51.7	97.4	54.3	99.8	57.4	102.6	58.9	104.0	63.8	108.2	67.1	111.0
116	44.1	90.0	47.6	93.5	51.1	96.8	53.7	99.3	57.0	102.3	60.1	105.0	63.3	107.8	64.2	110.2	69.3	112.8
120	45.5	91.4	49.7	95.5	51.8	97.5	55.8	101.2	58.9	104.0	62.4	107.0	65.5	109.6	69.0	112.5	72.1	115.0
124	47.1	93.0	49.8	95.6	53.8	99.4	54.7	102.0	60.1	105.0	63.6	108.0	64.2	110.2	69.3	112.8	73.3	116.0
128	47.6	93.5	51.3	97.0	55.4	100.8	58.7	103.8	61.8	106.5	65.5	109.6	68.1	111.8	72.3	115.2	76.3	118.3
132	49.7	95.5	53.0	98.6	56.3	101.6	60.1	105.0	64.2	108.5	68.4	112.0	71.8	114.8	74.6	117.0	78.5	120.0
136	50.3	96.1	54.9	100.4	58.9	104.0	62.2	106.8	64.7	109.0	69.6	113.0	72.8	115.6	76.7	118.6	81.5	122.3
140	52.4	98.1	55.6	101.0	60.4	105.3	63.8	108.2	67.8	111.5	71.4	114.5	75.6	117.8	79.1	120.5	82.7	123.2
144	54.1	99.6	57.8	103.0	61.2	106.0	64.7	109.0	69.1	112.6	73.3	116.0	78.0	119.6	81.1	122.0	85.2	125.0

*Circular equivalent diameter (d_e). Calculated from $d_e = 1.3 \frac{(ab)^{.625}}{(a+b)^{.25}}$

†Large numbers in table are duct class.

Table 15. Recommended Maximum Duct Velocities for Low Velocity Systems (FPM)

APPLICATION	CONTROLLING FACTOR NOISE GENERATION Main Ducts	CONTROLLING FACTOR—DUCT FRICTION			
		Main Ducts		Branch Ducts	
		Supply	Return	Supply	Return
Residences	600	1000	800	600	600
Apartments Hotel Bedrooms Hospital Bedrooms	1000	1500	1300	1200	1000
Private Offices Directors Rooms Libraries	1200	2000	1500	1600	1200
Theatres Auditoriums	800	1300	1100	1000	800
General Offices High Class Restaurants High Class Stores Banks	1500	2000	1500	1600	1200
Average Stores Cafeterias	1800	2000	1500	1600	1200
Industrial	2500	3000	1800	2200	1500

Source : Carrier, Handbook of Air Conditioning System Design, p. 2-37

Table 16. Velocity Pressure

VELOCITY PRESSURE (in. wg)	VELOCITY (Ft/Min)	VELOCITY PRESSURE (in. wg)	VELOCITY (Ft/Min)	VELOCITY PRESSURE (in. wg)	VELOCITY (Ft/Min)	VELOCITY PRESSURE (in. wg.)	VELOCITY (Ft/Min)
.01	400	.29	2150	.58	3050	1.28	4530
.02	565	.30	2190	.60	3100	1.32	4600
.03	695	.31	2230	.62	3150	1.36	4670
.04	800	.32	2260	.64	3200	1.40	4730
.05	895	.33	2300	.66	3250	1.44	4800
.06	980	.34	2330	.68	3300	1.48	4870
.07	1060	.35	2370	.70	3350	1.52	4930
.08	1130	.36	2400	.72	3390	1.56	5000
.09	1200	.37	2440	.74	3440	1.60	5060
.10	1270	.38	2470	.76	3490	1.64	5120
.11	1330	.39	2500	.78	3530	1.68	5190
.12	1390	.40	2530	.80	3580	1.72	5250
.13	1440	.41	2560	.82	3620	1.76	5310
.14	1500	.42	2590	.84	3670	1.80	5370
.15	1550	.43	2620	.86	3710	1.84	5430
.16	1600	.44	2650	.88	3750	1.88	5490
.17	1650	.45	2680	.90	3790	1.92	5550
.18	1700	.46	2710	.92	3840	1.96	5600
.19	1740	.47	2740	.94	3880	2.00	5660
.20	1790	.48	2770	.96	3920	2.04	5710
.21	1830	.49	2800	.98	3960	2.08	5770
.22	1880	.50	2830	1.00	4000	2.12	5830
.23	1920	.51	2860	1.04	4080	2.16	5880
.24	1960	.52	2880	1.08	4160	2.20	5940
.25	2000	.53	2910	1.12	4230	2.24	5990
.26	2040	.54	2940	1.16	4310	2.28	6040
.27	2080	.55	2970	1.20	4380		
.28	2120	.56	2990	1.24	4460		

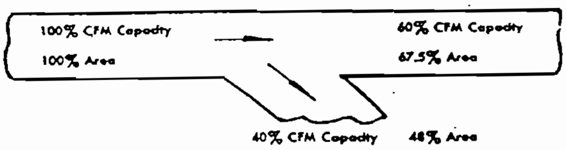
NOTES: 1. Data for standard air (29.92 in. Hg and 70 F)

2. Data derived from the following equation: $h_v = \left(\frac{V}{4005}\right)^2$ where: V = velocity in fpm.
h_v = pressure difference termed "velocity head" (in. wg).

Source : Carrier, Handbook of Air Conditioning System Design, p. 2-37

Table 17. Percent Section Area in Branches for Maintaining Equal Friction

CFM CAPACITY %	DUCT AREA %	CFM CAPACITY %	DUCT AREA %	CFM CAPACITY %	DUCT AREA %	CFM CAPACITY %	DUCT AREA %
1	2.0	26	33.5	51	59.0	76	81.0
2	3.5	27	34.5	52	60.0	77	82.0
3	5.5	28	35.5	53	61.0	78	83.0
4	7.0	29	36.5	54	62.0	79	84.0
5	9.0	30	37.5	55	63.0	80	84.5
6	10.5	31	39.0	56	64.0	81	85.5
7	11.5	32	40.0	57	65.0	82	86.0
8	13.0	33	41.0	58	65.5	83	87.0
9	14.5	34	42.0	59	66.5	84	87.5
10	16.5	35	43.0	60	67.5	85	88.5
11	17.5	36	44.0	61	68.0	86	89.5
12	18.5	37	45.0	62	69.0	87	90.0
13	19.5	38	46.0	63	70.0	88	90.5
14	20.5	39	47.0	64	71.0	89	91.5
15	21.5	40	48.0	65	71.5	90	92.0
16	23.0	41	49.0	66	72.5	91	93.0
17	24.0	42	50.0	67	73.5	92	94.0
18	25.0	43	51.0	68	74.5	93	94.5
19	26.0	44	52.0	69	75.5	94	95.0
20	27.0	45	53.0	70	76.5	95	96.0
21	28.0	46	54.0	71	77.0	96	96.5
22	29.5	47	55.0	72	78.0	97	97.5
23	30.5	48	56.0	73	79.0	98	98.0
24	31.5	59	57.0	74	80.0	99	99.0
25	32.5	50	58.0	75	80.5	100	100.0



Source : Carrier, Handbook of Air Conditioning System Design, p. 2-46

Table 18. Weights of Duct Material

WEIGHT (lb/sq ft)	GAGE (THICKNESS) (in.)	WEIGHT PER SHEET (lb) 36 x 96	48 x 96	48 x 120
GALVANIZED STEEL, U.S. GAGE				
.906	26 ga. (.022)	21.8	29.0	36.2
1.156	24 ga. (.028)	27.7	37.0	46.2
1.406	22 ga (.034)	33.8	45.0	56.2
1.656	20 ga. (.040)	39.7	53.0	66.2
2.156	18 ga. (.052)	51.6	70.0	86.2
2.656	16 ga. (.064)	63.6	85.0	102.2
3.281	14 ga. (.080)	78.8	105.0	131.2
HOT ROLLED STEEL, U.S. GAGE				
.750	26 ga. (.0179)	18.0	24.0	30.0
1.000	24 ga. (.0239)	24.0	32.0	40.0
1.250	22 ga. (.0299)	30.0	40.0	50.0
1.500	20 ga. (.0359)	36.0	48.0	60.0
2.000	18 ga. (.0478)	48.0	64.0	80.0
2.500	16 ga. (.0596)	60.0	80.0	100.0
3.125	14 ga. (.0747)	78.0	104.0	130.0
5.625	10 ga. (.1345)	135.0	180.0	225.0
ALUMINUM, B & S GAGE ([illegible])				
.288	24 ga. (.020)	6.9	9.2	11.5
.355	22 ga. (.025)	[illegible]	11.3	14.2
.456	20 ga. (.032)	11.0	14.6	18.2
.575	18 ga. (.040)	13.8	18.4	23.0
.724	16 ga. (.051)	17.4	23.2	29.0
.914	14 ga. (.064)	22.0	29.2	36.6
1.03	12 ga. (.071)	24.7	33.0	41.3
STAINLESS STEEL, U.S. GAGE (302)				
.66	28 ga. (.016)	15.8	21.1	26.4
.79	26 ga. (.019)	18.9	25.2	31.6
1.05	24 ga. (.025)	25.2	33.6	42.0
1.31	22 ga. (.031)	31.5	42.0	52.5
1.58	20 ga. (.038)	37.8	50.4	63.0
2.10	18 ga. (.050)	50.4	61.2	84.0
2.63	16 ga. (.063)	63.0	84.0	105.0
3.28	14 ga. (.078)	78.7	104.9	131.2
COPPER, OZ/SQ FT				
1.00	16 oz. (.0216)	24.0	32.0	40.0
1.25	20 oz. (.027)	30.0	40.0	50.0
1.50	24 oz. (.0323)	36.0	48.0	64.0
2.00	32 oz. (.0432)	48.0	64.0	80.0
2.25	36 oz. (.0486)	54.0	72.0	90.0
2.50	40 oz. (.0540)	60.0	80.0	100.0

Source : Carrier, Handbook of Air Conditioning System Design, p. 2-63

Table 19. Properties of R134a (Saturated Temperature)

Temp., °F	Pressure, psia	Density, lb/ft³	Volume, ft³/lb	Enthalpy, Btu/lb		Entropy, Btu/(lb · °K)	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor
−140	0.145	97.834	231.846	−25.470	80.935	−0.068975	0.263886
−130	0.243	96.942	142.352	−23.144	82.373	−0.061811	0.258258
−120	0.394	96.043	90.331	−20.773	83.829	−0.054726	0.253225
−110	0.620	95.135	59.058	−18.354	85.301	−0.047710	0.248727
−100	0.947	94.218	39.675	−15.887	86.787	−0.040755	0.244711
−90	1.410	93.292	27.321	−13.370	88.283	−0.033854	0.241128
−85	1.705	92.825	22.867	−12.092	89.035	−0.030422	0.239485
−80	2.050	92.355	19.242	−10.801	89.788	−0.027001	0.237935
−75	2.452	91.883	16.275	−9.497	90.542	−0.023591	0.236472
−70	2.917	91.408	13.833	−8.179	91.297	−0.020192	0.235092
−65	3.452	90.930	11.813	−6.849	92.052	−0.016802	0.233791
−60	3.067	90.448	10.134	−5.506	92.808	−0.013423	0.232564
−55	4.768	89.964	8.730	−4.149	93.563	−0.010053	0.231408
−50	5.566	89.476	7.552	−2.779	94.318	−0.006692	0.230319
−45	6.470	88.985	6.559	−1.396	95.071	−0.003341	0.229294
−40	7.490	88.490	5.717	0.000	95.823	0.000000	0.228329
−39	7.709	88.391	5.565	0.281	95.973	0.000667	0.228143
−38	7.933	88.291	5.417	0.562	96.123	0.001334	0.227960
−37	8.162	88.191	5.274	0.844	96.273	0.002000	0.227778
−36	8.396	88.091	5.136	1.126	96.423	0.002666	0.227599
−35	8.636	87.991	5.002	1.409	96.573	0.003332	0.227422
−34	8.882	87.891	4.872	1.692	96.723	0.003997	0.227247
−33	9.133	87.791	4.746	1.976	96.873	0.004662	0.227074
−32	9.389	87.690	4.624	2.261	97.022	0.005327	0.226903
−31	9.652	87.590	4.506	2.546	97.172	0.005991	0.226735
−30	9.920	87.489	4.391	2.831	97.321	0.006655	0.226568
−29	10.195	87.388	4.280	3.117	97.471	0.007318	0.226404
−28	10.475	87.287	4.172	3.404	97.620	0.007981	0.226242
−27	10.762	87.185	4.067	3.691	97.769	0.008644	0.226081
−26	11.054	87.084	3.966	3.978	97.918	0.009306	0.225923
−25	11.354	86.982	3.868	4.266	98.067	0.009968	0.225766
−24	11.659	86.880	3.772	4.555	98.216	0.010629	0.225612
−23	11.972	86.778	3.679	4.844	98.365	0.011290	0.225459
−22	12.290	86.676	3.589	5.133	98.513	0.011951	0.225309
−21	12.616	86.574	3.502	5.423	98.662	0.012611	0.225160

Source : Ari Rabl, Heating and Cooling for Buildings, p. 795

Table 19. Properties of R134a (Saturated Temperature) (Continued)

Temp., °F	Pressure, psia	Density, lb/ft^3	Volume, ft^3/lb	Enthalpy, Btu/lb		Entropy, Btu/(lb · °R)	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor
15	29.756	82.761	1.553	16.196	103.905	0.036124	0.220902
16	30.410	82.651	1.521	16.505	104.047	0.036769	0.220810
17	31.075	82.541	1.490	16.814	104.189	0.037415	0.220719
18	31.751	82.430	1.459	17.123	104.331	0.038060	0.220630
19	32.438	82.319	1.430	17.433	104.473	0.038705	0.220542
20	33.137	82.208	1.401	17.743	104.614	0.039349	0.220455
21	33.848	82.097	1.373	18.054	104.755	0.039993	0.119268
22	34.570	81.986	1.345	18.365	104.896	0.040636	0.220284
23	35.304	81.874	1.318	18.677	105.037	0.041279	0.220210
24	36.051	81.762	1.292	18.989	105.177	0.041922	0.220118
25	36.809	81.650	1.267	19.302	105.317	0.042564	0.220037
26	37.580	81.537	1.242	19.615	105.457	0.043206	0.219957
27	38.363	81.425	1.217	19.929	105.597	0.043848	0.219878
28	39.159	81.312	1.193	20.243	105.736	0.044489	0.219799
29	39.967	81.198	1.170	20.557	105.876	0.045130	0.219722
30	40.789	81.085	1.147	20.873	106.015	0.045770	0.219647
31	41.623	80.971	1.125	21.188	106.153	0.046410	0.219572
32	41.470	80.857	1.104	21.504	106.292	0.047050	0.219498
33	43.331	80.742	1.082	21.831	106.430	0.047689	0.219425
34	44.205	80.628	1.062	22.138	106.568	0.048328	0.219353
35	45.093	80.513	10.41	22.456	106.705	0.048967	0.219282
36	45.994	80.397	1.022	22.774	106.843	0.049605	0.219212
37	46.909	80.282	1.002	23.092	106.980	0.050243	0.219143
38	47.838	80.166	0.983	23.411	107.117	0.050880	0.219075
39	48.781	80.049	0.965	23.731	107.253	0.051517	0.219007
40	49.739	79.933	0.947	24.051	107.389	0.052154	0.218941
42	51.697	79.699	0.912	24.693	107.661	0.053426	0.218811
44	53.713	79.464	0.879	25.336	107.932	0.054697	0.218685
46	55.789	79.227	0.847	25.982	108.201	0.055967	0.218561
48	57.926	78.989	0.816	26.629	108.469	0.054235	0.218441
50	60.125	78.750	0.787	27.279	108.736	0.058502	0.218325
52	62.388	78.510	0.759	27.930	109.001	0.049768	0.218211
54	64.714	78.268	0.732	28.584	109.265	0.061032	0.218100
56	67.107	78.024	0.707	29.240	109.528	0.062295	0.217992
58	69.566	77.779	0.682	29.897	109.790	0.063556	0.217887

Source : Ari Rabl, Heating and Cooling for Buildings, p. 796

Table 19. Properties of R134a (Saturated Temperature) (Continued)

Temp., °F	Pressure, psia	Density, lb/ft³	Volume, ft³/lb	Enthalpy, Btu/lb		Entropy, Btu/(lb · °R)	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor
−20	12.949	86.471	3.417	5.714	98.810	0.013271	0.225013
−19	13.288	86.369	3.335	6.005	98.958	0.013931	0.224868
−18	13.635	86.266	3.255	6.296	99.106	0.014590	0.224725
−17	13.988	86.163	3.177	6.588	99.254	0.015249	0.224583
−16	14.349	86.060	3.102	6.881	99.402	0.015907	0.224444
−15	14.718	85.956	3.029	7.174	99.550	0.016565	0.224306
−14	15.094	85.853	2.957	7.467	99.697	0.017223	0.224170
−13	15.477	85.749	2.888	7.762	99.845	0.017880	0.224035
−12	15.868	85.645	2.821	8.056	99.992	0.018537	0.223903
−11	16.267	85.541	2.755	8.351	100.139	0.019194	0.223772
−10	16.674	85.436	2.692	8.647	100.286	0.019850	0.223642
−9	17.089	85.332	2.630	8.943	100.433	0.020505	0.223515
−8	17.512	85.227	2.570	9.239	100.580	0.021161	0.223389
−7	17.943	85.122	2.511	9.536	100.726	0.021815	0.223264
−6	18.383	85.017	2.455	9.834	100.873	0.022470	0.223141
−5	18.831	84.912	2.399	10.132	101.019	0.023124	0.223020
−4	19.288	84.806	2.345	10.431	101.165	0.023778	0.222900
−3	19.754	84.701	2.293	10.730	101.311	0.024431	0.222782
−2	20.228	84.595	2.242	11.029	101.456	0.025084	0.222666
−1	20.711	84.488	2.192	11.329	101.602	0.025737	0.222551
0	21.204	84.382	2.144	11.630	101.747	0.026389	0.222437
1	21.705	84.276	2.097	11.931	101.892	0.027041	0.222325
2	22.216	84.169	2.051	12.232	102.037	0.027692	0.222214
3	22.736	84.062	2.007	12.534	102.182	0.028343	0.222105
4	23.266	83.955	1.963	12.837	102.327	0.028993	0.221997
5	23.805	83.847	1.921	13.140	102.471	0.029644	0.221891
6	24.354	83.740	1.880	13.443	102.616	0.030293	0.221786
7	24.913	83.632	1.839	13.747	102.760	0.030943	0.221683
8	25.482	83.524	1.800	14.052	102.904	0.031592	0.221480
9	26.062	83.415	1.762	14.357	103.047	0.032240	0.221479
10	26.651	83.307	1.725	14.662	103.191	0.032889	0.221380
11	27.251	83.198	1.689	14.968	103.334	0.033536	0.221282
12	27.861	83.089	1.654	15.274	103.477	0.034184	0.221185
13	28.482	82.980	1.619	15.581	103.620	0.034831	0.221089
14	29.114	82.870	1.586	15.889	103.763	0.035477	0.220995

Source : Ari Rabl, Heating and Cooling for Buildings, p. 797

Table 19. Properties of R134a (Saturated Temperature) (Continued)

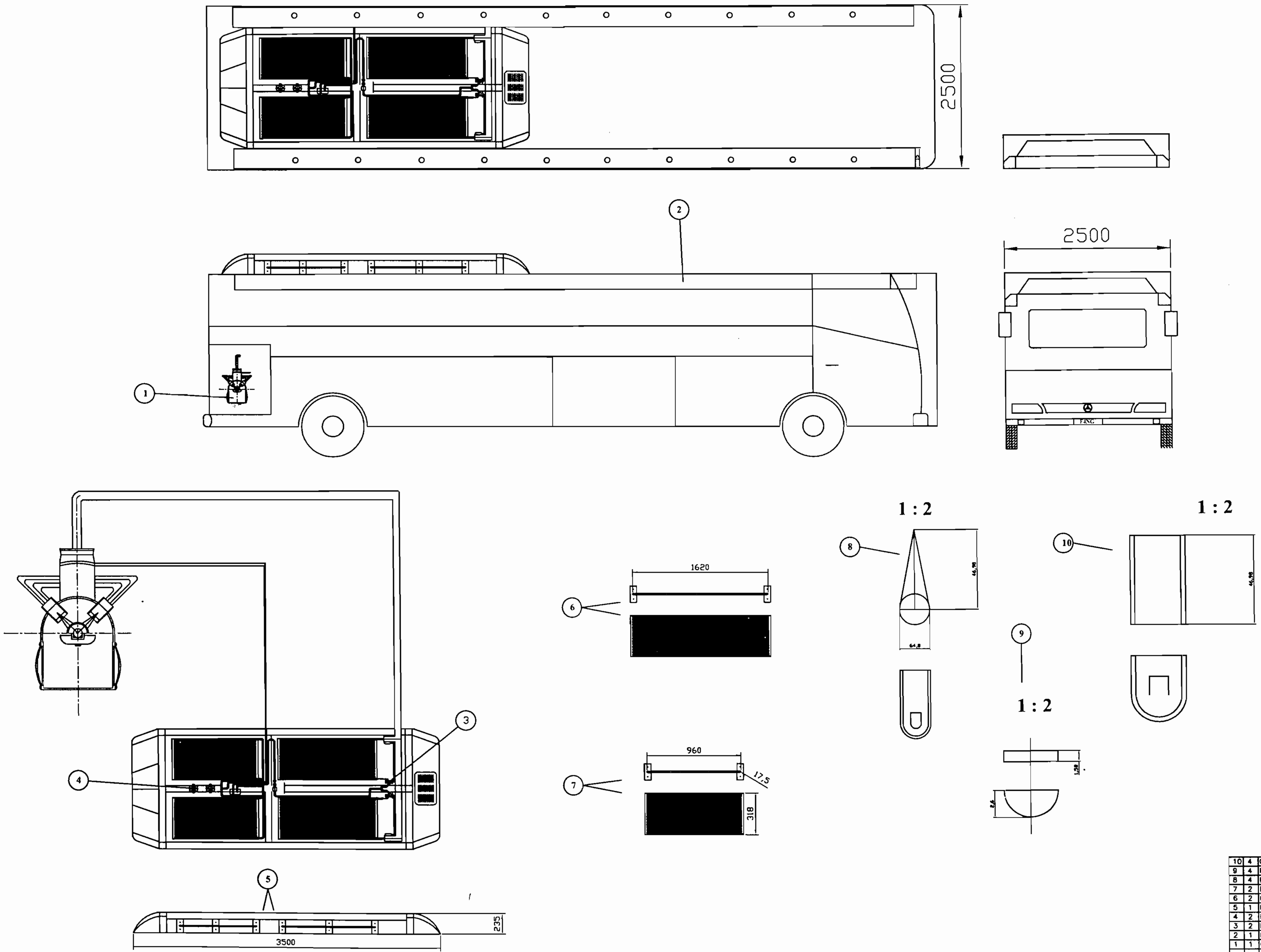
Temp., °F	Pressure, psia	Density, lb/ft³	Volume, ft³/lb	Enthalpy, Btu/lb		Entropy, Btu/(lb · °R)	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor
60	72.093	77.533	0.658	30.557	110.050	0.064817	0.217784
62	74.689	77.285	0.636	31.219	110.308	0.066076	0.217684
64	77.355	77.035	0.614	31.883	110.565	0.067335	0.217586
66	80.093	76.784	0.593	32.549	110.821	0.068592	0.217490
68	82.904	76.531	0.573	33.218	111.075	0.069848	0.217397
70	85.789	76.276	0.554	33.888	111.327	0.071103	0.217305
72	88.749	76.020	0.535	34.561	111.578	0.072357	0.217216
74	91.786	75.762	0.518	35.236	111.827	0.073611	0.217128
76	94.901	75.502	0.500	35.913	112.074	0.074863	0.217042
78	98.095	75.240	0.484	36.593	112.320	0.076115	0.216957
80	101.369	74.976	0.468	37.275	112.564	0.077366	0.216874
82	104.725	94.710	0.453	37.959	112.805	0.078616	0.216793
84	108.164	74.442	0.438	38.646	113.045	0.079865	0.216712
86	111.688	74.172	0.424	39.335	113.283	0.081115	0.216633
88	115.297	73.900	0.411	40.026	113.519	0.082363	0.216555
90	118.993	73.626	0.398	40.720	113.753	0.083611	0.216477
95	128.623	72.930	0.367	42.467	114.328	0.086730	0.216286
100	138.827	72.219	0.339	44.230	114.888	0.089848	0.216096
105	149.626	71.491	0.313	46.011	115.432	0.092967	0.215907
110	161.044	70.745	0.290	47.811	115.959	0.096087	0.215704
115	173.102	69.980	0.268	49.630	116.467	0.099212	0.215516
120	185.825	69.194	0.248	51.471	116.954	0.102342	0.215309
125	199.236	68.385	0.230	53.334	117.419	0.105481	0.215090
130	213.362	67.550	0.213	55.222	117.859	0.108630	0.214854
135	228.227	66.688	0.197	57.136	118.271	0.111793	0.214599
140	243.859	65.795	0.183	59.078	118.653	0.114972	0.214318
150	277.537	63.900	0.157	63.061	119.306	0.121399	0.213656
160	314.635	61.828	0.134	67.198	119.780	0.127952	0.212808
170	355.426	59.521	0.114	71.532	120.014	0.134691	0.211687
180	400.223	56.881	0.096	76.130	119.906	0.141711	0.210147
190	499.412	53.729	0.080	81.117	119.274	0.149191	0.207924
200	503.522	49.645	0.065	86.776	117.690	0.157357	0.204401
210	563.510	42.940	0.048	94.272	113.450	0.168450	0.197088

Source : Ari Rabl, Heating and Cooling for Buildings, p. 798

Table 20. Properties of Solid Metals

Composition	Melting point, K	Properties at 300 K				Properties at various temperatures (K), k(W/m · K)/c_p(J/kg · K)					
		ρ kg/m^3	c_p J/kg · K	k W/m · K	$\alpha \times 10^6$ m^2/s	100	200	400	600	800	1000
Aluminum:											
Pure	933	2702	903	237	97.1	302	237	240	231	218	
						482	798	949	1033	1146	
Alloy 2024-T6	775	2770	875	177	73.0	65	163	186	186		
(4.5% Cu, 1.5% Mg, 0.6% Mn)						473	787	925	1042		
Alloy 195, Cast (4.5% Cu)		2790	883	168	68.2			174	185		
Beryllium	1550	1850	1825	200	59.2	990	301	161	126	106	90.8
						203	1114	2191	2604	2823	3018
Bismuth	545	9780	122	7.86	6.59	16.5	9.69	7.04			
						112	120	127			
Boron	2573	2500	1107	27.0	9.76	190	55.5	16.8	10.6	9.60	9.85
						128	600	1463	1892	2160	2338
Cadmium	594	8650	231	96.8	48.4	203	99.3	94.7			
						198	222	242			
Chromium	2118	7160	449	93.7	29.1	159	111	90.9	80.7	71.3	65.4
						192	384	484	542	581	616
Cobalt	1769	8862	421	99.2	26.6	167	122	85.4	67.4	58.2	52.1
						236	379	450	503	550	628
Copper:											
Pure	1358	8933	385	401	117	482	413	393	379	366	352
						252	356	397	417	433	451
Commercial bronze	1293	8800	420	52	14		42	52	59		
(90% Cu, 10% Al)							785	160	545		
Phosphor gear bronze	1104	8780	355	54	17		41	65	74		
(89% Cu, 11% Sn)							—	—	—		
Cartridge brass	1188	8530	380	110	33.9	75	95	137	149		
(70% Cu, 30% Zn)							360	395	425		
Constantan	1493	8920	384	23	6.71	17	19				
(55% Cu, 45% Ni)						237	362				
Germanium	1211	5360	322	59.9	34.7	232	96.8	43.2	27.3	19.8	17.4
						190	290	337	348	357	375
Gold	1336	19,300	129	317	127	327	323	311	298	284	270
						109	124	131	135	140	145
Iridium	2720	22,500	130	147	50.3	172	153	144	138	132	126
						90	122	133	138	144	153
Iron:											
Pure	1810	7870	447	80.2	23.1	134	94.0	69.5	54.7	43.3	32.8
						216	384	490	574	680	975
Armco											
(99.75% pure)		7870	447	72.7	20.7	95.6	80.6	65.7	53.1	42.2	32.3
						215	384	490	574	680	975
Carbon steels:											
Plain carbon (Mn ≤ 1%,		7854	434	60.5	17.7			56.7	48.0	39.2	30.0
Si ≤ 0.1%)								487	559	685	1169

Source :çengel, Heat Transfer A Practical Approach, p. 948



10	4	Crossheads	Carbon Steel		
9	4	Piston Rings	Cr & Mo		
8	4	Piston Rods	Carbon Steel		
7	2	Evaporator	Copper		
6	2	Condenser	Copper		
5	1	Rooftop Unit			
4	2	Fan Condenser	Aluminum		
3	2	Expansion Valve	Steel		TEV
2	1	Duct	Galvanized steel		
1	1	Compressor	Copper		
No.	Qty.	Parts	Materials	Normalization	Note

Scale : 1:10	Drawn by : Nadia	Note	
Dimension : mm	NIM : 995214099		
Date :04.09.03	Checked by : Ir. G. Harianto		
MECH. ENGR. ED-USD	Executive Bus Air Conditioning	01	A3