ENGINEERING STANDARD

FOR

BUILDING AIR CONDITIONING SYSTEMS

CONTENTS :

PAGE No.

0. INTRODUCTION

With the advent of the 1990s new issues and terminologies are setting trends in the entire HVAC&R industry. Environmental authorities are diffusing new culture format in the industry with particular emphasis to substances that posses ozone depletion potential. With a result indoor air quality, air contaminants, clean air act, Chloroforocarbons (CFC) issues, safety codes etc. are now either being introduced or redefined. Changes on each of these issues have a direct effect on HVAC&R engineering, installation, operation, maintenance and replacements.

This Standard has been developed to help facilitate the requisites of building air conditioning, its selection procedures and system layout required to provide a comfort and healthy atmosphere in an air conditioning space. A simple approach to load calculation per ARI and ASHRAE procedures has been included as a ready reckoner for the design engineer.

In view of the Montreal Protocol which also addresses the production phaseout of CFC by Dec. 31, 1995, special importance has been provided towards the status of Refrigerants as described in Clause 10. However due to its continuous development changes, from HCFC to HFC to blending technique, the subject on Refrigerants are based on purely information gathering-stage complying to ANSI/ASHRAE Standard 34-1992.

1. SCOPE

This Standard covers the minimum requirement for practical approach towards, design, application and various methods of building air conditioning system together with relevant automatic controls.

System components are not individually described, but guidelines for selection procedure of equipment and supporting components are covered in this Standard. It also includes a section on "central chillers" to describe the importance on the combination of absorption and centrifugal chiller system in the HVAC&R industry.

This Standard shall be used, when required, in conjunction with relevant standard drawings.

This Standard does not cover the following subjects which are covered in relevant IPS as shown below:

a) Building Heating System (IPS-E-AR-100).
b) Venting, Ventilation and Pressurizing System (IPS-E-AR-160).
c) Humidification and Dehumidification System (IPS-E-AR-130).

2. REFERENCES

Throughout this Standard the following standards and codes are referred to. The editions of these standards and codes that are in effect at the time of publication of this Standard shall, to the extent specified herein, form a part of this Standard. The applicability of changes in standards and codes that occur after the date of this Standard shall be mutually agreed upon by the Company and the Consultant.

SMACNA (SHEET METAL AND AIR CONDITIONING CONTRACTOR'S NATIONAL ASSOCIATION)

SMACNA "Duct Fabrication Method"

ARI (AIR CONDITIONING AND REFRIGERATION INSTITUTE)

ARI 550-90 "Performance Rating of Chillers"
ARI 550-88 "Fouling Factor Limitation in the Evaporator and Condenser of Chillers"
ARI 575-87 "Method of Measuring Machinery Sound Within an Equipment Space"

ASHRAE (AMERICAN SOCIETY OF HEATING REFRIGERATING AND AIR CONDITIONING ENGINEERS, INC)

BSR/ASHRAE 135 P "Data Communication Protocols for Building Automation and Control Networks (BACnet)"

ANSI/ASHRAE 15-1992 "Safety Code for Mechanical Refrigeration"

ANSI/ASHRAE 34-1992 "Number Designation and Safety Classification of Refrigerants"

ANSI (AMERICAN NATIONAL STANDARDS INSTITUTE)

ANSI/ASME B31.5-1983 "Refrigeration Piping"

3. DEFINITIONS AND TERMINOLOGY

3.1 Air Washer

Unit for spraying or atomizing clean water into an air supply system; capable of heating, cooling, humidifying, or dehumidifying the air, depending on whether the water is heated or chilled.

3.2 Air-Conditioning System

Assembly of equipment for air treatment to control simultaneously its temperature, humidity, cleanliness, contamination, and distribution to meet the requirements of a conditioned space.

3.3 Azeotrope

A blend of two or more components whose equilibrium vapor phase and liquid phase compositions are the same at a given pressure.

3.4 Ceiling Diffuser

Round, square, rectangular, or linear air diffuser in the ceiling which usually provides a horizontal distribution pattern of primary and secondary air over the occupied zone and induces low velocity secondary air motion through the zone.

3.5 Central Fan Air-Conditioning System

System in which air is treated at a central plant and carried to and from the rooms by one or more fans and a system of ducts.

3.6 Chiller

1) Refrigerating machine used to transfer heat between fluids.

2) Complete, indirect refrigerating system of compressor, condenser, and evaporator with all operating and safety controls.

3.7 Comfort Air-Conditioning

It is the control of ventilation, air circulation, air cleaning, temperature, and humidity to provide maximum human efficiency, health, and comfort.

3.8 Conditioned Space

Space within a building provided with heated or cooled air, or both (or surfaces); and, where required, with humidification or dehumidification means, to maintain conditions for an acceptable thermal environment.

3.9 Design Load

It is the capacity required to produce specified inside conditions when specified outside conditions of temperature and humidity prevail, and when all sources of load are taken at the maximum that will then occur coincidentally.

3.10 Direct Digital Controls

Micro-processor based controller integrating automation and control in a stand alone or system network operation. It collects all the input and accumulates it in a computer which relays each individual output by digitally based controllers as required.

3.11 Heat Pump

Thermodynamic heating/refrigerating system to transfer heat. The condenser and evaporator may change roles to transfer heat in either direction. By receiving the flow of air or other fluid, a heat pump is used to cool or heat.

3.12 HVAC Systems

System that provides either collectively or individually the processes of comfort heating, ventilating, and/or air conditioning within, or associated with, a building.

3.13 Induction Unit

Air terminal device that delivers a small quantity of conditioned (primary) air through high-velocity jets, to induce a large quantity of room (secondary) air into the supply air stream. A heating coil may be located in the primary or secondary air stream. Primary air often is predominantly outside air. Units generally discharge directly through a grille into the space due to limited downstream static pressure capability.

3.14 Optimization

It is defined as an act, process or methodology of making something - design, systems or decision - as fully perfect, functional or effective as possible.

3.15 Psychrometric Chart

Graphical representation of the properties of moist air, usually including wet and dry bulb temperatures, specific and relative humidities, enthalpy, and density.

3.16 Relative Humidity

1) Ratio of the partial pressure or density of water vapor to the saturation pressure or density, respectively, at the same dry-bulb temperature, and barometric pressure of the ambient air.

2) Ratio of the mole fraction of water vapor to the mole fraction of water vapor saturated at the same temperature and barometric pressure.

3.17 Variable Air Volume

Variable air volume (VAV) system maintains room temperature by supplying a variable volume of constant temperature supply air, wherein

- the volume is modulated by a damper in response to room temperature;
- the damper can be powered by duct pressure powered controls, pneumatic controls, or electric controls.

3.18 Zoning

1) Division of a building or group of buildings into separately controlled spaces (zones), where different conditions can be maintained simultaneously.

2) Practice of dividing a building into smaller sections for control of heating and cooling. Each section is selected so that one thermostat can be used to determine its requirements.

4. UNITS

This Standard is based on International System of Units (SI) except where otherwise is specified.

5. BASIC DESIGN REQUIREMENTS

5.1 General

5.1.1 A reasonable and accurate requirement of air conditioning system shall be capable to promote physical well-being of human or provide improvement to industrial processes.

5.1.2 Temperature, relative humidity, motion of the air, and the temperature of the surrounding surfaces are important determining factors in the sensation of warmth and comfort because they directly influence the dissiption of body heat.

5.1.3 In industrial processes, the temperature and relative humidity of the air have a great deal to do with the rate of production, period of storage, and the weight, strength, appearance and quality of the product.

5.2 Design Load Factors

5.2.1 The following shall be specified as a basis for the calculation of design loads:

a) Design outside conditions (including temperature, relative humidity and other factors).
b) Design inside conditions.
c) Average number of occupants.
d) Other sources of substantial load from with-indoors when design conditions prevail.
e) Quantity of air assumed for ventilation.
f) Time of day at which design load from indoor sources is estimated to occur.
g) Class of activity assumed for occupants.
h) Size and physical characteristics of enclosure.
i) Quantity of air assumed for infiltration, including infiltration due to door usage.
j) Hours of operation.

5.2.2 Calculations of design loads (summer) should include the following sources of heat gain:

a) Conduction through physical barriers, such as walls, doors, windows, floors, ceilings, etc.
b) Heat from sunshine:
  1) Direct effect through glass areas.
  2) Additional conduction through opaque barriers, such as walls, roofs, etc., when these heat gains contribute to the design load.
c) Heat and moisture introduced by incoming outdoor air.
d) Heat and moisture liberated by occupants.
e) Heat and moisture liberated by appliances, office equipment, illumination, combustion, electric motors etc.

5.3 Design Outside Conditions

The design outside dry-bulb and wet-bulb temperatures, prevailing wind velocity including elevation and other factors for calculating cooling loads shall be not lower than those shown in Attachments 1-2 and 3 for the cities listed and for localities that are the same climatically. (The designer should exercise judgment to insure that results are consistent with expectations).

5.4 Design Inside Conditions

5.4.1 It is recommended that the design inside temperature and humidity for calculation of the cooling load should not be higher than 26.7°C (80°F) dry-bulb temperature and 50 percent relative humidity. The ventilation rate, permissible variations and control limits should also be inclusive.

5.4.2 For installations which have a load that is predominantly a "people" load, a slightly-lower design dry-bulb temperature, along with an increase in relative humidity, is permissible to attain a more economical equipment selection. Under these conditions, the design inside dry-bulb temperature may decrease to 25°C (77°F) and the design inside humidity may increase to 55 percent.

5.5 Solar Heat Gain

For factors on sensible heat gain transmitted through glass and absorbed solar energy, reference is made to ASHRAE Fundamental 1989 Guidebook Chapter 26.

5.6 Transmission Heat Gain

Transmission factors shall be used for estimating the heat gain or heat loss due to transmission through exterior walls, partitions, roofs, ceilings, and floors; and the heat loss due to transmission through windows. For the usual construction, wall transmissions are only a small proportion of the over-all load and, therefore, need not be treated as carefully as glass or roofs exposed to solar radiation.

5.7 Design Occupancy

The design-load calculations should be based on the stated average occupancy of the building during the time of maximum design conditions. The heat given off by each occupant should be calculated as not less than that in Table 1.

TABLE 1 - RATES OF HEAT GAIN FROM OCCUPANTS OF CONDITIONED SPACES a.b.c

Degree of activity	Typical Application	Total Heat Adults, Male, Btu/h	Total Heat Adjusted [d], Btu/h	Sensible Heat, Btu/h	Latent Heat, Btu/h
Seated at theater	Theater - Matinee	390	330	225	105
Seated at theater	Theater - Evening	390	350	245	105
Seated, very light work	Offices, hotels, apartments	450	400	245	155
Moderately active office work	Offices, hotels, apartments	475	450	250	200
Standing, light work; walking	Department store, retail store	550	450	250	200
Walking; standing	Drug store, bank	550	500	250	250
Sedentary work	Restaurant[e]	490	550	275	275
Light bench work	Factory	800	750	275	475
Moderate dancing	Dance hall	900	850	305	345
Walking 3 mph; light machine work	Factory	1000	1000	375	625
Bowling[e]	Bowling alley	1500	1450	580	870
Heavy work	Factory	1500	1450	580	870
Heavy machine work; lifting	Factory	1600	1600	635	965
Athletics	Gymnasium	2000	1800	710	1090

Notes:

a) Tabulated values are based on 24°C (75°F) room dry-bulb temperature. For (80°F) room dry-bulb, the total heat remains the same, but the sensible heat values should be decreased by approximately 20%, and the latent heat values increased accordingly.

b) Reference is also made to chapter 8 of ASHRAE 1989 Fundamental Guidebook.

c) All values are rounded to nearest 5 W.(5 Btu/h).

d) Adjusted heat gain is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85% of that for an adult male, and that the gain from a child is 75% of that for an adult male.

e) Adjusted total heat gain for sedentary work, restaurant, includes 18 W (60 Btu/h) for food per individual (9 W (30 Btu/h) sensible and 9 W latent).

f) For Bowling, figure one person per alley actually bowling, and all others as sitting (117 W 400 Btu/h) or standing and walking slowly (231 W- 550 Btu/h).

5.8 Heat from Appliances & Other Equipment

For recommended values of heat gain in Btu per hour, for a number of commonly-used appliances, heat gains from selected office equipment, hospital equipment and electric motors, reference is made to Attachments 4, 5, 6 and 7.

5.9 Ventilation and Infiltration

5.9.1 Ventilation

5.9.1.1 For Ventilation requirements the Table No. 8 mentioned in IPS-E-AR-160 shall apply. For maintaining positive pressure the quantity of air shall not be less than that drawn from the space by any exhaust fan that may be used.

5.9.1.2 The amount of outdoor air depends on the number of occupants and on the materials and apparatus that may give off odors within the space. Generally outdoor air for ventilation is introduced at the air conditioning apparatus rather than directly into the conditioned space, thus it becomes a cooling coil load instead of a space load component.

5.9.2 Infiltration

The infiltration through doors and windows shall be calculated per formula mentioned in ASHRAE Fundamental 1989 Guidebook and the larger of the heat gain for ventilation or the infiltration quantities shall be considered, but not both.

5.10 Air Motion in Conditioned Spaces

5.10.1 The air quantity and temperature of the treated air and the method of introducing it to the conditioned space shall be designed to limit to 1.6°C (3°F) or less, the simultaneous variation in dry-bulb temperature at the same level throughout that portion of a single room which is normally frequented by persons.

5.10.2 In spaces normally frequented by persons not moving about, it is recommended to avoid air velocities exceeding 0.25 m/s (50 fpm) in the zone between the floor and the 1.5 meter level.

5.10.3 Exceptions shall be made to the vicinity of a supply or return grille when the construction requires it to be located below the 1.5 meter level and in a space normally frequented by persons.

5.10.4 Noise level, odor level and indoor air quality shall be given due considerations in the equipment selection.

5.11 Safety Provisions

Comfort air conditioning systems shall be installed in accordance to safety requirements called for by relevant codes of OSHA and NFPA. The applicable code of ANSI/ASHRAE 15-1992 (Safety Code for Mechanical Refrigeration) shall also apply to this Standard.

5.12 Equipment Service Life

The selected equipment and system shall be governed by proper maintenance requirements such that life expectancy of the equipment meets minimum requirements of those recommended by ASHRAE as indicated in Attachment 8.

6. AIR CONDITIONING SYSTEM CLASSIFICATION

6.1 General

6.1.1 Varying air conditioning demands are met in residential, commercial, institutional, medical and office buildings. These demands are met primarily through the use of one of the systems mentioned below or through combination of these systems, wherein classification shall be based on type of refrigeration.

6.1.2 The coefficient of performance (COP) based on recent development addresses the performance of an air conditioning equipment, and can be determined by the following formula:

$$\mathbf{COP} = \frac{\mathbf{TR \times 12000}}{\mathbf{HP \times 2544\ Btu/HP}}$$

6.2 Packaged-Unitary System

6.2.1 Packaged direct expansion system

These are packaged self-contained units working on the principles of direct expansion refrigerating cycle, either single package for indoor mount or split-system requiring one indoor and one outdoor unit. These units either distribute the air through an integrated discharge plenum or circulate the air through fabricated sheet metal ducting or fiber glass duct-board.

Note:

For various ARI standard classification reference is made to IPS-M-AR-125. For window air conditioners or thru the wall room air conditioners reference is made to IPS-M-AR-245. Subject to the configuration of the computer system, computer room air conditioning also fall within this category.

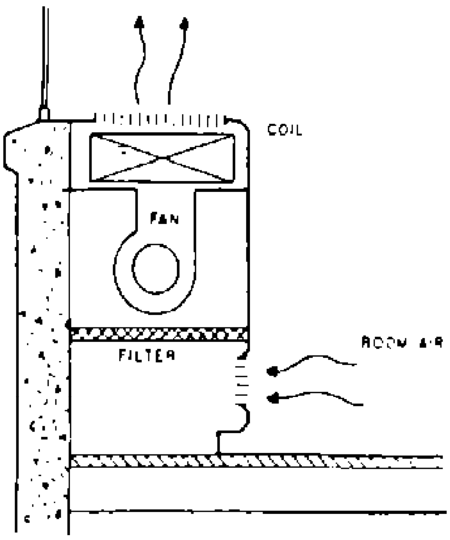
6.2.2 Unitary system

These are distribution systems where room units may be used in the central fan system and classified as non-refrigerant type, into three major areas:

a) All water and no air.
b) Part water and part air.
c) All air and no water.

6.2.2.1 All water and no air system

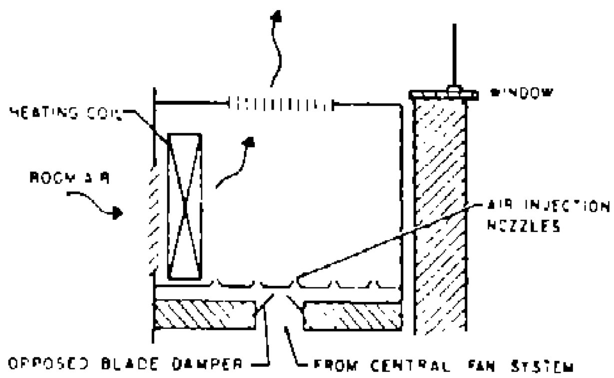
Commonly referred to as fan coil units located under perimiter windows, ceiling or wall suspended, controlled by room thermostat and/or motorized valves. Fig. 1 illustrates this system.



TYPICAL ALL-WATER NO-AIR UNITARY SYSTEM
Fig. 1

6.2.2.2 Part water and part air system

These system are frequently called induction systems, because part of the air is supplied from central fan (air handling) system which induces room air into the unit, whether high or low pressure. Fig. 2 illustrates this system. The water for the heating is supplied by the boiler through the heat exchanger, to the heating coil. (Ducting for air can be designed by static regain in high velocity ducts). These system are recommended for location under the window around the perimeter areas of multi-story multi-room buildings.

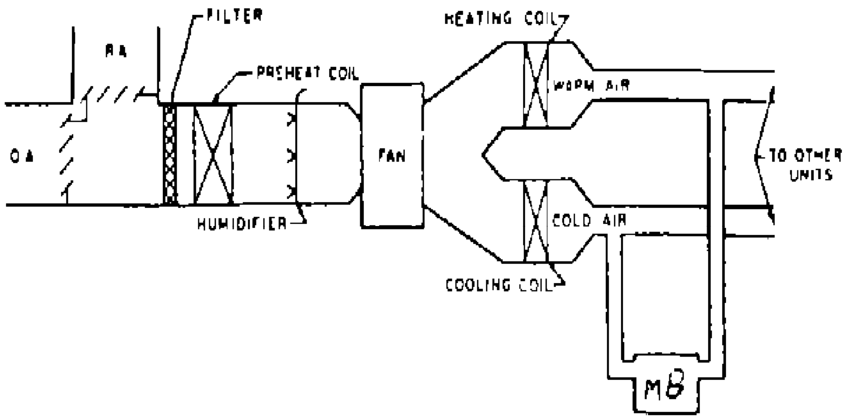


TYPICAL PART-WATER PART-AIR UNITARY SYSTEM
Fig. 2

6.2.2.3 All air and no water systems

6.2.2.3.1 These systems are a form of reheat system, condition the air at a central location (central fan system) and deliver the air through ducts. The system may be low/medium pressure, single duct or high pressure dual duct (representing hot and cold deck).

6.2.2.3.2 A variable air volume mixing box may be used for maintaining room conditions through control dampers/valves to proportion the amount of hot or cold air required at the mixing box. Fig. 3 illustrates the high velocity dual duct system.



TYPICAL ALL-AIR NO-WATER UNITARY SYSTEM
Fig. 3

6.3 Heat Pump System

6.3.1 General

6.3.1.1 A heat pump is a refrigeration cycle-either reciprocating or centrifugal-in which the cooling effect as well as the heat rejected is used to furnish cooling or heating to the air conditioning units either simultaneously or separately.

6.3.1.2 Any thermodynamic cycle that is capable of producing a cooling effect may theoretically be used as heat pump.

6.3.2 Advantages

The heat pump system operation has the advantages of:

- **a)** Over all first cost saving,
- **b)** space saving,
- **c)** nuisance elimination,
- **d)** single energy source,
- **e)** increased safety and,
- **f)** fire insurance rate reduction.

6.3.3 Heat pump types

Heat pumps for air conditioning service are further classified by:

a) Type of heat source and sink.
b) Heating and cooling distribution fluid.
c) Type of thermodynamic cycle.
d) Type of building structure.
e) Size and configuration.
f) Limitation of the source and the sink.

6.3.4 Unitary heat pumps

The two basic types of unitary heat pumps are:

a) Air-to-air: These equipment consists of factory matched refrigerant cycle components which are applied in the field to fulfill requirements of the user. These can be either in preassembled unit or as a split system.

b) Water-to-air: These use water as the heat source when in the heating mode and as the heat sink when in the cooling mode. The water supply may be closed water loop, a lake or a well (ground source).

7. METHODS OF FLUID AND AIR DISTRIBUTION

7.1 Methods of Distributing Fluid Media

7.1.1 General

7.1.1.1 In an air conditioning system, each coil, washer or heat exchanger are an arrangement of piping that will convey the cooling or heating fluid in an effective manner to the refrigeration machine.

7.1.1.2 When water is used, this arrangement shall be used with black carbon steel or galvanized pipes in the form of supply and return water piping along with the necessary pump to circulate the water through the piping and other equipment.

7.1.2 Pipe sizing

7.1.2.1 General

Pressure drop caused by Newtonian fluids often presented in head or specific energy form is described by the following Darcey-Weisbach equation:

$$\Delta h = f \frac{L}{D} \times \frac{V^2}{2g}$$

Where:

Δh = head loss, feet
f = friction factor, dimensionless
L = length, feet

D = Internal diameter, feet
V = average velocity, ft/s
g = acceleration due to gravity, ft/s^2

Note:

For further information, reference is made to ASHRAE 1989 Fundamentals, Chapter 33.

7.1.2.2 Sizing cold water pipes (open system)

1) Sketch the main lines, risers, and branches, and identify the fixtures to be served.

2) Compute the demand weight of the fixtures.

3) Determine the total demand in fixture units, and find the expected demand in L/s.

4) Determine the equivalent length of pipe in the main lines, risers, and branches.

5) Determine the average minimum pressure in the street main and minimum pressure required for the operation of the top most fixture. This should be between 55 kPa (8 psi) to 172 kPa (25 psi).

6) Using the equation in Clause 7.1.2.1, calculate the approximate pressure drop value in 100 ft.

7) From the expected rate of flow and pressure drop, size of pipes shall be determined as mentioned in the ASHRAE 1989 Fundamentals (Chapter 33).

7.1.2.3 Sizing chilled/hot water pipes (closed system)

The procedure recommended for sizing chilled and hot water system suitable for air handling units, fan coil units, radiators and plant room piping is indicated, as a ready reckoner, in the Pipe Size Chart mentioned in Table 2.

TABLE 2 - PIPE SIZING CHART

SIZE	350 ML"/FT FL→FAN COILS ΔT=10°F			250 ML"/FT FL→RADIATORS ΔT=20°F		
	GPM	FPS 3FT/100FT VELOCITY	MBH	GPM	FPS 2FT/100FT VELOCITY	MBH
½"	1.5	1.7	7.5	1.2	1.3	12
¾"	3.2	2.0	16	2.6	1.6	26
1"	6.3	2.3	31.5	5.0	1.3	50
1¼"	14	2.8	70	11	2.3	110
1½"	20	3.2	100	16	2.5	160
2"	39	3.8	195	32	3.0	320
2½"	64	4.3	320	52	3.4	520
3"	112	5.0	560	92	4.0	920
4"	240	6.2	1200	190	4.8	1900
5"	440	7.3	2200	360	5.8	3600
6"	750	8.3	3750	580	6.7	5800
8"	1600	10.2	8000	1300	8.2	13000
10"	3000	13.4	15000	2400	9.7	24000
12"	4800	13.6	24000	3800	10.8	38000

7.1.2.4 Steam pipe sizing

It shall be determined by the following procedures:

1) The initial pressure and the total pressure drop shall be calculated between the source of supply and at the end of return pipe.

2) The minimum velocity of steam allowable for quiet and dependable operation of the system (with consideration of the direction of condensate flow) shall be determined.

3) The equivalent length of the run from boiler or source of steam supply to the farthest heating units.

4) With the rate of steam consumption known, the pipe size shall be determined by referring to the chart of ASHRAE 1989 Fundamentals (Chapter 33).

7.1.2.5 Halocarbon refrigerant pipe sizing

a) For halocarbon refrigerant pipe sizing reference is made to Attachment 9 (for discharge and liquid lines) and Attachment 10 (for suction lines) of IPS-E-AR-180 standard.

b) The pipe types used for direct expansion system shall be either K or L copper pipes, suitable for interconnecting the complete refrigeration circuit as follows:

- The hot gas discharge line connecting the compressor to the condenser.
- The liquid line connecting the condenser to the liquid receiver.
- The liquid line connecting the liquid receiver to the cooling coil.
- The suction line connecting the cooling coil to the compressor.

7.2 Methods of Air Distribution and Circulation

Central fan or unit air conditioning distribute and circulate their air through:

a) Ducting, wherein the air is transmitted through any of the following:
- sheet metal ductwork;
- fiberglass ductboard;
- unplasticised PVC ducts.

b) Air outlet, devices, where distribution takes place through various grilles, registers and diffusers.
c) Discharge plenum which is generally integral part of a packaged unitary air conditioning equipment.

7.2.1 Ducting

7.2.1.1 The recommended practice shall be through rectangular fabrication on low velocity arrangement not exceeding 10 m/s (2000 FPM).

7.2.1.2 High velocity system shall be for 15 m/s (3000 FPM) and higher, wherein use of round ducts are recommended. These round ducts provide the following advantages:

a) Use of thinner gage material.
b) Exclusion of reinforcing braces.
c) Creates less turbulence.
d) Requires less head room.

7.2.1.3 The common method of air duct system design which are recommended for use are as follows:

a) Equal friction method

In this system the ductwork is sized for a constant pressure loss per unit length of duct. However higher airflow rates may require velocity limitations to prevent objectionable noise level. This method is the recommended practice for design engineers.

b) Velocity reduction method

This method consists of selecting the fan discharge velocity, then designing for progressively lower main duct velocities at each junction.

c) Static regain method

In this method the static pressure increase (static regain) at each take-off, offsets the pressure loss of the succeeding sections of ductwork. This method is specially suited to supply systems having long runs with many registers and diffuser located at take-offs. With this design procedure, approximately the same static pressure exists at the entrance of each branch, simplifying outlet selection.

d) Constant velocity method

This method is generally applied to exhaust systems.

7.2.1.4 The duct sizing (supply, return and exhaust) shall be conducted in accordance with AMCA-500 standard conforming to the latest issue of NFPA pamphlet 90 A (standards for installation of air conditioning and ventilating system).

7.2.1.5 Based upon specified design method the automated computerized duct sizing approach through Autocad are recommended for use. This procedure shall be capable to perform an analysis of trunk and runout lengths, fittings, runout air quantities and noise program. (The duct design method through Autocad shall use the methodology of the 1985 ASHRAE fundamentals and the noise control calculations shall be based on the 1981 HVAC system duct design published by SMACNA.)

7.2.1.6 Any of the following traverse joints used in ducting shall be considered:

- pocket or slip joint;
- flanged or 'S' slip joint;
- sleeve or drive slip joint.

Note:

For types of various joints and its application of use, reference is made to IPS-E-AR-160 and relevant ASHRAE guidebook.

7.2.1.7 The following recommended fabrication schedule for galvanized sheet metal ducting shall be used, where applicable:

DUCT WIDTH	THICKNESS	FLAT BAR SIZE (SLIP)	ANGLE IRON (FOR BRACING)	SPACING AT
Up to				
76 cm	0.6 mm (24 ga)	2.54 cm (1")	2.5 × 2.5 × 3 mm thick	4 Ft
107 cm	0.8 mm (22 ga)	1.54 cm (1")	3 cm × 3 cm × 3 mm	4 Ft
137 cm	0.8 mm (22 ga)	3.81 cm (1½")	4 cm × 4 cm × 3 mm	4 Ft
157-214 cm	1.0 mm (20 ga)	3.81 cm (1½")	4 cm × 4 cm × 3 mm	2 Ft
244 cm	1.20 mm (18 ga)	3.81 cm (1½")	4 cm × 4 cm × 5 mm	2 Ft
Above 245 cm	1.20 mm (18 ga)	3.81 cm (1½")	5 cm × 5 cm × 6 mm	2 Ft

7.2.1.8 Recommended minimum zinc coating requirement on each standard 1000 × 2000 mm galvanized sheets shall be per the following schedule:

0.26 kg/m^2 (0.85 ounce per sqft) for 0.5 mm thick sheets
0.315 kg/m^2 (1.05 ounce per sqft) for sheets 0.6 mm to 1 mm thick
0.375 kg/m^2 (1.25 ounce per sqft) for sheets 1.20 to 1.25 mm thick

7.2.1.9 Smooth transitions and long radius fittings with properly sized take-offs shall be provided for the duct system. The provision of splitter damper near each take-offs (branch line) shall be made where applications demand. All sharp 90° elbows shall be provided with turning vanes.

7.2.1.10 Duct insulation

7.2.1.10.1 Thermal insulation shall be able to retard the flow of heat energy and its materials can be fibrous sheet or monolithic, open or closed cell, or composite of these materials that can be chemically or mechanically bound or supported. It is provided for ducts to reduce heat leakage into the air passing to the conditioned area or to prevent condensation of moisture on the exterior of the duct. The need for insulation shall depend upon the peculiarities of each installation including provisions for fire protection.

7.2.1.10.2 The insulating material shall not cause moisture absorption and be used on all ducts running outside of building, on cool air supply ducts running through unconditioned spaces, and on ducts running through hot spaces such as boiler rooms.

7.2.1.10.3 Insulation is optional on supply ducts running through conditioned spaces, and not recommended for return or exhaust ducts.

Note:

For type and material specification of insulating materials, reference is made to IPS-M-AR-235.

7.2.2 Air outlet devices

7.2.2.1 The distributed air from the ducting are circulated to the space through a properly designed supply air outlets. Due consideration shall be given to throw, drop, rise and spread on air outlets' capabilities.

7.2.2.2 Each outlets, grilles, registers and diffusers shall have sponge rubber gasket around it and shall be installed with a moth-proof supportive wooden frame between the duct neck and air outlets.

Note:

For additional information, on air outlet devices, reference is made to IPS-E-AR-160.

7.2.3 Variable air volume (VAV)

7.2.3.1 General

7.2.3.1.1 The VAV system design can air condition both perimeter and interior spaces. The internal sensible cooling load in interior spaces when occupied remains relatively constant the year round, while the internal sensible load (people, light, solar and transmission) in perimeter spaces can require either heating or cooling depending upon the outside weather conditions and the internal load.

7.2.3.1.2 Conditioned air from the VAV terminal units can be supplied to multiple T-Bar slot air diffusers through flexible ductwork between the terminal unit and the diffusers. The diffusers are set over the T-Bar ceiling framework. Some manufacturers provide integral diffuser with their terminal unit (s). Control for conditioning an area is achieved by the terminal unit (s) regulating the amount of cool air entering the room through the T-Bar slot air diffuser (s). The terminal unit shall be equipped with motor controlled throttling damper arrangement activated by a room thermostat to admit more or less air to the conditioned space.

7.2.3.2 Types of airside system

7.2.3.2.1 Single supply air duct

The single supply air duct system (with single or multizone air handling unit) are typical for VAV, wherein the terminal unit modulates the supply air volume. In this system the refrigeration and air handling equipment are sized according to the building block load, typically 15 to 20% less than the sum of the peak loads. (For example one zone may be at peak load in the morning while the other may be at part load. The reverse holds true in the afternoon yet the same amount of air would be handled by the fan since the air not required by one zone would be used for another).

7.2.3.2.2 Double duct

The dual duct operates as one for cold air and another for hot air which are supplied through mixture terminal unit. This system is available in many variation but the common type supplies a constant volume of room supply air by mixing cold air and hot air, thus varying the supply air temperature in response to room temperature.

Ducting design shall be based on velocity over 15 m/s and over 76 mm water (over 3000 FPM and pressures above 3" water), where pressure and power requirements are greater than those of the conventional systems. As the duct velocity increases, duct friction and total pressure increases, requiring additional pressure at the fan. (Use of dual duct system are not recommended for Iran.)

Note:

The recommended duct sizing method can also be through low friction combined with static regain. This method is capable to maintain the static pressure relatively constant at all points in the system, thus facilitating balancing, increasing stability and providing for greater flexibility.

7.2.4 Methods of Filtering Air

7.2.4.1 Filtered air in air conditioning shall be used to maintain a clean atmosphere in the conditioned space. The concentration of contaminants in the air and the degree of cleanliness required in the conditioned space determines the type of filters required.

Note:

For additional information on type of filters, its selection method and air resistance capability, reference is made to IPS-E-AR-160.

8. CENTRAL HVAC SYSTEM

8.1 General

The central HVAC system generally represent an arrangement of equipment which includes air conditioning through means of refrigeration, one or more heat transfer units, room or space air terminal units, pumping units, air filtration assembly, a means of air distribution, an arrangement for piping the refrigerant and heating medium, and suitable controls to regulate the proper capacity and function of these components.

8.2 Refrigeration Machines

8.2.1 Common type of refrigeration machines, classified according to their type of operation are mechanical compression, absorption and vacuum. Mechanical compression machines may be divided into reciprocating, rotary and scroll types.

8.2.2 Methods for the measurement of unit sound levels shall be based on ARI Standard 575-87.

8.3 Central Chillers

The two central chiller unit commonly used on large commercial and industrial installations (preferably over 100 TR cooling capacity) are:

- Absorption water chillers (its condenser-evaporator represents refrigeration cycle and absorber-generator represents power cycle).

- Centrifugal water chillers (consists basically of a centrifugal compressor, a shell and tube cooler and a condenser).

8.3.1 Absorption chillers

8.3.1.1 The absorption machine which can be fired by natural gas, oil, steam or even waste heat, employs environmentally safe lithium bromide with anti-corrosive inhibitor in the cooling process has its cycle based on two principles:

a) Lithium bromide solution has the ability to absorb water vapor.
b) Water as refrigerant, will boil,or flash cool itself, at low temperatures when it is subjected to a high vacuum.

8.3.1.2 A direct fired preferably gas absorption-type chilled hot water generator (also called chiller-heater assembly) single or dual effect factory assembled tested and installed on a rugged steel base is a recommended updated technology to be used on all large and major installations. While providing round-the-year air conditioning with one unit it saves machine room space, requires lower initial investment, prevents ozone depletion problem, reduces peak power consumption and costs for electricity, operation and maintenance are low.

8.3.1.3 To provide proper operation and eliminate hazards, a suitable microprocessor technology shall be used. These should preferably be supplied with microprocessor-based burner control, lithium bromide control to prevent crystallization, a bearing-wear monitoring system on pumps with variable speed drives.

8.3.2 Centrifugal chillers

a) These chillers have compressors which are not constant displacement type; it offers a wide range of capacities continuously modulated over a limited range of pressure ratios.

b) By altering built-in design items (including number of stages, compressor speed, impeller diameters, and choice of refrigerant), it can be used in liquid chillers having a wide range of design chilled liquid temperatures and design cooling fluid temperatures providing much faster cooling requirement for space than absorption machines.

c) Its ability to vary capacity continuously to match a wide range of load conditions with nearly proportionate changes in power consumption makes it desirable for both close temperature control and energy conservation. Its ability to operate at greatly reduced capacity makes for more on-the-line time with infrequent starting.

d) Its capacity can be varied to match the load by means of constant speed drive with variable inlet guide vanes or suction damper control, or a variable speed drive with the suction damper control.

e) These machines are available in water cooled or air cooled, and classified by either open or closed type of compressor unit, operating with either direct or external power sources through electric, steam or diesel powered operation. Generally the evaporators operate flooded and the flow control chamber acts as the expansion device.

8.3.3 Combination centrifugal-absorption system

8.3.3.1 In choosing an air conditioning system of large tonnage, it is common practice to settle upon either a steam absorption system or an electric or steam driven centrifugal chiller. However by using a combination of absorption machine and steam turbine drive centrifugal chillers, it may be possible to obtain a more efficient system.

8.3.3.2 The combination system provides lower steam cost and operating cost. Also it has lower heat rejection rate than low pressure absorption systems, requiring smaller cooling towers, pumps, piping and wire sizes for the units electrical power. The centrifugal chiller can be electric powered. When steam powered full load steam rates of 10 to 12 lbs/hr/ton can be obtained with 896 kpag (130 psig) supply steam.

8.3.3.3 This combination for cooling load 700 ton of refrigeration and above shall be applied on ratio of 1:2 respectively (one centrifugal and two absorption), for providing lower operating costs modulated through control of steam and chilled water circuits.

8.3.3.4 It is recommended that with one centrifugal and one absorption combination system, the piping for chilled water circuit should be arranged in a series fashion where all the water flow through the absorption chiller first and the centrifugal chiller second. On such cases the cooling water for the condensers shall be piped in parallel.

8.3.3.5 In the event one centrifugal is to be combined with two absorption chillers, arranging of chilled water circuits in and out of the combination shall affect the steam circuit and flow. The principle recommended arrangements are as follows:

a) Water in and out, in parallel with each of the three machines handling separate load. This arrangement is seldom used.

b) Water in and out in parallel but with a common load and common piping connections. This arrangement is frequently used.

c) Water in and out of the absorption unit in parallel, then in and out of the centrifugal in series. This arrangement is commonly used.

d) Water in and out of all units in series. This method is not in common use.

8.3.3.6 Where stand-by applications are required, it is recommended that capacity of each of the three machines be evenly split into one-third of the total load. Therefore should the centrifugal chiller be shut down for some reason, the remaining units could still handle two-third of the total load.

Note:

For special applications, provision of stand-by facilities shall be given due consideration.

8.4 Air Cooled Versus Water Cooled System

8.4.1 General

8.4.1.1 Depending on economic and operating advantages, the selection of air cooled or water cooled system shall be carefully evaluated. In places where water is insufficient or expensive, where a high ambient wet bulb temperature exists, where cooling water pumping or water cooling costs are uneconomical, water cooled systems are not recommended.

8.4.1.2 The air cooled system is applicable where the ambient dry bulb temperature is below the desired condensing temperature. On sites with abusive conditions the air cooled condenser specification must be protected with specific metals and control requirements, other than standard.

8.4.1.3 In general, air-cooled units are suitable choice for wet bulb temperatures above 26.7°C(80°F) and water cooled temperatures are suitable choice for high ambient dry bulb with low relative humidity.

8.4.1.4 Since both system are exposed to sun, wind, storm, rain, dust etc., a suitable site location and structure away from any hindrances shall be considered for economical operation.

8.4.2 Condensing mediums

Any one of the following condensing medium shall be considered for refrigeration equipment:

I) Water-cooled based on well or city water. A wet bulb temperature not less than the design outside wet bulb shall be considered for the selection of cooling towers.

II) Air-cooled, where the condenser air temperature shall not be less than the design dry bulb.

III) Evaporative condenser cooling, where the condenser shall be selected for a wet bulb temperature not less than the design.

8.4.3 Economic comparison

8.4.3.1 Economic comparison for the cooling tower should include the initial cost, make-up water facilities, wind velocity, blowdown facilities, piping length, water treatment, and the overall available power supply.

8.4.3.2 The application evaluation on either air cooled or water cooled system shall depend upon, but not limited to, the following conditions:

a) Ambient dry bulb and wet bulb temperatures.
b) Quantity and quality of available water.
c) Available space for installations.
d) Component replacement facilities.
e) Operating pressure limitations.
f) Wind and storm velocity.
g) Initial and shipping costs.
h) Noise environmental vibration limitations.
i) Maintenance and operating costs.
j) Site corrosive conditions.

Note:

On both systems the fabrication capabilities in Iran shall be given due consideration and a feasibility chart shall be compiled for optimization, to try and make the designed system fully perfect, functional and effective, cost and efficiency wise.

8.5 Chilled/Hot Water Piping

8.5.1 It is used in large air conditioning systems where chilled/hot water flows through black carbon steel pipes providing the cooling media from the chillers through the pumps to the terminal units and air handling equipment.

8.5.2 The following steps shall be considered in the design of the chilled/hot water piping circuit:

1) Local building codes and ordinances shall be studied and complied with.

2) Shut-off valves shall be provided at all individual pieces of equipment to enable normal servicing of unit without draining system.

3) For remote room unit installations, the reversed return piping system shall be used to secure an inherently balanced system. A properly designed constant flow control valves can be used in lieu of reverse return system.

4) A dirt leg and blow-off valve shall be provided on both supply and return risers.

5) All high points in the piping circuit shall be properly vented to prevent air lock in the lines.

6) In systems designed for heating and cooling, pipe expansion caused by the change in water temperature shall be provided for.

7) Where noise is a factor, it is recommended that water velocities in pipes up to 2" size should not exceed 1.25 m/s (four feet per second), and maximum 3.10 m/s in pipes from 2½" through 8" including headers.

8) The total friction loss of the piping circuit shall be carefully calculated to determine the pumping head.

9) Where individual unit flow control or shut-off is to be provided, by-pass or three-way valves shall be used to limit system water pressures and provide adequate flow-rate through the water cooler.

10) A properly designed closed or open type expansion tank shall be provided for the system.

11) All supply and return chilled/hot water piping shall be insulated to prevent condensation.

12) A strainer of at least 40 mesh per square inch is recommended to be installed in the piping circuit at the pump inlet.

13) Thermometers and pressure gages shall be strategically located to aid in start-up and test work and normal service checks. Necessary tappings for testing, adjusting and balancing (TAB) procedures shall be provided.

14) The maximum pump pressure or the maximum pressure created by a static head should not exceed the design working pressure of the water cooler or the maximum pressure ratings of the accessories.

8.6 Water Treatment

8.6.1 General

8.6.1.1 For proper and efficient operation of a central HVAC system, adequate water treatment facilities shall be provided for condenser water and chilled water circuit. (For further information, reference is made to ASHRAE 1987 Guidebook, Systems and Application, Chapter 53).

8.6.1.2 Besides chemical water treatment, modern integrated circuitry and signal processing to produce a deionizing effect for increasing the solubility of minerals in the liquid can be used.

8.6.1.3 A fouling allowance of 0.044 m^2.K/kW is included in manufacturer's rating based on ARI Standard 590-92.

8.6.2 Chilled water treatment

Limescale causes fouling in the chiller tubes, reducing efficiency, increasing use of energy and causing high head pressure. It is recommended that a closed system treatment be added to the chilled water, preferably nitrite-type inhibitor which does not effect the pump glands, water seals, chiller tubes or other types of commonly used material.

8.6.3 Condenser water treatment

The condenser water circuit shall be either an open one in the case of a cooling tower, or a once through circuit in the case of well water. Treating well water is not practical, but the water should be analyzed for its scale forming potential, including these caused by water from the chillers draining down through the polyplastic fill. If this analysis indicates that carbonate or other scale build-up on the inside of the tubes can be expected, the condenser shall be selected on the basis of a 0.176 m^2 °C/kW (001 ft^2/°F/h/Btu) fouling factor.

Notes:

1) The water treatment for the cooling towers and air handling units (even domestic water supplies) shall be capable to control and prevent legionnaires (pneomonic) disease caused through the bacteria in the water which lurks in the atmosphere (creating devasting illness and death). Such water has been shown to be a common habitat for this organism, and research by ASHRAE aimed at improving control and preventive measures are being conducted.

2) To provide saving in energy, water,chemical treatment and maintenance on large installation, the suction and the pressure side of the pump (on chilled/hot water and condenser water circuits) shall be incorporated with suitable sized centrifugal-action filtration system (separators) to remove particles as small as 74 microns for one single passage, and 44 microns if two separators are used.

9. EQUIPMENT SELECTION GUIDELINES

9.1 General

9.1.1 The calculated net capacity shall be based on individual manufacturer's tables, charts and performance curve conforming to certified ratings of ARI, AHAM, AMCA, CTI, NEC, etc. Necessary correction factors shall be applied wherever deemed essential.

9.1.2 In selecting types of relevant equipment, careful consideration shall be given to its feasibility of usage, initial cost, available facilities such as necessary spare parts and performance guarantee in Iran and the owning and operating cost.

9.1.3 Since each manufacturer have different selection procedures, hence the overall selection method shall be based on procedures outlined by individual manufacturer. The procedures mentioned in this Standard shall therefore be considered as recommended guidelines.

Note:

The description on selection procedures covered in this Standard are based on conventional method suitable for locations devoid of computer facilities. Computation methods shall be used wherever computer facilities are available.

9.2 Liquid Chiller Selection

9.2.1 General

9.2.1.1 These packaged water chillers operate with broad class of positive displacement compressors such as:

a) Reciprocating (open, hermetic or semi-open type).
b) Rotary (rotating vane, scroll or rolling piston type).
c) Helical screw type.

9.2.1.2 The packaged water chiller may be available in any one of the following types:

1) Complete with water cooled condenser.
2) Complete with integrated air cooled condenser.
3) For use with remote air cooled condenser up to maximum 62.8°C (145°F) condensing temperature.

9.2.2 Selection steps

9.2.2.1 Chilled water temperature leaving machine

Chilled water temperature is selected during system design. A considerable range of choice exists for comfort conditioning applications. A higher chilled water temperature permits a more economical machine selection but generally results in more expensive airside equipment. On the other hand, low chilled water temperature allows a higher water temperature rise thorough airside coils, less water is pumped and pump horsepower and piping costs are reduced.

9.2.2.2 Chilled water quantity and range

9.2.2.2.1 Refrigeration capacity and chilled water range fix quantity of water to be circulated in accordance with the equation:

$$\text{Chilled Water } M^3 / hr = \frac{\text{kcal / hr (cooling effect)}}{1000 \times \text{chill water range (°C)}} \quad \text{or}$$

$$GPM = \frac{\text{Btuh (cooling effect)}}{500 \times \text{chill water range (°F)}}$$

9.2.2.2.2 A system should be designed for constant water flow through the cooler. Either three way control valves or two way control valves at the cooling coils with an automatically controlled pump bypass shall be used.

9.2.2.2.3 Careful consideration must be given in selecting design ambient. For instance, air temperature above a roof is frequently 3°C above recorded design dry bulb temperatures.

9.2.2.2.4 The minimum outside air temperature at which the system will be operated must also be determined. Air cooled condensers used with chillers must always be provided with discharge damper head pressure control.

9.2.2.3 Fouling factors

9.2.2.3.1 Rating tables shall include an allowance for a 0.0005 fouling factor in the cooler and water cooled condenser. The chilled water circuit is closed and there should be no need to increase the fouling factor for the cooler. However, if a 0.001 cooler fouling factor is desired, multiply manufacturer’s table capacity by 0.97 and table kilowatts by 1.03, or follow manufacturer’s instructions.

9.2.2.3.2 When well or river water is used for the condenser, a 0.001 condenser fouling factor may be desirable. In this case, multiply manufacturer’s table capacity by 0.97 and table KW input by 1.03 or follow manufacturer’s instructions.

9.2.2.4 Interpolation

Manufacturer’s generally permit interpolation within their published ratings but extrapolation is not permitted.

9.2.2.5 Condenser water range

A 5.5°C condenser water range is generally considered as the best compromise between the most economical cooling tower and chiller selection. Exact flow rates for a temperature range are given in manufacturer’s rating tables at each selection point providing m^3/hr (gpm) and capacity multipliers for different ranges.

9.2.2.6 Condenser temperature and head pressure control

9.2.2.6.1 Entering condenser water temperature is fixed by climatic conditions or temperature of available water sources. If a cooling tower is used condenser water 4°C above ambient wet bulb is obtainable. In many parts of the world where design wet bulb is 25.5°C a properly sized tower will provide 29.5°C condenser water.

9.2.2.6.2 For proper operation of a water cooled chiller it is necessary to maintain a leaving condenser water temperature no lower than 30°C.

This means that condenser water range and entering water temperature may require control at part load conditions. Where condenser water is supplied from a cooling tower, the most commonly used method of control is to cycle the cooling tower fan when the outside temperature is below design conditions in order to maintain satisfactory entering water temperature.

9.2.3 Selection procedures

9.2.3.1 Water cooled

9.2.3.1.1 Establish machine requirements as follows:

- cooling capacity in KW (TR);
- chilled water temperature leaving cooler;
- chilled water range;
- chilled water pressure drop;
- chilled water quantity;
- cooler fouling factor;
- water temperature entering condenser;
- water temperature leaving condenser;
- condenser fouling factor;
- electrical characteristics.

9.2.3.1.2 Select and determine from manufacturer's rating tables:

- unit capacity, chilled water flow, brake horse power;
- cooler and condenser pressure drop.

9.2.3.2 Air cooled

9.2.3.2.1 Establish machine requirements as mentioned in Clause 9.2.3.1.1 including maximum and minimum outside temperature at which system will run.

9.2.3.2.2 Select and determine from manufacturer's rating tables per Clause 9.2.3.1.2.

9.3 Absorption Chiller Selection

9.3.1 General

9.3.1.1 The absorption water chiller and the mechanical compressor water chiller accept heat to evaporate a refrigerant at low pressure in the evaporator, thereby creating cooling effect. They condense the vaporous refrigerant at higher temperature in the condenser.

9.3.1.2 The absorption and compression cycle function similar, taking low pressure refrigerant vapor from the evaporator and delivering high pressure refrigerant vapor to the condenser. The only difference being in the method of transporting the vapor from the low to high pressure side.

9.3.1.3 In an absorption machine lithium bromide (LiBr) salt solution is the agent and water acts as refrigerant.

9.3.1.4 The two basic limitation in the absorption chiller application is the temperature limitation of leaving chilled water being above 4°C (40°F) and the critical balance of solution concentration in the cycle.

9.3.1.5 The fouling factor requirements in the evaporator and condenser shall be based on ARI standard 560-92.

9.3.2 Selection procedures

9.3.2.1 Since each manufacturer have different selection procedure for steam and hot water absorption machines, hence selections shall be made per steps outlined by individual manufacturers.

9.3.2.2 The following data shall be provided to the manufacturer for proper unit selection.

1. Job name/Location	
2. Application	Air conditioning, Industrial process etc.
3. Quantity	Number of units
4. Cooling capacity	UST or Kcal/h
5. Chilled water inlet temp.	°C
6. Chilled water outlet temp.	°C
7. Cooling water inlet temp.	°C
8. Cooling water outlet temp. or flow rate	°C or m'/h
9. Heat source : Steam-pressure	Kg/cm'G (PSIG)
Heat source : Steam-consumption	Kg/hr (Lbs/hr)
Heat source : Hot water-temperature	°C
Heat source : Hot water-Flow rate	m'/hr
10. Operation of control valve	Electric or Pneumatic (Pressure)
11. Electricity for controls/pump motors	Voltage Hertz, Phase
12. Installation location	Indoor or outdoor etc.
13. Nozzle arrangement	
14. Options or special features	

9.3.3 Application limitations

Design parameters for using absorption chillers shall be based on following limitation.

1) Min. leaving chilled water temperature 4.5°C (40°F)
2) Max. entering cooling water temperature 34°C (93.2°F)
3) Max. inlet steam pressure at the generator 1.5 kg/cm^2 G
4) Min. inlet steam pressure at the control valve 0.4 kg/cm^2 G
5) Max. inlet steam temperature at the control valve 170°C (338°F)
6) Max. leaving condenser water temperature 40°C (104°F)

Notes:

1) Some manufacturers' absorption machines may be capable to operate without a cooling tower bypass valve up to minimum 15°C entering condenser water temperature.

2) It is recommended that the chilled water leaving temperature be limited between 6.7°C (44°F) to 7.8°C (46°F)

9.4 Centrifugal Chiller Selection

9.4.1 General

9.4.1.1 From a thermodynamic stand-point the centrifugal refrigeration cycle is identical to the vapor compression system operating with reciprocating compressor with only difference in means of compressing the refrigerant.

9.4.1.2 The centrifugal machine is a variable displacement with one or more impellers, spinning in specifically formed housings that impart centrifugal force to the gas. The velocity energy resulting from this centrifugal force is then converted to pressure.

9.4.1.3 They handle large volumes of refrigerant gas and are commonly available in capacities over 100 tons of refrigeration. Its compressor uses treated halocarbons refrigerants with varying physical properties.

9.4.1.4 The fouling factor in the evaporator and condenser shall be based on ARI Standard and the performance in these machines shall be rated in accordance to ARI 550-92 and service access shall be per ANSI/ASHRAE 15-1989, and NFPA 70 (NEC). The factory insulation on the evaporator shell shall conform to UL standard 94 classification 94 HBF.

9.4.2 Selection procedures

a) Establish cooling capacity in kcal/hr and chilled water range in cubic meters per hour to be circulated in accordance with the following equation. (The system shall be designed for constant water flow through the cooler).

$$\textbf{Chilled Water } M^3 / hr = \frac{\text{kcal / hr (cooling capacity)}}{1000 \times \text{chill water range (°C)}} \quad \textbf{or}$$

$$\textbf{GPM} = \frac{\text{Btuh (cooling effect)}}{500 \times \text{chill water range (°F)}}$$

b) Condenser water temperature entering machine shall be determined. This is done by site ambient temperature and available water sources such as city, wells or rivers. When using cooling tower, the entering condenser water temperature shall be minimum 4°C above ambient wet bulb temperature.

c) The condenser water range shall be calculated according to equation mentioned in item (a) above.

Note:

Experience has shown that a 5.5 to 6°C condenser water range is generally recommended for obtaining most economical cooling tower and machine selection.

d) From manufacturer's table determine fouling factor allowance. Unless special conditions exist a fouling factor of 0.0001°C/hr/m^2/kcal is adequate for both cooler (being a closed circuit) and condenser when city water is used.

e) It is recommended that for every one point increase in the fouling factor for the cooler, the desired machine selection shall be based on 1°C below actual (design) chilled water temperature and for the condenser on 1.5°C above actual design condenser water temperature.

f) From manufacturer's tables, select number of condenser and cooler passes within the calculated chilled (item a) and condenser (item c) water range. Maximum number of passes provide for greater efficiency.

g) From manufacturer's chart check the pressure drop in the cooler and condenser within velocity limitation in (m/s) tube velocity.

h) Interpolation between chilled water temperature, condenser water temperature and chilled water range are generally permissible.

9.5 Air Handling Unit Selection

9.5.1 The proper selection of an air handling unit is conducted through the conditions entered in the psychrometric chart.

9.5.2 For the selection of these units the air volume, ambient conditions, inside design conditions, total load and sensible cooling load shall be available for coil selection. The sensible load shall be divided by total load to obtain the sensible heat factor.

9.5.3 The following known data shall be furnished to the manufacturer:

I) Operating conditions:

- Air inlet conditions:	DB/WB
- Air outlet conditions:	DB/WB
- Chilled water entering temp.:	°C/°F

II) Capacity requirements:

- Total air volume (circulated plus fresh air)	m^3/hr(cfm)
- Fan total static pressure	"WG
- Cooling coil capacity	Kw (MBH)
- Chilled water flow	L/S (USGPM)
- Fan motor	BHP

III) Correction factors (Derivation of air constants if other than sea level).

9.5.4 The air handling unit shall conform to certified ratings of ARI Standard 430. The coils used in conjunction with the air handling unit shall be certified in accordance with ARI Standard 410.

9.6 Fan Coil Selection

9.6.1 The required room sensible cooling capacity is usually the basis for selection of a fan coil unit, however total cooling capacity should be checked to ensure that the selection will meet all conditions of service.

9.6.2 For cooling, the unit is usually selected at fan high speed, however except for extreme conditions a unit so selected shall meet room conditions at normal or slow fan speed.

9.6.3 The manufacturer's cooling capacity tables allows for interpolation between flow rates, water temperatures and air temperatures, but extrapolation are not recommended.

9.6.4 From capacity tables, multiply the rating obtained from high speed table by cooling capacity multiplier to obtain ratings at other speeds. Manufacturer's tabulation also provides fan capacity in cfm at various speeds and external static pressure.

Note:

To prevent motor overload, a minimum static pressure loss in filters, grilles, plenum etc. are generally foreseen by the manufacturer.

9.6.5 The data required for proper selection shall be as follows:

- entering air temperature;
- available water temperature;
- sensible load;
- total load;
- type of discharge and mountings.

9.7 Air Induction Unit Selection

9.7.1 General

9.7.1.1 In an air induction terminal units air and water are two medias used to handle the total air conditioning job to be done.

a) Air

Only a relatively small volume of air is used. This air, called Primary air, provides ventilation, dehumidifying or humidifying capacity, plus the motive power at the room terminal unit for effective distribution in the conditioned space.

b) Water

The water flowing through the terminal unit coil does the sensible cooling job-that of offsetting sun load, plus heat generated within the space by people, lights, or other internal sources.

9.7.1.2 Central apparatus

Centrally located equipment provides the air-induction units with air and water at the proper conditions. Since air is distributed at relatively high velocity, a minimum of space is required for the duct distribution system.

Notes:

1) Maintenance is centralized remotely from the conditioned spaces since the air-induction units contain no moving parts.

2) For further information on induction units and its capacity curve on cooling-heating functional mode, reference is made to relevant ASHRAE Application Guidebook.

9.7.2 Selection procedure

The following selection procedure is recommended:

1) System parameters shall be established as follows:
- **a)** Application.
- **b)** Total cooling load.
- **c)** Ventilation requirement.
- **d)** Maximum nozzle pressure.
- **e)** Room temperature.
- **f)** Primary air temperature.
- **g)** Entering water temperature-cooling.
- **h)** Entering water temperature-heating.
- **i)** Gpm.

2) For cooling selection

a) Calculate the primary air cooling capacity.

b) Required cooling capacity of the coil (by subtracting the primary air cooling capacity from the total cooling load).

c) Capacity curve between entering air and entering water.

d) Select the smallest unit (normally with a 1-row coil) which will meet the requirement of capacity, nozzle pressure and noise level.

e) Determine the secondary water pressure drop.

3) For heating selection

Proceed with the above procedure and calculate heating capacity by multiplying the cooling curve capacity by the rates of delta T (heating - between entering air and entering water which is generally maintained at 13.9°C (25°F) delta T).

4) For gravity heating

Adjust the hot water temperature during unoccupied periods of space.

Note:

Reference is made to individual manufacturer's capacity curves for proper selection procedure.

9.8 Cooling Tower Selection

9.8.1 General

Psychrometry is a subject relating to the measurement of atmospheric conditions, and in particular, the moisture content of air. Since most of the heat lost by the water in a cooling tower is absorbed by direct contact with ambient atmospheric air, some knowledge of psychrometry and thermodynamic is desirable.

9.8.2 Selection Procedures

The data required for the cooling tower selection shall be as follows:

a) Ambient wet-bulb temperature	(W.B.T)	°C (°F)
b) Circulating water flow rate	(L)	LPM (GPM)
c) Outlet water temperature	(C.W.T)	°C (°F)
d) Inlet water temperature	(H.W.T)	°C (°F)
e) Heat rejection load (capacity)	(Q)	kcal/hr (BTU/hr)
f) Noise allowance		dB (A)
g) Electrical characteristics	cycle × voltage × phase	
h) Location		
i) Water quality		

9.9 Pump Selection

9.9.1 Types

Common types of centrifugal pumps used in the HVAC&R industry are:

a) Close coupled end suction pumps.
b) Base mounted end suction pump with flexible coupling.
c) In-line pumps or circulators.
d) Vertical or horizontal split case single or multi - stage.
e) Vertical or horizontal single or double - stage turbine pumps.

Notes:

1) All centrifugal pumps shall be with volute casing.
2) The number of impellers indicate the stage characteristic of a pump.
3) For construction description of pumps reference is made to IPS-M-AR-225.

9.9.2 Selection procedures

9.9.2.1 For proper selection of pumps the following information shall be required:

- Maximum flow in system.
- System head at maximum flow.
- System operating pressures and temperatures.
- Pump environmental conditions including ambient temperature.
- Electrical current characteristic and RPM.
- Electrical service starting limitations.
- Special electrical control.
- Location
- Water quality.

9.9.2.2 Pumps shall be selected from manufacturer's performance curves on 50 Hz operation. The prime objective in the selection approach shall be efficiency, quiet operation, lowest initial and operating costs and close conformance to actual needs.

9.9.2.3 The selection of pumps shall take into consideration the changes of flow in the system and the point of operation of a pump on its head-capacity curve.

9.10 Cooling Coil Selection

9.10.1 General

9.10.1.1 Coil circuiting are based on air and water counterflow where water enters on the leaving air side and leaves on the entering air side. These must be properly installed in order that ratings are met.

9.10.1.2 Coil circuiting should be selected to produce a water velocity as high as permitted by pressure drop limitations. A pressure drop of 6 to 10 meters of water is generally not considered excessive.

9.10.1.3 Entering water temperatures should be at least 3 to 4°C below leaving air dry bulb. In order to keep required system GPM to a minimum, it may be desirable to design for a water rise through a coil in excess of 6°C. Water temperature rise can be varied from one coil bank to another within a system for best overall performance.

9.10.1.4 Normal air face velocities used are between 2 to 3 m/s (400 and 600 fpm). Moisture carry over depends upon three factors-fin spacing, face velocity, and the degree of condensation. Since the degree of condensation varies with entering air conditions, it is advisable to select a conservative face velocity so that carryover does not become a problem regardless of entering air conditions.

9.10.2 Selection procedures

9.10.2.1 To select a coil, the following information must be determined from system requirements:

a) Air quantity to be cooled.
b) Entering air wet bulb and dry bulb temperature.
c) Total cooling load to be handled by the coil.
d) Entering water temperature.
e) Water temperature range (outlet minus inlet).
f) Maximum allowable face velocity across coil in m/s (fpm).
g) Coil pressure loss
h) Water flow rate

9.10.2.2 Select coil size from manufacturer's table by determining approximate face area required and space limitations.

9.10.2.3 Calculate required Kw (mbh) per square meter (square feet) of face area.

9.10.2.4 Calculate water flow rate and determine sensible heat capacity of coil, air resistance through coil and water pressure drop.

9.11 Heating Coil Selection

9.11.1 General

9.11.1.1 Like the cooling coil, heating coils are based on air and water counterflow. Therefore these must be properly installed in order for the ratings to be met.

9.11.1.2 Coil circuiting should be selected to produce a water velocity as high as permitted by system pressure drop limitations.

9.11.1.3 Water temperature should be as high as practical to permit a greater water temperature drop and reduced water flow. While a 20°F drop is often used a 40°F drop will halve gpm with negligible effect on coil surface. Sometimes water temperature is a function of the outdoor temperature and is changed by the control system as outdoor temperature varies.

9.11.1.4 Recommended face velocities are between 3.5 to 4.5 m/s (700 to 900 fpm). Higher velocities may be used keeping in mind airside static pressure drop limitation.

9.11.2 Selection procedures: (hot water)

9.11.2.1 To select hot water heating coil, the following information must be determined from system requirements:

a) Quantity of standard air to be heated.
b) Entering air db temperature.
c) Leaving air db temperature.
d) Entering water temperature.
e) Water temperature range or drop.
f) Desired coil face velocity in m/s (fpm).
g) Heating load
h) Dimension limitation

9.11.2.2 Clauses 9.10.2.2 through 9.10.2.4 shall apply.

9.11.2.3 Determine total water pressure drop by adding water pressure drop through tubes from manufacturer's table. Apply correction factors where required.

9.11.3 Selection procedures: (steam)

a) To select a steam coil, the steam pressure, entering air temperature, face velocity and required minimum leaving air temperature must be known. If heat load is given instead of leaving air temperature, use the following relationship:

$$\textbf{Quantity of air} = \frac{\textbf{Heating load}}{\textbf{1.08} \times \textbf{Temperature range}}$$

b) Obtain adjusted air temperature rise for actual steam pressure and entering air conditions:

$$\textbf{Adjusted temperature rise (TR)} = \frac{\textbf{Specified TR}}{\textbf{Factor from chart}}$$

c) Pick coil fin spacing from manufacturer's chart.
d) Obtain condensate rate by taking latent heat from manufacturer's load and dividing it into heat load.
e) Find air resistance from manufacturer's table.

9.11.4 Altitude corrections for coils

9.11.4.1 Manufacturer's data and calculations are based on standard air, that is 1.2 kg/m^3 at 21°C (0.075 lbs per cubic feet at 70°F). Conditions at other than those for standard air will give incorrect results unless proper corrections are made.

9.11.4.2 Normally it is not considered necessary to correct conditions for altitudes under 606 meters (2000 feet). Refer to manufacturer's altitude and temperature correction chart to obtain actual coil capacity at altitude conditions.

9.12 Condensing Unit Selection

Following informations shall be provided to the manufacturer for proper unit selection:

- total cooling load;
- outside ambient temperature;

- required heat rejection capacity;
- condensing temperature;
- type of compressor and refrigerant gas;
- required suction temperature;
- coil air resistance;
- corrosive atmosphere.

Note:

Refer to manufacturer's selection tables and interpolate whenever necessary. Extrapolations are not recommended.

10. REFRIGERANTS

10.1 General

10.1.1 Since the status of present refrigerant gas represent an industry in transition responding to external forces, product and market categories are being identified and their technical option outlined, it is imperative that design engineers be introduced to these changes. Also the future prospect represent promising new materials which can assist the design engineer in the selection of suitable compressor and chillers.

10.1.2 In order to make the right choice when making a selection on ideal refrigerant, it is recommended that the performance ratings of each refrigerant should be compared as indicated in Table 3.

TABLE 3 - COMPARATIVE REFRIGERANT PERFORMANCE RATINGS[a]

Refrigerant	Evaporator Pressure at 5°F (-14.9°C) psig (kPag)	Condensing Pressure at 86°F (29.7°C) psig (kPag)	Ratio of Compression	Net Refrigerating Effect Btu/lb (kJ/kg)	Refrigerant Circulated per ton (per kW) lb/min (g/s)	Liquid Circulated per ton (per kW) in³/min (mL/s)	Specific Volume of Vapor 5°F (-14.9°C) ft³/lb (L/g)	Compressor Displacement per ton (per kW) cfm (L/s)	Horsepower per ton (kW per kW) hp/ton (kW/kW)	Coefficient of Performance	Temperature of Compressor Discharge °F (°C)
Refrigerant 11 Trichloromonofluoromethane	24.0* (610)**	3.6 (24.8)	6.24	67.5 (157)	2.96 (6.36)	56.0 (4.35)	12.27 (0.766)	36.32 (0.871)	0.935 (0.198)	5.04	112 (44.[illegible])
Refrigerant 12 Dichlorodifluoromethane	11.8 (81.4)	93.3 (643)	4.08	50.0 (116)	4.00 (8.60)	85.6 (6.64)	1.46 (0.091)	5.83 (0.782)	1.002 (0.212)	4.70	101 (38.[illegible])
Refrigerant 22 Monochlorodifluoromethane	28.3 (195)	159.8 (1102)	4.06	69.3 (161)	2.89 (6.21)	68.0 (5.28)	1.25 (0.078)	3.60 (0.483)	1.011 (0.214)	4.66	131 (54.[illegible])
Refrigerant 113 Trichlorotrifluoroethane	27.9* (709)**	13.9* (353)**	8.02	53.7 (125)	3.73 (8.02)	66.8 (5.16)	27.04 (1.68)	100.76 (13.512)	0.960 (0.204)	4.92	86 (29.[illegible])
Refrigerant 114 Dichlorotetrafluoroethane	16.1* (409)**	22.0 (152)	5.42	43.1 (100)	4.64 (9.98)	89.2 (6.92)	4.22 (0.263)	19.59 (2.627)	1.015 (0.215)	4.64	86 (30.5)
Refrigerant 114a Dichlorotetrafluoroethane	15.6* (396)	22.7 (156)	5.33	43.0 (100)	4.65 (10.0)	86.7 (6.88)	4.04 (0.252)	18.78 (2.518)	1.025 (0.217)	4.60	88 (30.[illegible])
Refrigerant 500 Azeotrope of Dichlorodifluoromethane and Difluoroethane	16.4 (113)	113.4 (782)	4.12	61.2 (143)	3.27 (7.03)	79.4 (6.15)	1.52 (0.094)	4.97 (0.666)	1.011 (0.214)	4.66	105 (40.[illegible])
Refrigerant 717 Ammonia	19.6 (135)	154.5 (1065)	4.94	474.4 (1105)	0.422 (0.907)	19.6 (1.52)	8.15 (0.508)	3.44 (0.461)	0.989 (0.210)	4.76	210 (9[illegible])

*** Inches of mercury below one atmosphere.**

**** Millimeters of Mercury below one Atmosphere.**

a Calculated values based on 5°F (-15°C) evaporation and 86°F (30°C) condensing temperature.

10.2 Properties

The requirements for desirable properties of a good refrigerant shall be:

a) Low boiling point.
b) Safe, non-toxic and non-flammable.
c) Easy to liquefy at moderate pressure and temperature.
d) High latent heat value.
e) Operate on a positive pressure and stable inside a system.
f) Have no effect on moisture and ozone.
g) Mix well and be compatible with oil and lubricants.
h) Non-corrosive to metal and environmentally benign even with respect to decomposition products.
i) Abundantly available and easy to manufacture, handle and detect.

10.3 Safety Group Classification

10.3.1 ANSI/ASHRAE Standard 34-1992 defines Safety Group Classifications for refrigerants according to their toxicity and flammability. Toxicity classifications are based on the Threshold Limit Value and Time Weighted Average (TLV-TWA) established for each refrigerant. (Safety is defined as being free from harm or the risk of injury or loss).

10.3.2 The designations for toxity and flammability defined in six possible combination are shown in the matrix (something within which something else originates or develops) as indicated in Table 4.

TABLE 4 - ASHRAE STANDARD 34-1992 REFRIGERANT SAFETY CLASSIFICATIONS

INCREASING FLAMMABILITY	LOWER TOXICITY	HIGHER TOXICITY
Higher Flammability	Group A3	Group B3
Lower Flammability	Group A2	Group B2
No Flame Propagation	Group A1	Group B1

INCREASING TOXICITY

10.3.3 Refrigerants are assigned to one of the two classes "A" or "B" and are based on the following criteria:

a) Class A

Signifies refrigerants for which toxicity has not been identified at concentrations less than or equal to 400 ppm, based on data used to determine Threshold Limit Value-Time Weighted Average (TLV-TWA) or consistent indices.

b) Class B

Signifies refrigerants for which there is evidence of toxicity at concentrations below 400 ppm, based on data used to determine TLV-TWA or consistent indices.

10.4 Refrigerant Recovery and Recycling

10.4.1 General

Effective July 1,1992 EPA's "No Venting Law" requires all contractors and owners to posses a suitable refrigerant recovery and recycling unit conforming to the requirements of ARI 740. This approach is essential to minimize the CFC contaminant level.

10.4.2 Contaminant level

To comply with ARI 700-88 (allowable contaminant level in used refrigerant) standard, the physical properties of fluorocarbon refrigerants and the the maximum contaminant level shall be as indicated in Table 5.

TABLE 5 - CONTAMINANT LEVEL

PHYSICAL PROPERTIES	REFRIGERANTS								
	R11	R12	R13	R22	R113	R114	R500	R502	R503
Boiling point °F@ 29.9 in Hg	74.9	-21.6	—	-41.4	117.6	38.8	-28.3	-49.8	-127.6
Boiling range °F for 5% to 85% by vol. distilled	0.5	0.5	0.9	0.5	0.5	0.5	0.9	0.9	0.9
Vapor phase contaminants air and other non-condensables, max. % by vol.	—	1.5	1.5	1.5	—	1.5	1.5	1.5	1.5
Liquid phase contaminants water-ppm by weight	10	10	10	10	10	10	10	10	10
Chloride ION-NO turbidity to pass by test	Pass	Pass	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Acidity-Max. ppm by weight	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
High boiling residues-max. % by volume	0.01	0.01	0.05	0.01	0.03	0.01	0.05	0.01	0.01
Particulate/Solids-Visually clean to pass	Pass	Pass	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Other refrigerants-max. % by weight	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5

11. AUTOMATION AND CONTROLS

11.1 Automatic Controls

11.1.1 General

Applications of automatic control systems range from simple domestic and commercial temperature regulation to precision control of industrial processes. Automatic controls can be used wherever a variable condition must be controlled. That condition may be pressure, temperature, humidity, or rate and volume of flow, and it may exist in a liquid, a solid, or a gas. In controlling these conditions, the most important consideration is in the operation of the controlling and the controlled devices.

Note:

For further information on automatic controls, reference is made to ASHRAE 1991 Application Guidebook, Chapter 41.

11.2 Actuators

11.2.1 An actuator is a controlled motor, relay or solenoid in which the electric or pneumatic energy is converted into a rotary, linear or switching action. An actuator can effect a change in the control variable by operating a number of kinds of final control elements, such as valves and dampers.

11.2.2 Pneumatic motors or actuators are proportioning and modulating in action, that they can assume any position between and including both extremes, depending on the pressure of the air delivered to them. To secure two-position or on-off action, relays must be used to supply either zero air pressure or full air pressure to the motor.

11.2.3 Electric motors are two-position, floating, or proportional position. Some of these motors are uni-directional and rotate through 360°, while others have a limited stroke and two directions of travel.

11.3 Controllers

11.3.1 General

11.3.1.1 A controller is a device which (1) senses and measures changes in the controlled variable, and (2) uses an impulse received from sensing and measuring the controlled variable to meter energy of a form usable in the control circuit. The metered energy actuates the control equipment which then corrects a change or prevents a further change in the controlled variable.

11.3.1.2 The sensing and measuring functions are performed by the primary element of the controller. The material and construction of the primary element must be such that the primary element will respond to changes in the controlled condition. Electric and pneumatic controls use essentially the same kinds of primary elements.

11.3.1.3 The primary elements of a typical controller should have the capability to measure the following:

- temperature sensing primary elements
- pressure sensing primary elements
- humidity sensing primary elements

11.3.1.4 The controller takes the information in either the form of temperature sensing or humidity sensing. It gives command which can turn 'off' and 'on' the pilot lights, start and stop a fan or provide overall heating or cooling as required. Recent system allows personal computers (PC) to be tied to the controller and this PC can act as the customers' window into the system.

11.3.2 Controller mechanisms

11.3.2.1 The translation of the measured change in the controlled variable into a form of energy which can be used by the control system. In the primary element, the measurement of the controlled condition has been transformed into an impulse. This impulse then acts on the controller mechanism.

11.3.2.2 In an electric controller the impulse from the primary element is used to open or close an electric circuit or set up a varying resistance in an established circuit. For pneumatic controllers the mechanism is usually a system of valves which are opened and closed, or a vane which regulates the air pressure to the final control element by bleeding air to the atmosphere.

11.3.2.3 The mechanism of the controller is, in effect an amplifier. This is even more apparent in electronic control devices, as the sensing and measuring functions are carried out in terms of electronic energy and amplified by an electronic amplifier.

11.4 Modes of Control

All automatic control systems do not employ the same kind of action to accomplish their purposes. The method by which a control system acts are called "control mode", the commonly used of which are as follows;

a) Two position control, which is further divided into:
- Simple two-position control
- Timed two-position control

b) Floating control, which is further divided into:
- Proportional plus reset control
- Direct digital control (DDC)

c) Proportional control which may be proportional or single speed.

12. BUILDING AUTOMATION AND CONTROL INTEGRATION

12.1 Integration and Automation Control

12.1.1 Integrated controls

12.1.1 Microprocessor based controllers should be used to integrate communications through centralize monitoring and control of single or multiple equipment rooms or fan systems. The concept of multiplexing signals shall be used to reduce the wiring and tubing requirement. Through automation, this system provides means of accessibility to control system.

12.1.1.2 Control panels tie the component mounted microprocessor together over a local area network (LAN), so that overall system performance can be monitored and controlled. The panels may provide alarms when components fail or slip beyond predetermined parameters which provide significant diagnostic capabilities.(manufacturers are developing protocols to communicate with building fire, security, elevators and lighting control systems).

12.1.1.3 Integrating the automation and controller functions improves the hardware utilization, functionality and operator interface capabilities. Therefore to increase the potential for providing better environmental control, a typical building energy management system (BEMS) in a building should be coordinated with relevant manufacturers of controls.

12.2 Direct Digital Controls (DDC)

12.2.1 General

The first "D" in DDC can stand for two concepts: Direct or Distributed. Direct (the definition accepted in the industry today) signifies that system devices are monitored and controlled by digital electronics. Distributed control means that system devices interact among themselves to control a mechanical system, without reference to an over-seeing device.

12.2.2 DDC and personal computers

As the power of personal computers (PCs) increases and their cost decreases, DDC central data gathering functions have migrated to an affordable PC platform for nearly every market segment. These central data gathering stations can be of great use in the commissioning procedure. Not only can they monitor physical data, but they can also indicate whether or not a particular application is behaving correctly. In addition, they can supply the following diagnostic and trouble-shooting information, and document test results.

- Trend logging
- Dynamic trending
- Command trace

12.2.3 DDC commissioning stages

For DDC operation, the first stage of commissioning involves testing, calibrating and verifying the sensors and actuators installed on the job. Verifying wiring terminations, checking the placement of the sensors and actuators, calibrating sensor reading against known values and stroking actuators should be done before DDC control loops are commissioned.

12.3 'Gateways' to Integration

The integrated and automated control facilities available with modern distributed intelligence microprocessor based building automation system (BAS) may include but not limited to, the following:

- Plant alarm and status monitoring and logging
- Optimization of boilers and chillers
- Optimum start/stop of heating and cooling plant
- Security and fire alarm monitoring
- Plant maintenance scheduling
- Full DDC control loops
- Control of remote building via telephone
- Energy consumption logging
- Electrical load cycling and maximum demand control
- Metering of electricity, gas, oil and water
- Lighting control

12.4 Communication Protocol Standards

12.4.1 A data communication protocol for Building Automation energy management and Control network (BACnet)procedures shall be based on ASHRAE Standard *BSR/ASHRAE 135-P. The proposed BACnet standard defines data communication services and protocol for computer equipment used for monitoring and control of HVAC and other building systems. It defines an abstract object-oriented representation of information communicated between equipment, thereby facilitating the application and use of digital control technology in buildings.

12.4.2 To develop mechanisms by which computerized equipment of arbitrary function can exchange information, regardless of the particular building service it performs, the SPC 135P (Standards Project Committee) recommends the following four key components to the development process required to be tackled:

a) How to represent the internal functioning of a Vendors' equipment in a common, network-visible way, recognizing both proprietary nature of Vendor's internal design and the diversity of fuctionility involved.

*** BSR = Board of Standards Review**

b) To agree on a set of common commands or services that could be used between devices to get them to carry out the functions of distributed monitoring and control. (The Standard currently provides 30 services in fire areas: alarm and event services; object access services; remote device management services; and virtual terminal services).

c) To agree how to encode the messages defined above in a standard way. How should the messages be represented as binary zeros and ones on the communications media.

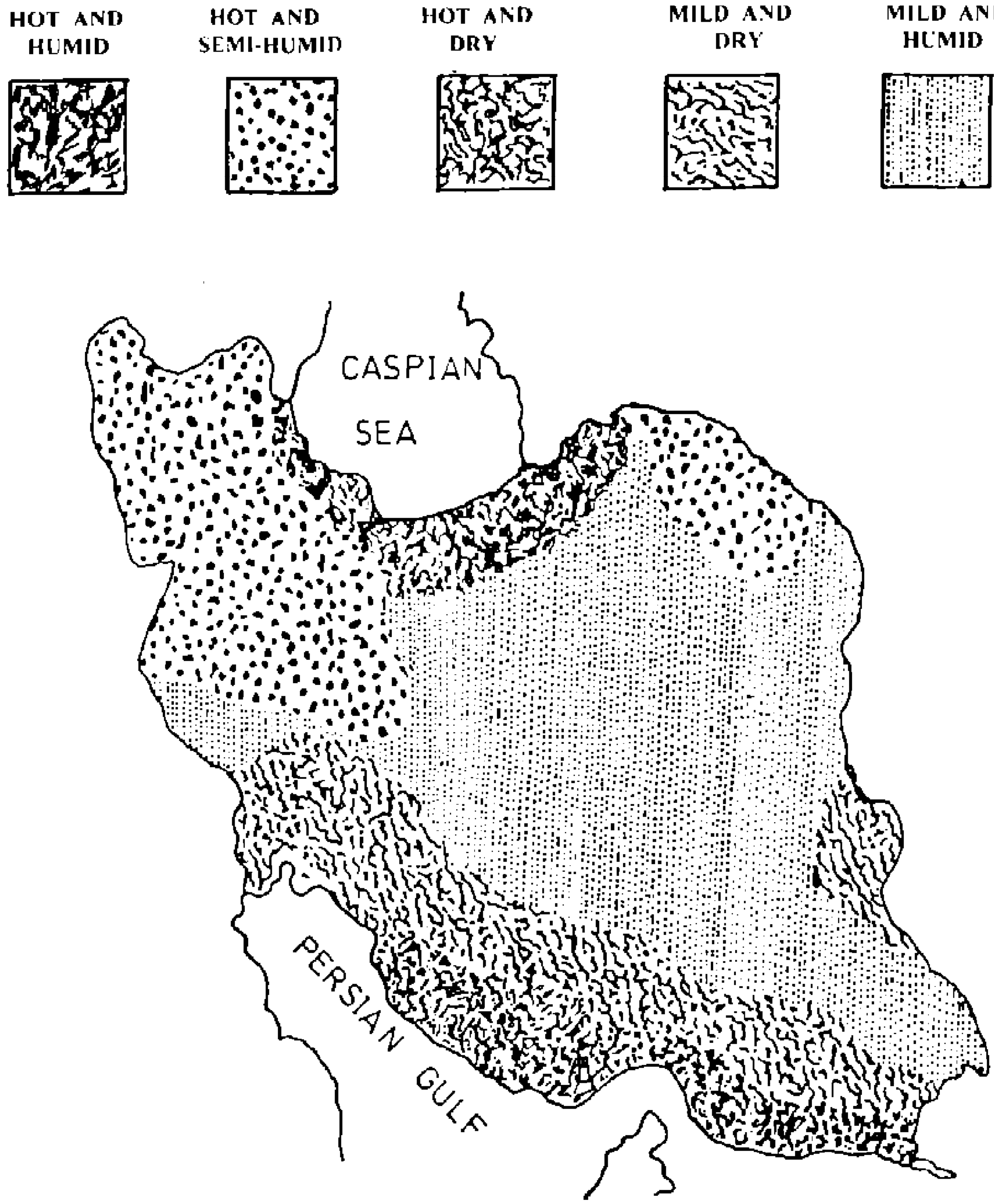
d) What network technologies should be used to actually get the BACnet messages from one device to another.

12.4.3 Computers used in the HVAC industry have their own defacto standards. To define this model for protocol communications, the International Standards Organization (ISO) proposed that manufacturers comply to Open Systems Interconnect (OSI) model. This follows the ISO recommendations for the provision of "gateways"to other networks. Within the LON (Local Operating Network) protocol the OSI has defined its workings in 7 layers (sub-task).

ATTACHMENTS

(These Attachments are not part of this Standard and are included for information purpose only).

ATTACHMENT 1



WEATHER ZONING OF IRAN

ATTACHMENT 2
ZONING CLASSIFICATION PER CLIMATIC CONDITION IN IRAN

ZONING AREA			OUTSIDE AIR CONDITIONS									GEOGRAPHICAL SITUATION		INSIDE DESIGN CONDITIONS (PROPOSED)							
			SUMMER						WINTER					SUMMER						WINTER	
TYPE	SUMMER	WINTER	DRY BULB TEMP.		WET BULB TEMP.		REL HUM	HUM LEVEL	DAILY RANGE	MINIMUM TEMP.		LAT	ELEV ASL	DRY BULB TEMP.		WET BULB TEMP.		REL HUM	HUM LEVEL	TEMP. DIFFERENCE	
			°C	°F	°C	°F	%RH	W gr/lb	DR °F	°C	°F	DEG-REE	(m)	°C	°F	°C	°F	%RH	W gr/lb	°C	°F
1	HOT AND HU-MID	HOT	43.5	110	33.5	92	50%	200	20	6.5	42	24	10	26.5	80	19.5	67	50%	77	16	24
2	HOT AND SEMI HU-MID	MILD	47	115	26	78	20%	87	30	0	32	30	20	26.5	80	19.5	67	50%	77	22	40
3	HOT AND DRY	COLD	41	105	21	70	17%	55	20	-8	18	30-32	1500	25	77	17	62.5	50%	65	30	54
4	MILD AND DRY	VERY COLD	37	99	20.5	69	22%	28	25	-15	5	36	1500	25	77	17	62.5	50%	65	38	
5	MILD AND HU-MID	COLD	35	95	29	84	65%	160	20	-4	25	37	-20	25	77	17	62.5	50%	65	28	50

ELEV = ELEVATION
ASL = ABOVE SEA LEVEL
GR/LB = GRAINS PER POUND
REL HUM = RELATIVE HUMIDITY

ATTACHMENT 3
SCHEDULE OF IRANIAN CITIES PER ZONING AREA

TYPE 3 - HOT & DRY	TYPE 2 - HOT & SEMI-HUMID
ANEN	ABADAN
ARDEKAN	AGAJARI
BAFQ	AHWAZ
BAFT	ANDIMESHK
BEERJAND	BEHBAHAN
DAMGHAN	BUM
EMAM SHAHR	DASHTE ABAS
ESFAHAN	DEZFUL
FESA	DOW GONBADAN
FIRDAUS	EEZEH
GARMSAR	GACHSAR
GONABAD	HOVEIZEH
JAHROM	IRANSHAHR
JEEROFT	KAHNOOJ
KAHRIZAK	LAR
KASHAN	MASJID SULAIMAN
KASHMAR	RAMHORMOZ
KERMAN	SHOOSH
KERMANSHAH	SHOOSHTAR
KHORAMABAD	SUSANGERD
NAJAF ABAD	ZABOL
NEIREEZ	
PASSARGAD	
RAFSANJAN	
RAVAND	
SABZEVAR	
SEERJAN	
SEMNAN	
SHAHR BABAK	
SHIRAZ	
TAFTAN	
TEHRAN	
YASOOJ	
YEZD	
ZAHEDAN	

(to be continued)

ATTACHMENT 3 (continued)

TYPE 1 - HOT & HUMID	TYPE 4 - MILD & DRY
ABU MUSA ISLAND BANDAR ABAS BANDAR AMIR BANDAR BOOSHER BANDAR DYLAM BANDAR GONAVEH BANDAR JASK BANDAR IMAM KHOMEINI BANDAR MAHSHAR CHAHBAHAR HENGAM PORT HORMOZ PORT KHARG ISLAND KISH ISLAND LARAK PORT LAVAN PORT MINAB MINOO PORT TOMB BOZORG TOMB KOOCHEK QESHM ISLAND	ABHAR ALI GOODARZ ARAK ARDABIL AZARSHAHR BANEH BAZARGAN BOEEN ZAHRA BIJAR BISETOON BOJNOORD BOROOJERD BUKAN DAMAVAND DEHLORAN DOOZ DOOZAN ESLAM ABAD HAMEDAN HESARAK KARADJ KHOY MAHALAT MAKOO MALAYER MARAGEH MARIWAN MASHED MESHKEEN SHAHR MIYANDOAB MIYANEH NEISHABOOR NOUSOOD OSHANOOYIEH PAVEH PIRANSHAHR QAZVIN QOOCHAN RAVANSAR REZAYEH ROODE HEN SALMAZ SANANDAJ SAQEZ SARAB SARDASHT SHABESTAR SHAHIN DEJ SHAHR-E-KORD SHEERVAN TABRIZ TAKAB ZANJAN

(to be continued)

ATTACHMENT 3 (continued)

TYPE 5 - MILD & HUMID
AZADSHAHR
ASTARA
ASTANEH
AMOL
BABOL
BABOLSAR
BANDAR ANZALI
BANDAR TORKAMAN
BANDAR GAZ
BEHSHAHR
CHABOKSAR
CHALOOS
FOOMEN
GONBAD KAVOOS
HASHTPAR
KHOLAK CHAL
LAHIJAN
LANGAROOD
MANJIL
NEKA
NOOR
NOUSHAHR
QAEM SHAHR
RAMSAR
RASHT
ROOD-BAR
ROOD-SAR
SARI
SIYAKHAL
SOMEH SARA
TONKABON

ATTACHMENT 4
RECOMMENDED RATE OF HEAT GAIN FROM SELECTED RESTAURANT EQUIPMENT[a]

		Input Rating, Btu/h		RECOMMENDED RATE OF HEAT GAIN, Btu/h			
				Without Hood			With Hood
Appliance	Size	Max	Standby[b]	Sens.	Latent	Total	Sensible
Electric, No Hood Required							
Blender, per quart of capacity	1 to 4 qt	1550		1000	520	1520	480
Cabinet (large hot holding)	16.2 to 17.3 ft³	7100		610	340	960	290
Cabinet (small hot holding)	3.2 to 6.4 ft³	3070		270	140	410	130
Coffee brewer	12 cups 2 brnrs	5660		3750	1910	1910	1810
Coffee brewing urn (large),							
Per quart of capacity	23 to 40 qt	2130		1420	710	2230	680
Coffee heater, per warming burner	1 to 2 brnrs	340		230	110	340	110
Dishwasher (hood type chemical sanitizing),							
Per 100 dishes/h	950 to 2000 dishes/h	1300		170	370	540	170
Dishwasher (conveyor type water sanitizing),							
Per 100 dishes/h	5000 to 9000 dishes/h	1160		150	370	520	170
Display case (refrigerated), per ft³ of interior	6 to 67 ft³	154		62	0	62	0
Food warmer (infrared bulb), per lamp	1 to 6 bulbs	850		850	0	850	850
Food warmer (well type), per ft³ of well	0.7 to 2.5 ft³	3620		1200	610	1810	580
Freezer (large)	73 ft³	4570		1840	0	1840	0
Griddle/grill (large), per ft² of cooking surface	4.6 to 11.8 ft²	9200		620	340	960	340
Hot plate (high speed double burner)		16720		7810	5430	13240	6240
Ice maker (large)	220 lb/day	3720		9320	0	9320	0
Microwave oven (heavy duty commercial)	0.7 ft³	8970		8970		8970	0
Mixer (large), per quart of capacity	80 qt	94		94	0	94	0
Refrigerator (large), per 100 ft³ of space	25 to 74 ft³	750		300	0	300	0
Rotisserie	300	10920		7200	3720	10920	3480
Serving cart (hot), per ft³ of well	1.8 to 3.2 ft³	2050		680	340	1020	320
Steam kettle (large), per quart of capacity	80 to 320 qt	300		23	16	40	13
Toaster (large pop-up)	10 slice	18080		9590	8500	18080	5800
Electric Exhaust Hood Required							
Charbroiler, per ft² of cooking surface	1.5 to 4.6 ft²	7320					3310
Fryer (deep fat), per lb of fat capacity	15 to 15 to 70 lb	1270					14
Fryer (pressurized), per lb of fat capacity	13 to 33	1570					59
Oven (large convection), per ft³ of oven space	7 to 19 ft³	4450					180
Oven (small convection), per ft³ of oven space	1.4 to 5.3 ft³	10340					150
Range (burners), per 2 burner section	2 to 10 burners	7170					2660
Gas, No Hood Required							
Broiler, per ft² of broiling area	2.7 ft²	14770	61	5310	2860	8170	1220
Dishwasher (hood type chemical sanitizing),							
Per 100 dishes/h	950 to 2000 dishes/h	1740	660[b]	510	200	710	230
Dishwasher (conveyor type water sanitizing)							
Per 100 dishes/h	5000 to 9000 dishes/h	1370	660[b]	370	80	450	140
Griddle/grill (large), per ft² of cooking surface	4.6 to 11.8 ft²	17000	330	1140	610	1750	460
Oven (pizza), per ft² of hearth	6.4 to 12.9 ft²	4740	61[b]	620	220	840	84
Gas, Exhaust Hood Required		9840	620				2430
Braising pan, per quart of capacity	105 to 140 qt	16440	510				790
Charbroiler (large), per ft² of cooking area	4.6 to 11.8 ft²	2270	300[b]				160
Fryer (deep fat), per lb of fat capacity	11 to 70 lb	8670	19[b]				250
Oven (convection), per ft³ of oven space	7.4 to 19.4 ft³	7240	61[b]				130
Oven (pizza), per ft² of oven hearth	9.3 to 25.8 ft²	33600	1325				6590
Range (burners), per 2 burner section	2 to 10 burners	11800	330				3390
Range (hot top fry top), per ft of cooking surface	3 to 8 ft²						
Steam							
Compartment steamer, per lb of food/h	46 to 450 lb	280		22	14	36	11
Dishwasher (hood type chemical sanitizing),							
per 100 dishes/h	950 to 2000 dishes/h	3150		880	380	1260	410
Dishwasher (conveyor water sanitizing),							
per 100 dishes/h	5000 to 9000 dishes/h	1180		150	370	520	170
Steam kettle, per quart capacity	13 to 32 qt	500		39	25	64	19

a) In cases where heat gain is given per unit of capacity the heat gain is calculated by multiplying the capacity by the recommended heat gain per unit of capacity.

b) Standby input rating is for the entire appliance regardless of size.

ATTACHMENT 5
RECOMMENDED RATE OF HEAT GAIN FROM SELECTED OFFICE EQUIPMENT

Appliance	Size	Maximum Input Watts	Maximum Input Btu/h	Standby Input Watts	Standby Input Btu/h	Recommended Rate of Heat Gain Watts	Recommended Rate of Heat Gain Btu/h
Computer Devices							
Communication/ transmission		1800-4600	6140-15700	1640-2810	5600-9600	1640-2810	5600-9600
Disk drives/mass storage		1000-10000	3400-34100	1000-6600	3400-22400	1000-6600	3400-22400
Microcomputer/ wordprocessor	16 - 640 kbytes[a]	100-600	340-2050	90-530	300-1800	90-530	300-1800
Minicomputer		2200-6600	7500-15000	2200-6600	7500-15000	2200-6600	7500-15000
Printer (laser)	8 pages / min	870	3000	180	600	300	1000
Printer (Line, high speed)	5000 - more pages / min	1000-5300	3400-18000	500-2550	2160-9040	730-3800	2500-13000
Tape drives		1200-6500	4100-22200	1000-4700	3500-15000	1000-4700	3500-15000
Terminal		90-200	300-700	80-180	270-600	80-180	270-600
Copiers / Typesetters							
Blue print		1150-12500	3900-42700	500-5000	1700-17000	1150-12500	3900-42700
Copiers (large)	30 - 67[a] copies / min.	5800-22500	1700-6600	5800-22500	900	3100	1700-6600
Copiers	6 - 30[a] copies / min.	1570-5800	460-1700	1570-5800	300-900	1000-3100	460-1700
Phototypesetter		1725	5900			1520	5200
Mailprocessing							
Inserting machine 3600 - 6800 pieces/h		600-3300	2000-11300			390-2150	1300-7300
Labeling machine 1500 - 30000 pieces/h		600-6600	2000-22500			390-4300	1300-14700
Miscellaneous							
Cash register		60	200			48	160
Cold food / beverage		1150-1920	3900-6600			575-960	1960-3280
Coffee maker	10 cup	1500	5120		sensible	1050	3580
					latent	450	1540
Microwave oven	$1\ ft^3$	600	2050			400	1360
Paper shredder		250-3000	850-10200			200-2420	680-8250
Water cooler	8 gal/h	700	2400			1750	6000

[a] Input is not prportional to capacity

ATTACHMENT 6
RECOMMENDED RATE OF HEAT GAIN FROM HOSPITAL EQUIPMENT LOCATED IN THE AIR CONDITIONED AREA

Appliance Type	Size	Maximum Input Rating, Watts (Btu/h)	Recommended Rate of Heat Gain Watts [a] (Btu/h)
Autoclave (bench)	0.02 m^3 (1 ft^3)	1250 (4266)	140 (1778)
Bath, hot or cold circ., small	3.7 to 36.7 litres, - 30 to 100°C (1 Gal) (10 Gal) (86 to 212°F)	750 to 1800 (2560 to 6143)	130 to 310[f] (444 to 1058) 250 to 590[g] (853 to 2014)
Blood analyser	120 samples/hour	735 (2508)	735 (2509)
Blood analyser with CRT screen	115 samples/hour	1500 (5119)	1500 (5119)
Centrifuge (large)	8 to 24 places	1100 (3754)	1050 (3584)
Chromatograph	---	2000 (6826)	2000 (6826)
Cytometer (Cell sorter/analyser)	1000 cells/second (-36 cells/Hr)	21 460 (73242)	21 460 (71242)
Hot plate, concentric ring	4 holes, 100°C (212 °F)	1100 (3454)	870 (2969)
Incubator, CO_2 (5 to ft^3)	0.14 to 0.28 m^3, up to 55 °C (131 °F)	2830 (9659)	1410 (4812)
Incubator, forced draft (10 ft^3)	0.28 m^3, 27 to 60 °C (80 to 140 °F)	720 (2557)	360 (1229)
Incubator, general application (2 to 11 ft^3)	0.04 to 0.31 m^3, up to 70 °C (158 °F)	1660 to 2260[b] (5666 to 7713)	850 to 1130[b] (2901 to 3857)
Magnetic stirrer	---	600 (17048)	600 (12048)
Microcomputer	16 to 256 kbytes [c]	100 to 600 (341 to 2048)	88 to 528 (300 to 1802)
Oven, general purpose, small (2 to 4 ft^3)	0.04 to 0.08 m^3, 240 °C (464 °F)	21 900[b] (74744)	2970[b] (10136)
Refrigerator, laboratory	0.63 to 3.0 m^3, 4 °C(39 °F)	880[d] (3003)	350[d] (1194)
Refrigerator, blood, small (22 to 106 ft^3)	0.20 to 0.56 m^3, 4°C(39°F)	2680[b] (9147)	1060[b] (3618)
Spectrophotometer	---	500 (1706)	500 (1706)
Sterilizer, freestanding (4 ft^3)	0.11 m^3, 100 to 132 °C(212 to 270°F)	20 900 (71331)	2370 (8089)
Washer, glassware (8 ft^3)	0.22 m^3 load area	4460 (15221)	2930 (70000)

a For hospital equipment installed under a hood, the heat gain is assumed to be zero.

b Heat gain per cubic meter of interior space.

c Input is not proportional to memory size

d Heat gain per 10 m3 of interior space

e Heat gain per liter of capacity

f Sensible heat

g Larent heat

ATTACHMENT 7
HEAT GAIN FROM TYPICAL ELECTRIC MOTORS

				Location of Motor and Driven Equipment with Respect Conditioned Space or Airstream		
				A	B	C
Motor Name-plate or Rated Horse-power	Motor Type	Nom-inal rpm	Full Load Motor Effici-ency in Percent	Motor in, Driven Equip-ment in Btu/h	Motor out, Driven Equip-ment in Btu/h	Motor Driven Equip-ment out Btu/h
0.05	Shaded Pole	1500	35	360	130	240
0.08	Shaded Pole	1500	35	580	200	380
0.125	Shaded Pole	1500	35	900	320	590
0.16	Shaded Pole	1500	35	1160	400	760
0.25	Split Phase	1750	54	1180	640	540
0.33	Split Phase	1750	56	1500	840	660
0.50	Split Phase	1750	60	2120	1270	850
0.75	3- Phase	1750	72	2650	1900	740
1	3 - Phase	1750	75	3390	2550	850
1	3- Phase	1750	77	4960	3820	1140
2	3- Phase	1750	79	6440	5090	1350
3	3- Phase	1750	81	9430	7640	1790
5	3- Phase	1750	82	15,500	12,700	2790
7.5	3- Phase	1750	84	22,700	19,100	3640
10	3- Phase	1750	85	29,900	24,500	4490
15	3- Phase	1750	86	44,400	38,200	6210
20	3- Phase	1750	87	58,500	50,900	7610
25	3- Phase	1750	88	72,300	63,600	8680
30	3- Phase	1750	89	85,700	76,300	9440
40	3- Phase	1750	89	114,000	102,000	12,600
50	3- Phase	1750	89	143,000	127,000	15,700
60	3- Phase	1750	89	172,000	153,000	18,900
75	3- Phase	1750	90	212,000	191,000	21,200
100	3- Phase	1750	90	283,000	255,000	28,300
125	3- Phase	1750	90	353,000	318,000	35,300
150	3- Phase	1750	91	420,000	382,000	37,800
200	3- Phase	1750	91	569,000	509,000	50,300
250	3- Phase	1750	91	699,000	636,000	62,900

ATTACHMENT 8
EQUIPMENT SERVICE LIFE

Equipment Item	Median Years	Equipment Item	Median Years	Equipment Item	Median Years
Air conditioners		**Air terminals**		Air-cooled condensers	20
Window unit	10	Diffusers, grilles, and registers	27	Evaporative condensers	20
Residential single or split package	15	Induction and fan-coil units	20	**Insulation**	
Commercial through-the-wall	15	VAV and double-duct boxes	20	Molded	20
Water-cooled package	15	Air washers	17	Blanket	24
Heat pumps		Duct work	30	**Pumps**	
Residential air-to-air	15[b]	Dampers	20	Base-mounted	20
Commercial air-to-air	15	**Fans**		Pipe-mounted	10
Commercial water-to-air	19	Centrifugal	25	Sump and well	10
Roof-top air conditioners		Axial	20	Condensate	15
Single-zone	15	Propeller	15	Reciprocating engines	20
Multizone	15	Ventilating roof-mounted	20	Steam turbines	30
Boilers, hot water (steam)		**Coils**		Electric motors	18
Steel water-tube	24 (30)	DX, water, or steam	20	Motor starters	17
Steel fire-tube	25 (25)	Electric	15	Electric transformers	30
Cast iron	35 (30)	**Heat Exchangers**		**Controls**	
Electric	15	Shell-and-tube	24	Pneumatic	20
Burners	21	Reciprocating compressors	20	Electric	16
Furnaces		**Package chillers**		Electronic	15
Gas- or oil-fired	18	Reciprocating	20	**Valve actuators**	
Unit heaters		Centrifugal	23	Hydraulic	15
Gas or electric	13	Absorption	23	Pneumatic	20
Hot water or steam	20	**Cooling towers**		Self-contained	10
Radiant heaters		Galvanized metal	20		
Electric	10	Wood	20		
Hot water or steam	25	Ceramic	34		

[a] Obtained from a nation-wide survey conducted by ASHRAE TC 1.8 (Akalin 1978). Data changed by TC 1.8 in 1986.

[b] See Lovora and Hiller (1985) and Easton Consultants (1986) for further information.