

An Introduction to Air Conditioning System Design



Guyer Partners
44240 Clubhouse
Drive
El Macero, CA 95618
(530)7758-6637
jpguyer@pacbell.net

J. Paul Guyer, P.E., R.A.

Paul Guyer is a registered Mechanical Engineer, Civil Engineer, Fire Protection Engineer and Architect with over 35 years experience in the design of buildings and related infrastructure. For an additional 9 years he was a principal staff advisor to the California Legislature on infrastructure and capital outlay issues. He is a graduate of Stanford University, and has held numerous national, state and local positions with the American Society of Civil Engineers and National Society of Professional Engineers.

CONTENTS

1. INTRODUCTION
2. LOAD CALCULATIONS
3. AIR CONDITIONING EQUIPMENT
4. AIR DISTRIBUTION
5. RULES OF THUMB

This course is adapted from Department of Defense MIL-HDBK-1003/3 HEATING, VENTILATING, AIR CONDITIONING AND DEHUMIDIFYING SYSTEMS which is in the public domain, has unlimited distribution and is not copyrighted.

1. INTRODUCTION

This is an introduction to air conditioning systems (frequently referred to as HVAC systems – heating, ventilating and air conditioning systems). It is intended for those engineers, architects and construction professionals who are only peripherally involved with HVAC systems in their professional activities, but would like to learn more about HVAC concepts, principles, systems and equipment. It is not a design manual, but will give design and construction professionals a step forward in understanding this area of building technology. Design information presented here is presented in a “manual” form, that is, calculations are presented as if calculated manually, although, of course, this is done in most cases in practice by computer programs. This manual presentation will give a better understanding of the underlying principles rather than just leaving the matter of load calculations as a simple data input exercise.

2. LOAD CALCULATIONS

2.1 General. The first step in HVAC system design is to select indoor and outdoor summer and winter design conditions. There are various sources for this information, but among the best are DOD Military Handbook MIL-HDBK-1190 and Naval Facilities Engineering Command NAVFAC Publication P-89, Engineering Weather Data Manual procedures provided below for determining heating and cooling loads are for illustration and training purposes only, but may be used for small systems (e.g., heating systems less than 200,000 Btu per hour and cooling systems less than 10 tons). Computer programs are available that will provide more precise load determinations and the time of day with the highest cooling load. The highest heating load is assumed to occur just before dawn; therefore, this should be considered in the design heating load.

2.1 Heating Load. Heating load.... the amount of heating that must be provided given the assumed outside air temperature and desired inside air temperature....is calculated as described below. Heating load is due to transmission, infiltration and ventilation.

2.2.1 Transmission. Heating load due to transmission is calculated using Eq 2.1.

$$Q = U \times A \times (T_i - T_o) \quad (\text{Eq 2.1})$$

where:

Q = Btu/hr heat loss by transmission,

U = heat transfer coefficient (look this up in a handbook for your particular wall, floor, roof, etc. construction)

A = area of the surface (wall, window, roof, etc.),

T_i = inside design temperature, and

T_o = outside design temperature.

Use this formula to compute heat transmission losses from each element of the building skin (e.g., walls, windows, roof, etc.). Note that attic and crawl space and ground temperature are different from outdoor temperatures.

2.2.2 Infiltration and Ventilation. To determine the heating load use the larger of the infiltration and ventilation loads. Outdoor air provided for ventilation should exceed the air exhausted by 10 to 15 percent to minimize infiltration. The designer must use judgment on the amount of excess supply air to include based on number and type of windows and doors.

$$Q = 1.10 \times \text{CFM} \times (T_i - T_o) \quad (\text{Eq 2.2})$$

where:

CFM = cubic feet per minute of outdoor air, and

Q = the sensible heat loss, Btu/hr.

This calculation does not apply to industrial ventilation systems, e.g., systems to control fumes, vapors, and dust from such processes as plating, painting, welding, and woodworking. Refer to American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) Handbook, HVAC Systems and Applications, for guidance on design of these systems.

2.2.3 Total Heating Load. Sum the transmission loads with infiltration and ventilation loads to get the total heating load. To this computed total heating load, add the following to size central equipment (do not apply these factors when sizing terminal equipment such a finned-tube radiation, fan-coil units, etc.):

2.2.3.1 Exposure factor (prevailing wind side) up to 15 percent.

2.2.3.2 Pickup (for intermittently heated buildings with primary heat sources such as boilers, steam-to-water heat exchangers, etc.) 10 percent.

2.2.3.3 Buildings with night setback. A building with 10 degrees F setback may require up to 30 percent oversizing for acceptable pickup and minimum energy requirements.

2.3 Cooling Load. Computation of the peak cooling load can be a difficult effort. Heat gain (heat gain = cooling load) is composed of or influenced by the conduction heat gain through opaque portions of the building skin; the conduction plus solar radiation through windows and skylights; the building internal loads such as people, lights, equipment, motors, appliances, and devices; and outdoor air load from infiltration. For sizing variable air volume (VAV) systems, calculation of loads has more stringent requirements, not addressed here.

2.3.1 Transmission and Glass Solar Gain. Cooling load is heat gain from transmission, solar heat gain through glass, infiltration and ventilation, and internal loads. It is calculated as discussed below.

2.3.1.1 Walls and Roof Transmission. Cooling load due to transmission through walls and roof is calculated using Eq 2.3.

$$Q = U \times A \times (T_o - T_i) \quad (\text{Eq 2.3})$$

2.3.1.2 Glass Transmission and Solar Gain. Heat gain (cooling load) due to transmission and solar gain through glass is calculated as shown below.

2.3.1.2.1 Transmission. Heat gain by transmission through glass is calculated using Eq 2.4.

$$Q = U \times A \times (T_o - T_i) \quad (\text{Eq 2.4})$$

2.3.1.2.2 Solar Heat Gain. Solar heat gain through glass is calculated using (Eq 2.5).

$$Q = A \times SC \times SHGF \quad (\text{Eq 2.5})$$

where:

SC = shading coefficient, and

SHGF = solar heat gain factor (look up the SHGF in a handbook (e.g., ASHRAE Handbook, Fundamentals) for each exposure and type of glass).

2.3.2 Infiltration and Ventilation. Sensible and latent heat gains from infiltration and ventilation are calculated using Eqs 2.6 and 2.7. The concepts of sensible and latent heat require an understanding of psychometrics, which is beyond the scope of this discussion. The question of latent heat usually comes into play only in particularly humid climates. In this simplified presentation we will assume a Mediterranean climate (i.e. like California) and generally not be concerned with latent heat.

2.3.2.1 Sensible

$$Q_S = 1.10 \times \text{CFM} \times (T_o - T_i) \quad (\text{Eq 2.6})$$

2.3.2.2 Latent

$$Q_L = 4840 \times \text{CFM} \times W \quad (\text{Eq 2.7})$$

where:

W = change in humidity ratio (lb water/lb air).

2.3.2.3 Ventilation Rates. Refer to ASHRAE Standard 62 for ventilation requirements or the typical values below:

Auditoriums, theaters - 15 cfm/person

Sleeping rooms - 15 cfm/person

Bedroom - 30 cfm/room
 Classroom - 15 cfm/person
 Communication centers - 20 cfm/person
 Conference rooms - 20 cfm/person
 Corridors - 0.1 cfm/sq ft
 Dining - 20 cfm/person
 Lobbies - 15 cfm/person
 Locker, dressing rooms - 0.5 cfm/sq ft
 Lounges, bars - 30 cfm/person
 Offices - 20 cfm/person
 Toilet, bath (private) - 35 cfm/room
 Toilet (public) - 50 cfm/water closet or urinal

The total corrected outdoor air requirement for central systems supplying spaces with different ratios of outdoor-air-to-supply-air is determined from the following:

$$CFM_{ot} = Y \times CFM_{st} \quad (\text{Eq 2.8})$$

where:

CFM_{ot} = corrected total outdoor air quantity,

CFM_{st} = total system airflow (i.e., sum of air supplied to all spaces), and

Y = corrected fraction of outdoor air, or

$$Y = X / (1 + X - Z) \quad (\text{Eq 2.9})$$

where:

$$X = CFM_{oa} / CFM_{st},$$

$$Z = CFM_{oc} / CFM_{sc},$$

where:

CFM_{oa} = uncorrected sum of outdoor airflow rates for spaces on the system,

CFM_{oc} = outdoor air required for critical space, and

CFM_{sc} = supply air to the critical space.

The critical space is that space with the greatest required fraction of outdoor air in the supply to that space.

2.3.3 Internal Loads

2.3.3.1 People Loads. Adjusted (normal male/female/child), per person.

Sensible/Latent

Office (seated light work, typing) 255 Btu/hr 255 Btu/hr

Factory (light bench work) 345 Btu/hr 435 Btu/hr

Factory (light machine work) 345 Btu/hr 695 Btu/hr

Gymnasium athletics 635 Btu/hr 1165 Btu/hr

2.3.3.2 Lights and Equipment

2.3.3.2.1 Lights

$$Q = 3.41 \times W \times F_{ul} \times F_{sa} \quad (\text{Eq 2.10})$$

where:

W = total light wattage,

F_{ul} = use factor, and

F_{sa} = special allowance factor for fluorescent fixtures or for fixtures that release only part of their heat to the conditioned space.

2.3.3.2.2 Equipment

(1) Motors within conditioned space or within airstream.

$$Q = 2545 \times \text{HP} / (E_m \times F_{lm} \times F_{um}) \quad (\text{Eq 2.11})$$

where:

HP = motor horsepower,

E_m = motor efficiency,

F_{lm} = motor load factor, and

F_{um} = motor use factor.

(2) Appliances and equipment, such as business machines and computers. Refer to ASHRAE Handbook, Fundamentals and manufacturer's data to determine sensible and latent heat gains from equipment.

$$Q_s = 3.41 \times W \times F_{ue} \quad (\text{Eq 2.12})$$

where:

Q_s = sensible load,

W = appliance wattage, and

F_{ue} = equipment use factor.

2.3.3.3 Heat Gain From Miscellaneous Sources

2.3.3.3.1 HVAC Fan Motors (Outside the Airstream). Typically, thirty-five percent of the input to an HVAC fan motor is converted to heat in the airstream because of fan inefficiency.

2.3.3.3.2 HVAC Fan Motors (Within the Airstream). The motor load is converted to heat.

2.3.3.3.3 Duct Leakage. Loss of supply air due to duct leakage shall be compensated by system capacity as follows:

(1) Well designed and constructed system: increase fan capacity by 3 percent.

(2) Poorly designed and constructed system: increase fan capacity by 10 percent.

3. AIR CONDITIONING EQUIPMENT

A detailed discussion of air conditioning equipment is beyond the scope of this presentation, but a few comments can be offered.

3.1 Cooling Systems

3.1.1 Central Air Conditioning Systems. Use these systems for applications where several spaces with uniform loads will be served by a single apparatus and where precision control of the environment is required. Cooling coils can be direct expansion or chilled water. Select air cooled or evaporative condensers, cooling towers, and ground-loop systems based on life cycle economics considering operating efficiencies and maintenance costs associated with outdoor design conditions and environment, e.g., high ambient temperatures and dusty conditions could adversely impact the operation of air cooled condensers. Consider temperature rise of chilled water supply when selecting chilled water coils, especially for applications requiring precision humidity control.

3.1.2 Unitary Air Conditioning Systems. These systems should generally be limited to loads less than 100 tons. Unitary systems are packaged in self-contained or split configurations. Self-contained units incorporate components for cooling or cooling and heating in one apparatus. Thermostatic expansion valves are preferred over capillary tubes and orifices for refrigerant control when available as a manufacturer's option since expansion valves provide better superheat control over a wide range of operating conditions. *Split systems* may include the following configurations:

- a) Direct expansion coil and supply fan combined with a remote compressor and condensing coil; or
- b) Direct expansion coil, supply fan, and compressor combined with a remote condenser, cooling tower, or ground-loop system.

These systems generally have lower first cost than central systems but may have higher life cycle costs. If part load operation is anticipated for a majority of equipment operating life, consider multiple unitary equipment for superior operating efficiencies and added reliability. Refer to ASHRAE Handbook, Equipment for size and selection criteria.

3.1.3 Room Air Conditioning Units. These units are self-contained units serving only one space. These units are typically referred to as window or through-the-wall type air conditioners. Rooms served by these units should have a separate HVAC unit to provide ventilation air for a group of rooms. Use them when they are life cycle cost effective, and in accordance with MIL-HDBK-1190. Refer to ASHRAE Equipment Handbook.

3.1.4 Built-up Systems. These systems consist of individual components assembled at the building site. Generally, use them when a large volume of air is handled. These systems may be used as remote air handling systems with a central cooling plant. unitary air handling units. Determine the number of air handling units by an economic division of the load, considering: (a) the value of space occupied by equipment; (b) the extent of ductwork and piping; (c) the multiplicity of control, maintenance, and operating points; and (d) energy conservation factors.

3.2 Heating Systems. Heating sources can be either steam, hot water, natural gas, oil, electricity, or a renewable resource. Select these sources based on life cycle cost. Heating systems may be combined with ventilating systems when feasible. Heating-dominated climates require perimeter radiation at windows in office spaces.

3.2.1 Individual Heating Plants. Locate individual heating plants in the building they serve or in a separate, adjoining building.

3.2.2 Central Heating Plants. Base the total heating system capacity on normal demand rather than total connected load.

3.3 All-Air Systems. Refer to ASHRAE Systems Handbook. In humid climates, provide all-air systems for air conditioning. These systems are central systems which provide complete sensible and latent heating and cooling of the air supply. These systems are either single path or dual path. Single-path systems have heating and cooling elements in a series configuration. Dual path system elements are arranged in parallel. Consolidation of system components at a central location provides increased opportunity for energy conservation.

3.3.1 Constant-Volume Systems. Use where room conditions are to be maintained by supplying a constant volume of air to the space and varying supply air temperature in response to demands for net space heating or cooling.

a) Applications. In addition to multi-zone systems, this includes single-zone or single space applications in auditoriums, meeting rooms, cafeterias, restaurants, and small retail stores.

b) Multi-zone Systems. Use these systems to provide individual temperature control of a small number of zones, maximum 10 zones, from a central air handler. For normal comfort cooling applications, place cooling and heating coils in the air handler. For applications where humidity control is critical, place coils in series so that air is conditioned by the cooling coil prior to passing to the hot deck. Provide cooling by direct-expansion or chilled-water coils. Provide heating by steam coils, hot water coils, or electric coils.

c) Terminal Reheat Systems. These systems overcome zoning limitations by adding individual heating coils in each zone's branch duct to compensate for areas of unequal heating load. Heat, whether in the form of hot water, steam, or electrical resistance heaters, is applied to either preconditioned primary air or recirculated room air.

(1) These systems waste energy because supply air is cooled to a low enough temperature to serve the zone needing the coolest air, but then supply air must be reheated for other zones to avoid overcooling. Where constant volume is maintained, the waste of energy can be even more significant. Reset cold deck temperature to meet

cooling requirements of the room with the largest load or to satisfy humidity requirements. This cold deck temperature control reduces energy consumption.

(2) Due to high energy consumption, limit these systems to applications requiring close control of temperature and humidity, such as hospital intensive care areas and laboratories. When economically feasible, use heat recovered from the refrigeration cycle in heating coils.

3.3.2 Variable Air Volume (VAV) Systems. Use VAV systems for buildings with sufficient zones (11 or more zones) and load variation to permit reduction of fan capacity for significant periods during the day. Do not use bypass VAV systems. The complexity of systems should be consistent with minimum requirements to adequately maintain space conditions.

3.3.3 Economizer Cycle. The economizer cycle should not be used in humid climates and for spaces where humidity control is critical, such as computer rooms. Problems have been experienced with linkage corrosion, excessive damper leakage, jammed linkage on large dampers, and inadequate maintenance. Outdoor air dampers should be located away from the intake louver and after duct transition to minimize exposure to weather and size of dampers. Provide outdoor air dry bulb changeover rather than enthalpy or outdoor air/return air comparator changeover. With VAV systems, return or relief fans shall not be used. An economizer should only be used when it can be designed with gravity relief through the building envelope. Size gravity relief dampers to prevent building over pressurization.

3.4 System and Equipment Performance. For size and selection criteria of systems and equipment, refer to ASHRAE Equipment Handbook. HVAC systems shall be able to dehumidify supply air under loading conditions, provide reliable operations, and tolerate reasonable variations in chilled-water temperatures. Air conditioning systems generally operate at part load conditions most of the time. This is particularly true of comfort air conditioning systems which often operate at less than 50 percent of their design load

capacity for more than 50 percent of the time. Since high part load efficiencies are desirable to conserve energy, the selection of equipment and step starting and sequencing controls shall be made with an emphasis on reducing life-cycle costs at part load conditions. Verify and document the equipment operation in accordance with ASHRAE Guideline 1, Commissioning of HVAC Systems.

4. AIR DISTRIBUTION

4.1 Duct Design for HVAC Systems

4.1.1 Sizing General. ASHRAE Handbook, Fundamentals recognizes three methods of sizing ductwork: the equal friction method, the static regain method, and the T-method. The ASHRAE Handbook also provides a commonly used chart for sizing ducts using the equal friction method. For design of small simple systems, the equal friction method will suffice. Use the static regain method for VAV.

4.1.2 Equal Friction Method Sizing. Select a constant pressure loss in inches of water per 100 foot length of duct from the preferred part of the ASHRAE equal friction sizing chart. The preferred part is between 0.08 and 0.6 inches of water per 100 feet friction loss for air quantities up to 18,000 cfm, and between 1800 fpm and 4000 fpm for air quantities greater than 18,000 cfm. Use low velocities and a low friction drop for small projects, or where ductwork is cheap and energy is expensive. For systems of 18,000 cubic feet per minute and over, use a friction loss of 0.08 and velocities of 1800 to 3000 feet per minute. After sizing the entire system at the selected unit pressure drop, go back and adjust velocities and pressure drops in the shorter branches to equalize the pressure drops at each duct branch junction.

4.1.3 Ductwork, General

4.1.3.1 Round Ducts. Use round ducts wherever possible. Under normal applications, the minimum duct size shall be 4 inches in diameter. Use smooth curved elbows as much as possible. If these are not available, use three-piece elbows for velocities below 1600 feet per minute and five-piece elbows for velocities above 1600 feet per minute. The throat radius shall not be less than 0.75 times the duct diameter.

4.1.3.2 Rectangular Ducts. Use a minimum duct size of 6 inches by 6 inches. Where possible, keep one dimension constant in transitions and do not make transitions in elbows. Make transitions in sides and bottom of the duct keeping top level to maintain

maximum clearance above ceiling. The transition slope shall be 30 degrees on the downstream. Where ductwork is connected to equipment fittings such as coils, furnaces, or filters, the transition shall be as smooth as possible. Drawings shall indicate ductwork pitch, low spots, and means of disposing of the condensate. Elbows shall be smooth, with an inside radius of 1.0 times the width of the duct. Where space constraints dictate use of mitered elbows, such elbows shall have single thickness turning vanes. Using double thickness turning vanes instead of single thickness vanes increases the pressure loss of elbows by as much as 300 percent. Use the circular equivalents table in ASHRAE Handbook, Fundamentals instead of matching areas when you change aspect ratios. The aspect ratio is the ratio of larger to smaller rectangular duct dimension. Try to use an aspect ratio of 3 to 1 with a maximum aspect ratio of 6 to 1 or less.

4.1.3.3 Access Doors. Show access doors or panels in ductwork for apparatus and devices for maintenance, inspection, and servicing.

4.1.3.4 Flexible Ducts. To save construction expense, flexible duct may be used to connect ceiling outlets. Limit the length of flexible ducts to straight runs of 5 feet. Seek self-balancing by having equal lengths of flexible ducts instead of long and short lengths on the same branch. Do not use flexible ducts for elbows, including connection to diffusers; provide elbows at ceiling diffusers. Do not use flexible ducts in industrial ventilation systems.

4.1.3.5 Rooftop Ductwork. Rooftop ducts exposed to the weather can leak rain water. Exterior insulation tends to have a short life. One way to avoid such problems is to put insulation inside the duct, and then use galvanized steel ductwork with soldered joints and seams. Exterior insulation shall have weatherized coating and wrapping throughout, where it must be used; such as on kitchen exhaust hoods containing grease.

4.1.3.6 Glass Fiber Ductwork. Investigate the bidding climate in your local area before deciding that ductwork made from glass fiber panels will always be less expensive than galvanized steel ductwork. Fiberglass ductwork should be coated inside to avoid bacteria growth. In some parts of the country the sheet metal subcontractor can make or buy metal ducts made on an automatic machine at competitive prices.

4.1.3.7 Balancing Dampers for HVAC. Provide balancing dampers on duct branches and show dampers on drawings. See Figure 4-1 for damper installation. Use extractors or volume dampers instead of splitter dampers at branch connections. Do not use splitter dampers since they make ductwork more difficult to balance than a job with volume dampers. Provide access in the ceiling and clamping quadrants for dampers or use a type with a remote control that extends through the ceiling. Outdoor air dampers should be located away from the intake louver and after the duct transition to minimize exposure to weather and oversizing of dampers. Avoid using balancing dampers for industrial ventilation (IV) systems. Design IV ductwork so that the system will function properly without balancing dampers. Do not use balancing dampers when designing a VAV system. A VAV system with ductwork designed using the static regain method and properly sized VAV terminal units is inherently self-balancing.

4.1.3.8 Fire Dampers and Smoke Dampers

a) Fire Dampers. The term "fire damper" usually means a curtain type damper which is released by a fusible link and closes by gravity or a mechanical spring. Fire dampers are mounted in walls of fire rated construction to ensure integrity of the space. Fire dampers should be installed where the passage of flame through a fire rated assembly is prohibited.

b) Combination Fire and Smoke Dampers. The term "combination fire and smoke damper" usually means a fire damper which is automatically controlled by an external source (such as a fire alarm control panel or energy management system) to stop passage of both fire and smoke. Combination fire and smoke dampers should be installed where passage of fire or smoke is prohibited. Activation of combination fire and smoke dampers can be by several methods including pneumatic damper operators, electric damper operators, and electro-thermal links. Electro-thermal links include explosive squibs which are not restorable and McCabe type links which are restorable. Pneumatically operated dampers are the preferred method of damper activation, and should be configured in the fail-safe mode such that loss of pneumatic pressure will result in dampers closure.

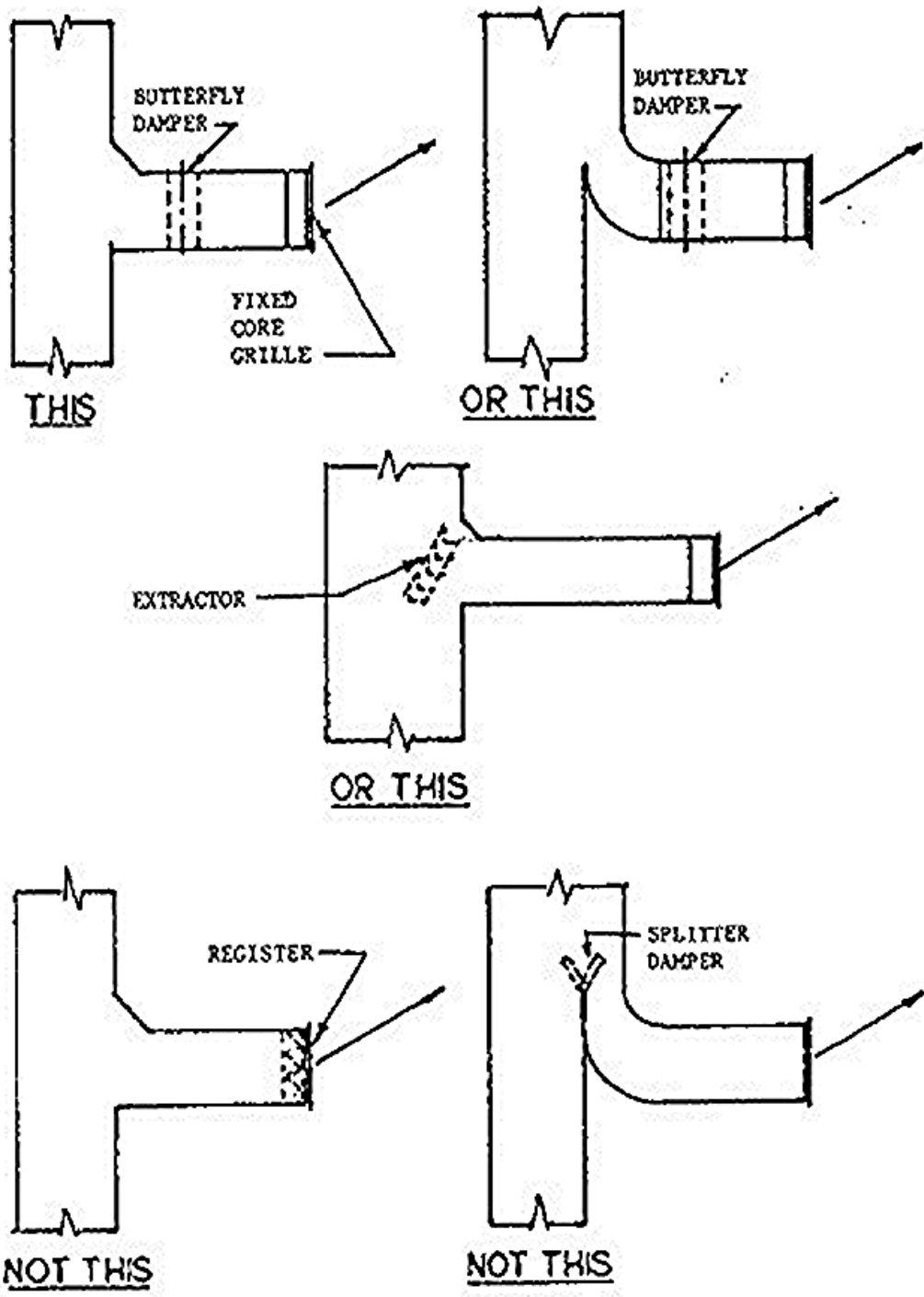


Figure 4-1

In electronic data processing rooms, combination fire and smoke dampers should be installed in walls with a fire resistance rating of 1 hour or greater. In other type spaces, either fire dampers or combination fire and smoke dampers should be installed in walls with a fire resistance rating of 2 hours or greater. Where a smoke damper is required to stop passage of smoke through a barrier (e.g., hospitals), the installation of a combination fire and smoke damper is required.

c) Mounting Details. Fire dampers and combination fire and smoke dampers must remain in the wall during a fire. Though ductwork may collapse, the damper should remain in the fire rated assembly, therefore, indicate on drawings the details for attaching dampers to the wall. Use UL listed firestopping materials between the damper collar and the wall, floor, or ceiling assembly where penetrated.

4.1.3.9 Fan System Effect Factors. Fans are tested and rated based upon a certain standard ductwork arrangement. If installed ductwork creates adverse flow conditions at the fan inlet or fan outlet, loss of fan performance is defined as a system effect factor. The system effect factor can be caused by obstructions or configurations near the fan inlet and outlet. For example, failure to recognize the affect on performance of swirl at the fan inlet will have an adverse effect on system performance. Refer to Air Movement and Control Association (AMCA) 201, Fans and Systems for additional information on fans and system effects.

4.1.4 Ductwork Details

4.1.4.1 Branches. See Figure 4-2 and Figure 4-3.

4.1.4.2 Elbows. See Figure 4-4.

4.1.4.3 Offsets and Transitions. See Figure 8/4-5.

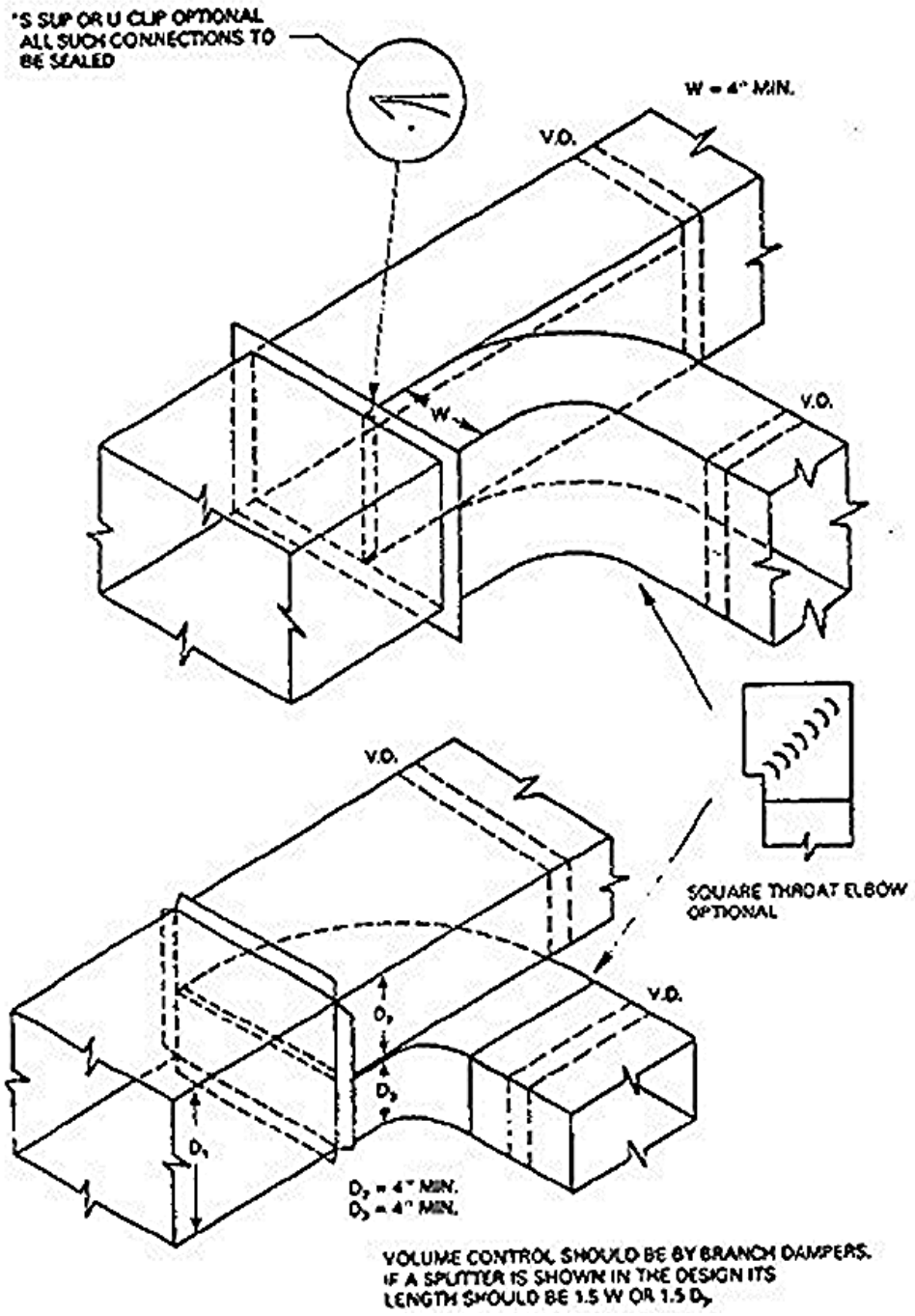
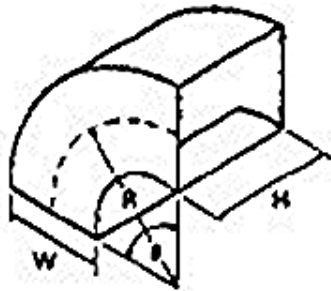
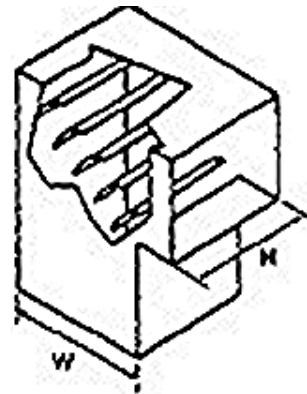


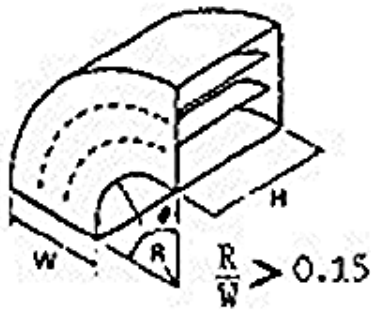
Figure 4-2



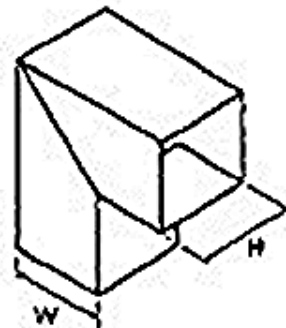
**TYPE 1
RADIUS ELBOW
(CENTERLINE $R = \frac{3W}{2} = \text{STD RADIUS}$)**



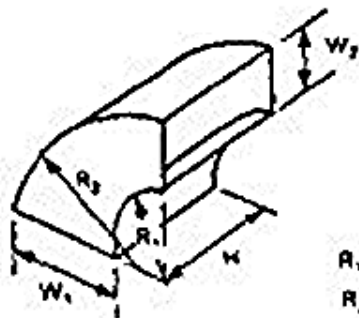
**TYPE 2
SQUARE THROAT ELBOW
WITH VANES**



**TYPE 3
RADIUS ELBOW
WITH VANES**



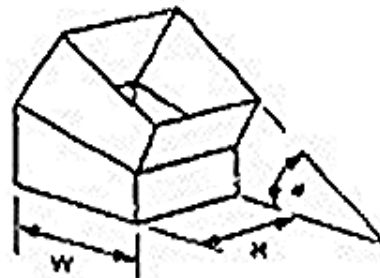
**TYPE 4
SQUARE THROAT ELBOW
WITHOUT VANES
(1000 FPMs MAXIMUM VELOCITY)**



**TYPE 5
DUAL RADIUS ELBOW**

$$R_1 = \frac{1}{2}W_1$$

$$R_2 = R_1 + W_2$$



MITERED ELBOW

BEAD, CROSSBREAK AND REINFORCE FLAT SURFACES AS IN STRAIGHT DUCT

Figure 4-4

OFFSETS 2 AND 3 AND TRANSITIONS MAY HAVE EQUAL OR UNEQUAL INLET AND OUTLET AREAS. TRANSITIONS MAY CONVERT DUCT PROFILES TO ANY COMBINATION FOR RECTANGULAR, ROUND OR FLAT OVAL SHAPES.

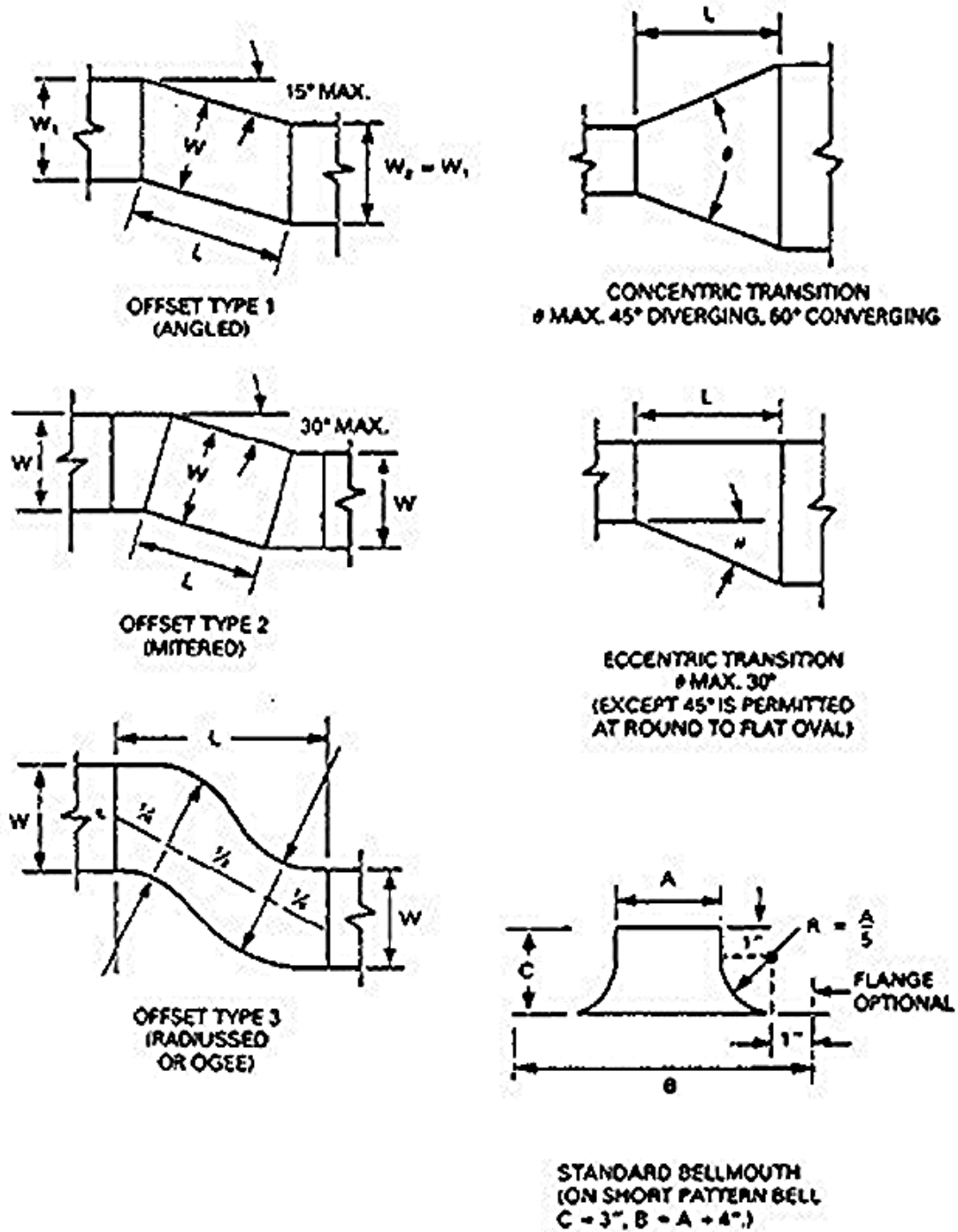


Figure 4-5

5. RULES OF THUMB

5.1 General. The following information provides guidance that could be used in planning to estimate utility requirements and to assess the adequacy of equipment sizing during design reviews. Note that it is preferable to do a quick block load calculation instead of using these rules of thumb.

5.2 Air Conditioning Capacity. See Table 5-1.

5.3 Heating Capacity. 35 to 40 Btu per square foot for mild climate region (less than 4,000 degree days), no fresh air load.

5.4 Chilled Water Circulation. 2.5 to 3.0 gallons per minute per ton.

5.5 Hot Water

Gallons per minute (20 degree drop) = Btu/h/10,000

Gallon per minute = (Btu/h)/(500 x TD) (temperature drop)

5.6 Condenser Water. Required thermal capacity of cooling water = 15,000 Btu/h per ton, or = 3 gpm per ton

5.7 Steam. 1 pound of steam per 1,000 Btu.

5.8 Condensate. 120 gallons per 1,000 pounds steam.

APPLICATIONS	AIR CONDITIONING FLOOR AREA SQ. FT./TON (EXCEPT WHERE NOTED)
ADMINISTRATION BUILDING	450-600
AUDITORIUMS, THEATERS	0.004 TO 0.08 TONS/SEAT
BOWLING ALLEYS	0.8 TO 1.4 TONS/ALLEY
COMPUTER ROOMS	50 TO 150
DINING ROOMS	175 TO 450
DISPENSARIES	450 TO 550
ENLISTED MEN'S AND OFFICER'S CLUBS	275 TO 375
HOSPITAL PATIENT ROOMS	450 TO 550
MULTIPLE FAMILY HOUSING UNITS	900 TO 1275
RECREATION ROOMS	375 TO 450
RELIGIOUS FACILITIES	0.02 TO 0.03 TONS/SEATS
HOPS (PRECISION EQUIPMENT)	450 TO 550
TRAINING FACILITIES	400 TO 500
BACHELOR QUARTERS	725 TO 900

Table 5-1
Air Conditioning Load Estimating Factors